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Power Plant Engineering

Third Edition

P K Nag

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Dr. P K Nag had been with the Indian Institute of Technology, Kharagpur, for about forty years, almost since his graduation. After retirement from IIT, he was an Emeritus Fellow of AICTE, New Delhi, stationed at Jadavpur University, Kolkata, till June, 2005. He was a Visiting Professor in Technical University of Nova Scotia (now Dalhousie University), Halifax, Canada, for two years during 1985–86 and 1993–94. He has authored four books, including this one, and about 150 research papers in several national and international journals and proceedings. His research areas include circulating fluidized bed boilers, combined cycle power generation, second law analysis of thermal systems, and waste heat recovery. He was the recipient of the President of India medal (1995) from the Institution of Engineers (India). He is a Fellow of the National Academy of Engineering (FNAE) and a Fellow of the Institution of Engineers (India). He is a Life Member of the Indian Society for Technical Education, Indian Society for Heat and Mass Transfer, and the Combustion Institute, USA (Indian section). He was formerly a Member of the New York Academy of Sciences, USA.

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Power Plant Engineering

Third Edition

P. K. Nag

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Preface

Convinced by the suggestions made by the reviewers and users of the book, two new chapters, viz., *Non-conventional Power Generation: Direct Energy Conversion,* and *Environmental Degradation and Use of Renewable Energy,* have been added in this revised edition. Considering the current global environmental scenario, the last chapter is particularly important and considerable emphasis has been stressed on fuel cells and solar-hydrogen system, which seem to be the most significant future contributors in ridding us from the dual crises of energy and environment.

Fluidized beds, besides being one of the best Clean Coal Technologies, are also proving to be the most practical option for biomass conversion. The book boasts of providing an up-to-date information on fluidized bed technology, both in bubbling and circulating modes of combustion.

Meant for the undergraduate course on ***Power Plant Engineering*** studied by the mechanical engineering students, this book is a comprehensive and up-to-date offering on the subject. It has detailed coverage on hydroelectric, diesel engine and gas turbine power plants.

The book is divided into fourteen chapters. The first chapter introduces the economics of power generation. Chapter 2 is on the analysis of steam cycles. Chapters 3 and 4 discuss combined cycle power generation, and fuels and combustion respectively. Chapter 5 further discusses topics on combustion mechanism, combustion equipment and firing methods. Chapters 6 and 7 are on steam generators and steam turbines. Chapter 8 is on condenser, feedwater and circulating water systems. Chapter 9 is on nuclear power plants, Chapter 10 is on hydroelectric power plants and Chapter 11 is on diesel engine and gas turbine power plants. Chapter 12 discusses energy storage. As already mentioned, two new chapters have been added at the end. Chapter 13 is on non-conventional power generation and direct energy conversion while Chapter 14 discusses environmental degradation and use of renewable energy.

The book has about a **hundred solved examples** and **two hundred practice problems.** Moreover it has over **750 short-answer questions** to give the students a thorough practice of the concepts learnt. The numerous self-explanatory **illustrations** makes this a very student-friendly text. The key features are as follows:

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- In-depth coverage of Hydroelectric, Diesel Engine and Gas Turbine Power Plants
- Analytical and theoretical treatment of concepts with the right blend of theory, design and practice of power stations
- Unique chapter on Energy Storage Mechanisms
- Detailing of combined cycle mode of power generation and fluidized bed boilers

In addition, instructors can get help from the **solutions manual** and also from **Power Point slides** which can be used as teaching aides. These can be downloaded from the Online Learning Centre of the book.

The book has become quite voluminous and it would be difficult to complete it in one semester. The teacher has to adopt judiciously the portions necessary for the curriculum on the subject.

I am highly grateful to Ms. Shukti Mukherjee of McGraw-Hill Education for the exhaustive review of the book. I would like to express my sincere thanks to Prof. A Valan Arasu, Thiagarajar College of Engineering, Madurai; Ms S Supriya, Govt. College of Engineering, Tirunelveli; Dr G Kamaraj, Annamalai University; Dr Niranjan Sahoo, IIT Guwahati; Dr Suman Chakraborty, IIT Kharagpur; Prof. B K Mondal, Bengal Engineering and Science University; Dr R R Kulkarni, Vishwakarma Institute of Technology, Pune; and Dr B J Ranganath of Vidya Vikas Institute of Engineering and Technology, Mysore, for taking pains to go through the book and provide useful comments. He is also thankful to Ms. Vibha Mahajan for the overall supervision and all others of McGraw-Hill who helped in the publication of this new edition.

The author would appreciate if readers point out any mistakes and graciously make suggestions for further improvement of the book, which would be gratefully acknowledged.

June, 2007 **P. K. Nag**

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1

Introduction: Economics of Power Generation

1.1 INTRODUCTION

Electricity is the only form of energy which is easy to produce, easy to transport, easy to use and easy to control. So, it is mostly the terminal form of energy for transmission and distribution. Electricity consumption per capita is the index of the living standard of people of a place or country.

Electricity in bulk quantities is produced in power plants, which can be of the following types: (a) Thermal, (b) Nuclear, (c) Hydraulic, (d) Gas turbine and (e) Geothermal. Thermal, nuclear and geothermal power plants work with steam as the working fluid and have many similarities in their cycle and structure. Gas turbine plants are often used as peaking units. They run for short periods in a day to meet the peak load demand. They are, however, being increasingly used in conjunction with a bottoming steam plant in the mode of combined cycle power generation. Hydraulic power plants are essentially multipurpose. Besides generating power, they also cater for irrigation, flood control, fisheries, afforestation, navigation, etc. They are, however, expensive and take long time to build. There is also considerable opposition against their erection due to the ecological imbalance they produce. Geothermal power plants can be built only in certain geographical locations.

Thermal power plants generate more than 80% of the total electricity produced in the world. Fossil fuels, viz. coal. fuel oil and natural gas are the energy source, and steam is the working fluid. Steam is also required in many industries for process heat. To meet the dual need of power and process heat, cogeneration plants are often installed.

There has been an exponential growth in the production of electricity. If the electricity production E increases at the same fractional rate, i, each year, the rate of change of electricity production per year becomes

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$$\frac{dE}{dt} = Ei \tag{1.1}$$

After integration,

$$\ln \frac{E}{E_0} = i\,(t - t_0)$$

or,

$$E = E_0\, e^{i(t-t_0)} \tag{1.2}$$

where E_0 is the electricity production in the base year t_0. This exponential behaviour gives rise to the term called "doubling time" which indicates the time required for electrical power production to double. If E_1 is the power generation at time t_1 and E_2 is the power generation at time t_2, then from Eq. (1.2).

$$\frac{E_2}{E_1} = e^{i(t_2-t_1)} \tag{1.3}$$

If we denote the doubling time by t_d, then

$$t_d = t_2 - t_1 \qquad \text{and} \qquad \frac{E_2}{E_1} = 2$$

Therefore,

$$\ln 2 = it_d \qquad \text{or,} \qquad t_d = \frac{0.693}{i} \tag{1.4}$$

Using this simple expression, for various rates of increase in electricity production per year, the doubling time can be estimated. From the trend of production, it is observed that $i = 6.2\%$ and so the doubling time t_d is about 11.2 years.

It is found that the demand for electricity bears a linear relationship with the gross national product (GNP) of a country. Projection of future demand of electricity is thus tied to estimates of economic growth of the concerned region. With the increase in economic growth, the consumption of electricity also increases.

1.2 LOAD–DURATION CURVES

When planning a power plant, the two basic parameters to be decided are:

1. Total power output to be installed (kW_{inst}).
2. Size of the generating units.

The total installed capacity required can be determined from:

1. First demand (kW_{max}) estimated.
2. Growth of demand anticipated.
3. Reserve capacity required.

The size of the generating units will depend on

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1. Variation of load (load curve) during 24 hours (summer, winter, weekdays, holidays).
2. Total capacity of units connected to the electric grid.
3. Minimum start-up and shut-down periods of the units.
4. Maintenance programme planned.
5. Plant efficiency vs. size of unit.
6. Price and space demand per kW vs. size of unit.

For estimating the expected maximum load, the most useful tool is the construction of a hypothetical load curve. Figure 1.1 shows the typical hourly load curve in a metropolitan area, from which the maximum demand can be estimated. Several factors used in the electricity supply industry in this regard are defined below:

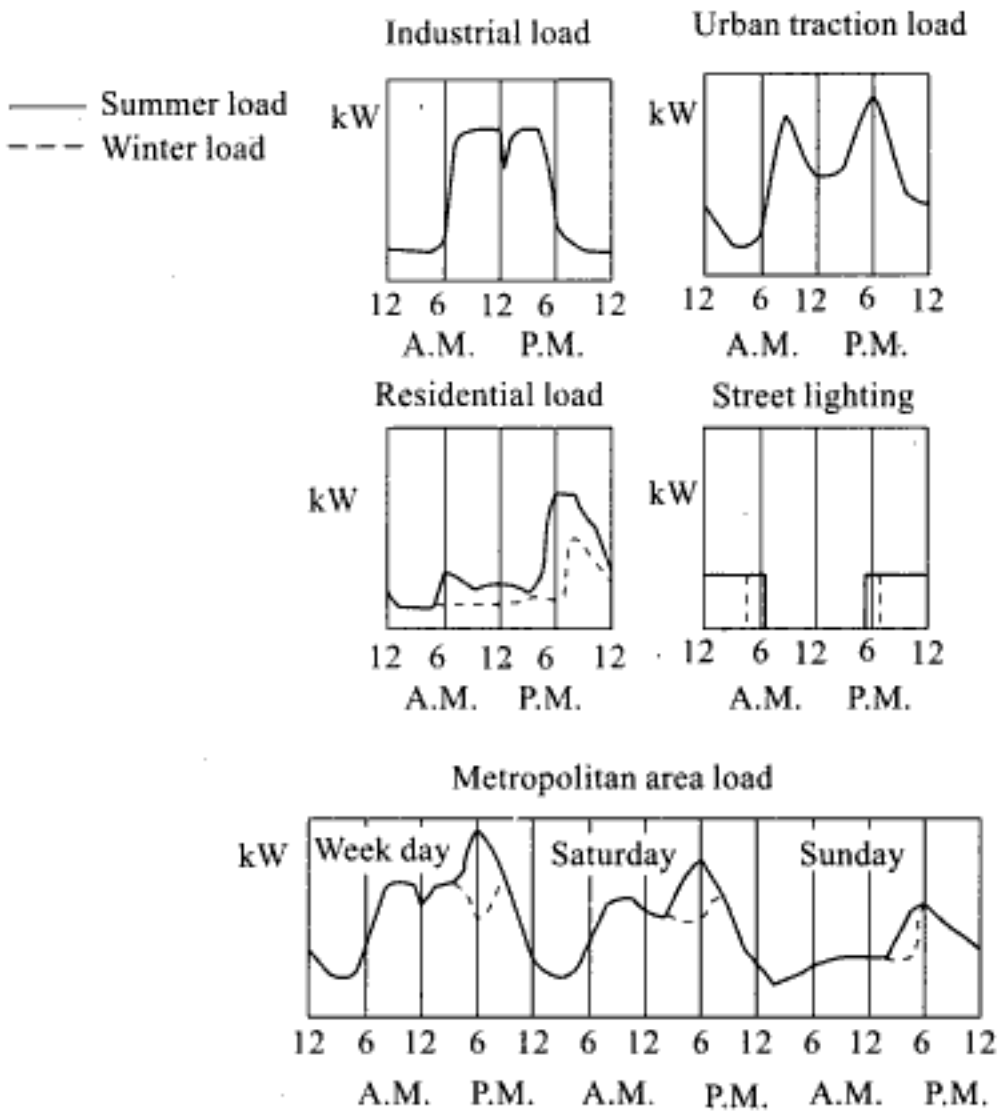

Fig. 1.1 *Typical hourly variation in energy demand for different types of electrical load and curves of total load in a metropolitan area*

1. Load factor, $$m = \frac{\text{average load over a given time interval}}{\text{peak load during the same time interval}}$$

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$$= \frac{\text{kWh(av.) in a year}}{\text{kW}_{max} \times 8760} \quad \text{(one year = 8760 hr)}$$

Thus, if a plant has a peak load of 1000 MW but the average annual load is 350 MW, its annual load factor would be only 0.35. Most of its capacity remains unutilized for major part of the year, and so the cost would be high.

The average load is estimated by dividing the area under the daily load curve by the time period considered (Fig. 1.2).

$$\text{Average load} = \frac{\text{area under load curve (kWh)}}{24 \text{ (h)}}$$

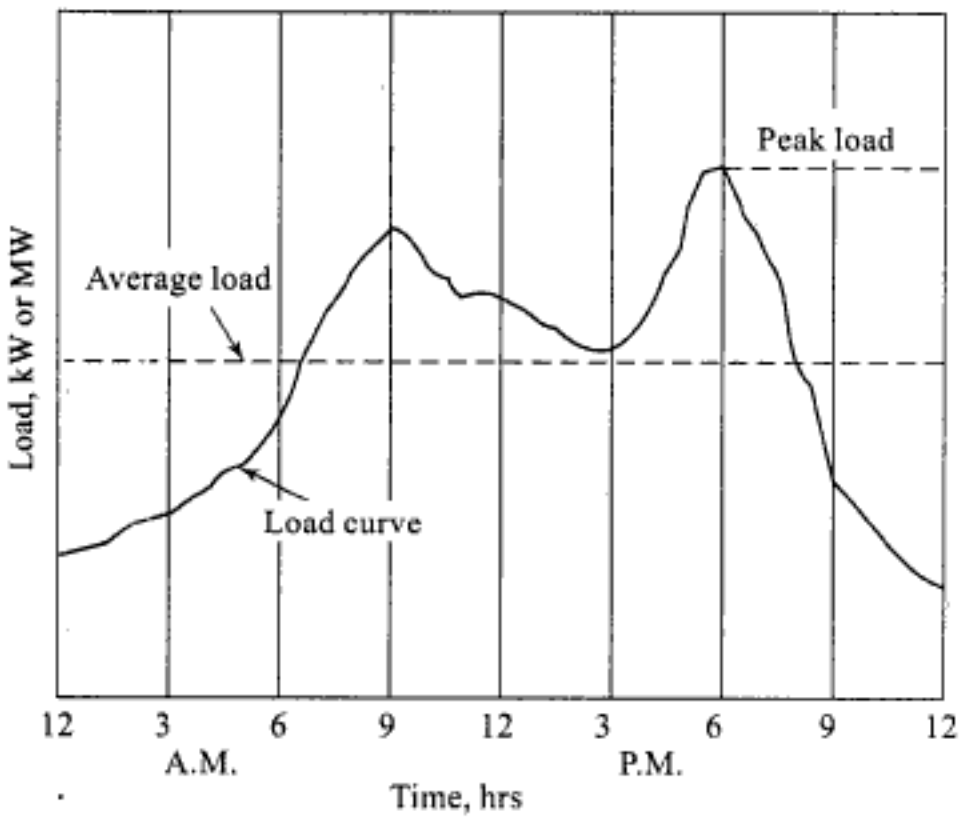

Fig. 1.2 *Load curve*

2. Capacity factor (or plant factor),

$$n = \frac{\text{average load}}{\text{rated capacity of the plant}}$$

$$= \frac{\text{kWh generated in a year}}{\text{kW}_{inst} \times 24 \times 365} = \frac{\text{kWh}_{gen} \text{ (year)}}{\text{kW}_{inst} \times 8760}$$

If the rated capacity of the plant is equal to the peak load, then the load factor and capacity factor will be numerically equal.

The difference between load factor and capacity factor is an indication of the reserve capacity.

3. Reserve factor, $$r = \frac{\text{kW}_{inst}}{\text{kW}_{max}} = \frac{m}{n} = \frac{\text{load factor}}{\text{capacity factor}}$$

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Each consumer has a "connected load" which is the sum of the continuous ratings of all the equipment and outlets on the consumer's circuits. The maximum demand is the maximum load which a consumer uses at any time. It is always less than, or equal to, the connected load.

4. Demand factor,

$$\text{dem} = \frac{\text{actual maximum demand}}{\text{total connected demand}} = \frac{\text{kW}_{\text{max}}}{\text{kW}_{\text{conn}}}$$

The time distribution of maximum demands for similar types of consumers is measured by a term called "diversity factor". It is the ratio of the sum of the maximum demands of the individual consumers and the simultaneous maximum demand of the whole group during a particular time.

5. Diversity factor,

$$\text{div} = \frac{\text{sum of individual consumer groups}}{\text{actual peak load of the system}}$$

The diversity factor is illustrated in Fig. 1.3(a). It is given to be

$$\text{div} = \frac{a+b+c}{d}$$

Diversity helps to improve the load factor and economic operation of the power plant.

Figure 1.3(b) shows the load curves of an industrialized country, the annual peak occurring in winter. The seasonal variation in the load influences the maintenance programme of the power plants. Major maintenance work (cleaning, overhauling etc.) has to be done during the off-peak season.

A high load factor is, in general, an indication of balanced load curve with relatively small load changes.

High values of demand factor, load factor, diversity factor and capacity factor are desired for economic operation of the plant and to produce electricity at less cost.

$$\text{Plant use factor, } u = \frac{\text{kWh}_{\text{gen}}}{\text{kW}_{\text{inst}} \times \text{operating hours}}$$

It is the ratio of energy produced in a given time to the maximum possible energy that could have been produced during the same time of operation. If the operating time is 1 year or 8760 hrs, the plant use factor is equal to the capacity factor (i.e., $u = n$)

As the plant-use factor approaches 1, it indicates the need for additional capacity of the plant. The plant capacity is always designed to be greater than the peak load to take extra loads coming in future. The high value of the plant use factor indicates that the plant is operating quite efficiently. In some inter-connected systems, the plant use factor may exceed unity (e.g., 1.1 or 1.2) indicating that the loads carried are in excess of the rated capacity since an equipment is always designed to take 10 to 20% more load than rated. The different factors are indicated in Fig. 1.3 (c).

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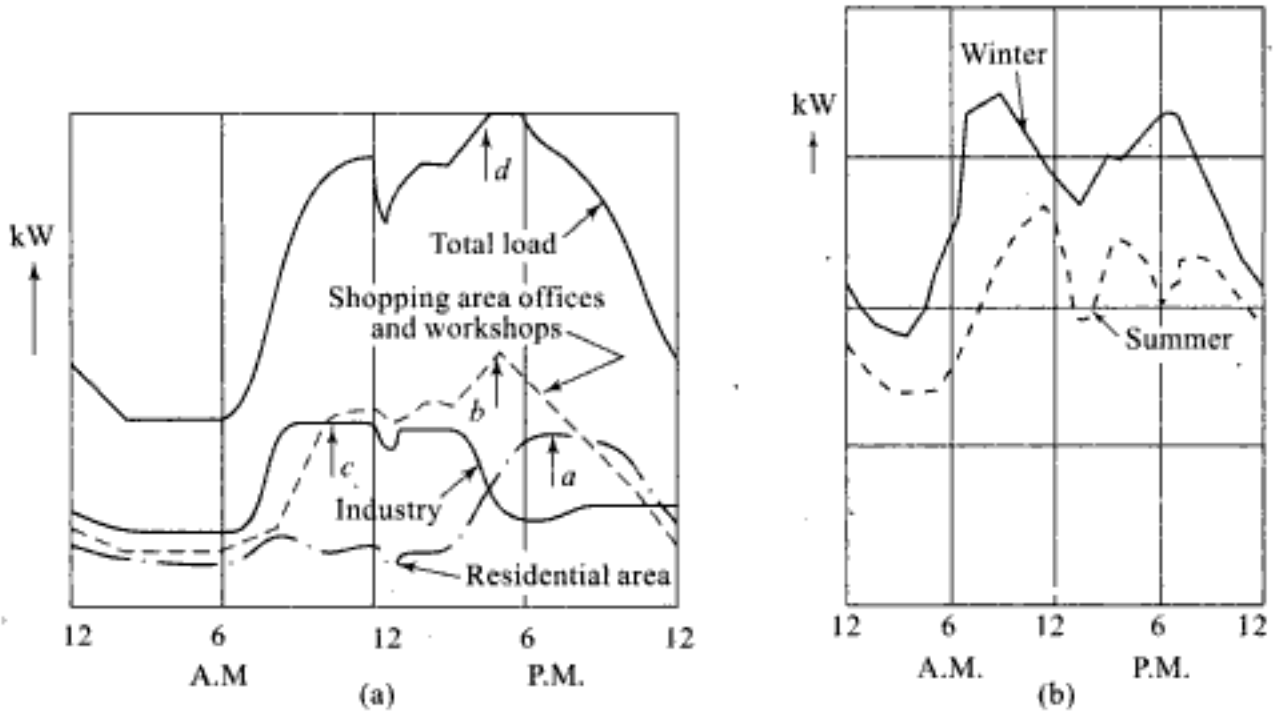

Fig. 1.3 *(a) Diversity of loads of consumer groups*
(b) Load curves of an industrialized country on typical winter and summer week-days

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load}}$$

$$\text{Use factor} = \frac{\text{Peak load}}{\text{Plant capacity}}$$

$$\text{Capacity factor} = \frac{\text{Average load}}{\text{Plant capacity}}$$

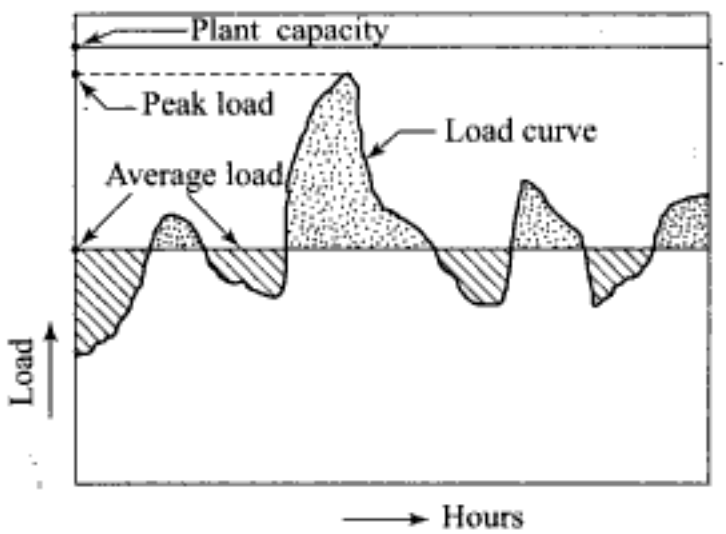

Fig. 1.3 *(c) Representation of different factors*

It can be seen that

$$\text{Load factor} \times \text{Use factor} = \text{Capacity factor}$$

Larger the unit size, less is the cost of electricity produced. Therefore, larger units are more economical than smaller units as regards investment per kW and operating expenses per kWh. Investigations should be made in each individual case, considering all operational, technical and economic factors involved, to determine the maximum unit size admissible. A reserve factor of 1.2 to 1.25 is often found to be satisfactory. The unit size must not exceed 20 to 25% of the total generating capacity of the plant.

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In order to obtain a basis for economical operation of the various types of generating units normally connected to a grid and also for planning of tariffs, it is useful to construct load duration curves for typical days which indicate the power demands at different intervals of time. There can be residential loads (domestic lights, heaters, refrigerator, TV, audio and video cassette recorders, etc.), commercial loads (shops, offices, advertisements, restaurants, etc.), industrial loads (various mills and factories), municipal loads (street lighting, power for water supply and drainage), irrigation loads and traction loads (railways, trams, trolleys, etc.).

The construction of a load duration curve for one day from the chronological load curve of this day is illustrated in Fig. 1.4(b). It rearranges the load elements of a chronological curve in the descending order of magnitude. The area under the curve is

$$\text{kWh} = \int_0^{24} \text{kW}\, dt$$

during the day. The load duration curve indicates for how many hours a certain load has been required in the course of the day. By compiling the load hours on various days, the load duration curves for one month (720 hr) or one year (8760 hr) can be obtained (Fig. 1.4). Daily, monthly and yearly load duration curves aid better planning for economical utilization of the power plants, existing or planned.

Figure 1.5 shows the construction of a load-duration curve using the daily load curve. The area under the annual load duration curve represents the total energy supplied by the utility's generating system during the year. It is usually divided into three parts:

1. Base load
2. Intermediate load
3. Peaking load

The base load is the load below which the demand never falls and is supplied 100% of the time. The peaking load occurs for about 15% of the time. The intermediate load represents the remaining load region.

Since peaking load plants are used only for a small fraction of time, the fuel cost is not of major importance. Minimum capital cost should be the criterion.

The base load plants are plants which are loaded very heavily. Operating costs of such plants are very important. A high capital cost is permissible if low operating costs can be maintained (e.g. new, large coal and nuclear power stations). Intermediate load plant can be somewhat smaller than older units. By judicious combination of all the tree types of generation—peak, intermediate and base—maximum economy can be achieved.

The variable load problem affects power plant design and operation as well as the cost of generation. A careful study of the load duration curve helps to decide the capacity of the base load plant and also of the peak load plant. The base load plant should be run at high load factor. The peak load plant should be of smaller capacity to reduce the cost of generation. It could be a gas turbine

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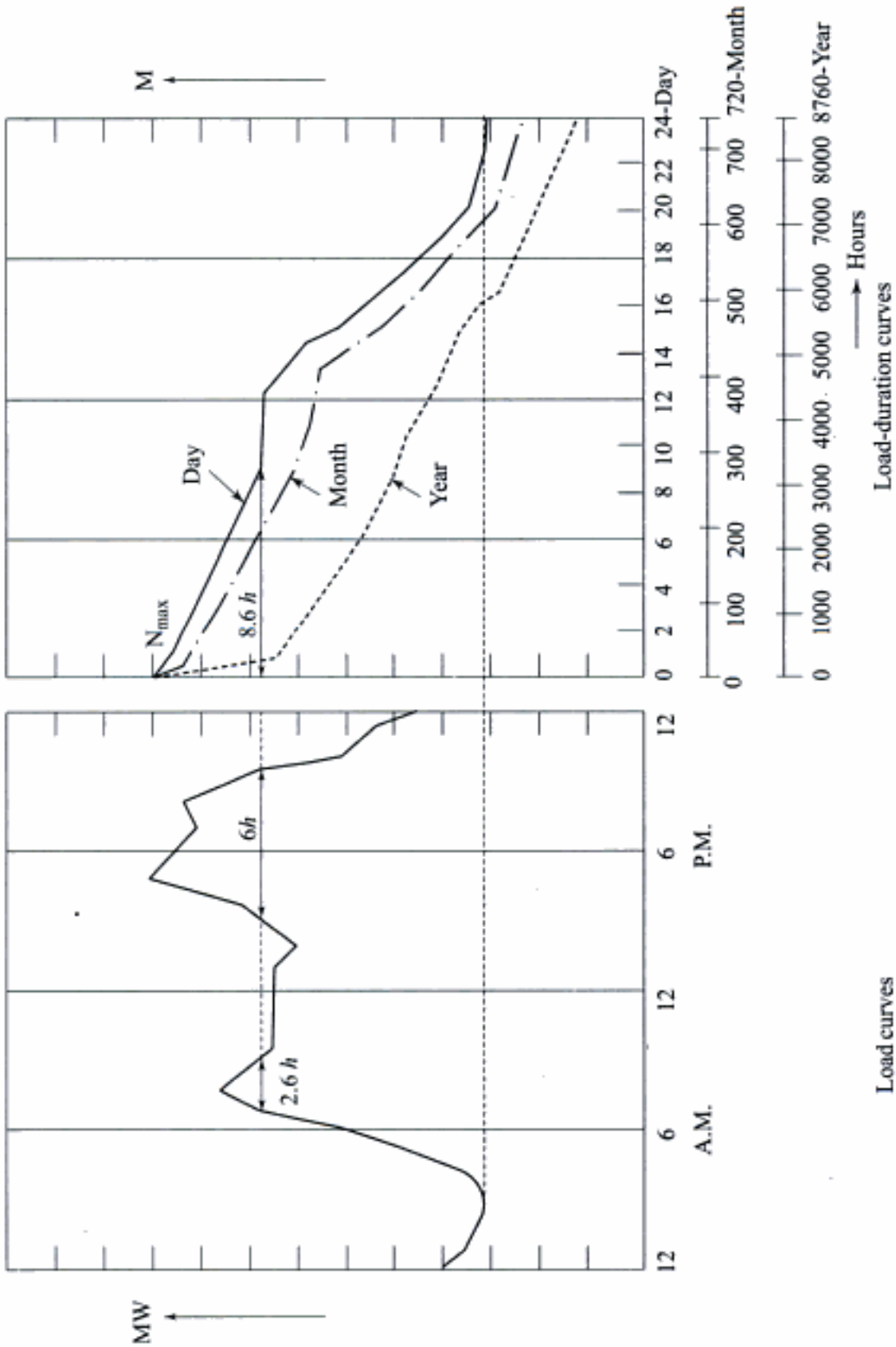

Fig. 1.4 *Load-duration curve constructed from the chronological load curve of a day*

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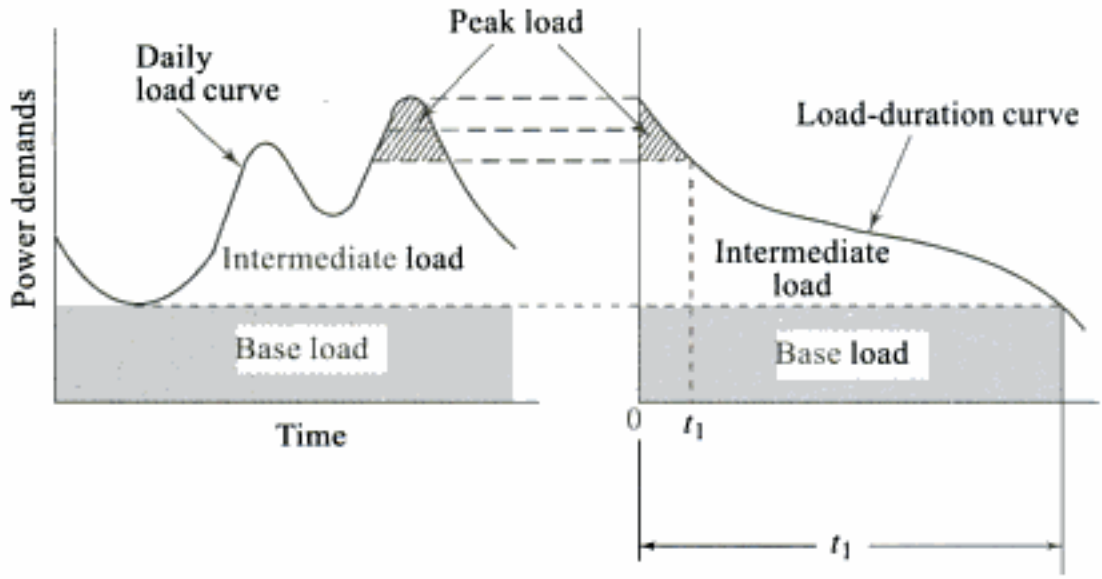

Fig. 1.5 *Construction of the load-duration curve*

unit, pumped hydro-system, compressed air energy storage system or a diesel engine depending on the size and scope of availability. If the whole of load is to be supplied by the same power plant then the prime movers and generators should act fairly quickly and take up or shed load without variation of the voltage or frequency of the system. It is the function of the governor to control the supply of fuel to the prime mover according to load. The capacity of the generators should be so chosen as to suit and fit into the portions of the predicted load curve. If the load conditions differ too much from this capacity, the cost of energy increases.

1.3 LOCATION OF POWER PLANTS

The location of hydroelectric power plants is usually predetermined by the availability of water and the water head which is utilized. For conventional base load thermal power plants, the following factors are to be considered:

1. Availability of cooling water (if cooling towers are used the possibility of adequate make-up water).
2. Availability of fuel (water, rail or pipe connection to the fuel source, and the cost of fuel transport).
3. Distance from the centre of gravity of load demand.
4. Cost of land (including space for extension, maintenance workshop and storage yard).
5. Character of soil.
6. Main wind direction and water currents in cooling water source (sea, lake or river) in order to minimize air and water pollution, and other ecological considerations.
7. With coal-fired stations, disposal of ash.
8. If the plant is erected far from a town, accommodation for staff.
9. Rail and road connections.
10. Security considerations.

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For nuclear power plants two additional factors are to be considered.

11. Density of population in the vicinity.
12. Danger of earthquakes.

1.4 POWER PLANT ECONOMICS

A power plant should provide a reliable supply of electricity at minimum cost to the consumer. The cost per kWh_{net} is determined by:

1. Fixed costs (FC), mainly interest, depreciation, insurance, taxes, depending on the capital invested, i.e. on the construction costs of the plant including the cost of the land.
2. Operation and maintenance (O & M) costs covering salaries and wages, overhauling of equipment, repairs including spare parts, water, lubricating oil, chemicals and miscellaneous expenses.
3. Fuel costs, dependent on the amount of electricity generated.
4. kWh_{net} of electricity sent out per year.

The total annual costs (C_t) in a power plant can be calculated from

$$C_t = \frac{I + D + T}{100} C_c + (W + R + M) + C_f \tag{1.5}$$

where I is the interest, %; D is depreciation, %; T is taxes and insurance, %; C_c the construction cost; W is wages and salaries; R is repairs (maintenance); M is miscellaneous; and C_f the fuel cost.

The annual amount of electricity sent out by a power plant (kWh_{net}) is given by

$$kWh_{net} = kW_{inst} \times 8760 \times \left(1 - \frac{L_{aux}}{100}\right) \times n \tag{1.6}$$

where kW_{inst} is the rated (installed) output of generators; L_{aux} the power consumption by the auxiliaries, %; n the plant capacity factor, and 8760 = 24 × 365 hours per year.

In order to calculate the electric power cost to a consumer, in addition to the production cost (fixed cost, operation and maintenance, and fuel cost), the transmission cost, distribution cost, administrative expenses, and return or profit on the investment have to be taken into consideration.

A measure for the reliability of a power plant is the forced outage rate defined by the annual ratio of

$$\frac{\text{forced outage hours}}{\text{service hours + forced outage hours}}$$

As forced outages raise operation and maintenance costs, proven reliability of equipment and regular preventive maintenance work are essential, not only to guarantee electric supply but also for the overall economy of the power plant.

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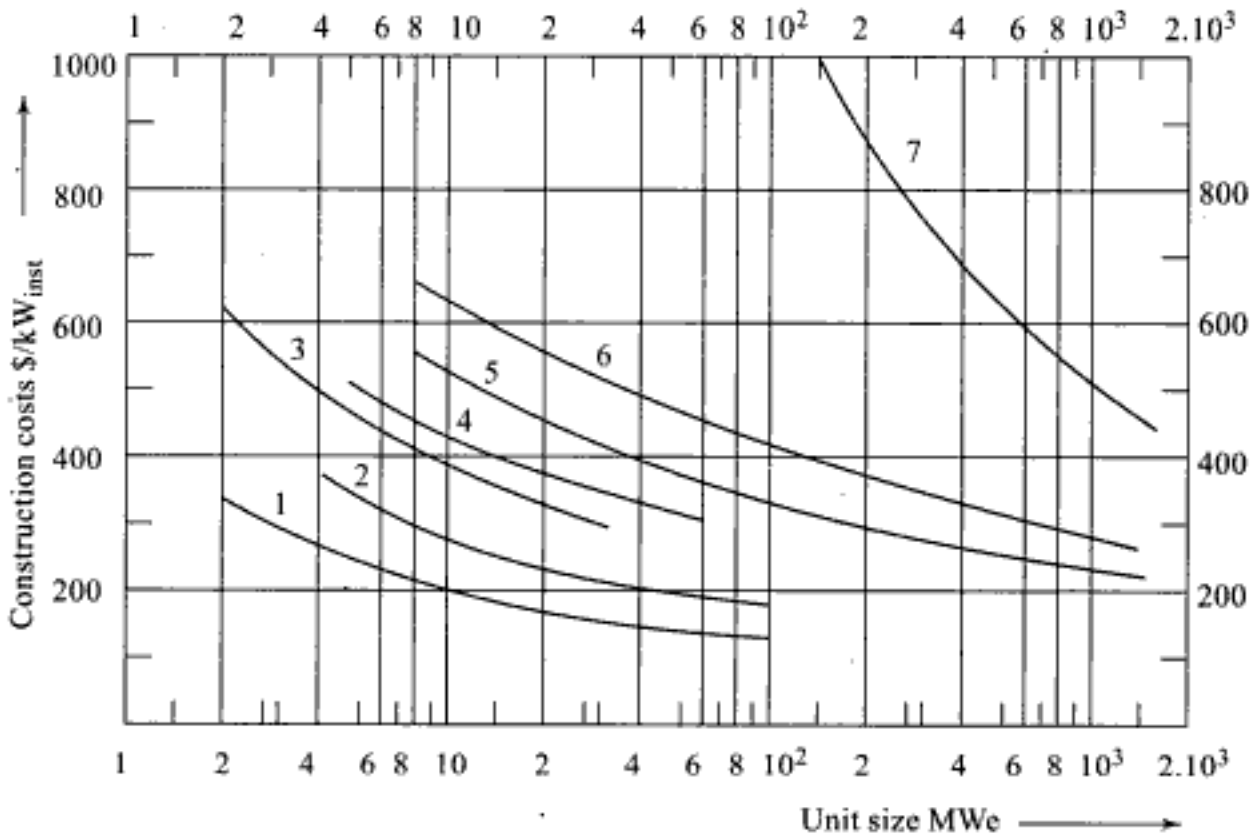
The costs, however, have a continuous upward trend due to monetary inflation, rising fuel prices and increased demand for measures protecting the environment.

The cost of power generation can be reduced by

(a) selecting equipment of longer life and proper capacities,
(b) running the power station at high load factor,
(c) increasing the efficiency of the power plant,
(d) carrying out proper maintenance of power plant equipment to avoid plant breakdowns,
(e) keeping proper supervision, since good supervision is reflected in lesser breakdowns and extended plant life,
(f) using a plant of simple design that does not need highly skilled personnel.

1.4.1 Construction Costs

The construction costs of a plant consist of expenses for:



1. Jet type gas-turbine plants without recuperator
2. Jet type gas-turbine plants with recuperator
3. Industrial type gas-turbine plants with recuperator
4. Diesel plants for continous service
5. Conventional steam power plants, oil fired
6. Conventional steam power plants, bituminous-coal fired
7. Nuclear LWR steam power plants

Fig. 1.7 *Approximate construction cost of power plants vs unit size (Aschner, 1978)*

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(i) planning and design;
(ii) land and preparation of site, including road (and rail) connections;
(iii) buildings and machinery foundations;
(iv) plant equipment, including transport to site;
(v) erection and testing of equipment;
(vi) interest during construction (for payments made before commissioning of the plant); and
(vii) administrative work.

In all types of power plants there is an economy of scale, i.e. the construction costs per kW decrease with size (Fig. 1.7). This effect varies with the type of equipment. The economy of scale of the construction costs C_c of two parts of equipment of the same type can be defined by

$$C_{c2} = C_{c1} (R_2/R_1)^k \tag{1.7}$$

where C_{c_1} and C_{c_2} are for parts with rated outputs of R_1 and R_2, respectively, $k < 1$. Average values of k for conventional steam generators and for steam turbo-generators and centrifugal pumps are 0.8, whereas $k = 0.6$ for nuclear steam supply systems and $k = 0.9$ for condensers and feedwater heaters (Aschner, 1978).

1.4.2 Fixed Costs and Depreciation

Fixed costs consist of annual costs for interest, depreciation, insurance and taxes on property and estimated income calculated as a percentage of the capital investment.

Interest rates I depend on general financial conditions at the time of construction and on the source of funding international, national, public, private or bond issues. Because of inflation, i.e., decrease in the value of the money, the concept of real and charged interest rates has been proposed. If, for instance, the real rate is fixed at 10% and a price inflation of 7% per year is forecast, the charged interest rate goes up to 17%.

Depreciation rates D are determined by the life expectancy of the equipment and the method used for calculating the depreciation.

The power plant and equipment in the station will have a certain period of useful life. After years of use, the equipment loses its efficiency or becomes obsolete and needs replacement. Sometimes, equipment may have to be changed even when fairly new, if more efficient equipment has come into the market. To enable this to be done when necessary, some money is put aside annually, and is known as the depreciation fund.

There are two methods of accumulating the money for the depreciation fund:

1. Straight-line method
2. Sinking-fund method

The straight-line method is based on the assumption that depreciation occurs uniformly every year according to a straight-line law. The money saved neglects

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any interest. If A is the capital cost of the equipment and G is its salvage value after N useful years, then the depreciation charge per year is

$$D = (A - G)/N$$

Figure 1.8 illustrates this simple, straight-line method of accumulating the depreciation reserve. It is seen that the value of the equipment, which is 100% initially, depreciates to only 15% at the end of its 100% useful years of life. The straight-line BE indicates the depreciation and the slope of the line indicates the rate at which the plant depreciates. The depreciation at the end of 50% of useful life is FG, and the total depreciation at the end of 100% of useful life is CE. The line AD represents the depreciation reserve to be accumulated so that at the end of useful life the total is DJ, which is equal to the total depreciation CE.

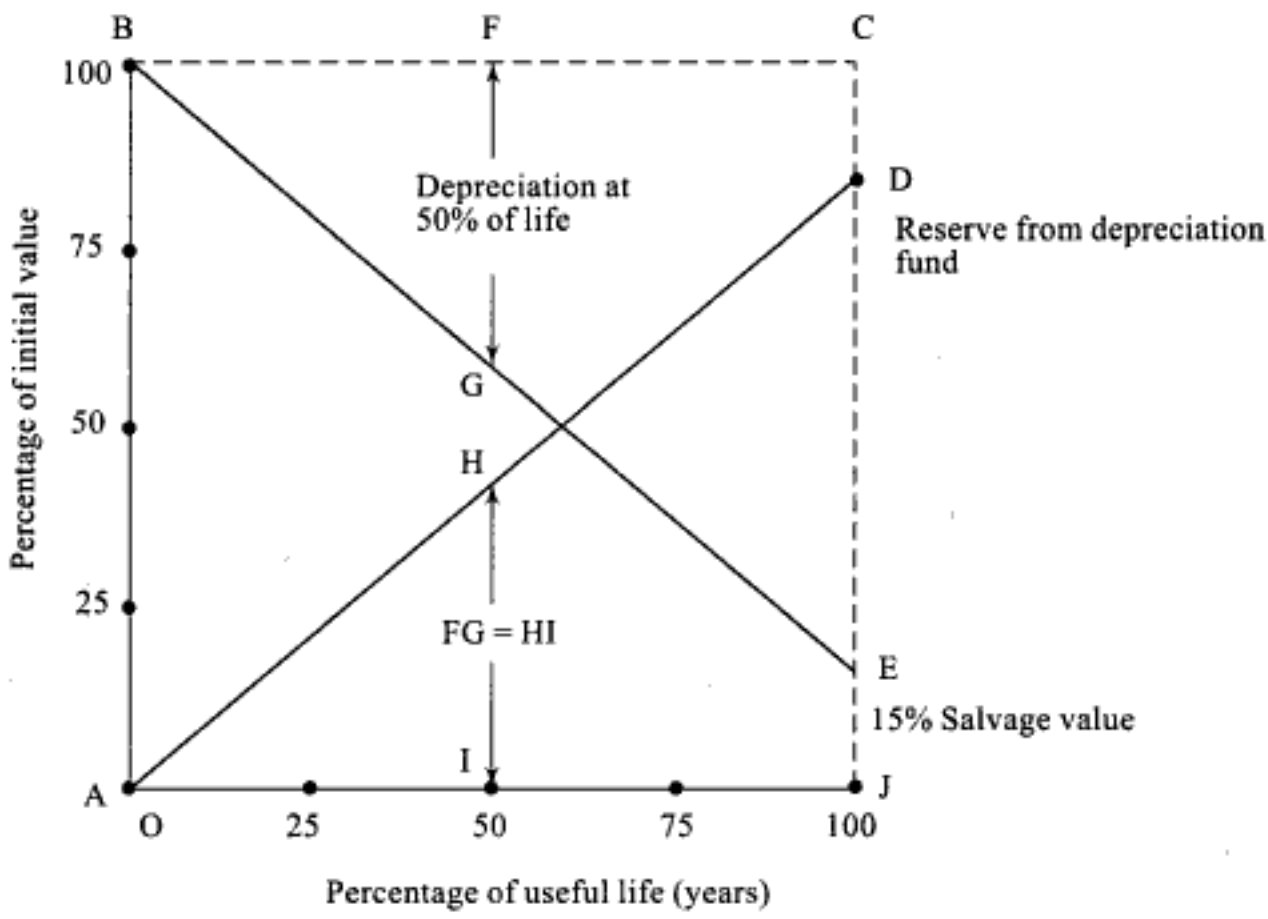

Fig. 1.8 *Straight-line method*

In the sinking-fund method, a sum of money is set aside every year for N years and invested to earn compound interest. Let us suppose P is the annual deposit and i is the interest compounded annually when the deposit is invested.

At the end of the first year, Rs. P is deposited. After $(N-1)$ years this will be worth (compounded annually)

$$\text{Rs. } P(1+i)^{N-1}$$

At the end of the second year, Rs. P is deposited. After $(N-2)$ years this will be worth

$$\text{Rs. } P(1+i)^{N-2}$$

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At the end of $(N-1)$ years, Rs. P is deposited. After 1 year this will be worth

$$\text{Rs. } P(1+i)$$

At the end of Nth year, Rs. P is deposited and this will be worth Rs. P
At the end of N years, the value of all the sums deposited will be

$$P(1+i)^{N-1} + P(1+i)^{N-2} + \ldots + P(1+i) + P$$
$$= P + P(1+i) + P(1+i)^2 + \ldots + P(1+i)^{N-1}$$

This is a geometric progression with a common ratio of $r = 1 + i$.
Therefore, the sum is

$$S = P\,\frac{(1+i)^{N-1}}{i}$$

This sum is equal to $A' = A - G$ = capital cost – salvage value

$$P = \frac{A'\,i}{(1+i)^{N-1}} \tag{1.8}$$

The sinking-fund method can also be represented by a diagram as shown in Fig. 1.9. The curve AHD shows the sinking fund accumulating at a certain rate of compound interest. The total amount at the end of the useful life is given by JD, which is the initial cost of the plant minus its salvage value. The rate at which the sum is saved annually is not, however, the same as in the straight-line method, as the final sum consists of annual instalments plus the compound interest on all instalments at the end of the whole period $JD = CE$. The depreciation at the end of 50% of useful life is shown by FG, while the sinking-fund saved at the end of 50% of useful life is shown by HI.

The amount which should be set aside annually is given by Eq. (1.8),

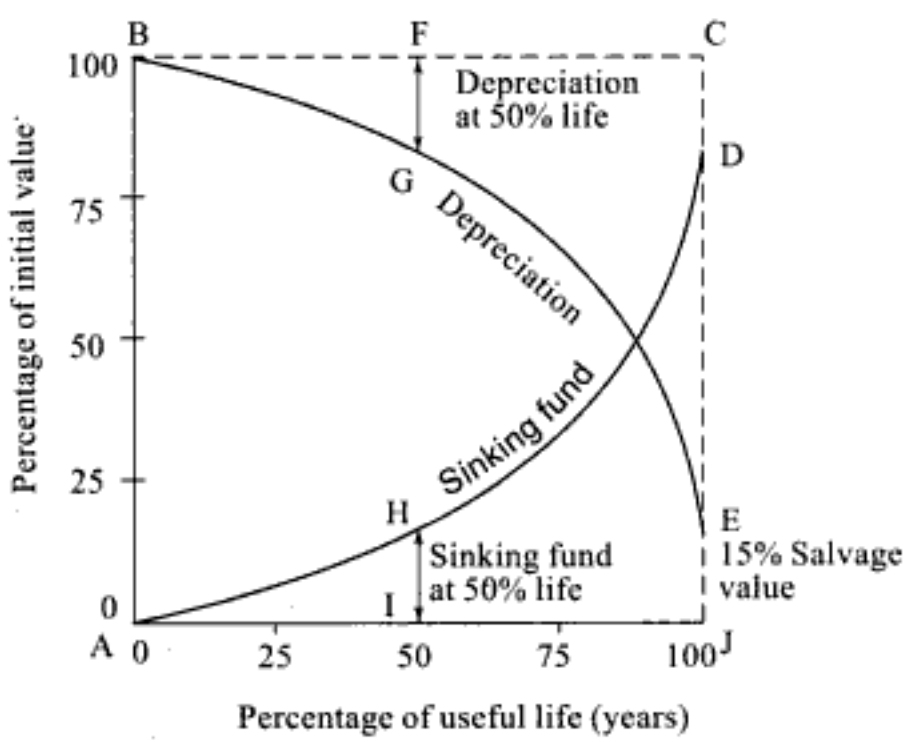

Fig. 1.9 *Sinking fund method*

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$$P = \text{annual payment to sinking fund}$$

$$= (\text{initial value} - \text{salvage value}) \frac{i}{(1+i)^{N-1}},$$

where i is the annual rate of compound interest and N is the number of years over which the total amount of depreciation is to be saved.

1.4.3 Fuel Cost

The fuel consumption depends on the amount of electrical energy produced. As load on the prime movers increases, the fuel consumption increases and so does the cost. The efficiency of the prime mover is the highest at the rated load. At lower loads, the efficiency decreases and so the fuel consumption increases. Therefore, the cost of electricity generation is the minimum at rated load and it increases as load decreases. The cost of fuel also depends on the calorific value, availability and transportation charges.

1.4.4 Present-Worth Concept

In the evaluation of power plant development plans or alternate payment schemes (cash flow schemes) for financing a power plant project, the influence of interest on the value of funds to be spent at a later date leads to the present-worth concept. The present worth is the value of a sum of money at the present time that, with compound interest, will have a specified value at a certain time in the future.

Compound interest payment at the interest rate i will increase the value of a fund by $(1 + i)^N$ within N years. The present worth (PW) of a payment S to be made after N years is therefore,

$$\text{PW} = \frac{S}{(1+i)^N} \tag{1.9}$$

The term $(1 + i)^{-N}$ is frequently referred to as the single payment present worth factor (PWF).

If $i = 10\%$ and $N = 4$ years,

$$\text{PW} = 0.683\, S$$

Also, if $i = 10\%$, $N = 1$ year and $S = \text{Rs. } 110$, then PW = Rs. 100.

On many occasions, equal amounts of annual expenses are required. Then the present worth of a uniform annual series of payments P after N years is calculated from Eqs (1.8) and (1.9),

$$\text{PW} = P\frac{(1+i)^{N-1}}{i} \times \frac{1}{(1+i)^N}$$

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$$= \frac{P}{i(1+i)^N}\left[(1+i)^N - 1\right] \tag{1.10}$$

If $i = 10\%$ and $Y = 12$ years

$$PW = \frac{P}{0.10(1.1)^{12}}\ [(1.1)^{12} - 1] = 6.8136\,P$$

1.4.5 Incremental Heat Rate

The performance of a generating plant can be expressed in terms of

$$\text{Plant net heat rate (PNHR) (expressed in kJ/kWh)} = \frac{\text{heat input } (I)}{\text{net kW output } (L)}$$

and

$$\text{Plant net efficiency } (n_{th}) = \frac{\text{net output } (L)}{\text{heat input } (I)}$$

Table 1.1 indicates typical performance of various generating systems. The given efficiencies are achieved only when the plants operate at full loads. The incremental heat rate (IR) is defined as

$$IR = dI/dL \tag{1.11}$$

It represents the amount of energy needed to generate an additional unit of output at any given load. In Figure 1.10, it is seen that as the plant output decreases, the plant net heat rate increases, whereas the incremental heat rate decreases.

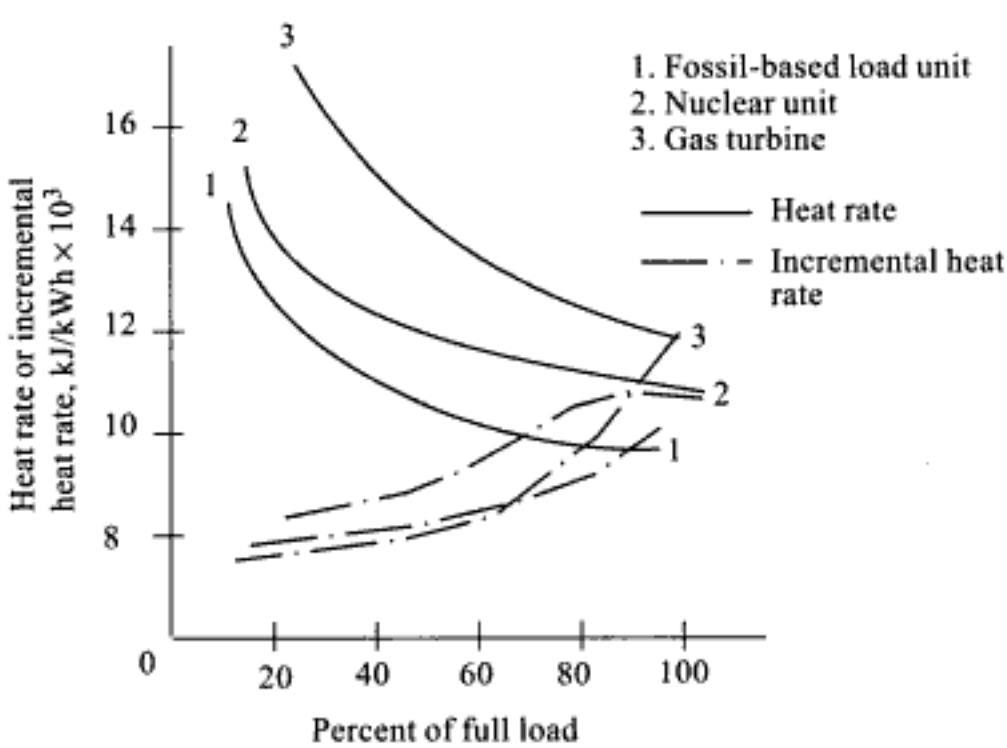

Fig. 1.10 *Plant net heat rate and incremental heat rate*

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Table 1.1 *Typical conversion efficiency of various generating systems (Li and Priddy, 1985)*

Generation type	Unit size (MW)	Thermal efficiency (%)
Steam	200 – 800	30 – 40
Nuclear (BWR & PWR)	500 – 1100	31 – 34
Gas turbine	50 – 100	22 – 28
Combined gas turbine/ steam turbine	300 – 600	36 – 50
Diesel engine	10 – 30	27 – 30

1.4.6 Economic Scheduling Principle

An utility system has always more than one generating unit. Proper distribution of load among the generating units is a problem frequently encountered by engineers. If the load is not properly distributed, it will result in a decrease of the thermal efficiency as a whole. According to the economic scheduling principle, the load is so distributed that at any moment all generating units will have the same incremental heat rates. As the system load increases, the incremental heat rate of each unit will increase. This principle is illustrated by the example involving two generating units in the network.

Let I_c be the combined input to units 1 and 2, and L_c be the combined output of units 1 and 2. The combined input, I_c, is a function of either L_1 or L_2 for a given combined output L_c. When the combined input I_c is at a minimum, it must hold

$$dI_c/dL_1 = 0 \tag{1.12}$$

Since $I_c = I_1 + I_2$

we have
$$dI_1/dL_1 + dI_2/dL_1 = 0 \tag{1.13}$$

Now, $dI_2/dL_1 = (dI_2/dL_2) \times (dL_2/dL_1)$

Since, $L_c = L_1 + L_2$

$$dL_2/dL_1 = -1$$

$$dI_2/dL_1 = -\,dI_2/dL_2 \tag{1.14}$$

From Eqs. (1.13) and (1.14),

$$dI_1/dL_1 = dI_2/dL_2 \tag{1.15}$$

Thus, the combined input I_c is a minimum only if the incremental heat rate of unit 1 is equal to that of unit 2. This principle can be extended to the network of multiple units.

Figure 1.11 represents the input-output curves of two generating units operating in parallel and supplying a common load. The corresponding heat rate curves are shown in Fig. 1.12. In dividing the load between these units to achieve maximum fuel economy, the principle of equal incremental heat rate is being used. Starting at zero load, the turbine B picks up the load up to L_{B1}, while

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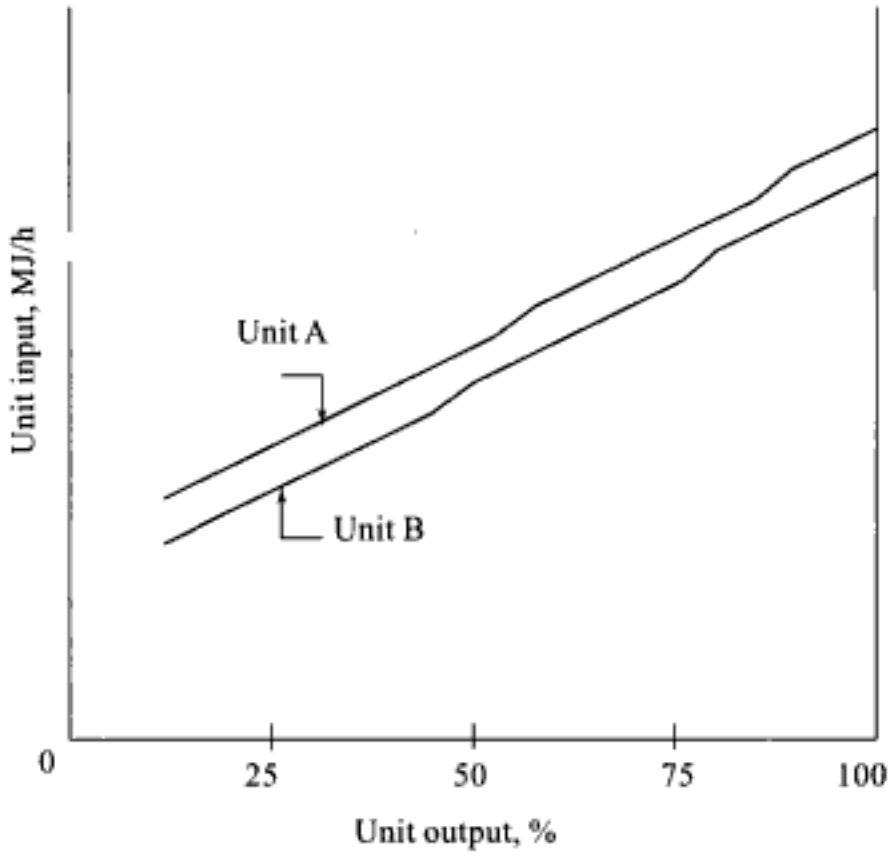

Fig. 1.11 *Input-output curves*

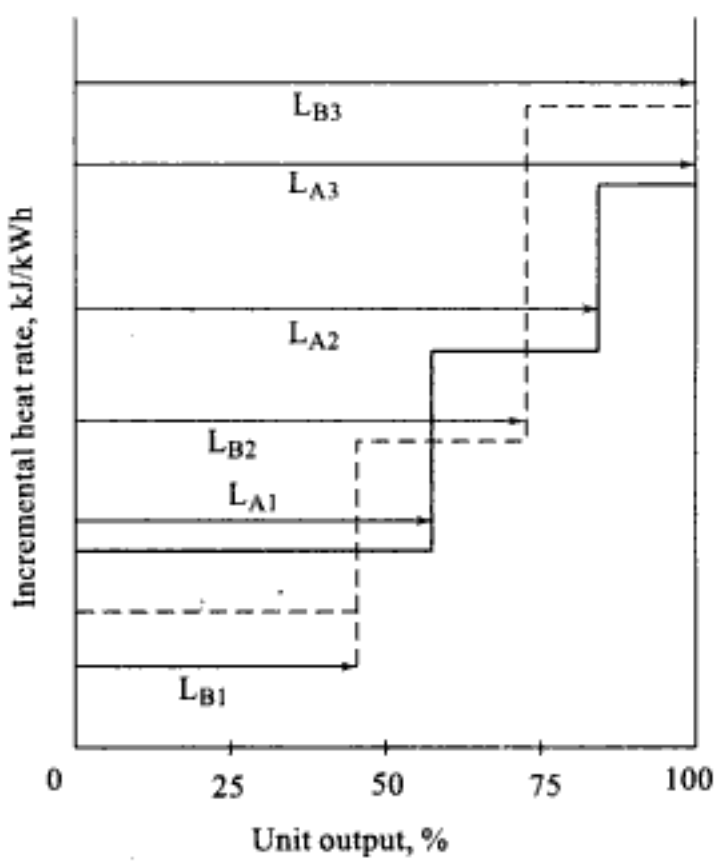

Fig. 1.12 *Incremental heat rate curves for unit A and B*

the turbine A remains at zero load. When combined load is greater than L_{B1}, the turbine A starts to pick up load until it reaches L_{A1}. At the combined load L_{A1} +

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L_{B1}, the second valve of turbine starts to open. As the load increases, the turbine B picks up the additional load, while the turbine A remains at the load L_{A1}. For any increase beyond $L_{A1} + L_{B2}$, the turbine A starts to open the second valve and later the third valve. At the combined load $L_{A3} + L_{B2}$, the turbine A would be at a full load L_{A3} and the turbine B at the load L_{B2}. If the combined load increases beyond this level, the turbine B starts to open the last valve to meet the additional demand.

In a utility system with many units, to maintain continuity of service, the generating capacity in operation must be greater than the system load. The difference between these two is defined as the *spinning reserve*. The minimum magnitude of spinning reserve varies from one utility to another, but it is usually equal to or greater than the capacity of the largest unit in operation. To minimize the total system fuel cost, the principle of equal incremental heat rate is to be observed in distributing the system load among the operating units.

1.4.7 Effect of Load Factor on Cost per kWh

The fixed costs are proportional to the MW capacity of a plant, while the operating costs are proportional to the total energy produced by the station annually. The fixed costs remain the same whether the station is running or idle. The fuel consumption and operating costs will, however, vary with the load. At 100% load factor the installed capacity is utilized to the best possible extent and the maximum possible amount of energy is generated during the year. The cost per kWh is thus a minimum in this case. When the load factor is lower than 100%, less energy is produced giving a higher cost per kWh, since the non-varying fixed cost and lower operating costs are distributed over fewer kWh generated. As the load factor decreases, the cost per kWh increases (Fig. 1.13).

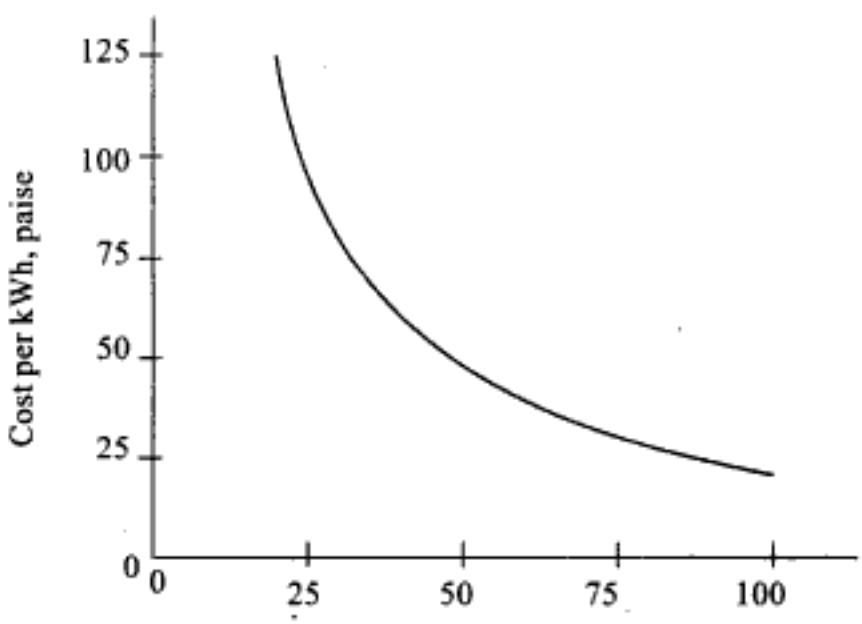

Fig. 1.13 *Effect of load factor on cost of energy*

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1.5 INDIAN ENERGY SCENARIO

Economic growth in India, being dependent on the power sector, has necessitated an enormous growth in electricity demand over the last two decades. In 1947, the total power generation capacity was only 1360 MW, and by 1991 it grew to 65,000 MW, of which 69% (45,000 MW) was generated in thermal plants.

Hydro developments, as stated are not so environmentally benign as was earlier believed. They can cause conflict over usage of lands and the damage to ecology. The total Indian hydro-potential, as assessed by the Central Electricity Authority, is 84,000 MW. The installed capacity is 18,443 MW (March, 1991) compared to 200 kW in 1897 (Darjeeling) and 508 MW at the time of independence. Most of the turbomachinery have been indigenously produced by the Bharat Heavy Electricals Ltd., the largest unit being 165 MW. However, the pace of hydro development has considerably slowed down recently for many reasons, and the focus is more on mini or small hydel plants in the mountainous regions. Tentatively, in the order of 26,000 MW of hydro capacity would be added by 2002 during the 9th five-year plan, as envisaged.

Nuclear power promotion has all but ceased in many areas of the world as a consequence of growing public concern regarding safety and regulatory hurdles. Nonetheless, in India, there appears to be a comparatively energetic nuclear programme. Presently, in India, about 2.3% of generating capacity is nuclear based. This corresponds to about 1500 MW of the installed capacity comprising 8 units. An additional 1320 MW is under construction, based on 220 MW units, and should be on line by 1997. According to the development plan, the total installed capacity by 2002 will be 5700 MW, including the first Indian 500 MW PHWR unit in Rajasthan. It is anticipated that following on the tail of the natural uranium fuelled reactor programme, a fast breeder programme using plutonium will be established, followed by a thorium based programme, thereby taking advantage of a large indigenous resource of thorium.

The state-of-the-art technology for the renewable energy utilization (i.e. solar, wind, biomass, tidal. OTEC, etc.) is such that it will hardly make any contribution to the total energy production. However, construction of a 100 MW ocean thermal energy conversion (OTEC) project has commenced in 1994, 40 km offshore from Kulasenkharapatnam in Tamil Nadu.

Coal–fired power plants are now getting maximum attention, since coal is abundantly available and the implementation time is relatively short. Tentatively, the additional thermal power generation capacity during the 8th Plan is 28,000 MW and during the 9th Plan (by 2002) further 32,000 MW has been added as shown in Table 1.2.

It has been planned to generate an additional 73,000 MW of power by the end of the 11th Plan (2007-2012) which excludes the proposed eight ultra megapower projects with a capacity of 4000 MW each (i.e. 105,000 MW total). The per capita energy consumption will be still 600 MW in India, while it is 12000 MW in China (in 2006).

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Table 1.2 *Indian generation capacity (in MW)*

	1991	8th Plan (1997)	9th Plan (2002)	10th Plan (2007)	Total (2007)
Thermal	45.000 (69.3%)	28,000[a]	32,000	58,000	1,63,000
Hydro	18,443 (28.5%)	8,680	26,000	23,000	76,123
Nuclear	1,500 (2.2%)	1,320	2,880	–	5,700
Addition		38,000	61,000	81,000	–
Total	65,000	1,03,000	1,64,000	2,45,000	2,45,000

[a] includes 7500 MW of gas-based combined cycle plant.

As in other parts of the world, natural gas-based combined cycle power plant is gaining prominence. Consequently, the 8th Plan envisages installation of 7500 MW capacity, using indigenous gas reserves.

Table 1.3 gives the growth pattern in unit sizes and steam parameters in coal-based power plants. In the 70s, the focus was on 100 and 200 MW units, while in the 80s it was on 210 and 500 MW units. Consideration is now being given to increasing the size of future units to 660,800 and 1000 MW to reduce installation cost per kW.

Table 1.3 *Growth pattern in unit sizes and steam parameters*

Unit size (MW)	Collaboration	Steam parameters	Station first installed	Year	No. of units till March, 1991
10–15		27.5 bar 400°C		1947–50	N.A.
30–37.5		59 bar, 482°C		1951–60	62
60–100		72.4–91 bar, 496–535°C		1961–76	56
100	USSR	91 bar, 535°C	Patratu	Dec. 71	79
200	LMW, USSR	132 bar, 535°C/535°C	Obra (UPSEB)	Dec. 77	12
210	KWU, Siemens	152 bar, 535°C/535°C	Korba (NTPC)	March 83	89
500	KWU, Siemens	172 bar, 538°C/538°C	Trombay-5 (Tata)	Dec. 84	13

With the liberalization of Indian economy, the power sector has been made open to the private sector, and it is expected that the rate of growth of generation capacity will be faster than that envisaged in the five-year plans (Table 1.2).

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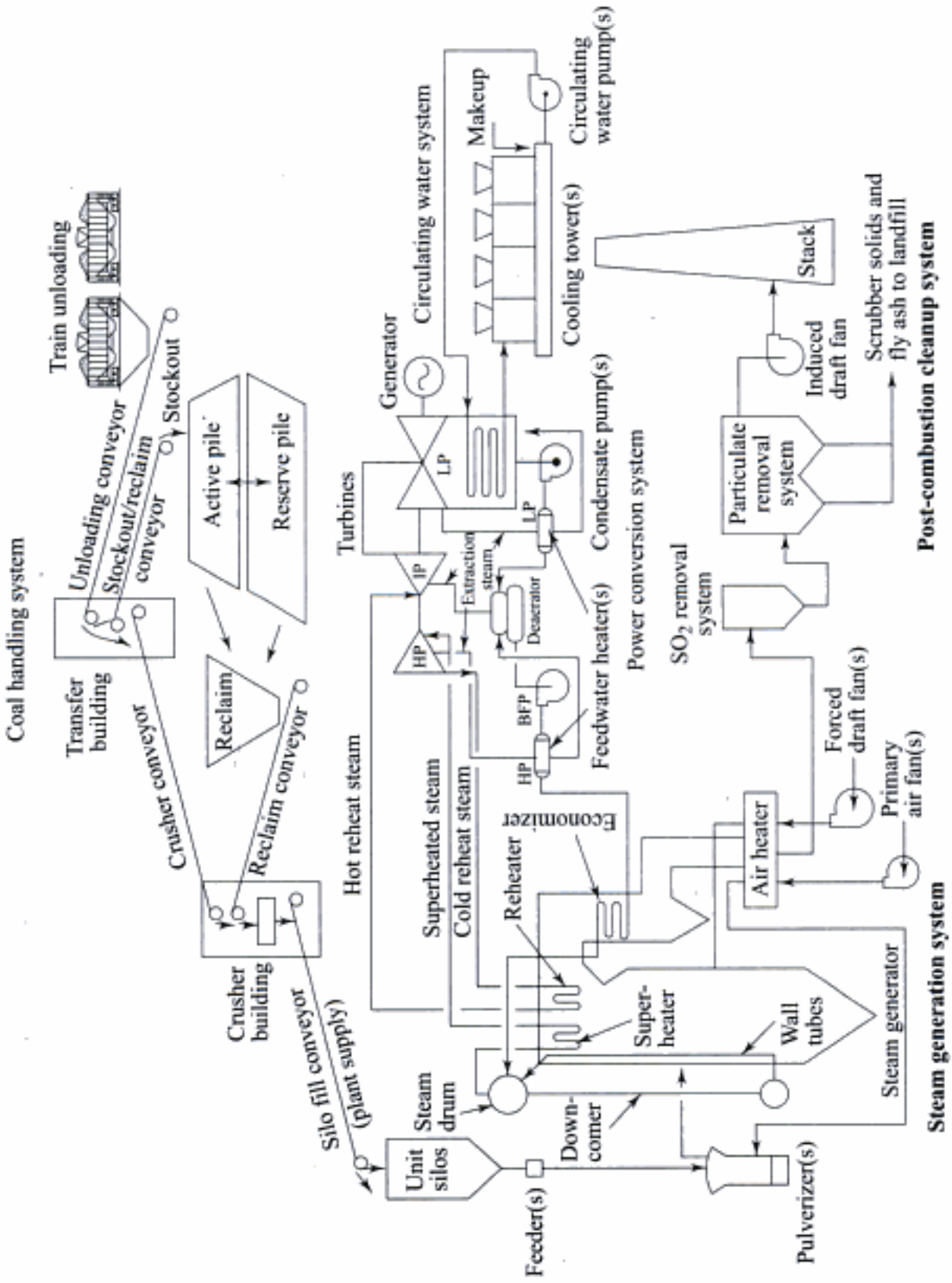

Fig. 1.14 *Modern Pulverized Coal Fuelled Electricity Generating Unit*

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1.6 COAL-FUELLED ELECTRICITY GENERATING UNIT

Figure 1.14 depicts a modern pulverized coal-fired electricity generation facility that uses the most commonly employed Rankine-based thermodynamic cycle. Steam at the desired pressure, temperature and mass flow is produced in a steam generator and expanded through a turbine generator. The exhaust steam is then condensed in a condenser and the condensed liquid is used again in the steam generator.

Coal is delivered to the facility by railway wagons, barges or trucks. The coal handling system unloads the coal, stocks, reclaims, crushes and conveys it to storage silos. Coal from the silos is then pulverized to a fine powder and blown into the steam generator, where it is mixed with air and combusted to release energy for the generation of steam. The steam generator produces, superheats and reheats steam as it proceeds through the cycle.

The steam turbine generator converts the thermal energy (enthalpy rise) of the superheated and reheated steam to electrical energy. Steam exhausted from the turbine is condensed to liquid in the condenser. The condensate pumps feed the water through the l.p. regenerative feedwater heaters to a deaerator. Boiler feed pumps move the deaerated liquid through the h.p. feedwater heaters back to the steam generator.

Forced draught (or draft) fans supply combustion air to the steam generator and the primary air fans transport pulverized coal into the burners. Induced draught fans remove the flue gases from the furnace and exhaust them through the stack into the atmosphere.

Cooling water for the condenser is supplied by the circulating water system, which takes the heat removed from the condenser and rejects it to cooling towers or another heat sink such as a cooling lake, river or sea.

Combustion gases coming out of the steam generator require additional treatment for removal of flyash particulates, sulphur dioxide and nitrogen oxides before the gases are released through the exhaust stack.

Example 1.1 A power station supplies the following loads to the consumers:

Time in hours	*0–6*	*6–10*	*10–12*	*12–16*	*16–20*	*20–22*	*22–24*
Load in MW	*30*	*70*	*90*	*60*	*100*	*80*	*60*

(a) Draw the load curve and estimate the load factor of the plant. (b) What is the load factor of a standby equipment of 30 MW capacity if it takes up all loads above 70 MW? What is its use factor?

Solution (a) The load curve is drawn in Fig. E1.1.

$$\text{Energy generated} = \text{area under the load curve}$$
$$= 30 \times 6 + 70 \times 4 + 90 \times 2 + 60 \times 4 + 100 \times 4 + 80 \times 2 + 60 \times 2$$

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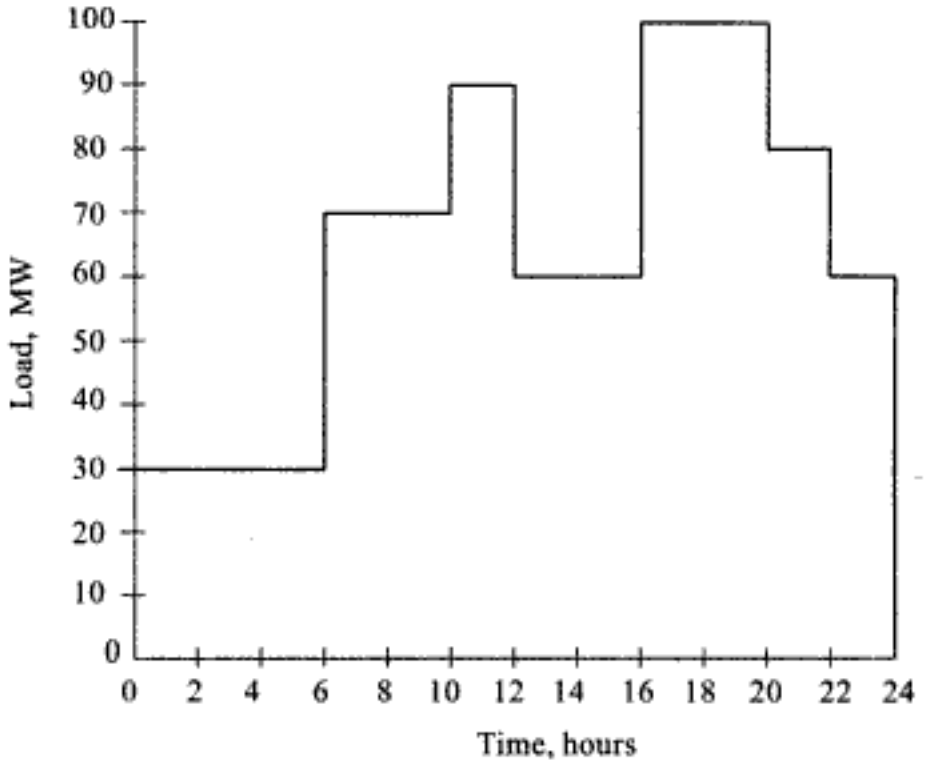

Fig. E1.1

$$= 180 + 280 + 180 + 240 + 400 + 160 + 120$$

$$= 1560 \text{ MWh}$$

$$\text{Average load} = \frac{1560 \text{ MWh}}{24 \text{ h}} = 65 \text{ MW}$$

$$\text{Load factor} = \frac{\text{average load}}{\text{peak load}} = \frac{65}{100} = 0.65 \qquad \textit{Ans.}$$

(b) If the load above 70 MW is supplied by a standby unit of 30 MW capacity, the energy generated by it

$$= 20 \times 2 + 30 \times 4 + 10 \times 2$$

$$= 40 + 120 + 20 = 180 \text{ MWh}$$

Time during which the standby unit remains in operation

$$= 2 + 4 + 2 = 8 \text{ h}$$

$$\text{Average load} = 180 \text{ MWh}/8\text{h} = 22.5 \text{ MW}$$

$$\text{Load factor} = 22.5/30 = 0.75 \textit{ Ans.}$$

$$\text{Use factor} = \frac{\text{energy generated}}{\text{plant capacity} \times \text{operating hours}}$$

$$= \frac{180 \times 10^3 \text{ kWh}}{30 \times 10^3 \text{ kW} \times 8 \text{ h}} = 0.75 \qquad \textit{Ans.}$$

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Example 1.2 The peak load on a power plant is 60 MW. The loads having maximum demands of 30 MW, 20 MW, 10 MW and 14 MW are connected to the power plant. The capacity of the power plant is 80 MW and the annual load factor is 0.50. Estimate (a) the average load on the power plant, (b) the energy supplied per year, (c) the demand factor, (d) the diversity factor.

Solution

(a) $$\text{Load factor} = \frac{\text{average load}}{\text{peak load}}$$

$$0.5 = \frac{\text{average load}}{60 \text{ MW}}$$

$$\text{Average load} = 30 \text{ MW} \qquad \textit{Ans.}$$

(b) Energy supplied per year

$$= \text{average load} \times 8760 \text{ h}$$

$$= 30 \times 8760 \text{ MWh} = 262.8 \times 10^6 \text{ kWh} \qquad \textit{Ans.}$$

(c) $$\text{Demand factor} = \frac{\text{maximum demand}}{\text{connected load}} = \frac{60}{30+20+10+14}$$

$$= \frac{60}{74} = 0.811 \qquad \textit{Ans.}$$

(d) $$\text{Diversity factor} = \frac{\text{sum of individual maximum demands}}{\text{simultaneous maximum demand}}$$

$$= \frac{30+20+10+14}{60} = \frac{74}{60}$$

$$= 1.233 \quad \textit{Ans.}$$

Example 1.3 A thermal power plant of 210 MW capacity has the maximum load of 160 MW. Its annual load factor is 0.6. The coal consumption is 1 kg per kWh of energy generated and the cost of coal is Rs 450.00 per tonne. Calculate (a) the annual revenue earned if energy is sold at Re 1 per kWh and (b) the capacity factor of the plant.

Solution Annual load factor = (average load)/(peak load)

$$\text{Average load} = 0.6 \times 160 = 96 \text{ MW}$$

Energy generated per year

$$= 96 \times 8760 \text{ MWh} = 840{,}960 \times 10^3 \text{ kWh}$$

$$\text{Coal required per year} = 840{,}960 \times 10^3 \text{ kg}$$

$$= 840{,}960 \text{ tonnes}$$

$$\text{Cost of coal per year} = 840{,}960 \times 450 = \text{Rs } 378.432 \times 10^6$$

$$\text{Cost of energy sold} = \text{Rs } 840{,}960 \times 10^3$$

$$= \text{Rs } 840.96 \times 10^6$$

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(a) Revenue earned by the power plant per year

$$= \text{Rs } 840.96 \times 10^6 - \text{Rs } 378.432 \times 10^6$$

$$= \text{Rs } 462.528 \times 10^6 = \text{Rs } 46.25 \text{ crore} \qquad \textit{Ans. (a)}$$

(b) $$\text{Capacity factor} = \frac{\text{average load}}{\text{capacity of plant}} = \frac{96 \text{ MW}}{210 \text{ MW}}$$

$$= 0.457 \qquad \textit{Ans. (b)}$$

Example 1.4 A power plant has the following annual factors: load factor = 0.75, capacity factor = 0.60, use factor = 0.65. Maximum demand is 60 MW. Estimate (a) the annual energy production, (b) the reserve capacity over and above the peak load, and (c) the hours during which the plant is not in service per year.

Solution

(a) $$\text{Load factor} = \frac{\text{average load}}{\text{peak load}}$$

$$\text{Average load} = 0.75 \times 60 = 45 \text{ MW}$$

$$\text{Annual energy production} = 45{,}000 \times 8760 = 394{,}200{,}000 \text{ kWh}$$

$$= 394.2 \times 10^6 \text{ kWh} \qquad \textit{Ans.}$$

(b) $$\text{Capacity factor} = \frac{\text{average load}}{\text{plant capacity}}$$

$$\text{Plant capacity} = 45/0.6 = 75 \text{ MW}$$

Reserve capacity over and above the peak load

$$= 75 - 60 = 15 \text{ MW} \qquad \textit{Ans.}$$

(c) $$\text{Use factor} = \frac{\text{energy generated per year}}{\text{plant capacity} \times \text{hours in operation}}$$

$$\text{Hours in operation} = \frac{394.2 \times 10^6 \text{ kWh}}{75 \times 10^3 \text{ kW} \times 0.65} = 8086 \text{ hrs}$$

Hours not in service in a year = 8760 – 8086 = 674 hrs *Ans.*

Example 1.5 It is proposed to supply a load with a maximum demand of 500 MW and load factor of 70%. Choice is to be made from a nuclear power plant, a hydraulic power plant and a steam power plant. Calculate the overall cost per kWh in case of each scheme as given below:

Cost	*Steam power plant*	*Hydroelectric power plant*	*Nuclear power plant*
1. Capital cost per MW installed	Rs 3 crore	Rs 4 crore	Rs 5 crore
2. Interest	6%	5%	5%
3. Depreciation	6%	4%	5%
4. Operating cost (including fuel) per kWh	30 paise	5 paise	15 paise
5. Transmission and distribution cost per kWh	2 paise	3 paise	2 paise

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Solution (a) Steam power plant:

$$\text{Capital cost} = \text{Rs } 3 \times 500 \text{ crore} = \text{Rs } 1500 \times 10^7$$

$$\text{Interest} = 0.06 \times 1500 \times 10^7 = \text{Rs } 90 \times 10^7$$

$$\text{Depreciation} = 0.06 \times 1500 \times 10^7 = \text{Rs } 90 \times 10^7$$

Annual fixed cost (interest + depreciation)

$$= \text{Rs } 180 \times 10^7$$

Energy generated per year = average load × 8760

= Load factor × maximum demand × 8760

$$= 0.7 \times 500 \times 10^3 \times 8760$$

$$= 306.6 \times 10^7 \text{ kWh}$$

Running cost/kWh = (operating cost / kWh)

+ (transmission and distribution cost/kWh)

= 30 + 2 = 32 paise

Overall cost/kWh = fixed cost/kWh + running cost/kWh

$$= \frac{\text{Rs } 180 \times 10^7}{306.6 \times 10^7 \text{ kWh}} + 0.32$$

$$= 0.59 + 0.32 = 0.91 \quad \text{or} \quad 91 \text{ paise}$$

(b) Hydroelectric power plant:

$$\text{Capital cost} = \text{Rs } 4 \times 500 \text{ crore} = \text{Rs } 2000 \times 10^7$$

$$\text{Interest} = 0.05 \times 2000 \times 10^7 = \text{Rs } 100 \times 10^7$$

$$\text{Depreciation} = 0.04 \times 2000 \times 10^7 = \text{Rs } 80 \times 10^7$$

Annual fixed cost = interest + depreciation

$$= \text{Rs } 180 \times 10^7$$

$$\text{Fixed cost/kWh} = \frac{\text{Rs } 180 \times 10^7}{306.6 \times 10^7} = 0.59$$

Running cost/kWh = 5 + 3 = 8 paise

Overall cost/kWh = 0.59 + 0.08 = Re 0.67

(c) Nuclear power plant:

$$\text{Capital cost} = \text{Rs } 5 \times 500 \text{ crore} = \text{Rs } 2500 \times 10^7$$

$$\text{Interest} = 0.05 \times 2500 \times 10^7 = \text{Rs } 125 \times 10^7$$

$$\text{Depreciation} = 0.05 \times 2500 \times 10^7 = \text{Rs } 125 \times 10^7$$

$$\text{Annual fixed cost} = \text{Rs } 250 \times 10^7$$

$$\text{Fixed cost/kWh} = \frac{\text{Rs } 250 \times 10^7}{306.6 \times 10^7 \text{ kWh}} = 0.82$$

Running cost/kWh = 15 + 2 = 17 paise

Overall cost/kWh = 0.82 + 0.17 = Re 0.99

Therefore, the overall cost/kWh is minimum in case of hydroelectric power plant.

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Example 1.6 A power plant of 210 MW installed capacity has the following particulars:

Capital cost = Rs 18,000/kW installed
Interest and depreciation = 12%
Annual load factor = 60%
Annual capacity factor = 54%
Annual running charges = Rs 200×10^6
Energy consumed by power plant auxiliaries = 6%
Calculate (a) the cost of power generation per kWh, and (b) the reserve capacity

Solution

$$\frac{\text{Load factor}}{\text{Capacity factor}} = \frac{\text{average load}}{\text{maximum demand}} \times \frac{\text{capacity of the plant}}{\text{average load}}$$

$$\frac{0.60}{0.54} = \frac{210 \text{ MW}}{\text{maximum demand}}$$

Maximum demand = $(210 \times 0.54)/0.60 = 189$ MW

Reserve capacity = $210 - 189 = 21$ MW *Ans.* (b)

Average load = load factor × maximum demand
= $0.6 \times 189 = 113.4$ MW

Energy produced per year
= $113.4 \times 10^3 \times 8760 = 993.384 \times 10^6$ kWh

Net energy delivered = $0.94 \times 993.384 \times 10^6$
= 933.781×10^6 kWh

Annual interest and depreciation (fixed cost)
= $0.12 \times 18000 \times 210 \times 10^3$ = Rs 453.6×10^6

Total annual cost = fixed cost + running cost
= $453.6 \times 10^6 + 200 \times 10^6$
= Rs 653.6×10^6

$$\text{Cost of power generation} = \frac{\text{Rs } 653.6 \times 10^6}{933.781 \times 10^6 \text{ kWh}}$$

= Rs 0.70 or 70 paise *Ans.* (a)

Example 1.7 The incremental fuel costs for two generating units a and b of a power plant are given by the following relations,

$$dF_a/dP_a = 0.065\, P_a + 25$$
$$dF_b/dP_b = 0.08\, P_a + 20$$

where F is the fuel cost in rupees per hour and P is the power output in MW. Estimate (a) the economic loading of the two units when the total load supplied by the power plants is 200 MW, (b) the loss in fuel cost per hour if the load is equally shared by both units.

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Solution (a) The given data are

$$P_a + P_b = 200 \tag{1}$$

The condition for economic loading demands equal incremental heat rates of the two units which leads to $dF_a/dP_a = dF_b/dP_b$

$$0.065\,P_a + 25 = 0.08\,P_b + 20 \tag{2}$$

Solving Eq. (1) and (2), we get

$$P_a = 75.86 \text{ MW} \qquad \text{and} \qquad P_b = 124.14 \text{ MW}$$

(b) If the load is equally shared by the units, i.e. $P_a = P_b = 100$ MW, then the increase in cost of fuel for unit *a* is given by

$$\int \frac{dF_a}{dp_a}\,dp_a = \int_{75.86}^{100} (0.065\,P_a + 25)\,dP_a$$

$$= \left[0.065\,\frac{P_a^2}{2} + 25\,P_a\right]_{75.86}^{100} = 741.47$$

i.e., Rs 741.47 per hour.
The increase in cost for unit *b* is

$$= \int_{124.14}^{100} (0.08\,P_b + 20)\,dP_b$$

$$= \left[0.08\frac{P_b^2}{2} + 20\,P_b\right]_{124.14}^{100} = -699.23$$

that is, the cost of fuel for unit *b* decreases by Rs 699.23 per hour.
Therefore, the net increase in fuel cost due to departure from economic distribution of load

$$= 741.47 - 699.23 = \text{Rs } 42.24 \text{ per hour} \qquad \textit{Ans.}$$

Example 1.8 Calculate the cost of generation per kWh for a power station having the following data:

Installed capacity of the plant = 200 MW
Capital cost = Rs 400 crores
Rate of interest and depreciation = 12%
Annual cost of fuel, salaries and taxation = Rs 5 crores
Load factor = 50%

Also estimate the saving in cost per kWh if the annual load factor is raised to 60%.

Solution Assuming the maximum demand equal to the capacity of the power plant,

$$\text{Load factor} = \frac{\text{average load}}{\text{maximum demand}}$$

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$$0.5 = \frac{\text{average load}}{200}$$

$$\text{Average load} = 200 \times 0.5 = 100 \text{ MW}$$

Energy generated per year $= 100 \times 10^3 \times 8760 = 87.6 \times 10^7$ kWh

Interest and depreciation (fixed cost)

$$= 0.12 \times 400 \times 10^7 = \text{Rs } 48 \times 10^7$$

$$\text{Total annual cost} = 48 \times 10^7 + 5 \times 10^7 = \text{Rs } 53 \times 10^7$$

$$\text{Cost per kWh} = \frac{\text{Rs } 53 \times 10^7}{87.6 \times 10^7 \text{ kWh}} = \text{Rs } 0.61 \quad \text{or} \quad 61 \text{ paise} \quad \textit{Ans.}$$

When the load factor is raised to 60%,

$$\text{Average load} = 0.60 \times 200 = 120 \text{ MW}$$

Energy produced per year $= 120 \times 10^3 \times 8760$

$$= 105.12 \times 10^7 \text{ kWh}$$

Total annual cost will remain the same.

$$\text{Cost per kWh} = \frac{\text{Rs } 53 \times 10^7}{105.12 \times 10^7} = \text{Re } 0.50 \quad \text{or} \quad 50 \text{ paise}$$

$$\text{Saving in cost per kWh} = 61 - 50 = 11 \text{ paise} \qquad \textit{Ans.}$$

Example 1.9 A 300 MW thermal power station is to supply power to a system having maximum and minimum demand of 240 MW and 180 MW respectively in a year. Assuming the load duration curve to be a straight line, estimate the (a) load factor, (b) capacity factor.

Solution The load duration curve is shown in Figure E1.9.

Energy supplied per year

$$= (180 \text{ MW} \times 8760 \text{ h}) + \frac{1}{2}(240 - 180) \text{ MW} \times 8760 \text{ h}$$

$$= 210 \times 8760 \text{ MWh} = 183.96 \times 10^7 \text{ kWh}$$

$$\text{Average load} = \frac{183.96 \times 10^7 \text{ kWh}}{8760 \text{ h}} = 210 \times 10^3 \text{ kW}$$

$$\text{Load factor} = \frac{\text{average load}}{\text{maximum demand}} = \frac{210 \text{ MW}}{240 \text{ MW}} = 0.875 \qquad \textit{Ans.}$$

$$\text{Capacity factor} = \frac{210 \times 8760}{300 \times 8760} = 0.70 \qquad \textit{Ans.}$$

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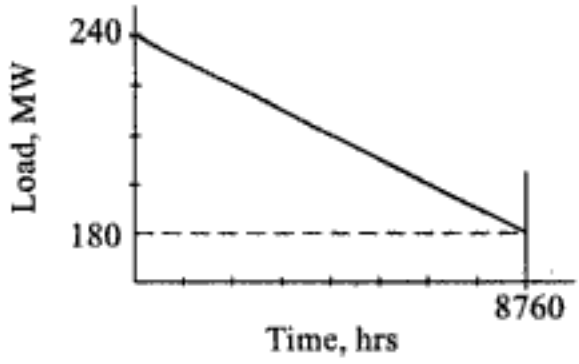

Fig. E1.9

Example 1.10 The maximum load on a thermal power plant of 60 MW capacity is 50 MW at an annual load factor of 60%. The coal consumption is 1 kg per unit of energy generated and the cost of coal is Rs 600 per tonne of coal. Find (a) the annual revenue earned if the energy is sold at Rs 2 per kWh and (b) the capacity of the plant.

Solution Average load = Peak load x Load factor

$$= 50 \times 0.6 = 30 \text{ MW}$$

Energy generated per year $= 30 \times 10^3 \times 8760 = 262.8 \times 10^6$ kWh

$$\text{Coal required per year} = \frac{(262.8 \times 10^6) \times 1}{1000} = 262.8 \times 10^3 \text{ tonnes}$$

Cost of coal per year $= 262.8 \times 10^3 \times 600 = 15{,}768 \times 10^4$ rupees

Cost of energy sold $= 262.8 \times 10^6$ kWh $\times 2 =$ Rs 525.6×10^6

Revenue earned by the power plant per year

$$= \text{Rs } 525.6 \times 10^6 - \text{Rs. } 157.68 \times 10^6$$

$$= \text{Rs } 367.92 \times 10^6 \qquad \textit{Ans.}$$

Example 1.11 A thermal power plant consists of two 60 MW units, each running for 8000 hours, and one 30 MW unit running for 2000 hours per year. The energy produced by the plant is 876×10^6 kWh per year. Determine the plant load factor and plant use factor. Consider the maximum load as equal to the plant capacity.

Solution Plant capacity $= 60 \times 2 + 30 = 150$ MW

$$\text{Average load} = \frac{\text{Energy produced per year}}{8760}$$

$$= \frac{876 \times 10^6}{8760} = 10^5 \text{ kW} = 100 \text{ MW}$$

$$\text{Load factor} = \frac{100}{150} = 0.67 \qquad \textit{Ans.}$$

$$\text{Plant use factor} = \frac{\text{Actual energy produced}}{\text{Max. possible energy that can be produced}}$$

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Max. possible energy that can be produced

$$= 60 \times 2 \times 8000 + 30 \times 1 \times 200 \text{ MWh}$$
$$= 1020 \times 10^6 \text{ kWh}$$

$$\therefore \quad \text{Plant use factor} = \frac{876 \times 10^6}{1020 \times 10^6} = 0.86 \qquad \textit{Ans.}$$

Example 1.12 The following loads are connected to a power plant:

Type of load	*Max. demand (MW)*	*Diversity factor*	*Demand factor*
Domestic	15	1.25	0.70
Commercial	25	1.20	0.90
Industrial	50	1.30	0.98

If the overall diversity factor is 1.5, determine (a) the maximum load, and (b) the connected load of each type.

Solution The maximum demand of the plant

$$= 15 + 25 + 50 = 90 \text{ MW}$$

Since the overall diversity factor is 1.5, the maximum load of the system

$$= \frac{90}{1.5} = 60 \text{ MW} \qquad \textit{Ans.} \text{ (a)}$$

$$\text{Max. domestic demand} = 15 \times 1.25 = 18.75 \text{ MW}$$

$$\text{Connected domestic load} = \frac{18.75}{0.70} = 26.78 \text{ MW}$$

Similarly, connected commercial load

$$= \frac{25 \times 1.2}{0.9} = 33.33 \text{ MW}$$

$$\text{Connected industrial load} = \frac{50 \times 1.3}{0.98} = 66.33 \text{ MW}$$

$\therefore$ Total connected load of the plant

$$= 26.78 + 33.33 + 66.33 = 126.44 \text{ MW} \qquad \textit{Ans.}$$

Example 1.13 A power station has to supply load as follows:

Time (hours)	*0–6*	*6–12*	*12–14*	*14–18*	*18–24*
Load (MW)	*30*	*90*	*60*	*100*	*50*

(a) Draw the load curve. (b) Draw the load duration curve. (c) Select suitable generating units to supply the load. (d) Calculate the load factor. (e) Calculate the capacity of the plant and the plant capacity factor.

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Solution The load curve and load duration curve have been drawn in Fig. E1.13.

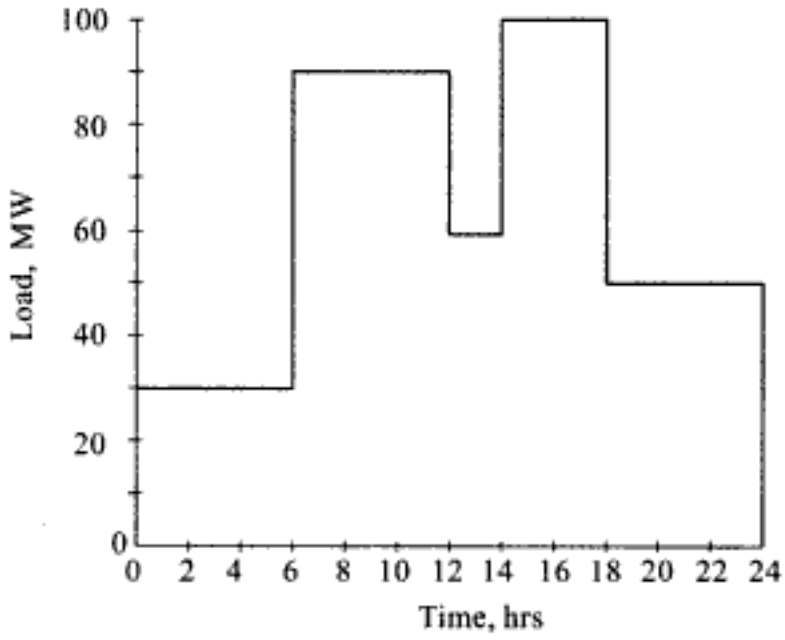
$$\text{Energy generated} = 30 \times 6 + 90 \times 6 + 60 \times 2 + 100 \times 4 + 50 \times 6$$
$$= 1540 \text{ MWh} = 1540 \times 10^3 \text{ kWh}$$

$$\text{Average load} = \frac{1540 \times 10^3}{24} \text{ kW}$$

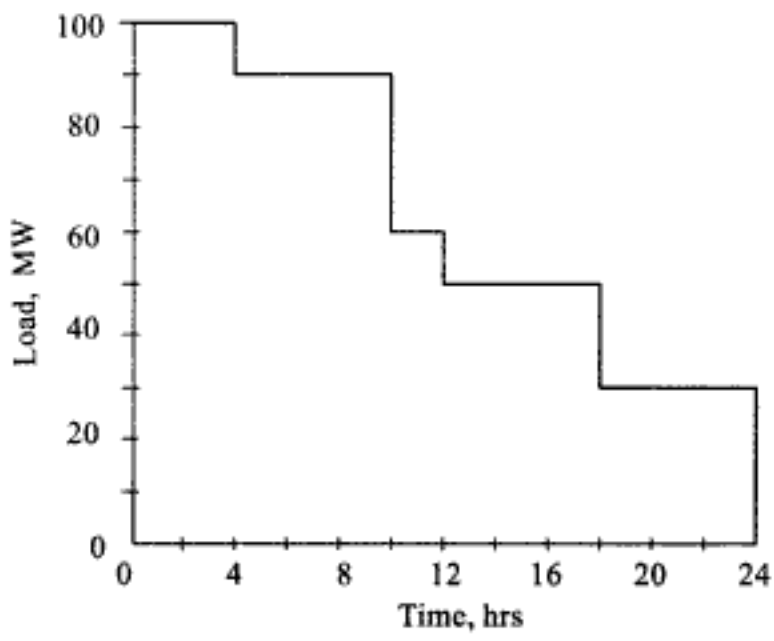
$$\text{Maximum demand} = 100 \times 10^3 \text{ kW}$$

$$\text{Load factor} = \frac{1540 \times 10^3}{24 \times 100 \times 10^3} = 0.64 \qquad \textit{Ans. (d)}$$

To supply the load, three generating units, each of 30 MW capacity, and one generating unit of 10 MW capacity will be selected. One additional unit will be kept as standby. Its capacity will be equal to that of the largest unit, i.e. 30 MW.



(a) Load curve



(b) Load duration curve

Fig. E1.13

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Load duration curve will indicate the operational schedule of different generating units, which will be as follows:

(i) One unit of 30 MW will run for 24 hours
(ii) Second unit of 30 MW will run for 18 hours
(iii) Third unit of 30 MW will run for 10 hours
(iv) Fourth unit of 10 MW will run for 4 hours *Ans.* (c)

$$\text{Plant capacity} = 30 \times 4 + 10 \times 1 = 130 \text{ MW} \qquad \textit{Ans.}\ (e)$$

$$\text{Capacity factor} = \frac{\text{energy generated}}{\text{capacity} \times \text{operating time}}$$

$$= \frac{1540 \times 10^3 \text{ kWh}}{130 \times 10^3 \times 24 \text{ kWh}} = 0.494 \qquad \textit{Ans.}\ (e)$$

Example 1.14 A generating unit of 10 MW capacity supplies the following loads:

(a) Domestic consumers with a maximum demand of 6 MW at a load factor of 20%.
(b) Small industrial load with a maximum demand of 3.6 MW at a load factor of 50%.
(c) Street-light load with a maximum demand of 400 kW at 30% load factor.

Find the overall cost of energy per kWh for each type of consumer using the following data:

Capital cost of the plant = Rs 10,000 per kW
Total running cost = Rs 36,00,000 per year

Annual rate of interest and depreciation on capital cost = 10%

Solution Energy supplied per year to all three consumers

$$= [(600 \times 0.2) + (3600 \times 0.5) + (400 \times 0.3)] \times 8760$$
$$= 3120 \times 8760 \text{ kWh}$$

$$\text{Operating charges per kWh} = \frac{36,00,000}{3120 \times 8760}$$

$$= 0.132 \quad \text{or} \quad 13.2 \text{ paise}$$

Capital cost of the plant = $10 \times 10^3 \times 10^4$ = Rs 10^8
Fixed charges per year = 0.1×10^8 = Rs 10^7
Fixed charges per kW = (Rs 10^7)/(10^4 kW) = Rs 1000

(a) For domestic consumers,

$$\text{total charges} = 1000 \times 6000 + 0.132 \times 6000 \times 0.2 \times 8760$$
$$= \text{Rs } 7,387,584$$

$$\text{Overall cost per kWh} = \frac{\text{Rs } 7,387,584}{1200 \times 8760 \text{ kWh}}$$

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$$= \text{Rs } 0.70 \quad \text{or} \quad 70 \text{ paise} \qquad \textit{Ans.}$$

(b) For industrial consumers,

$$\text{total charges} = 1000 \times 3600 + 0.132 \times 3600 \times 0.5 \times 8760$$
$$= \text{Rs } 5{,}681{,}376$$

$$\text{Overall cost per kWh} = \frac{\text{Rs } 5{,}681{,}376}{1800 \times 8760} = \text{Rs } 0.36, \quad \text{or} \quad 36 \text{ paise} \qquad \textit{Ans.}$$

(c) For street-lighting load,

$$\text{total charges} = 1000 \times 400 + 0.132 \times 400 \times 0.3 \times 8760$$
$$= \text{Rs } 5{,}387{,}584$$

$$\text{Overall cost per kWh} = \frac{\text{Rs } 5{,}387{,}584}{120 \times 8760} = \text{Rs. } 0.51 \quad \text{or} \quad 51 \text{ paise} \qquad \textit{Ans.}$$

Example 1.15 The capital cost of a power generating equipment in a steam power plant is Rs 80×10^6. The useful life of the plant is 30 years and its salvage value is 5% of the capital cost. Determine by the sinking-fund method the amount of money to be saved annually for replacement if the yearly rate of compound interest is 6%.

Solution From Eq. (1.8), the amount of money to be kept aside each year is given by

$$P = \frac{A'i}{(1+i)^N - 1}$$

Here,

$$A' = \text{capital cost} - \text{salvage value}$$
$$= 0.95 \times 80 \times 10^6 = \text{Rs } 76 \times 10^6$$
$$N = 30 \text{ years}; \quad i = 0.06$$

$$P = \frac{76 \times 10^6 \times 0.06}{(1.06)^{30} - 1} = \frac{4560000}{4.7435} = \text{Rs } 9{,}61{,}317 \qquad \textit{Ans.}$$

Example 1.16 The initial investment in a power plant is Rs 4000 crore. After 4 years, a further Rs 1200 crore have to be paid and from the 5th onwards to the 12th year Rs 400 crore per year are paid. The salvage value of the plant after 30 years will be Rs 600 crore. The interest rate is 10%. Estimate the present worth of these payments at the time of commissioning.

Solution By using Eqs. (1.9) and (1.10), the present worth of all payments at the time of commissioning is

$$\text{PW} = 4000 + 1200 \frac{1}{(1+0.1)^4} + \frac{400}{0.1(1+0.1)^{12}} [(1.1)^{12} - 1]$$
$$- \frac{400}{0.1(1+0.1)^5} [(1.1)^5 - 1] - 600 \frac{1}{(1+0.1)^{30}}$$
$$= 4000 + 819.62 + 2725.44 - 1516.31 - 34.39$$
$$= \text{Rs } 5994.36 \text{ crores} \qquad \textit{Ans.}$$

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Example 1.17 Two coal-fired generating units P and Q have the incremental heat rate defined by

$$(\text{IR})_P = 0.4818 \times 10^{-7} L_P^4 - 0.9089 \times 10^{-4} L_P^3 + 0.6842 \times 10^{-1} L_P^2 - 0.2106 \times 10_P^2 + 9860$$

for unit P, and

$$(\text{IR})_Q = 0.9592 \times 10^{-7} L_Q^4 - 0.7811 \times 10^{-4} L_Q^3 + 0.2625 \times 10^{-1} L_Q^2 - 0.2189 \times 10 L_Q + 9003$$

for unit Q; (IR) is in kJ/kWh and L is in MW. Determine the incremental heat rate at which the combined output of these two units is 1000 MW.

Solution

$$L = L_P + L_Q$$

where, L_P = load of unit P and L_Q = load of unit Q

$$L_P + L_Q = 1000$$

By making $(\text{IR})_P = (\text{IR})_Q$, and solving the above three equations by a numerical method such as Newton-Raphson algorithm, we get

$$L_P = 732.5 \text{ MW}$$

$$L_Q = 267.5 \text{ MW}$$

$$\text{IR} = (\text{IR})_P = (\text{IR})_Q = 9292 \text{ kJ/kWh} \quad \textit{Ans.}$$

Example 1.18 The fixed costs of a thermal station are Rs. 2700 per kW of installed capacity per year. The fuel and operating costs are 40 paise per kWh generated. Find the cost of electrical energy per kWh at station load factors of 100%, 75%, 50% and 25%.

Solution Fixed costs per kW per hour $= \dfrac{\text{Rs } 2700}{8760} = 31$ paise

At 100% load factor, 1 kW plant running for 1 hour produces 1 kWh. If run at 75% load factor, it can produce 0.75 kWh and so on. The cost per kWh at different load factors can therefore be found as follows:

Load factor (%)	*Energy produced in 1hr with 1-kW plant (kWh)*	*Fixed cost per hr (paise)*	*Fuel and operating cost (paise)*	*Total cost per hr (paise)*	*Cost per kWh (paise)*
100	1	31	40	71	71
75	0.75	31	30	61	81
50	0.50	31	20	51	102
25	0.25	31	10	41	164

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SHORT-ANSWER QUESTIONS

1.1 Why is electricity the most convenient form of energy?
1.2 What are the different types of power plants where electricity is produced in bulk quantities?
1.3 What is doubling time in electricity production?
1.4 How is the demand for electricity related to the GNP of a country?
1.5 What are the two basic parameters to decide while planning a power plant?
1.6 How is the total installed capacity of a power plant decided?
1.7 On what factors does the unit size of a power plant depend?
1.8 What do you understand by load factor and capacity factor? When are they numerically equal?
1.9 Define reserve factor. What is its usual value? How does it fix the maximum unit size?
1.10 What do you mean by diversity factor?
1.11 How is the load–duration curve constructed?
1.12 Explain what you understand by base load and peaking load. Why are base load plants loaded heavily?
1.13 What are the considerations to be made while selecting the suitable site for a thermal and a nuclear power plant?
1.14 What are the elements which contribute to the cost of electricity?
1.15 How is the total annual cost of electricity estimated?
1.16 What is the forced outage rate?
1.17 How can the cost of power generation be reduced?
1.18 What are the expenses which constitute the construction cost of a power plant?
1.19 How does the economy of scale affect the construction cost?
1.20 What are fixed costs?
1.21 What do you understand by depreciation? How is the depreciation rate determined?
1.22 What is the straight-line method of determining the depreciation charge?
1.23 What do you mean by the sinking-fund method of accumulating the money for the depreciation fund?
1.24 How does the fuel cost relate to the load and the cost of power generation?
1.25 What do you understand by the present-worth concept? How is it estimated?
1.26 What is incremental heat rate?
1.27 Explain the principle of economic scheduling.
1.28 Show that for two units running in parallel, the combined energy input will be minimum if the incremental heat rate of unit 1 is equal to that of unit 2.
1.29 Explain what you mean by spinning reserve.
1.30 Explain the effect of load factor on the cost of electricity generated.

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PROBLEMS

1.1 The loads on a power plant with respect to time for 24 hours are given as follows:

Time (hours)	0–6	6–8	8–12	12–14	14–18	18–22	22–24
Load (MW)	40	50	60	50	70	80	40

Draw the load curve and find out the load factor of the power station. If the loads above 60 MW are taken by a standby unit of 20 MW capacity, find out the load factor and the use factor of the standby unit.

[*Ans.* 0.71, 0.75 and 0.60]

1.2 A central power station has annual load factor, capacity factor and use factor as 60%, 40% and 45%, respectively. The maximum demand is 15 MW. Estimate (a) the annual energy production and revenue earned if the cost of energy is Re 1/kWh, (b) the reserve capacity over and above the peak load, and (c) the hours per year the station is not in service.

[*Ans.* (a) 78.84×10^6 kWh, Rs 78.84×10^6, (b) 7.5 MW, (c) 974 h]

1.3 A power station of 30 MW capacity has the maximum annual demand of 25 MW. It supplies loads having maximum demands of 10 MW, 8.5 MW, 5 MW and 4.5 MW. The annual load factor is 45%. Estimate (a) the average load, (b) the energy supplied per year, (c) the diversity factor, and (d) the demand factor.

[*Ans.* (a) 11.25 MW, (b) 98.55×10^6 kWh, (c) 1.12, (d) 0.89]

1.4 A power station is said to have a use factor of 47% and capacity factor of 40%. For how many hours in a year was the power station not in service?

[*Ans.* 1305 h]

1.5 Determine the generating cost per unit of 80 MW power station with the following data:

Capital cost = Rs 160×10^7
Annual cost of fuel = Rs 32×10^6
Annual wages and taxes = Rs 36×10^6
Interest and depreciation = 10% of capital cost
Annual load factor = 45%

[*Ans.* 72.3 paise]

1.6 The incremental fuel costs for two generating units *A* and *B* of a power plant are given by the following relations:

$dF_A/dP_A = 0.06\,P_A + 11.4$
$dF_B/dP_B = 0.07\,P_B + 10$

where *P* is the power in MW and *F* is the fuel cost in rupees per hour.

(a) Find the economic loading of the two units when the total load to be supplied by the power station is 150 MW.

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(b) Find the net increase in fuel cost per hour if the load is equally shared by the two units.

[*Ans.* (a) $P_A = 70$ MW, $P_B = 80$ MW, (b) Rs 1.63 per hour]

1.7 The following data pertain to a power plant:
Installed capacity = 200 MW; Capital cost = Rs 4000×10^6; Annual cost of fuel, taxes and salaries = Rs 600×10^6; Rate of interest = 5% of the capital; Rate of depreciation = 6% of the capital; Annual load factor = 65%; Capacity factor = 55%; Energy used in running the plant auxiliaries = 4% of total units generated. Determine (a) the reserve capacity, and (b) the cost of power generation.

[*Ans.* (a) 31 MW]

1.8 A power plant has the following annual factors:
Load factor = 70%; Capacity factor = 50%; Use factor = 60%; Maximum demand = 20 MW.
Estimate (a) annual energy production, (b) reserve capacity over and above the peak load, and (c) the hours per year during which the plant is not in service.

[*Ans.* (a) 122.8×10^6 kWh, (b) 8 MW, (c) 2060 h]

1.9 A 60 MW turbogenerator set has an overall efficiency of 25%. The calorific value of coal used is 24 MJ/kg. Estimate the consumption of coal per kWh and also per day of 24 h if the load factor is 30%.

[*Ans.* 0.6 kg, 259.2 tonnes]

1.10 Find the cost of generation per kWh from the following data:
Capacity of the plant = 150 MW; Capital cost = Rs 20,000 per kW installed; Interest and depreciation = 10% on capital; Fuel consumption = 1.2 kg/kWh; Fuel cost = Rs 400 per tonne; Salaries , wages, repair and maintenance = Rs 150×10^6 per year; Maximum demand = 120 MW; Load factor = 50%.

1.11 A plant costing Rs 65,000 has a useful life of 15 years. Find the amount which should be saved annually to replace the equipment at the end of that time (a) by the straight-line method, and (b) by the sinking-fund method if the annual rate of compound interest is 5%. Assume that the salvage value of the equipment is Rs 5000.

[*Ans.* (a) Rs 4000, (b) Rs 2780]

1.12 The connected loads of a consumer consist of 10 electric bulbs of 60 W each and 2 electric heaters of 1000 W each. His maximum demand is 1500 W. On an average, he uses 8 bulbs for 5 hours a day and one heater for 3 hours a day. Estimate his average load, monthly energy consumption and load factor.

[*Ans.* 350 W, 252 kWh, 23.33%]

1.13 A power generating station has a maximum demand of 10,000 kW and the daily load on the station is as follows:

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Time	kW	Time	kW
6 a.m. to 8 a.m.	3,500	5 p.m. to 7 p.m.	8,500
8 a.m. to 12 noon	8,000	7 p.m. to 9 p.m.	10,000
12 noon to 1 p.m.	3,000	9 p.m. to 11 p.m.	4,500
1 p.m. to 5 p.m.	7, 500	11 p.m. to 6 a.m.	2,000

(a) Draw the load curve and the load duration curve.
(b) Choose the size and number of generating units.
(c) Draw up the operating schedule of the units.
(d) What reserve plant would be necessary?
(e) Calculate the load factor, plant capacity factor and plant use factor of the station.

[*Ans.* (b) Three units of 5000, 2500 and 2500 kW capacity, (d) 5000 kW, (e) 55%, 36.66%, 86.5%]

1.14 Explain the method of constructing a load duration curve using a load curve. The following data were collected from the daily load curves of a power system during a year:

Load (kW)		Duration (hours)
15,000		87
12,000	and over	876
10,000	and over	1,752
8,000	and over	2,628
6,000	and over	4,380
4,000	and over	7,000
2,000	and over	8,760

Construct the annual load duration curve and find the load factor of the system.

[*Ans.* 46.8%]

1.15 A power station has an installed capacity of 210 MW. The cost of the station is Rs 2 crore per MW. The fixed costs are 14% of the cost of investment. On full load at 100% load factor, the variable costs of the station per year are 1.5 times the fixed costs. Assume that there is no reserve capacity of the plant and that the variable costs are proportional to the energy production. Find the cost of generation per kWh at load factors of 100%, 80%, 60%, 40% and 20%. Plot the curve.

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2

Analysis of Steam Cycles

2.1 STEAM POWER PLANT

A steam power plant continuously converts the energy stored in fossil fuels (coal, oil, natural gas) or fissile fuels (uranium, thorium) into shaft work and ultimately into electricity. The working fluid is water which is sometimes in the liquid phase and sometimes in the vapour phase during its cycle of operations. Figure 2.1 illustrates a fossil fuelled power plant as a bulk energy converter from fuel to electricity using water as the working medium. Energy released by the burning of fuel is transferred to water in the boiler (B) to generate steam at a high pressure and temperature, which then expands in the turbine (T) to a low pressure to produce shaft work. The steam leaving the turbine is condensed into water in the condenser (C) where cooling water from a river or sea circulates carrying away the heat released during condensation. The water (condensate) is then fed back to the boiler by the pump (P), and the cycle goes on repeating itself. The working substance, water, thus follows along the B–T–C–P path of the cycle interacting externally as shown. Since the fluid is undergoing a cyclic process, there will be no net change in its internal energy over the cycle ($\oint dE = 0$), and consequently, the net energy transferred to the unit mass of the fluid as heat during the cycle must equal the net energy transfer as work from the fluid.

$$\sum_{\text{cycle}} Q_{\text{net}} = \sum_{\text{cycle}} W_{\text{net}}$$

or,

$$Q_1 - Q_2 = W_T - W_P \tag{2.1}$$

where Q_1 = heat transferred to the working fluid, kJ/kg

Q_2 = heat rejected from the working fluid, kJ/kg

W_T = work transferred from the working fluid, kJ/kg

W_P = work transferred into the working fluid, kJ/kg

The efficiency of the vapour power cycle would thus be

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$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \tag{2.2}$$

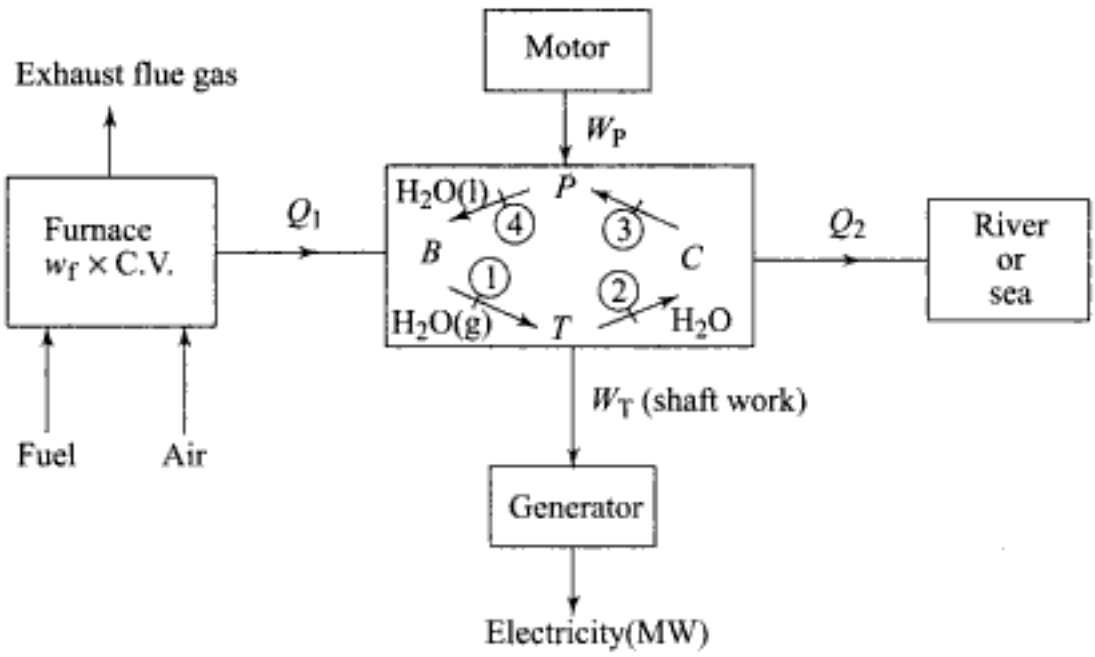

Fig.2.1 *Steam power plant—bulk energy converter from fuel to electricity*

2.2 RANKINE CYCLE

For each process in the vapour power cycle, it is possible to assume a hypothetical or ideal process which represents the basic intended operation and does not produce any extraneous effect (like heat loss). For the steam boiler, this would be a reversible constant pressure heating process of water to form steam, for the turbine the ideal process would be a reversible adiabatic expansion of steam, for the condenser it would be a reversible constant pressure heat rejection as the steam condenses till it becomes saturated liquid, and for the pump the ideal process would be the reversible adiabatic compression of this liquid ending at the initial pressure. When all these four processes are ideal, the cycle is an ideal cycle, called a Rankine cycle. This is a reversible cycle. Figure 2.2 shows the flow diagram of the Rankine cycle, and in Fig. 2.3, the cycle has been plotted on $p - v$, $T - s$ and h–s coordinates. Applying the steady flow energy equation (SFEE) to each of the processes on the basis of a unit mass of fluid and neglecting changes in kinetic and potential energy, the work and heat interactions can be evaluated in terms of the properties of the fluid.

For 1 kg fluid, the SFEE for the boiler as the control volume gives

$$h_4 + Q_1 = h_1$$

$$\therefore \quad Q_1 = h_1 - h_4 \tag{2.3}$$

The SFEE for the turbine as the control volume gives

$$h_1 = W_T + h_2$$

$$\therefore \quad W_T = h_1 - h_2 \tag{2.4}$$

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Similarly, the SFEE for the condenser is

$$h_2 = Q_2 + h_3$$

$$\therefore \quad Q_2 = h_2 - h_3 \tag{2.5}$$

and the SFEE for the pump gives

$$h_3 + W_P = h_4$$

$$\therefore \quad W_P = h_4 - h_3 \tag{2.6}$$

The efficiency of the Rankine cycle is then given by

$$\eta = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \tag{2.7}$$

The pump handles liquid water which is incompressible, i.e. its density or specific volume undergoes little change with an increase in pressure. For reversible adiabatic compression, by the use of the property relation

$$T\,ds = dh - v\,dp$$

Since $ds = 0$,

$$\therefore \quad \int_3^4 dh = \int_3^4 v\,dp$$

$$h_4 - h_3 = v_3(p_4 - p_3) = W_P \tag{2.8}$$

The pump work is usually small compared to the turbine work, and is often neglected.

The capacity of a steam plant is often expressed in terms of steam rate or specific steam consumption (s.s.c). It is defined as the rate of steam flow (kg/s) required to produce unit shaft output (1 kW).

$$\therefore \quad \text{Steam rate (S.R.)} = \frac{1}{W_{net}} \frac{\text{kg}}{\text{kWs}} \tag{2.9}$$

The cycle efficiency is sometimes expressed alternatively as heat rate which is the rate of heat input (kJ/s) required to produce unit shaft output (1 kW).

$$\text{Heat rate (H.R.)} = \frac{Q_1}{W_T - W_P} = \frac{1}{\eta} \frac{\text{kJ}}{\text{kWs}} \tag{2.10}$$

2.2.1 Economiser, Evaporator and Superheater

Heat transfer to water in the steam generator takes place in the three different regimes (Figs 2.2 and 2.3). Water is first heated sensibly in the economiser in the liquid phase at a certain pressure from state 4 to state 5 till it becomes saturated liquid.

In the evaporator or the boiler proper, there is phase change or boiling with the state changing from 5 to 6 by absorbing the latent heat of vaporization at that pressure. The saturated vapour at state 6 is further heated at constant pressure in the superheater to state 1 in the vapour or gaseous phase. For unit mass of fluid, heat transfer in these three types of heat exchangers is given by

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$$\left.\begin{aligned} Q_{Eco} &= h_5 - h_4 \\ Q_{Eva} &= h_6 - h_5 = h_{fg} \\ O_{SH} &= h_1 - h_6 \end{aligned}\right\} \qquad (2.11)$$

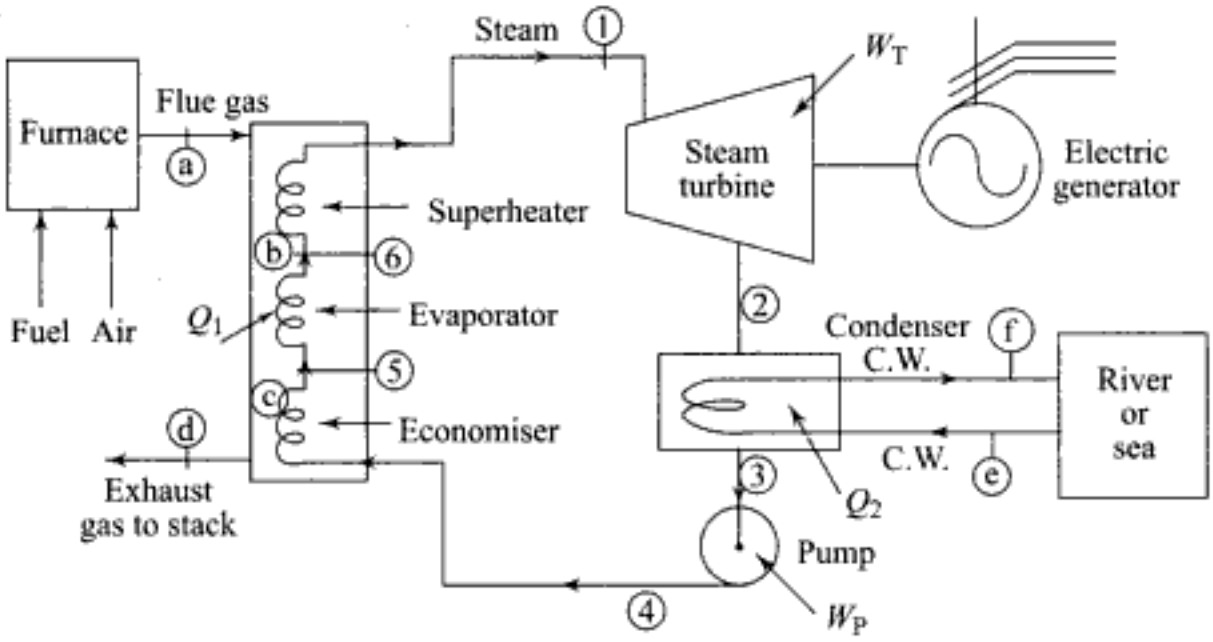

Fig. 2.2 *A simple steam plant representing Rankine cycle*

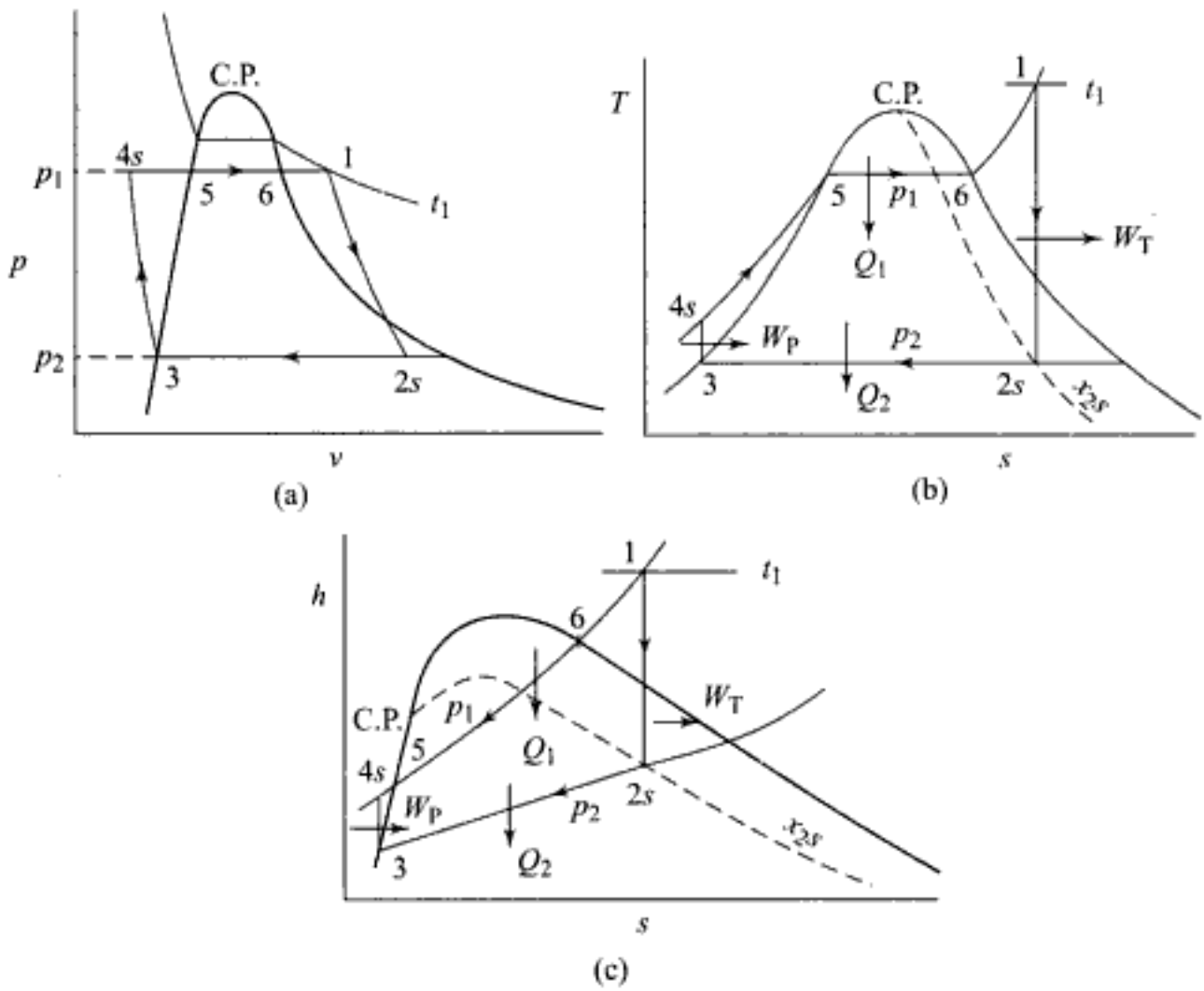

Fig.2.3 *Rankine cycle on p – v, T – s and h – s coordinates (a, b and c)*

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It is demonstrated in Fig. 2.4 by the areas under the respective processes at the particular isobar. The fractions of the total heat transfer absorbed in the economiser, evaporator and superheater are given by

$$\left.\begin{aligned}\frac{Q_{Eco}}{Q_1} &= \frac{h_5 - h_4}{h_1 - h_4} = \frac{\text{area under } 4-5}{\text{area under } 4-1}\\ \frac{Q_{Eva}}{Q_1} &= \frac{h_6 - h_5}{h_1 - h_4} = \frac{\text{area under } 5-6}{\text{area under } 4-1}\\ \frac{Q_{SH}}{Q_1} &= \frac{h_1 - h_6}{h_1 - h_4} = \frac{\text{area under } 6-1}{\text{area under } 4-1}\end{aligned}\right\} \quad (2.12)$$

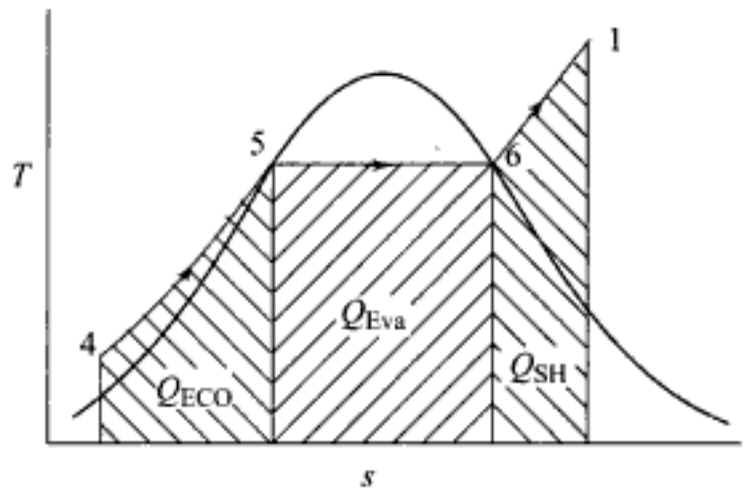

Fig. 2.4 *Fraction of total heat transfer absorbed in the economiser, evaporator and superheater*

It may be noted that as the pressure increases, the latent heat decreases and so the heat absorbed in the evaporator decreases and the fraction of the total heat absorbed in the superheater increases. In high pressure boilers, more than 40% of the total heat is absorbed in the superheaters. For steam generators operating above the critical pressure (approx 221 bar) there is no evaporator or boiling section. However, there is a transition zone where all the liquid on being heated suddenly flashes into vapour (Fig. 2.5).

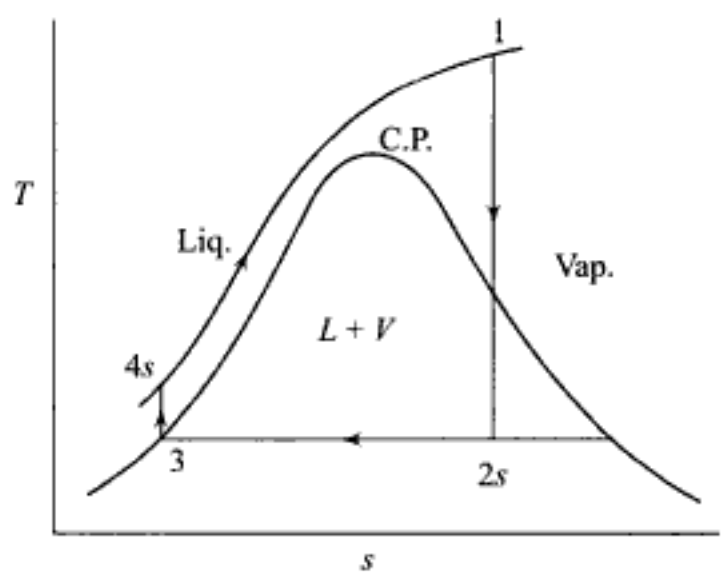

Fig. 2.5 *Rankine cycle with supercritical boiler pressure*

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2.2.2 Internally Irreversible Rankine Cycle

Internal irreversibility of Rankine cycle is caused by fluid friction, throttling and mixing. As the flow rates in the steam turbine as well as in the pumps are large, and the expansion and compression processes are quite rapid, the heat loss per unit mass may be considered negligible. Though the assumption of adiabatic flow in them is still valid, due to fluid friction the expansion and compression processes are not reversible and entropy of the fluid in both increases (Fig. 2.6). The internal or isentropic efficiency (η_T) of the turbine is given by

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{2.13}$$

The pumping process, being adiabatic and irreversible, also results in an increase in entropy, and the isentropic efficiency of the pump is given by

$$\eta_P = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{2.14}$$

The actual pump work would thus be

$$W_P = \frac{h_{4s} - h_3}{\eta_P} = \frac{v_3(p_4 - p_3)}{\eta_P} \tag{2.15}$$

One thus pays a penalty for irreversibility: the turbine produces less work and the pump absorbs more work.

The liquid leaving the pump must be at a higher pressure than at the turbine inlet because of the pressure drops due to friction etc. in boiler heat exchangers, feedwater heaters, pipes, bends, valves etc. Thus, p_4 represents the exit pump pressure, p_1 the turbine inlet pressure, and p_5 the steam generator exit pressure (Fig. 2.6). Steam leaves the boiler at state 5 and enters the turbine at state 1. The pressure of steam drops from p_5 to p'_5 (or p_1) due to friction in the pipeline and entropy decreases from 5′ to 1 due to heat loss.

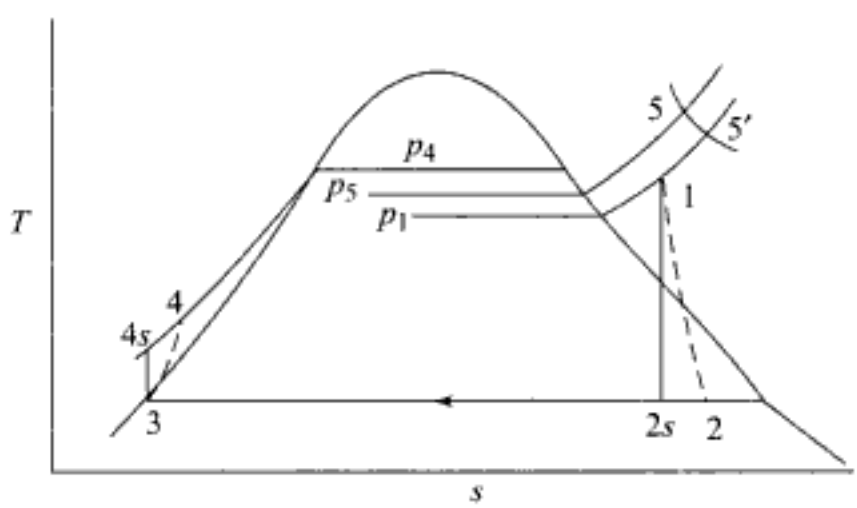

Fig. 2.6 *Internally irreversible Rankine cycle*

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2.2.3 Externally Irreversible Rankine Cycle

External irreversibility of the Rankine cycle is caused due to the temperature differences between the combustion gases and the working fluid on the source side, and the temperature differences between the condensing working fluid and the condenser cooling water on the sink side (Figs 2.2 and 2.7). The products of combustion or flue gases get cooled from *a* to *d* and the working fluid temperature rises from 4 to 1 in counterflow heat exchangers. The minimum temperature differences between the two fluids are $c - 5$ and $a - 1$, and the points where these occur are called *pinch points*. Too small a pinch point temperature difference causes a lower thermal (external) irreversibility and an increase in surface area resulting in a large expensive steam generator, whereas a large pinch-point temperature difference results in a small, inexpensive steam generator but with a reduced plant efficiency due to a large thermal irreversibility. The most economical pinch-point temperature difference is obtained by optimization which takes into account the fixed charges (based on capital costs) and operating costs (based on efficiency, and hence, fuel costs).

Line *e–f* represents the rise in temperature of the cooling water in the condenser, whereas the working fluid temperature remains constant during condensation process 2 – 3. When one of the fluids undergoes phase change, the direction of flow of the other fluid, whether parallel or counter-flow, is immaterial so far as the size of the heat exchanger and thermal irreversibility are concerned.

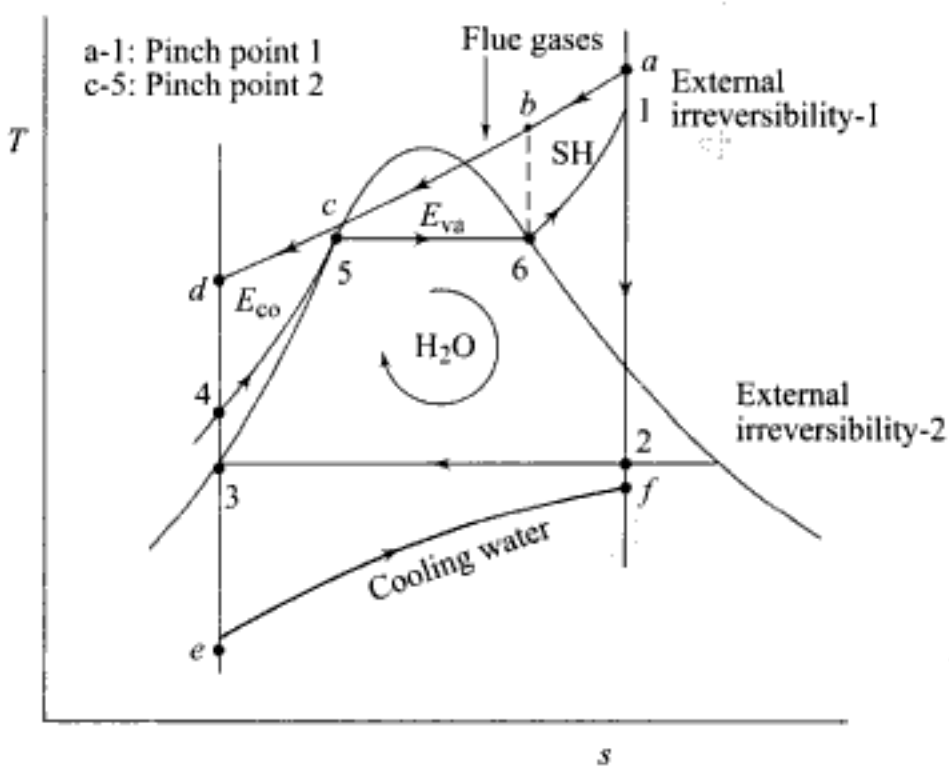

Fig. 2.7 *External irreversibility with Rankine cycle*

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2.3 CARNOT CYCLE

The Carnot cycle is an ideal but non-practical cycle giving the maximum possible thermal efficiency for a cycle operating on selected maximum and minimum temperature ranges. For the Carnot cycle 1–2–3–4 in Fig. 2.8, compressing a very wet steam at state 3 would require a compressor of size and cost comparable with the turbine; it would absorb work comparable to that developed by the turbine and its life would be short because of blade erosion and cavitation problems due to excessive moisture. For the cycle 1–2–5–6–1, the pump work $(h_6 - h_5)$ is again very high and it is impossible to supply heat at infinite pressures and at constant temperature from state 6 to state 4. So, the Carnot cycle cannot be realised in practice, but it sets the upper limit to which the cycle efficiency of any thermal plant can be raised. For both the cycles 1–2–3–4 and 1–2–5–6, the cycle efficiency is given by

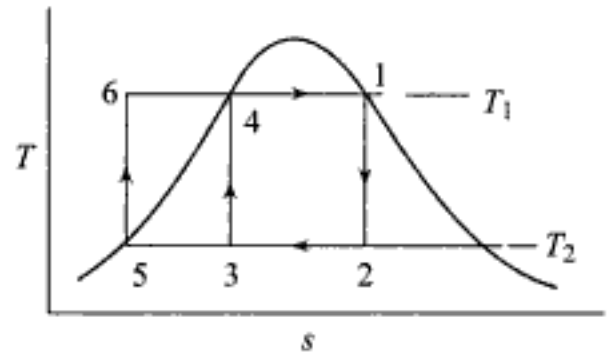

Fig. 2.8 *Carnot cycle*

$$\eta_{max} = 1 - \frac{T_2}{T_1} = \eta_{Carnot} \tag{2.16}$$

2.4 MEAN TEMPERATURE OF HEAT ADDITION

In the Rankine cycle, heat is added reversibly at a constant pressure but at infinite temperatures. If T_{m1} is the mean temperature of heat addition as shown in Fig. 2.9, so that the area under 4 and 1 is equal to the area under 5 and 6, then heat added is

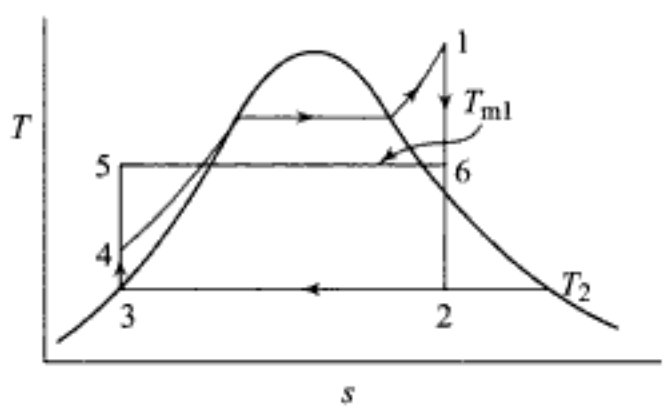

Fig. 2.9 *Mean temperature of heat addition*

$$Q_1 = h_1 - h_4 = T_{m1}(s_1 - s_4)$$

$$\therefore \quad T_{m1} = \text{mean temperature of heat addition} = \frac{h_1 - h_4}{s_1 - s_4} \tag{2.17}$$

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Since Q_2 = heat rejected = $h_2 - h_3 = T_2(s_1 - s_4)$,

$$\therefore \qquad \eta_{\text{Rankine}} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2(s_1 - s_4)}{T_{m_1}(s_1 - s_4)} = 1 - \frac{T_2}{T_{m_1}} \tag{2.18}$$

where T_2 is the temperature of heat rejection. The lower is the T_2 for a given T_{m1}, i.e. lower is the condenser pressure, the higher will be the efficiency of the Rankine cycle. But, the lowest practicable temperature of heat rejection is the temperature of the surroundings, T_o. The saturation pressure corresponding to this temperature T_o is the minimum pressure to which steam can be expanded in the turbine. This being fixed by the ambient conditions,

$$\eta_{\text{Rankine}} = f(T_{m1}) \quad \text{only} \tag{2.19}$$

The higher the mean temperature of heat addition, the higher will be the cycle efficiency.

2.4.1 Effect of Superheat

The effect of increasing the initial temperature at constant pressure on cycle efficiency is shown in Fig. 2.10. When the initial state changes from 1 to 1′, T_{m1} between 1 and 1′ is higher than T_{m1} between 4 and 1. So an increase in the superheat at constant pressure increases the mean temperature of heat addition and hence, the cycle efficiency. Moreover, with increase in superheat, the expansion line of steam in the turbine shifts to the right, as a result of which the quality of steam at turbine exhaust increases and performance of the turbine improves, as explained later. If hot flue gas is the primary fluid or heat source for steam generation in the power cycle, the use of superheat also reduces the thermal irreversibility. Since $c_p = T\left(\frac{\partial s}{\partial T}\right)_p$, we have the slope $\left(\frac{\partial T}{\partial s}\right)_p = \frac{T}{c_p}$. Now $(c_p)_{\text{water}} > (c_p)_{\text{gas}}$, therefore $\left(\frac{\partial T}{\partial s}\right)_{\text{gas}} > \left(\frac{\partial T}{\partial s}\right)_{\text{water}}$ If hot pressurized water is the heat source for steam generation, as in a pressurized water reactor (PWR), the superheat may not be practical, since the temperature differences between m–p and 5–1 vary little (Fig. 2.11). If the temperature at state 1 is fixed, to use superheat the pressure has to be reduced from p to p'. This increases the overall temperature difference resulting in reducing rather than increasing the cycle efficiency.

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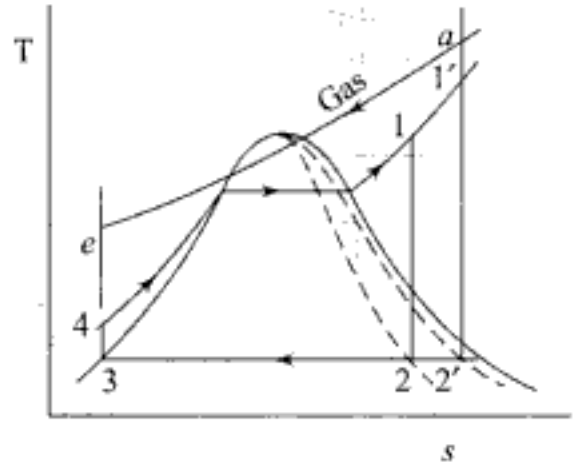

Fig. 2.10 *Effect of superheat on the mean temperature of heat addition*

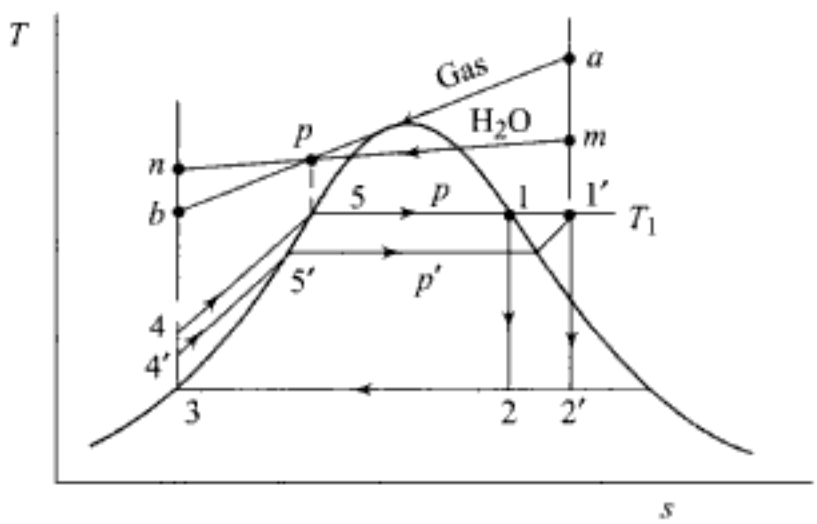

Fig. 2.11 *Superheat may not be practical with hot pressurized water as heat source as in a PWR*

Therefore, fossil fuel steam generators as well as gas-cooled and liquid metal cooled nuclear power plants employ superheat, while PWR power plants do not use superheat.

2.4.2 Effect of Inlet Pressure

The maximum temperature of steam that can be used is fixed from metallurgical considerations, i.e. the materials used for the manufacture of the components which are subjected to the high-pressure high-temperature steam like the superheaters, valves, pipelines, inlet stages of the turbine and so on. It is called the *metallurgical limit*. When the maximum temperature is fixed by this limit, as the operating steam pressure at which heat is added in the boiler increases from p_1 to p_2 (Fig. 2.12), the mean temperature of heat addition increases since T_{m1} between states 7 and 5 is higher than that between states 4 and 1. But when the turbine inlet pressure increases from p_1 to p_2, the ideal expansion line of steam shifts to the left and the moisture content at the turbine exhaust increases (because $x_6 < x_2$). If the moisture content of steam in the later stages of the turbine is high, the entrained water particles along with the vapour coming out

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of the nozzles with high velocity strike the blades and erode their edges, as a result of which the life of the blades decreases. From the consideration of the erosion of blades in the later stages of a turbine, the maximum moisture content at the turbine exhaust is not allowed to exceed 12%, or the quality of steam to fall below 88%. It is desirable that most of the turbine expansion should take place in the single phase or vapour region.

Therefore, with the maximum steam temperature at the turbine inlet, the minimum temperature of heat rejection, and the minimum quality of steam at the turbine exhaust being fixed by the materials used, the ambient conditions, and turbine blade erosion, respectively, the maximum steam pressure at the turbine inlet also gets fixed (Fig. 2.13). The vertical line drawn from state 2 fixed by T_2 and x_2, intersects the T_{max} line at state 1, which gives the maximum steam pressure at the turbine inlet. The irreversibility in the expansion process has, however, not been considered.

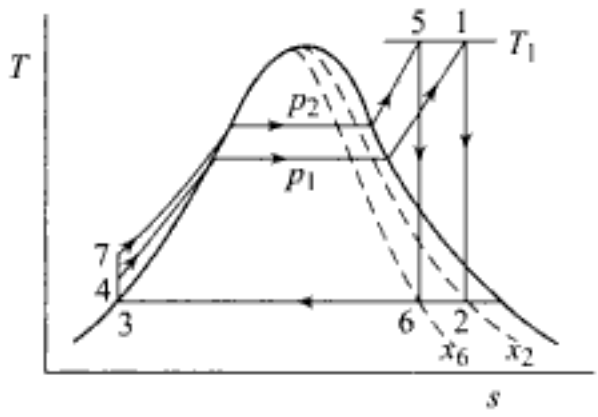

Fig 2.12 Effect of increase of pressure on Rankine cycle

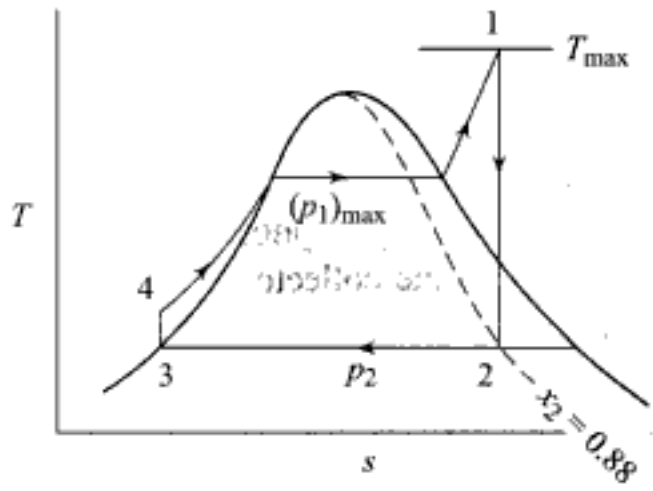

Fig. 2.13 Fixing of maximum steam pressure at turbine inlet

2.5 EFFECT OF VARIATION OF STEAM CONDITION ON THERMAL EFFICIENCY OF STEAM POWER PLANT

The variation of Rankine efficiency with the inlet steam pressure at a constant steam temperature of 470 °C and at three condenser pressures is shown in Fig. 2.14. It is seen that for inlet steam pressures above 100 bar, there is a continuing but decreasing rate of improvement of cycle efficiency. The increase in steam pressure is limited by considerations of mechanical stresses and the ensuing higher cost of equipment.

Figure 2.14 also demonstrates that there is a considerable improvement in cycle efficiency with the decrease of condenser pressure. Such a decrease mainly depends on the available cooling water temperature (t_{c1}) and thus on the climatic condition of the place. A lower cooling water temperature gives lower condenser pressure (higher vacuum). It follows that with identical steam

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conditions, and cycle and similar equipment, the thermal efficiency of a condensing steam power plant will be less in a warm region than in a cold region.

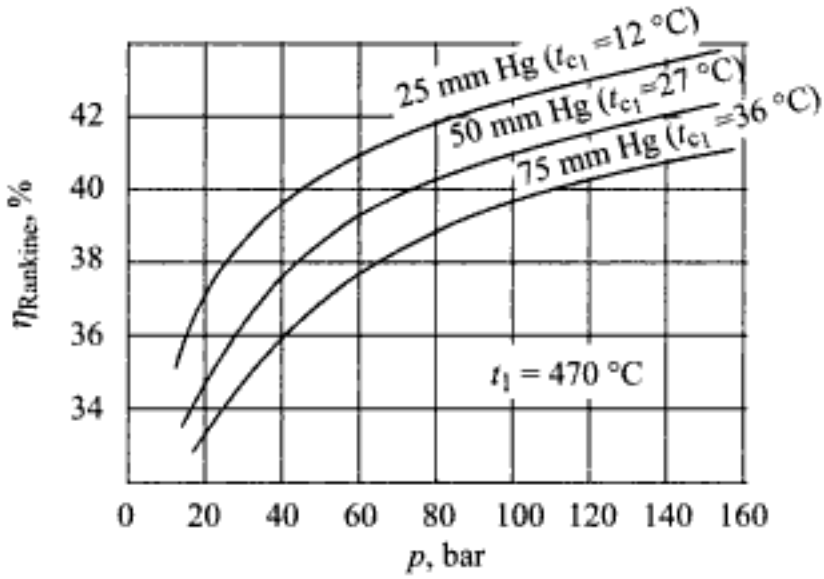

Fig.2.14 *Effect of inlet steam pressure (p_1) and condenser pressure on Rankine efficiency with constant inlet steam temperature of 470 °C*

An increase in inlet steam temperature, i.e. an increase in superheat at constant inlet steam pressure and condenser pressure gives a steady improvement in cycle efficiency and lowers the heat rate due to the increase in T_{m1}, as discussed in the earlier section. Raising the inlet steam temperature also reduces the wetness of the steam in the later stages of the turbine and improves the turbine internal efficiency. However, the increase in steam temperature is limited by the properties of the construction materials of boilers and turbines. The ultimate strength of unalloyed steels drops by about 30% as the steam temperature is raised from 400 to 500 °C. Alloying with chromium and molybdenum and eventually, the use of austenitic instead of ferritic steels increases the strength at high temperatures. Steam temperatures up to 620 °C have been used in some plants. The operating experience with the expensive high-temperature austenitic steels has, however, not been uniformly satisfactory. Recent practice in steam power plants generally limits steam temperatures to 538°C and in a few cases to 565 °C.

The maximum steam pressures (throttle pressures) that can be used at three different condenser pressures and two inlet steam temperatures are shown in Table 2.1 with turbine internal efficiency of 85% and quality at turbine exhaust of 88%. For other values of η_T, t_1 and x_2, the pressure limits can be readily determined by drawing the corresponding expansion line of the turbine on a Mollier diagram (Fig. 2.13).

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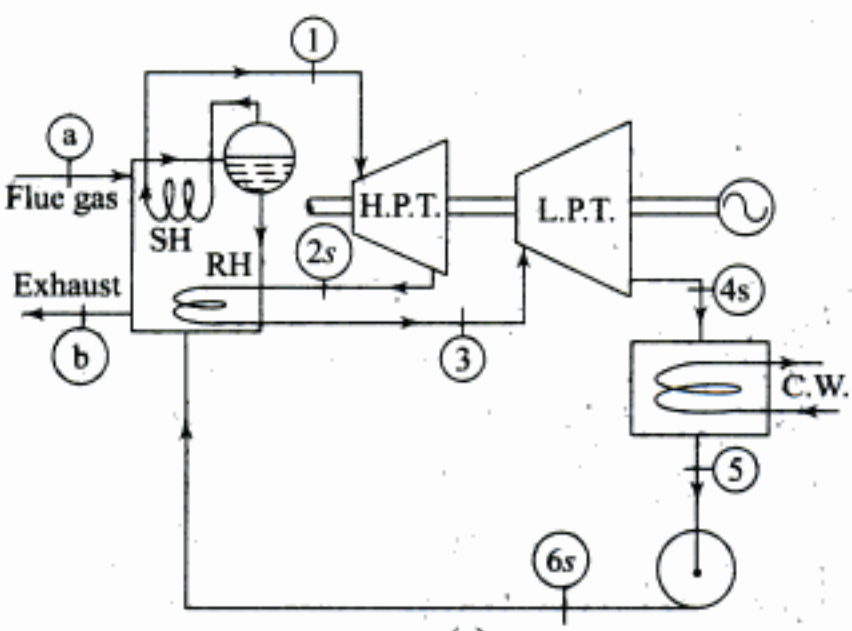
Table 2.1 *Maximum inlet steam pressure (without reheating) with $\eta_T = 0.85$ and $x_2 = 0.88$*

Condenser pressure	Turbine inlet steam temperature	
	$t_1 = 500$ °C	$t_1 = 550$ °C
$p_2 = 25$ mm Hg abs	$(p_1)_{max} = 60$ bar	$(p_1)_{max} = 90$ bar
$p_2 = 50$ mm Hg abs	$(p_1)_{max} = 80$ bar	$(p_1)_{max} = 115$ bar
$p_2 = 75$ mm Hg abs	$(p_1)_{max} = 95$ bar	$(p_1)_{max} = 135$ bar

2.6 REHEATING OF STEAM

If a steam pressure higher than $(p_1)_{max}$ is used (Fig. 2.13), in order to limit the quality to 0.88 at turbine exhaust, reheating of steam has to be adopted. In that case, all the steam after partial expansion in the turbine is brought back to the boiler, reheated by combustion gases and then fed back to the turbine for further expansion. The flow, *T–s* and *h–s* diagrams for the ideal Rankine cycle with reheat are shown in Fig. 2.15. In the *reheat cycle* the expansion of steam from the initial state 1 to the condenser pressure is carried out in two or more steps depending upon the number of reheats used. In the first step, steam expands in the high pressure (H.P.) turbine from the initial state to some intermediate pressure (process 1–2*s*). The steam is resuperheated (or reheated) at constant pressure in the boiler (process 2*s*–3) and the remaining expansion (process 3–4*s*) of steam is carried out in the low pressure (L.P.) turbine. For 1 kg of steam,



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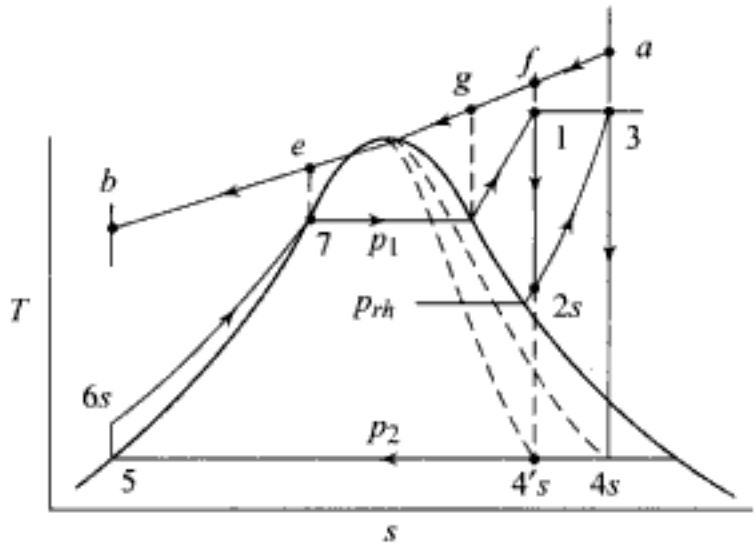

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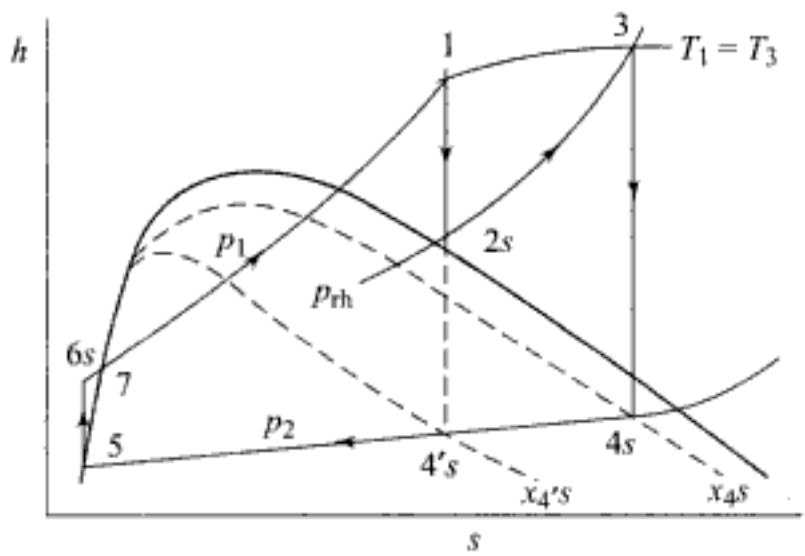

(c)

Fig. 2.15 *Reheat Cycle*

$$Q_1 = h_1 - h_{6s} + h_3 - h_{2s} \qquad Q_2 = h_{4s} - h_5$$

$$W_T = h_1 - h_{2s} + h_3 - h_{4s} \qquad W_p = h_{6s} - h_5$$

$$\eta = \frac{W_T - W_p}{Q_1} = \frac{(h_1 - h_{2s} + h_3 - h_{4s}) - (h_{6s} - h_5)}{h_1 - h_{6s} + h_3 - h_{2s}} \tag{2.20}$$

$$\text{Steam rate} = \frac{3600}{W_{net}} \text{ kg/kWh} \tag{2.21}$$

$$\text{Heat rate} = \frac{3600}{\eta} \text{ kJ/kWh} \tag{2.22}$$

Had the high pressure p_1 been used without reheat (Fig. 2.15), the cycle would have been 1–4′ s–5–6s with lot of moisture at turbine exhaust having quality $x_{4's}$. With the use of reheat, the area 2s–3–4s–4′s has been added to the basic cycle. It is seen that the net work output of the plant increases with reheat because $(h_3 - h_{4s})$ is greater than $(h_{2s} - h_{4's})$, and hence the steam rate decreases. Reheating also improves the quality at turbine exhaust from $x_{4's}$ to x_{4s}. Whether

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the cycle efficiency improves with reheat depends upon whether the mean temperature of heat addition in process 2*s*–3 is higher than that in process 6*s*–1.

By increasing the number of reheats, still higher steam pressures could be used, but the mechanical stresses increase in much higher proportion than the pressure because of the prevailing high temperature. The cost and fabrication difficulties will also increase. In that way the maximum steam pressure gets fixed, and more than two reheats have not yet been used. The use of more than two reheats results in cycle complication and increases capital costs that are not justified by improvement in the cycle efficiency.

The cycle efficiency in a single reheat plant is influenced by the pressure (p_{rh}) at which steam is reheated. The change in cycle efficiency $\Delta\eta$ per cent as a function of the ratio of reheat pressure to initial pressure p_{rh}/p_1 is plotted in Fig. 2.16, for the cycle with initial steam at 172 bar and 538 °C and steam reheat at 538 °C. For $p_{rh}/p_1 = 1$, $\Delta\eta = 0$ since no reheat is used. A reheat pressure too close to the initial pressure results in little improvement in cycle efficiency because only a small portion of additional heat is added at high temperature. The efficiency increases as the reheat pressure (p_{rh}) is lowered and reaches a peak at a pressure ratio p_{rh}/p_1 between 0.20 and 0.25. Lowering the reheat pressure further causes the temperature differences between the primary fluid (flue gases) and the working fluid to increase, and brings down the efficiency again since the mean temperature of heat addition during reheat is less than that of the basic cycle. Too low a reheat pressure (for $p_{rh}/p_1 < 0.025$) results in a negative $\Delta\eta$. The *optimum reheat pressure for most of the modern power plants is 0.2 to 0.25 of the initial steam pressure*. It may also be noted in Fig. 2.16 that as the reheat pressure decreases, the quality of steam at turbine exhaust (x_4) increases. For too low a reheat pressure the exhaust steam may even be in the superheated state, which is not good for the condenser.

2.7 REGENERATION

In order to increase the mean temperature of heat addition T_{ml} and reduce the external thermal irreversibility, attention was so far confined to increasing the amount of heat supplied at high temperatures, such as increasing superheat, using higher pressure and temperature of steam, and using reheat. The mean temperature of heat addition can also be increased by reducing the amount of heat added at low temperatures (in the liquid phase) in the economiser section of the steam generator with the attention focused in the process of heat transfer between the flue gas and feedwater, *c–d* and 4–5 in Fig. 2.7, and *e–b* and 6*s*–7 in Fig. 2.15(b). This irreversibility could be entirely eliminated if the feedwater could be entered into the steam generator at saturated liquid state 5 (Fig. 2.7) rather than at 4. This is possible by the process of regeneration in which energy is exchanged internally between the expanding fluid in the turbine and the compressed fluid before heat addition.

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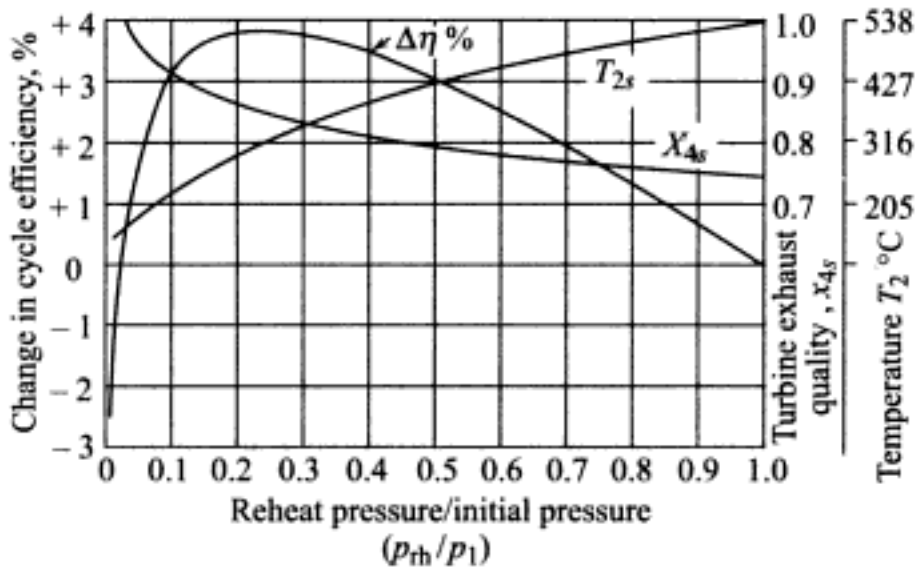

Fig.2.16 *Effect of reheat-to-initial pressure ratio on cycle efficiency, h.p. turbine exit temperature and l.p. turbine exhaust quality with initial steam at 172 bar, 538 °C and steam reheat to 538 °C*

2.7.1 Stirling Cycle

A well known gas cycle that uses regeneration is the Stirling cycle comprising two reversible isotherms and two reversible isochores. Heat addition at constant temperature T_1 from an external source and heat rejection at constant temperature T_2 to an external sink take place in the processes 4–1 and 2–3 respectively. Regeneration or heat exchange occurs reversibly between the constant volume processes 1–2 and 3–4 (Fig. 2.17). The areas under 1–2 and 3–4 denoting heat lost by the expanded fluid and gained by the compressed fluid are equal. Therefore, all the heat is added reversibly at T_1 and all the heat is rejected reversibly at T_2. So, the ideal Stirling cycle has the same efficiency as the Carnot cycle.

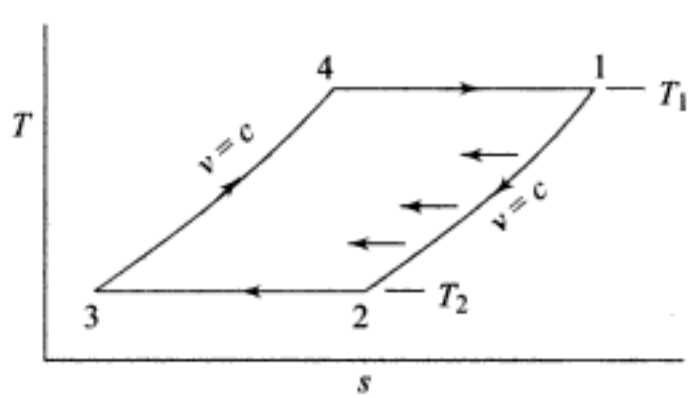

Fig. 2.17 *Regenerative Stirling cycle with reversible heat transfer between processes 1–2 and 3–4*

2.7.2 Ideal Regenerative Cycle

Following the Stirling cycle, in the ideal regenerative cycle (Fig. 2.18) the condensate after leaving the pump circulates around the turbine casing so that

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heat is transferred from the vapour expanding in the turbine to the condensate circulating around it. It is assumed that this heat transfer process is reversible, i.e. at each point the temperature of the vapour is only infinitesimally higher than the temperature of the liquid. The process 1–2 thus represents reversible expansion of steam in the turbine with reversible heat rejection to the surrounding liquid heated reversibly in the process 4*s*–5. For any small step,

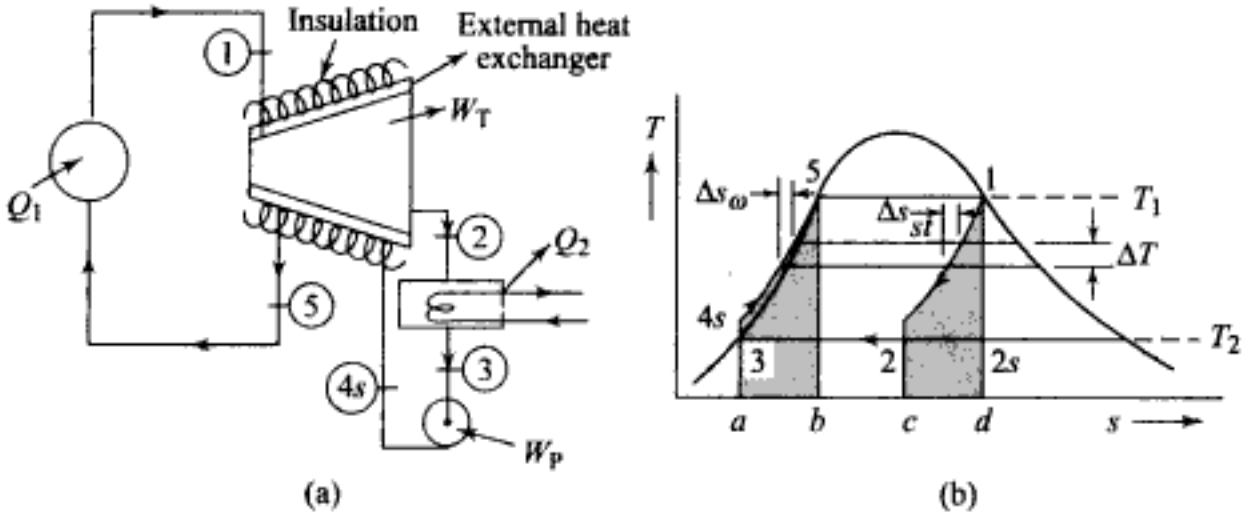

Fig. 2.18 *Ideal regenerative cycle*

$$\Delta T\,(\text{water}) = -\Delta T\,(\text{steam}) \qquad \text{and} \qquad \Delta s\,(\text{water}) = -\Delta s\,(\text{steam})$$

The slopes of the lines 1–2 and 5–4*s* (Fig. 2.18) are thus identical in contour. Areas 4*s*–5–*b*–*a*–4*s* and 1–*d*–*c*–2–1 are not only equal but also congruous. Therefore, all the heat added from an external source (Q_1) is at constant temperature T_1, and all the heat rejected (Q_2) is at constant temperature T_2, both being reversible. Then

$$Q_1 = h_1 - h_5 = T_1(s_1 - s_5)$$
$$Q_2 = h_2 - h_3 = T_2(s_2 - s_3)$$

Since for reversible heat transfer,

$$\Delta s_{\text{uni}} = \Delta s_{\text{water}} + \Delta s_{\text{steam}} = 0$$
$$\Delta s_{\text{water}} = -\Delta s_{\text{steam}} \qquad \text{or} \qquad s_5 - s_3 = s_1 - s_2$$
$$\therefore \qquad s_1 - s_5 = s_2 - s_3$$
$$\therefore \qquad \eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2}{T_1}$$

The efficiency of the ideal regenerative cycle like the Stirling cycle is thus equal to that of the Carnot cycle. Writing the steady flow energy equation for the turbine,

$$h_1 - W_T - h_2 + h_{4s} - h_5 = 0$$
$$\therefore \qquad W_T = (h_1 - h_2) - (h_5 - h_{4s}) \qquad (2.23)$$

The pump work remains the same as in the Rankine cycle, i.e.,

$$W_P = h_{4s} - h_3$$

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The net work output of the ideal regenerative cycle is thus less and hence, its steam rate will be more, although it is more efficient, compared to the Rankine cycle. However, the cycle is not practicable because:

(a) reversible heat transfer cannot be realized in finite time,
(b) heat exchanger in the turbine is mechanically impracticable, and
(c) the moisture content of the steam in the turbine is high, which leads to excessive erosion of turbine blades.

2.8 REGENERATIVE FEEDWATER HEATING

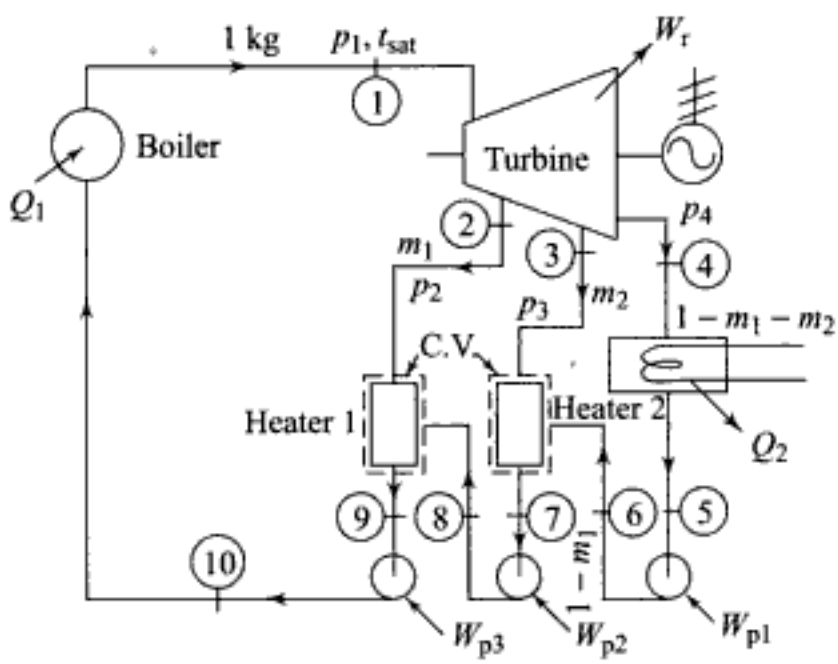
In the practical regenerative cycle, the feedwater enters the boiler at a temperature between states 5 and 4 *s* (Fig. 2.18), and it is heated by steam extracted or bled from intermediate stages of the turbine. The flow, *T–s* and *h–s* diagrams with saturated steam at turbine inlet and two regenerative direct-contact feedwater heaters are shown in Fig. 2.19. For 1 kg of steam at turbine inlet, m_1 kg of steam is extracted at pressure p_2 to mix adiabatically in the heater 1 with the $(1 - m_1)$ kg of feedwater as shown. The remaining $(1 - m_1)$ kg steam expands reversibly to pressure p_3 when m_2 kg steam is extracted to mix with $(1-m_1-m_2)$ kg feedwater in the heater 2. The remaining steam, $(1 - m_1 - m_2)$ kg, expands reversibly to the condenser pressure p_4. The heat and work transfer quantities of the cycle are

$$W_T = 1(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$

$$W_P = (1 - m_1 - m_2)(h_6 - h_5) + (1 - m_1)(h_8 - h_7) + 1(h_{10} - h_9)$$

$$Q_1 = 1(h_1 - h_{10}); \qquad Q_2 = (1 - m_1 - m_2)(h_4 - h_5)$$

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_P}{Q_1} \tag{2.24}$$



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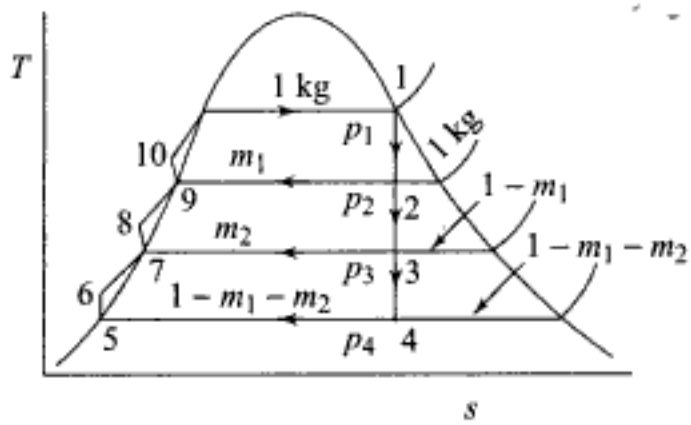

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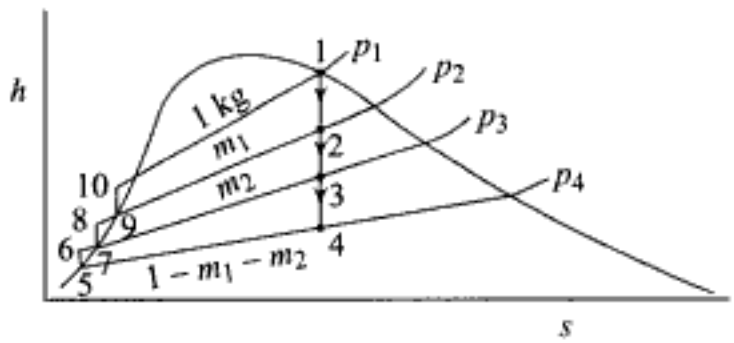

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Fig. 2.19 *Regenerative cycle with two direct-contact feedwater heaters*

In the Rankine cycle operating at the given boiler and condenser pressures p_1 and p_4, the heat addition would have been from state 6 to state 1. By using two stages of regenerative heating, feedwater enters at state 10 instead of state 6, and heat addition from an external heat source is now from state 10 to state 1. Therefore,

$$\left.\begin{aligned} (T_{m1}) \text{ with regeneration} &= \frac{h_1 - h_{10}}{s_1 - s_{10}} \\ \text{and} \qquad (T_{m1}) \text{ without regeneration} &= \frac{h_1 - h_6}{s_1 - s_6} \end{aligned}\right\} \tag{2.25}$$

Since (T_{m1}) with regeneration > (T_{m1}) without regeneration, the efficiency of the regenerative cycle is higher than that of the Rankine cycle.

The energy balance for heater 1 gives

$$m_1 h_2 + (1 - m_1)h_8 = 1\, h_9$$

$$\therefore \qquad m_1 = \frac{h_9 - h_8}{h_2 - h_8} \tag{2.26}$$

The energy balance for heater 2 gives

$$m_2 h_3 + (1 - m_1 - m_2)h_6 = (1 - m_1)h_7$$

$$\therefore \qquad m_2 = (1 - m_1)\frac{h_7 - h_6}{h_3 - h_6} \tag{2.27}$$

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The path 1–2–3–4 in Fig. 2.19 (b) represents the states of a decreasing mass of fluid. For 1 kg of steam, the path would be represented by 1–2–2′–3′–3″– 4′ as shown in Fig. 2.20. From Eq. (2.24),

$$W_T = h_1 - h_2 + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$
$$= (h_1 - h_2) + (h_{2'} - h_{3'}) + (h_{3''} - h_{4'}) \quad (2.28)$$

where $(1 - m_1)(h_2 - h_3) = 1(h_{2'} - h_{3'})$ and $(1 - m_1 - m_2)(h_3 - h_4) = 1(h_{3''} - h_{4'})$

The heat released by steam condensing from 2 to 2′, is utilized in heating up the water from 8 to 9.

$$1(h_2 - h_{2'}) = 1(h_9 - h_8)$$

Similarly, $$1(h_{3'} - h_{3''}) = 1(h_7 - h_6) \quad (2.29)$$

From Eqs (2.27) to (2.29)

$$W_T = (h_1 - h_{4'}) - (h_2 - h_{2'}) - (h_{3'} - h_{3''})$$
$$= (h_1 - h_{4'}) - (h_9 - h_8) - (h_7 - h_6) \quad (2.30)$$

The similarity of Eq. (2.23) and (2.30) can be noticed. It is seen that the stepped cycle 1–2–2′–3′–3″–4′–5–6–7–8–9–10 approximates the ideal regenerative cycle in Fig. 2.18, and a greater number of stages would give a closer approximation.

The heat rejected Q_2 in the cycle decreases from $(h_4 - h_5)$ to $(h_{4'} - h_5)$. There is also a loss in work output by the amount (Area under 2–2′ + area under 3′ –3″ – area under 4 – 4′), as shown by the hatched area in Fig. 2.20. So, the steam rate increases by regeneration, i.e. more steam has to circulate per hour to produce unit shaft output. It increases the boiler size and hence the capital cost. It reduces the operating cost due to higher cycle efficiency.

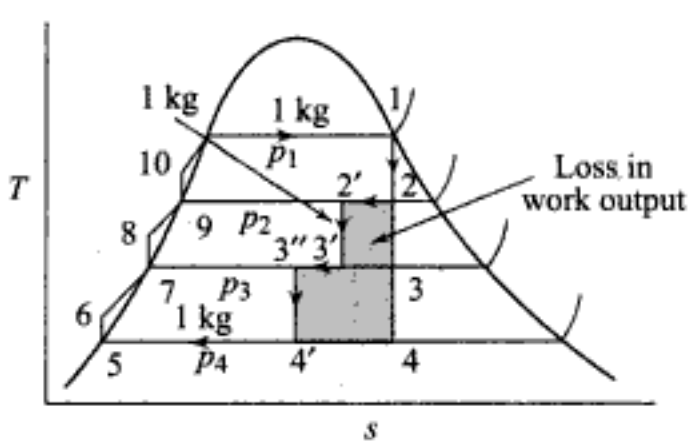

Fig. 2.20 *Regenerative cycle for unit mass of fluid*

From Eqs (2.26) and (2.27), m_1 and m_2 can be evaluated. Eqs (2.26) and (2.27) can also be written alternatively as

Energy given off by extracted steam in condensation = energy gain of feedwater

or $$m_1(h_2 - h_9) = (1 - m_1)(h_9 - h_8)$$

$\therefore$ $$m_2(h_3 - h_7) = (1 - m_1 - m_2)(h_7 - h_6)$$

Heaters have been assumed to be adequately insulated so that there is no heat loss to the surroundings.

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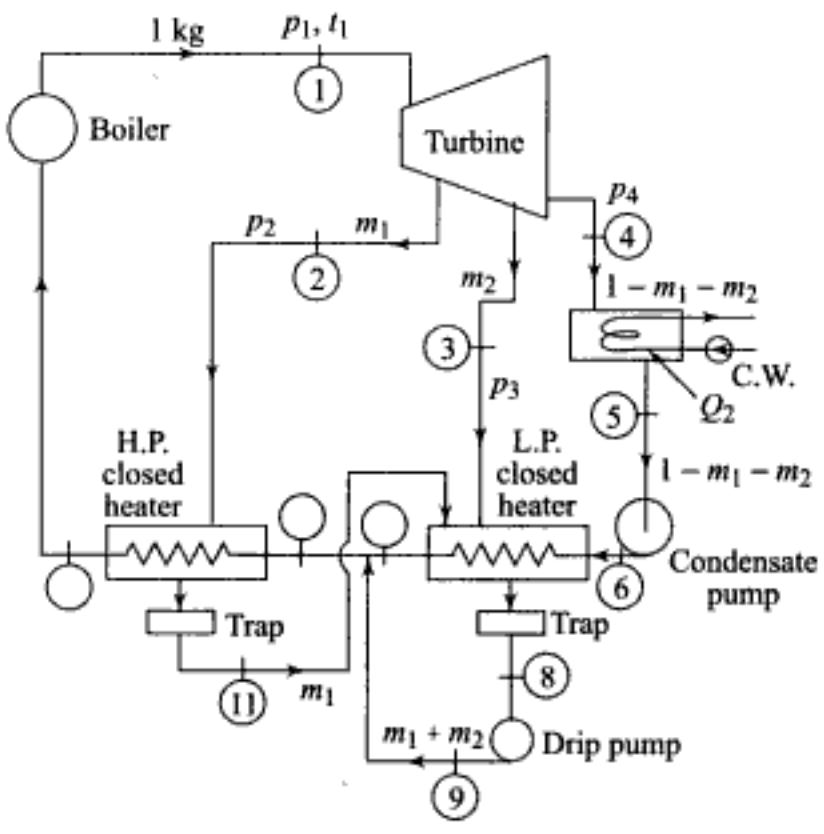
The effects of regenerative feedwater heating for the same turbine output may be summarized as follows:

1. It significantly increases the cycle efficiency and reduces the heat rate (reducing operating cost).
2. It increases the steam flow rate (requiring bigger boiler).
3. It reduces the steam flow to the condenser (needing smaller condenser).
4. If there is no change of boiler output, the turbine output drops.

2.9 FEEDWATER HEATERS

Feedwater heaters are of two types, viz., open heaters and closed heaters. In an open or contact type heater, the extracted steam is allowed to mix with feedwater and both leave the heater at a common temperature (Fig. 2.19a). In a closed heater, the fluids are kept separate and are not allowed to mix together (Fig. 2.21). Closed heaters are shell-and-tube heat exchangers where the feedwater flows through the tubes and the extracted steam condenses outside the tubes in the shell. The heat released by condensation is transferred through the walls of the tubes. The condensate (saturated water at the steam extraction pressure), sometimes called the *heater drip*, then passes through a trap into the next lower pressure heater. This, to some extent, reduces the steam required by that heater. The trap passes only liquid and no vapour. The drip from the lowest pressure heater could similarly be trapped to the condenser, but this would be throwing away energy to the condenser cooling water. To avoid this waste, a drip pump feeds the drip directly into the feedwater stream.

If the Fig. 2.19 and 2.21 are compared, it may be noted that the feedwater inlet temperature, Q_1, Q_2, W_T and so on would be marginally affected with the heaters being either open or closed. For the heaters, the energy balance gives



(a)

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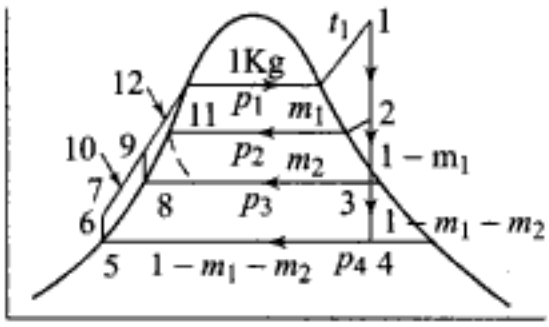


Fig. 2.21 *Regenerative feedwater heating with two closed heaters*

$$m_1(h_2 - h_{11}) = 1(h_{12} - h_{10}) = 1c_{pw}(t_{12} - t_{10})$$

$$m_2(h_3 - h_8) + m_1(h_{11} - h_8) = (1 - m_1 - m_2)(h_7 - h_6)$$

$$= (1 - m_1 - m_2)\, c_{pw}(t_7 - t_6) \tag{2.31}$$

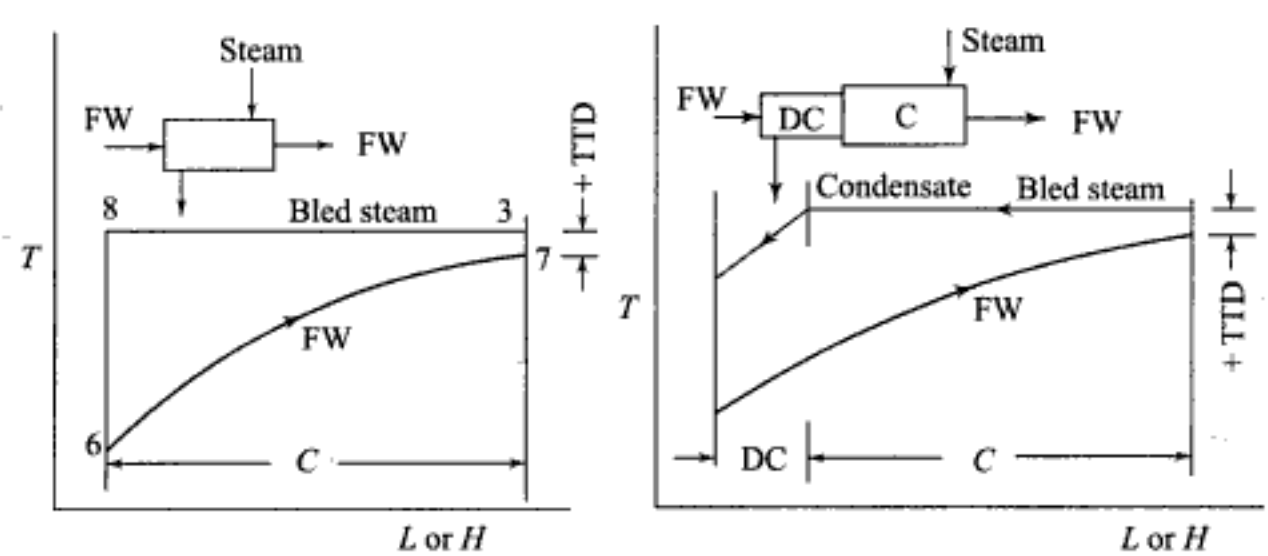
where c_{pw} is the specific heat of water.

Considering the $l - p$ heater of Fig. 2.21, wet steam at state 3 is used to heat up the h.p. subcooled feedwater at state 6. The temperature–length diagram is shown in Fig. 2.22(a). The water exit temperature at 7 cannot reach the inlet bled steam temperature at 3. *A terminal temperature difference* (TTD) is defined for all closed feedwater heaters as

$$\text{TTD} = \text{saturation temperature of bled steam} - \text{exit water temperature} \tag{2.32}$$

The value of TTD varies with heater pressure. For $l - p$ heaters receiving wet steam, the TTD is positive and often of the order of 3 °C. Too small a value, although good for plant efficiency, would require a larger heater. Too large a value would reduce the cycle efficiency.

If the extracted steam upon condensation gets subcooled, a *drain cooler* may be used. The heater would then have two sections, a condensing section and a draincooler section [Fig. 2.22(b)].



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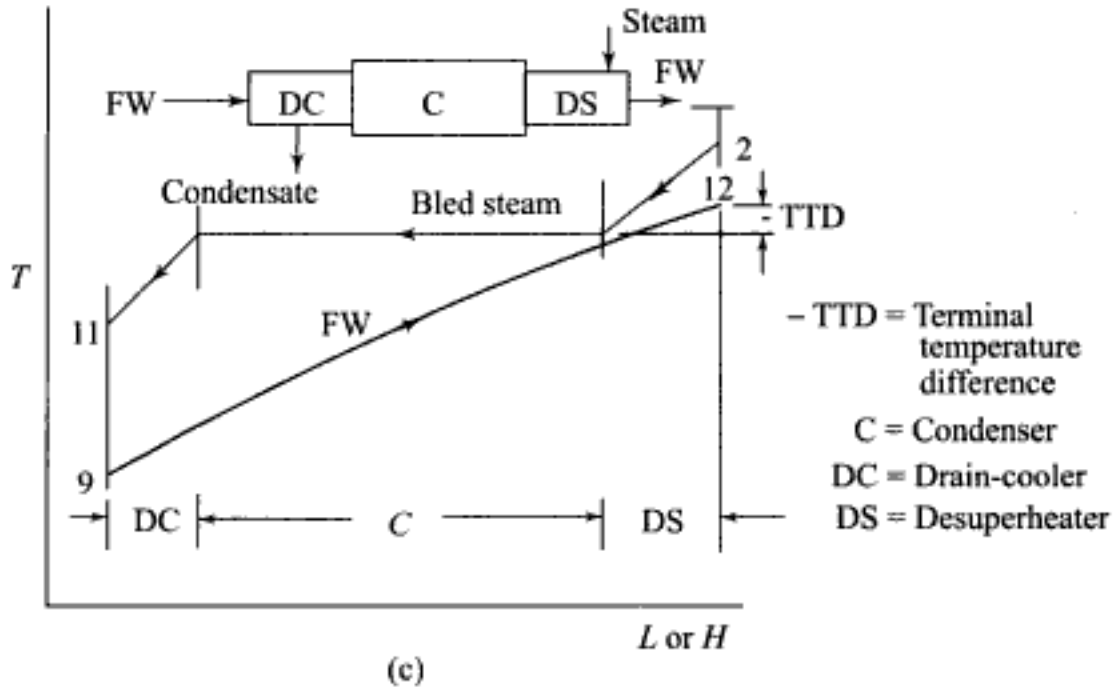

Fig. 2.22 *Temperature-length diagram for l.p. and h.p. feedwater heaters*

For the h.p. heater receiving superheated steam [Fig. 2.22(c)] bled from the turbine at state 2, the steam is first desuperheated, then condensed and finally subcooled to state 11, whereas the feedwater gets heated from 9 to 12. It may be noted that the exit water temperature (t_{12}) is higher than the saturation temperature at p_2, and the TTD is here negative. The heater is then composed of a desuperheating section, a condensing section and a drain cooler section.

The advantages of the open heater are simplicity, lower cost, and high heat transfer capacity. The disadvantage is the necessity of a pump at each heater to handle the large feedwater stream.

A closed heater requires only a single pump for the main feedwater stream regardless of the number of heaters. The drip pump, if used, is relatively small. Closed heaters are costly and may not give as high a feedwater temperature as do open heaters. In most steam power plants, closed heaters are favoured but at least one open heater is used, primarily for the purpose of feedwater *deaeration*. The open heater in such a system is called the deaerator. Closed heaters are mostly horizontal. Sometimes, they are made vertical to reduce the floor areas needed for their installations.

2.10 CARNOTIZATION OF RANKINE CYCLE

Figure 2.23 shows the flow diagram of a condensing steam power plant worked with saturated steam at the turbine inlet and four steam extractions. In the *T–S* diagram, abscissa represents the total entropy of the turbine steam. Without

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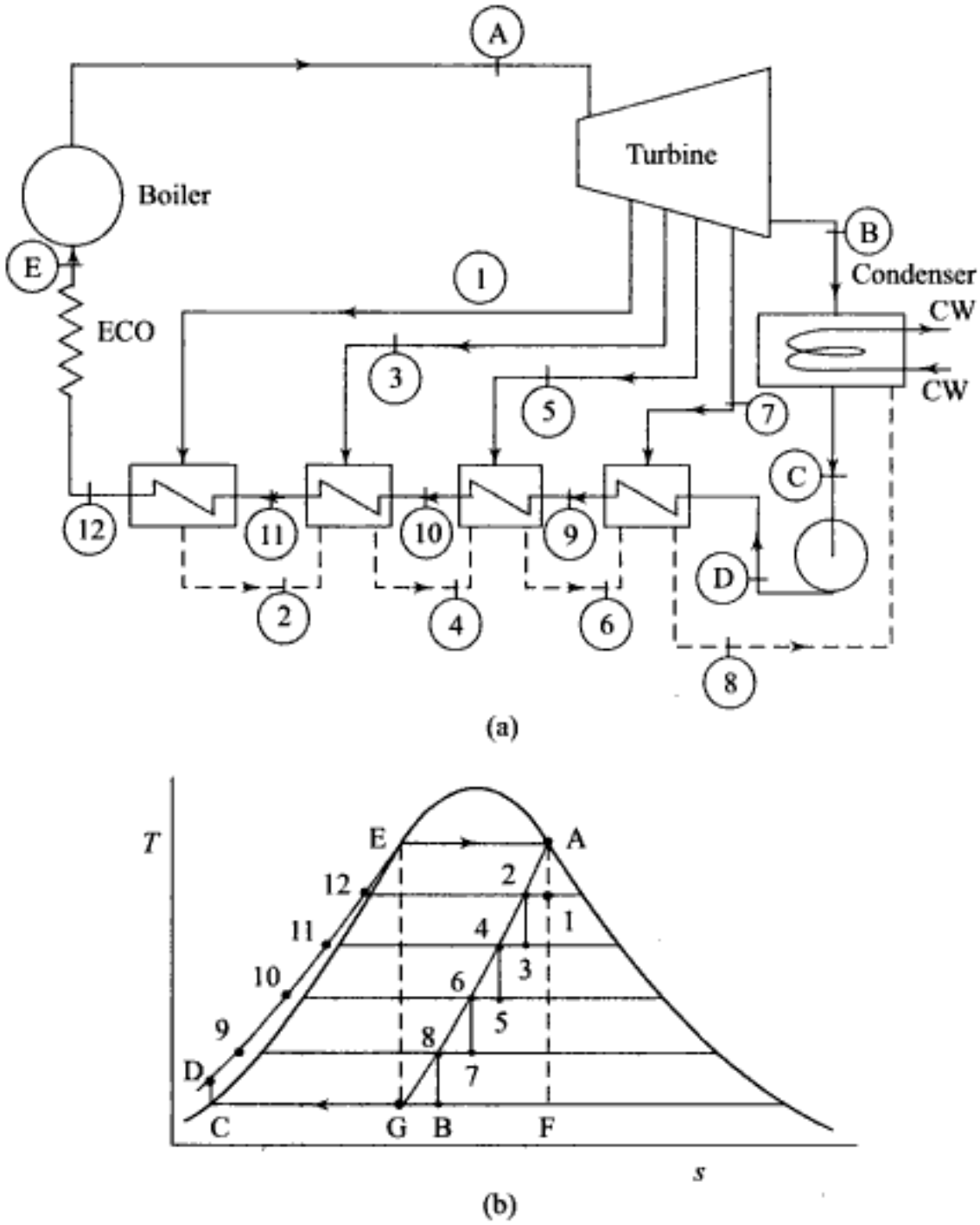

Fig. 2.23 *Regenerative feed cycle with four extractions*

extraction the ideal cycle is *AFCDEA*. After the throttle steam has expanded from *A* to 1, some steam is extracted so that the total entropy is reduced from 1 to 2. The heat given up by this steam is added to the feedwater, heating it from 11 to 12. This process is continued in four steps to the last extraction which heats the feedwater from *D* to 9. With an infinite number of extraction stages, *AB* would be parallel and equal in length to *DE*. The irreversible process *DE*—the heating from condenser to boiler saturation temperature–could thus be made reversible. The area of the parallelogram *CEAG*, which represents the cycle output, would be equal to the area of the rectangle *AFGE* which represents the output of the Carnot cycle. Regenerative feedwater heating by turbine extraction is, therefore, also termed the *carnotization of the Rankine cycle*. A regenerative feed heating cycle with an infinite number of feedwater heaters has thus an efficiency equal to that of Carnot cycle.

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2.11 OPTIMUM DEGREE OF REGENERATION

Complete carnotization of Rankine cycle is not possible with a finite number of heaters. If there is one feedwater heater used, m kg of steam is extracted from the turbine for each kg of steam entering it to heat the feedwater from state 5 to state 6 (Fig. 2.24) so that by energy balance,

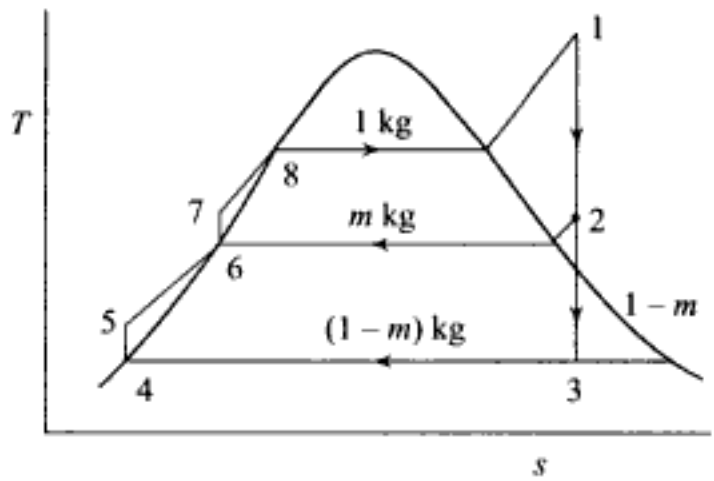

Fig. 2.24 *Optimization of regenerative feedwater heating*

$$m(h_2 - h_6) = (1 - m)(h_6 - h_5) \quad \text{or} \quad m = \frac{h_6 - h_5}{h_2 - h_5} = \frac{h_6 - h_4}{h_2 - h_4}$$

Therefore, the thermal efficiency of the cycle is

$$h = 1 - \frac{(1-m)(h_3 - h_4)}{h_1 - h_6} = 1 - \frac{\left(1 - \dfrac{h_6 - h_4}{h_2 - h_4}\right)(h_3 - h_4)}{h_1 - h_6}$$

$$= 1 - \frac{(h_2 - h_6)(h_3 - h_4)}{(h_2 - h_4)(h_1 - h_6)} \tag{2.33}$$

Following Salisbury [4], Horlock [1] and Haywood [5], it may be approximately assumed that the turbine expansion line follows a path on the diagram such that $(h - h_f)$ = constant = β, where h is the local enthalpy on the expansion line at a given pressure, and h_f is the enthalpy of saturated water at that pressure. Therefore, as seen in Fig. 2.24, $h_1 - h_8 = h_2 - h_6 = h_3 - h_4 = \beta =$ constant. Let the enthalpy rise of feedwater in the heater is γ, which is equal to $(h_6 - h_4)$.

Now, $$h_2 - h_4 = h_2 - h_6 + h_6 - h_4 = \beta + \gamma$$

If the total enthalpy rise of feedwater is equal to $\alpha = h_8 - h_4$, then

$$h_1 - h_6 = h_1 - h_8 + h_8 - h_4 + h_4 - h_6 = \beta + \alpha - \gamma$$

Therefore, Eq. (2.33) can be written in the form

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$$\eta = 1 - \frac{\beta^2}{(\beta+\gamma)(\alpha+\beta-\gamma)} \tag{2.34}$$

Here, α and β are fixed and γ is variable. So, there is an optimum value of γ for which η is a maximum. On differentiation,

$$\frac{d\eta}{d\gamma} = \beta^2[(\alpha+\beta-\gamma)-(\beta+\gamma)] = 0 \quad \therefore \quad \gamma = \frac{\alpha}{2} \tag{2.35}$$

The cycle efficiency is maximum when the total enthalpy rise of feedwater ($h_8 - h_4$) from the condenser temperature to the boiler saturation temperature is divided equally between the feedwater heater and the economiser (i.e. $h_8 - h_6 = h_6 - h_4$) in a single bleed cycle. So, the temperature rise of feedwater in the heater is

$$\Delta t = \frac{1}{2}(t_{\text{boiler saturation}} - t_{\text{condenser}})$$

and the corresponding cycle efficiency is

$$\eta = 1 - \frac{\beta^2}{\left(\beta+\frac{\alpha}{2}\right)\left(\alpha+\beta-\frac{\alpha}{2}\right)} = 1 - \frac{\beta^2}{\left(\beta+\frac{\alpha}{2}\right)^2} = \frac{\alpha^2+4\alpha\beta}{(\alpha+2\beta)^2} \tag{2.36}$$

For a non-regenerative cycle,

$$\eta_o = 1 - \frac{h_3 - h_4}{h_1 - h_4}$$

Now, $\quad h_3 - h_4 = \beta$ and $h_1 - h_4 = h_1 - h_8 + h_8 - h_4 = \beta + \alpha$

$$\therefore \quad \eta_0 = 1 - \frac{\beta}{\alpha+\beta} = \frac{\alpha}{\alpha+\beta} \tag{2.37}$$

The efficiency gain due to regeneration

$$\Delta\eta = \eta - \eta_0 = \frac{\alpha^2+4\alpha\beta}{(\alpha+2\beta)^2} - \frac{\alpha}{\alpha+\beta} = \frac{\alpha^2\beta}{(\alpha+\beta)(\alpha+2\beta)^2} \tag{2.38}$$

This is positive. This shows that the cycle efficiency has improved due to regeneration.

In the heater train, the feedwater enters the economiser section of the boiler at state F (Fig. 2.25), where feedwater is heated to the saturation temperature (G) at the boiler pressure. Assuming the economiser also as a feedwater heater (where feedwater is heated by the outgoing flue gases, instead of by the bled turbine steam, the total enthalpy rise ($h_G - h_3$) or temperature rise from the condenser to the boiler saturation is divided equally among the feedwater heaters for maximum gain in the efficiency. The enthalpy rise per heater (including the economiser) is thus,

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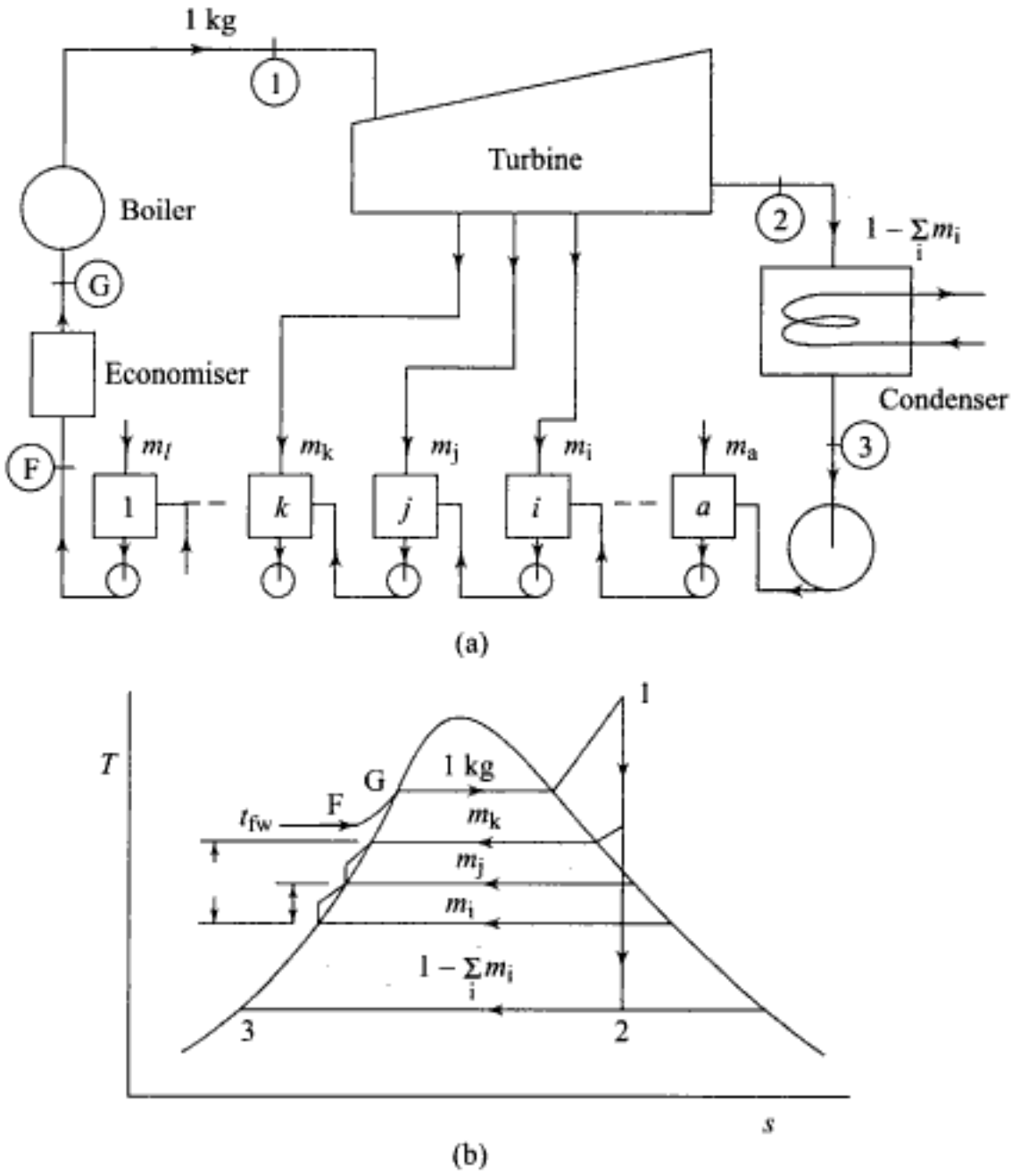

Fig. 2.25 *Heater train of a steam power plant* $\left(\gamma = \frac{\alpha}{2}\right)$

$$\Delta h_{\text{per heater}} = \frac{h_G - h_3}{n+1} \tag{2.39}$$

where n is the number of heaters and 1 stands for the economiser. Therefore, the total enthalpy rise of feedwater for n heaters by regenerative feedheating is

$$\Delta h_{\text{total}} = \frac{n}{n+1}(h_G - h_3) \tag{2.40}$$

Thus, the total temperature rise of feedwater, Δt_{fw}, due to regeneration for the maximum cycle efficiency is given by

$$\Delta t_{fw} = \frac{n}{n+1}\Delta t_{OA} \tag{2.41}$$

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where the overall temperature difference,

$$\Delta t_{OA} = \text{boiler saturation temperature} - \text{condenser temperature}$$

More is the number of heaters, more is the total temperature rise of feedwater, Δt_{fw}, by regeneration, less becomes the heat addition to water in the boiler, more becomes the mean temperature of heat addition, and more is the cycle efficiency. From Eq. (2.41),

If $n = 0$, $\Delta t_{fw0} = 0$

If $n = 1$, $\Delta t_{fw1} = \frac{1}{2}\Delta t_{OA}$

If $n = 2$, $\Delta t_{fw2} = \frac{2}{3}\Delta t_{OA}$

If $n = 3$, $\Delta t_{fw3} = \frac{3}{4}\Delta t_{OA}$ and so on.

By the use of the first heater, the gain is

$$\Delta t_{fw1} - \Delta t_{fw0} = \frac{1}{2}\Delta t_{OA}$$

By the use of the second heater, the gain over the first heater would be

$$\Delta t_{fw2} - \Delta t_{fw1} = \frac{2}{3}\Delta t_{OA} - \frac{1}{2}\Delta t_{OA} = \frac{1}{6}\Delta t_{OA}$$

By the use of the third heater, the gain over the second heater would be

$$\Delta t_{fw3} - \Delta t_{fw2} = \frac{3}{4}\Delta t_{OA} - \frac{2}{3}\Delta t_{OA} = \frac{1}{12}\Delta t_{OA}$$

Employing the fourth heater, the gain over the third heater becomes

$$\Delta t_{fw4} - \Delta t_{fw3} = \frac{4}{5}\Delta t_{OA} - \frac{3}{4}\Delta t_{OA} = \frac{1}{20}\Delta t_{OA} \quad \text{and so on.}$$

Since the gain in cycle efficiency is proportional to the gain in feedwater temperature, the efficiency gain follows the *law of diminishing return* with the increase in the number of heaters. In fact, the greatest increment in efficiency is brought by the first heater. The increments for each additional heater thereafter successively diminish (Fig. 2.26). The number of heaters is fixed up by the energy balance of the whole plant when it is found that the cost of adding another heater does not justify the saving in heat supply Q_1 or the marginal increase in cycle efficiency. An increase in feedwater temperature t_{fw} reduces the heat absorption from the outgoing flue gases in the economiser and may cause a reduction in boiler efficiency. The number of heaters and hence, the degree of regeneration thus get optimized. Five to seven points of extraction are often used in practice. Some cycles use as many as nine.

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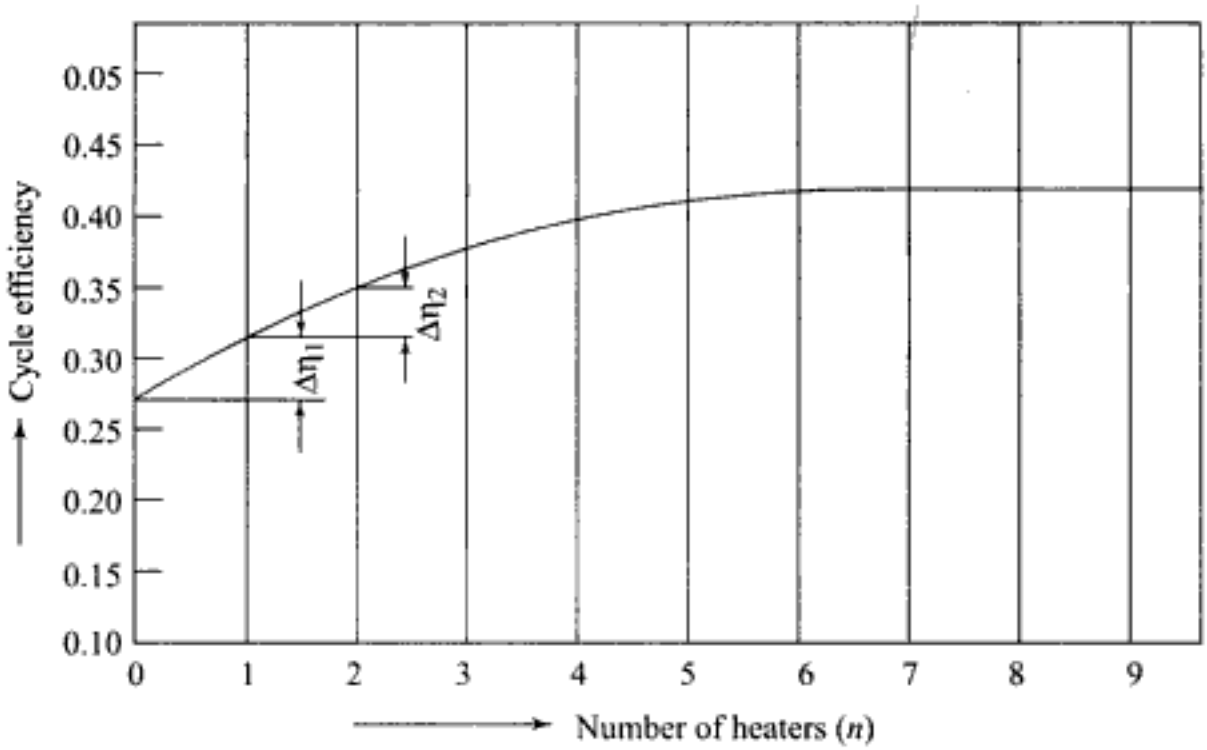

Fig. 2.26 *Efficiency gain (Δη) successively diminishes with the increase in the number of heaters*

Figure 2.27 shows the optimum locations of the heaters for a non-reheat steam power plant.

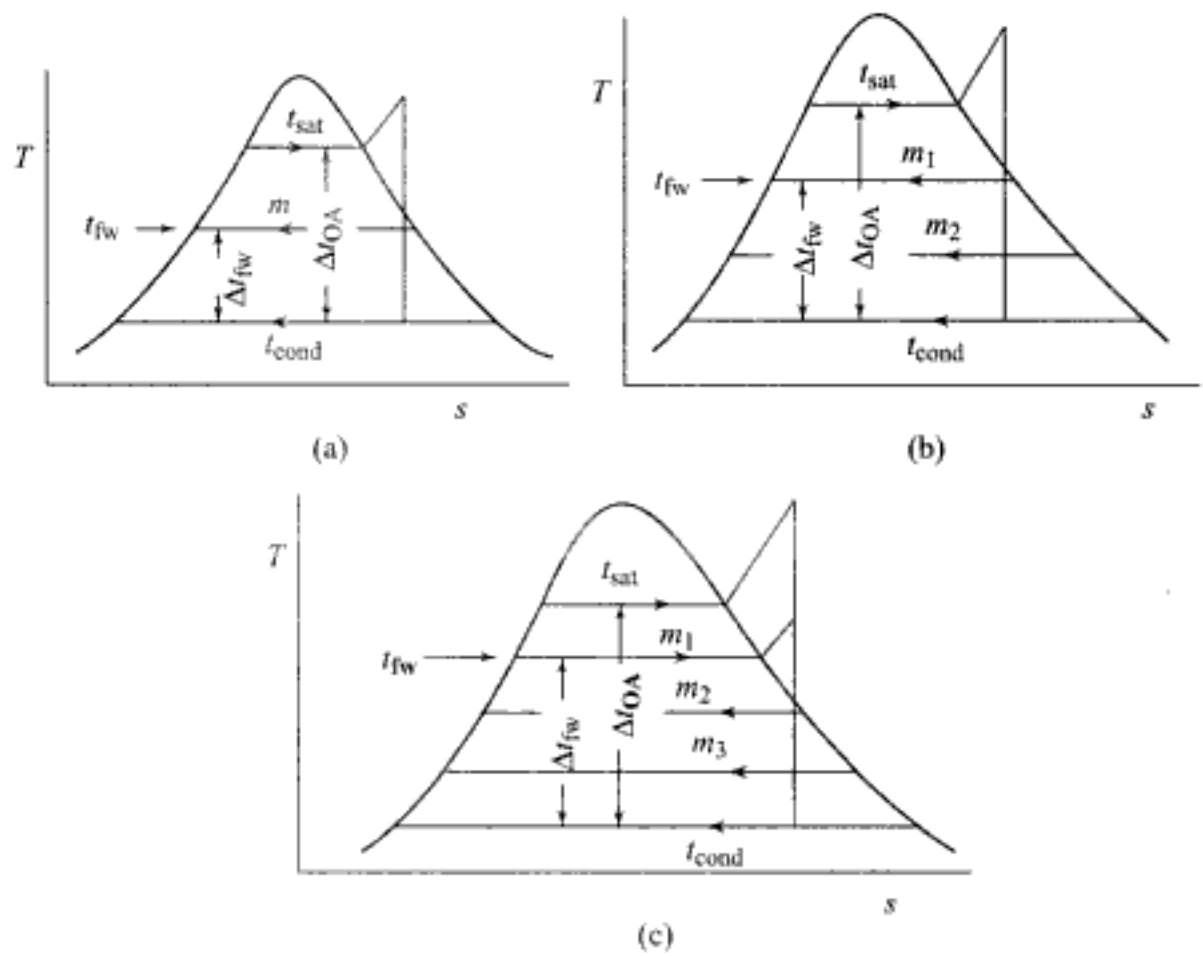

Fig. 2.27 *Optimum locations of heaters (a) For one heater, $\Delta t_{fw} = \frac{1}{2}\ \Delta t_{OA}$; (b) For two heaters, $\Delta t_{fw} = \frac{2}{3}\ \Delta t_{OA}$; and (c) For three heaters, $\Delta t_{fw} = \frac{3}{4}\ \Delta t_{OA}$*

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Figure 2.28 shows the reduction in heat rate by regenerative feedheating with different number of extractions for the same throttle steam conditions. With the use of four extractions the gain in heat rate is about 11%. The temperatures at which the heaters operate for maximum reduction in heat rate have been indicated in the figure and joined by a dotted line. The curve for a single feedwater heater peaks at a temperature rise of about halfway or 120 °C.

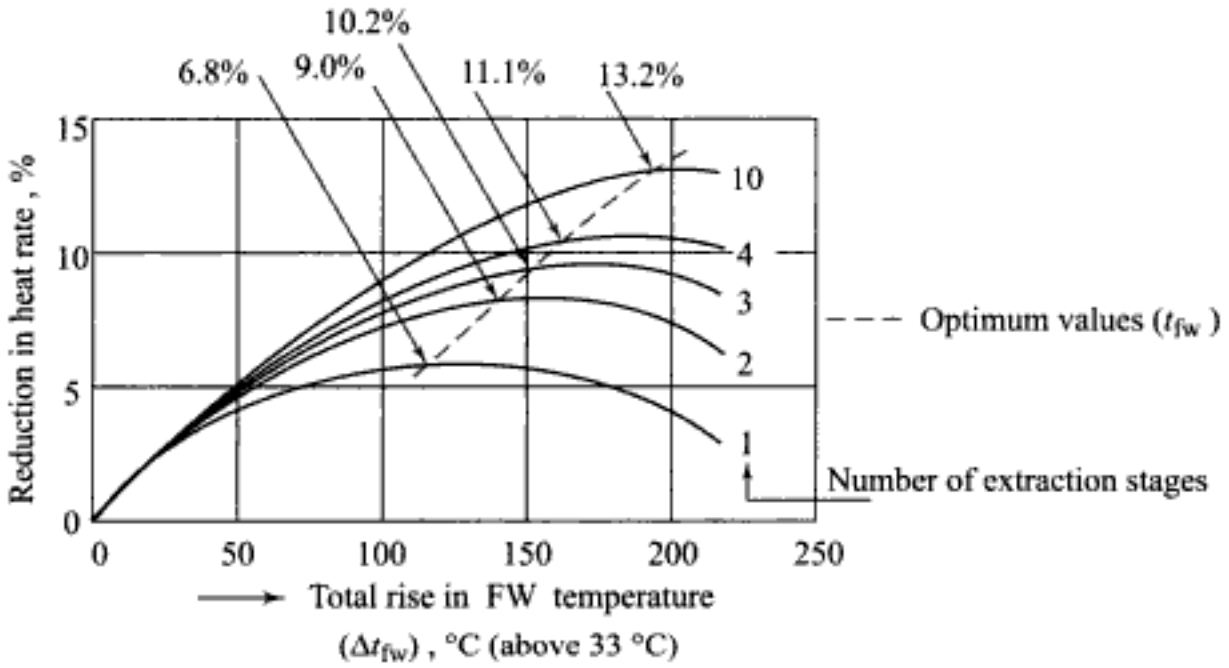

Fig. 2.28 *Reduction of heat rate by regenerative feedwater heating*

For two feedwater heaters, the peak occurs at $\frac{2}{3} \times 220$ or about 142 °C and so on. Small departures from the optimum values have no serious effect on the heat rate.

2.12 SUPERCRITICAL PRESSURE CYCLE

Figure 2.29 shows the plant arrangement and T–s diagram for a supercritical steam cycle. Steam is generated in a "once-through" boiler at a pressure above the critical point of 221.2 bar. If the plant incorporates reheat and several stages of feedheating, there is about a 2% gain in thermal efficiency compared with the corresponding subcritical cycle. However, such an increment is gained only at the expense of increased cost and complexity of the plant. Double reheat needs to be incorporated to prevent the l.p. turbine exhaust wetness from being excessive.

2.13 STEAM POWER PLANT APPRAISAL

Table 2.2 shows the past improvements in performance of steam power plants. These have been brought about by increasing the maximum pressure and temperature of the cycle, lowering the minimum pressure and introducing reheat and feedwater heating.

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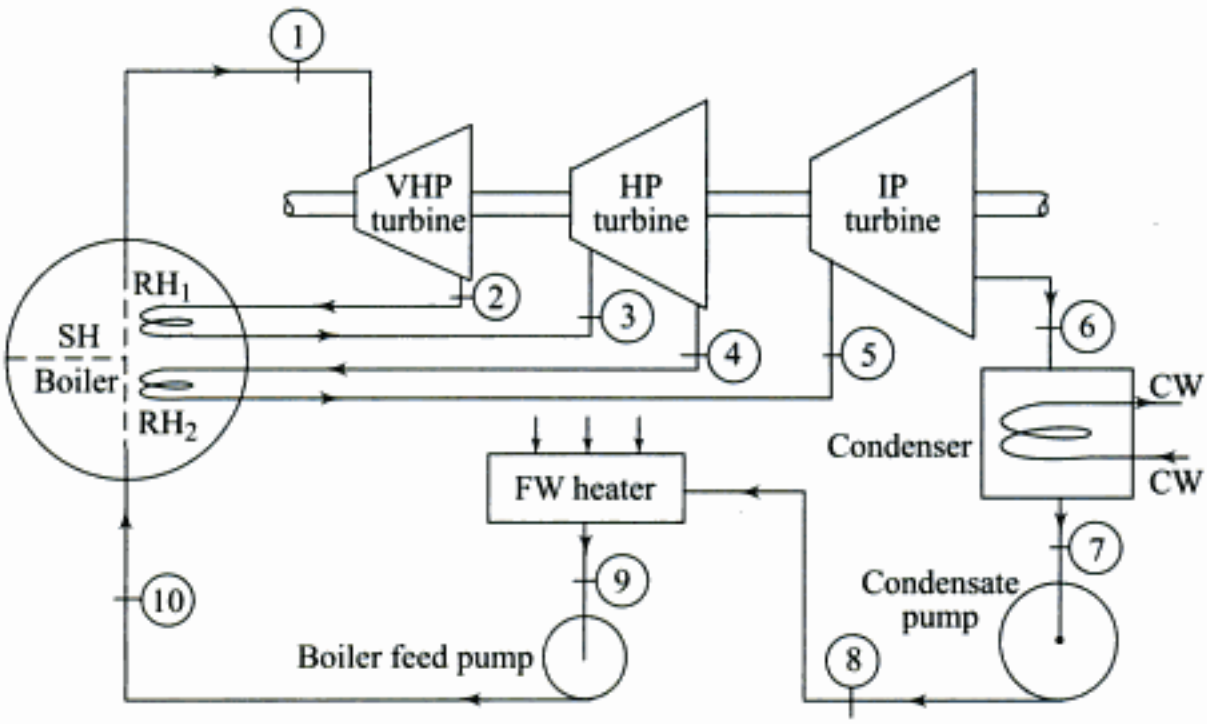
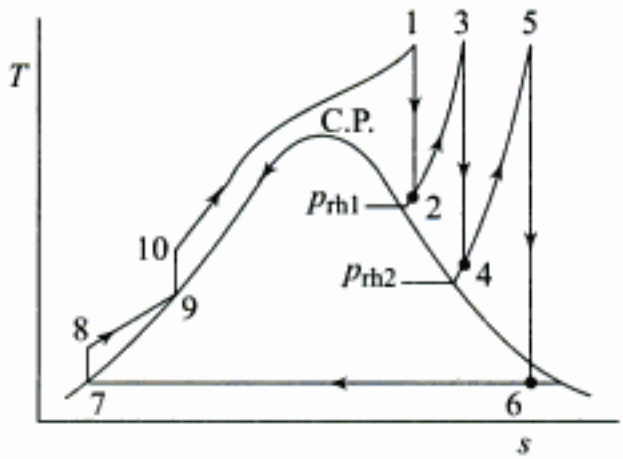
Table 2.2 *Approximate performance data of various steam power plants (Aschner)*

	Watt's beam engine	1 MW locomotive steam engine	2 MW marine steam engine	30 MW(e) steam power plant	660 MW(e) steam power plant
Thermal efficiency (%)	2	7	20	35	44
Initial pressure (bar)	2	15	15	40	160
Initial temperature (°C)	100	300	250	450	540
Reheat pressure (bar)	–	–	–	–	40
Reheat temperature (°C)	–	–	–	–	540
Exhaust pressure (bar)	near 1	1.2	0.1	0.045	0.045
Exhaust condition	wet	near sat	wet	0.9, dry	0.9, dry
Feedwater heaters	–	–	–	4	7



(a)



(b)

Fig. 2.29 *Supercritical steam cycle with double reheat*

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Units above 150 MW output are now most frequently built for h.p. steam conditions of 142 bar, 538 °C or 170 bar, 538 °C with reheating up to 538 °C. At supercritical pressures for more than 600 MW size, h.p. steam is often at 270 bar, 538 °C with double reheating to 538 °C. H.P. steam and reheat temperatures up to 565 °C have also been used. A landmark station, built in 1959, is the 325 MW Eddystone, Unit I of Philadelphia Electric Company, a double reheat plant with supercritical steam (325 bar, 610 °C/565 °C/565 °C) which had the highest steam conditions and lowest heat rate for any plant in the world.

Common unit size of a steam power plant is now 500 MW(e). Plant sizes, however, will continue to increase and 1300 MW(e) turbo generators are likely to be the standard unit sizes in the power station in near future. Large improvements through raising the metallurgical limit are unlikely in the near future because it would then have to use prohibitively expensive austenitic steels or complicated turbine blade cooling. The thermodynamic design of the steam cycle for a 1300 MW(e) turbo generator would have steam conditions similar to those of the present generation of 660 MW(e) sets, but the plant sizes would be correspondingly larger. Even larger unit sizes of 2000 and 3000 MW(e) have been proposed but the physical construction and operation of such plants would pose serious problems like the flexibility of the longer rotor shafts, construction of the huge generator stator, effect on the electricity grid system due to shutdown of one of these very large sets, and high loss of revenue due to a shutdown.

2.14 DEAERATOR

One of the feedwater heaters is a contact-type open heater, known as deaerator, others being closed heaters. It is used for the purpose of deaerating the feedwater.

The presence of dissolved gases like oxygen and carbon dioxide in water makes the water corrosive, as they react with the metal to form iron oxide. The solubility of these gases in water decreases with increase in temperature and becomes zero at the boiling or saturation temperature. These gases are removed in the deaerator, where feedwater is heated to the saturation temperature by the steam extracted from the turbine. Feedwater, after passing through a heat exchanger, called vent condenser, is sprayed from the top so as to expose large surface area, and the bled steam from the turbine is fed from the bottom (Fig. 2.30). By contact the steam condenses and the feedwater is heated to the saturation temperature. Dissolved oxygen and carbon dioxide gases get released from the water and leave along with some vapour, which is condensed back in the vent condenser, and the gases are vented out.

To neutralize the effect of residual dissolved oxygen and carbon dioxide gases in water, sodium sulphite (Na_2SO_3) or hydrazine (N_2H_4) is injected in suitable calculated doses into the feedwater at the suction of the boiler feed pump (BFP).

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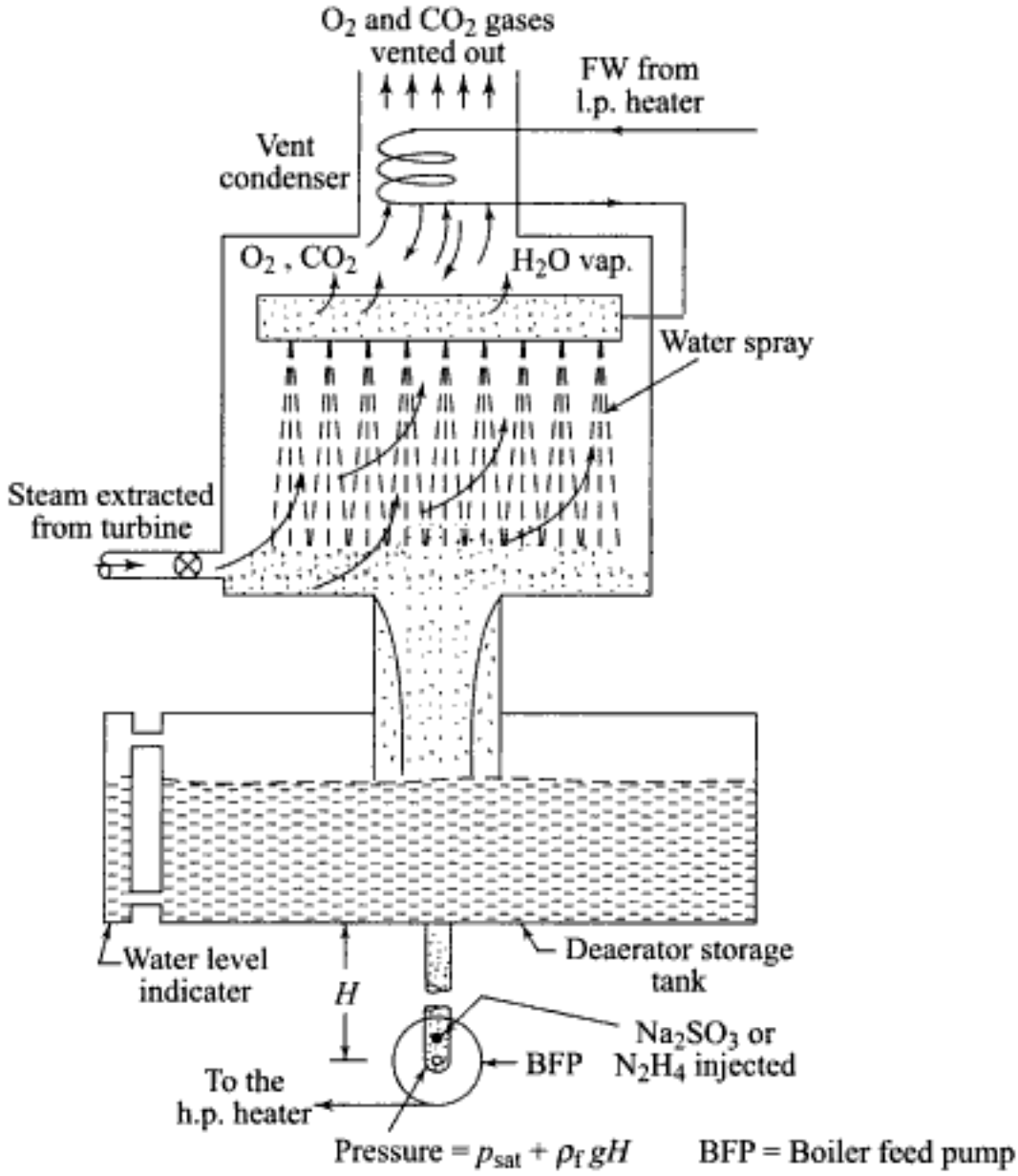

Fig. 2.30 *Deaerator with storage tank*

During suction of the BFP, some of the saturated feedwater may flash into vapour due to reduction in pressure causing vapour lock and cavitation problems in the pump. To prevent this from occurring and to provide a net positive suction head (NPSH) for the pump, the deaerator is located at a sufficient height (H) from the basement where the pump is installed so that the pressure before suction is ($P_{sat} + \rho_f gH$). When this water is sucked by the pump, the pressure does not fall below P_{sat} and there is no flashing of any water into vapour, which protects the BFP from any damage due to vapour lock and cavitation.

The deaerator is usually placed near the middle of the feedwater system so that the total pressure difference between the condenser and the boiler is shared equitably between the condensate pump and boiler feed pump. The feedwater heaters before the deaerator are often termed as high pressure (h.p.) heaters and those after the deaerator are termed as low pressure (l.p.) heaters.

The deaerator is not used in water-cooled-and-moderated nuclear power plants because of the concern regarding radioactivity release with deaeration.

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2.15 TYPICAL LAYOUT OF STEAM POWER PLANT

Figure 2.31(a) shows the typical layout of a steam power plant demonstrating all the relevant components in it. The heat balance diagram of a 215 MW reheat power plant is shown in Fig. 2.31(b).

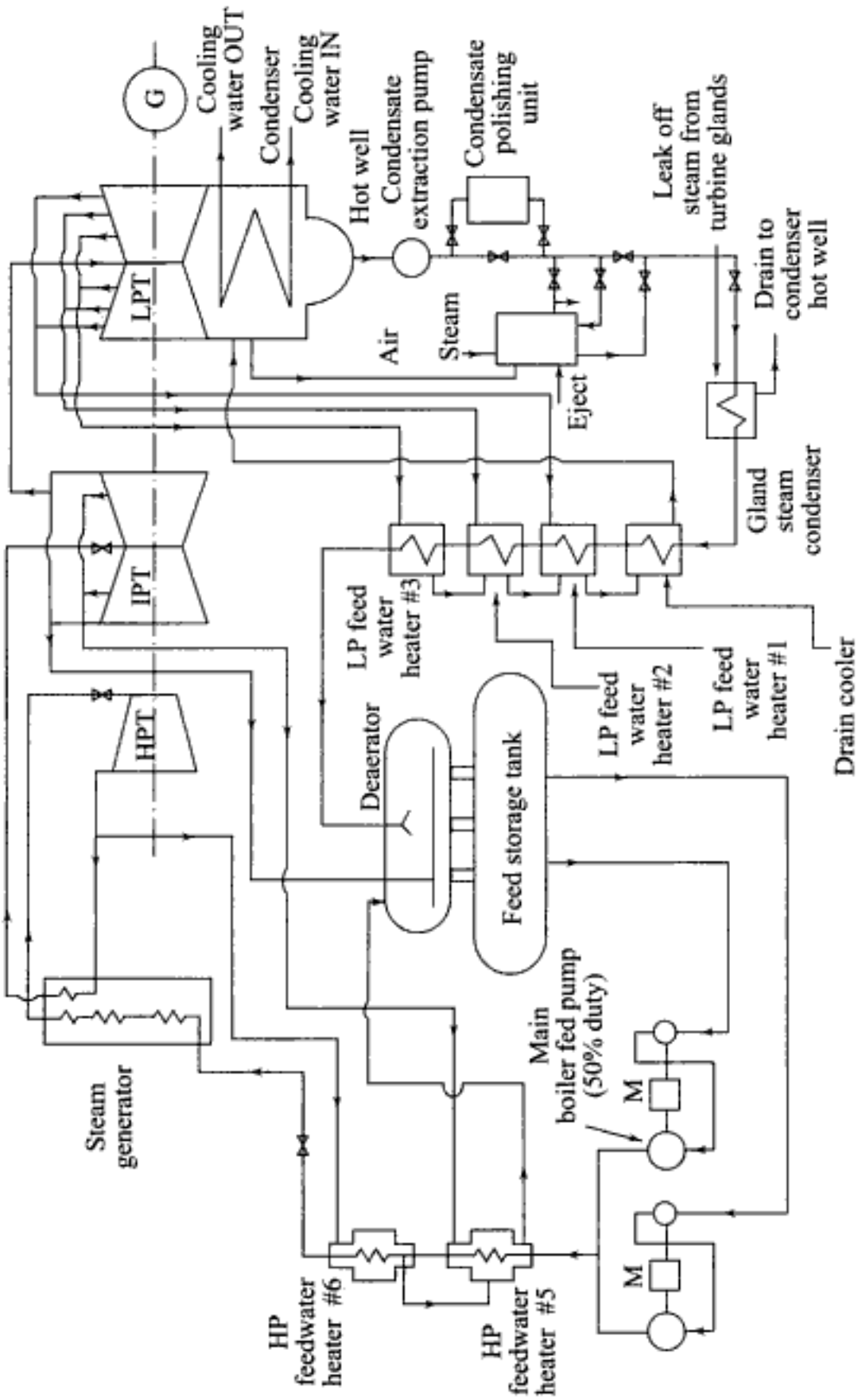

Fig. 2.31 (a) *Typical layout of a steam power plant*

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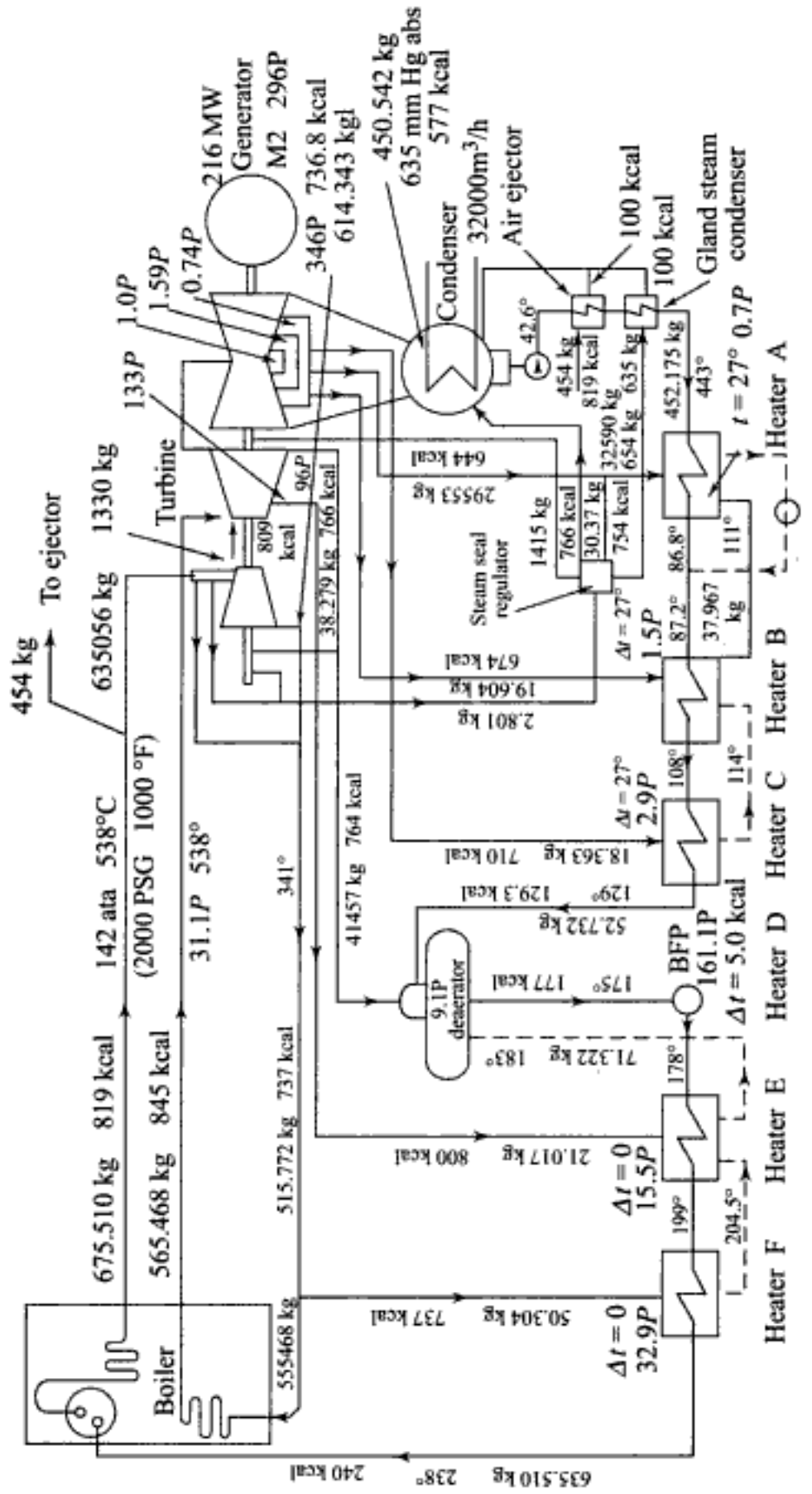

Heat rate of turbo-generator

$$h_{TG} = \frac{635{,}510(819 - 246) + 565{,}468(845 - 737)}{214{,}000} = 1{,}987 \text{ kcal/kWh}$$

kg = kg/hr
P = kg/cm² abs = ata
kcal = kcal/kg
° = °C

Fig. 2.31 (b) *Heat balance diagram of a 214 MW reheat unit (1 kcal = 4.187 kJ)*

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2.16 EFFICIENCIES IN A STEAM POWER PLANT

A steam power plant is a bulk energy coverter from fuel to electricity (Fig. 2.32). The overall efficiency of a power plant is defined as

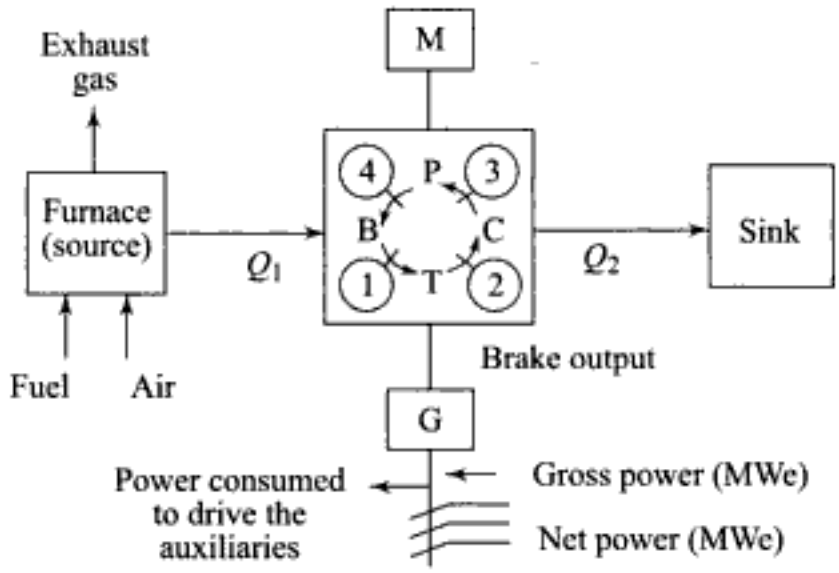

Fig. 2.32 *A power plant converts energy in fuel to electricity*

$$\eta_{\text{overall}} = \frac{\text{power available at the generator terminals}}{\text{rate of energy release by the combustion of fuel}}$$

$$= \frac{\text{MWe} \times 10^3}{w_f \times \text{C.V.}} \tag{2.42}$$

where w_f is the fuel burning rate and C.V. is the calorific value of fuel.
The boiler (or the steam generator) efficiency is defined by

$$\eta_{\text{boiler}} = \frac{\text{rate of energy absorption by water to form steam}}{\text{rate of energy release by the combustion of fuel}}$$

$$= \frac{w_s\,(h_1 - h_4)}{w_f \times \text{C.V.}} \tag{2.43}$$

where w_s is the steam generation rate.
The cycle efficiency is given by

$$\eta_{\text{cycle}} = \frac{h_1 - h_2}{h_1 - h_4} \tag{2.44}$$

The pump work has here been neglected.
The mechanical efficiency of the turbine will be

$$\eta_{\text{turbine(mech)}} = \frac{\text{brake output of the turbine}}{\text{internal output of the turbine}}$$

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$$= \frac{\text{brake output}}{w_s(h_1 - h_2)} \tag{2.45}$$

The generator efficiency of the electric alternator is

$$\eta_{\text{generator}} = \frac{\text{electrical output at generator terminals}}{\text{brake output of the turbine}}$$

$$= \frac{\text{MWe} \times 10^3}{\text{brake output in kWe}} \tag{2.46}$$

Multiplying Eq. (2.43) to Eq. (2.46),

$\eta_{\text{boiler}} \times \eta_{\text{cycle}} \times \eta_{\text{turbine(mech)}} \times \eta_{\text{generator}}$

$$= \frac{w_s(h_1 - h_4)}{w_f \times \text{C.V.}} \times \frac{w_s(h_1 - h_2)}{w_s(h_1 - h_4)} \times \frac{\text{brake output}}{w_s(h_1 - h_2)} \times \frac{\text{MWe} \times 10^3}{\text{brake output}}$$

$$= \frac{\text{MWe} \times 10^3}{w_f \times \text{C.V.}} = \eta_{\text{overall}}$$

Therefore,

$$\eta_{\text{overall}} = \eta_{\text{boiler}} \times \eta_{\text{cycle}} \times \eta_{\text{turbine(mech)}} \times \eta_{\text{generator}} \tag{2.47}$$

There are certain auxiliary equipment in the power plant like F.D. and I.D. fans, pulverizers, crushers, conveyors and so on which are driven by electricity taken from the generated power of the plant. The net power transmitted from the generator will be the gross power produced minus the power consumed by the internal auxiliaries of the plant. The efficiency of the auxiliaries, η_{aux}, is given by

$$\eta_{\text{aux}} = \frac{\text{net power transmitted by the generator}}{\text{gross power produced by the plant}} \tag{2.48}$$

Therefore, the overall efficiency of the plant is the product of five component efficiencies as given by

$$\eta_{\text{overall}} = \eta_{\text{boiler}} \times \eta_{\text{cycle}} \times \eta_{\text{turbine(mech)}} \times \eta_{\text{generator}} \times \eta_{\text{aux}} \tag{2.49}$$

For a modern power plant, the typical values are

$$\eta_{\text{boiler}} = 0.92,\ \eta_{\text{cycle}} = 0.44,\ \eta_{\text{turbine(mech)}} = 0.95,\ \eta_{\text{generator}} = 0.93,$$
$$\eta_{\text{aux}} = 0.95,\ \eta_{\text{overall}} = 0.92 \times 0.44 \times 0.95 \times 0.93 \times 0.95 = 0.34$$

Only 34% of the energy in fuel is converted to electricity and 66% of the energy is lost. The maximum loss of energy takes place in the condenser where heat is rejected to cooling water. This is the loss due to heat to work energy conversion in the cycle or the loss due to the second law.

The parameter which readily reflects the fuel economy is the heat rate, which is inversely proportional to the efficiency and hence, the lower its value the better. It broadly indicates the heat added per unit of work produced. There are various heat rates corresponding to the work used in the denominator. Thus,

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$$\text{Net cycle heat rate (HR)} = \frac{\text{rate of heat addition to cycle}}{\text{net cycle work output}} = \frac{Q_1}{W_{net}}$$

$$\text{Gross cycle HR} = \frac{\text{rate of heat addition}}{\text{turbine output}} = \frac{Q_1}{W_T}$$

$$\text{Net station (HR)} = \frac{\text{rate of heat addition to boiler}}{\text{net station output}} = \frac{Q_1}{\text{net station output}}$$

$$\text{Gross station (HR)} = \frac{Q_1}{\text{gross generation output}} \tag{2.50}$$

2.17 COGENERATION OF POWER AND PROCESS HEAT

There are several industries such as paper mills, textile mills, chemical factories, jute mills, sugar factories, rice mills and so on where saturated steam at the desired temperature is required for heating, drying etc. For constant temperature heating (or drying), steam is a very good medium since isothermal condition can be maintained by allowing saturated steam to condense at that temperature and utilizing the latent heat released for heating purposes. Apart from the process heat, the factory also needs power to drive various machines, for lighting and other purposes.

Earlier, steam for power purposes was generated at a moderate pressure and saturated steam for process work was generated separately at a pressure which gave the desired heating temperature. Having two separate units for process heat and power is wasteful, for of the total heat supplied to the steam generator for power purposes, a greater part will normally be carried away by the cooling water in the condenser.

2.17.1 Back Pressure Turbine

By modifying the initial steam pressure and exhaust pressure, it is possible to generate the required power and make available the required quantity of exhaust steam at the desired temperature for process work. In Fig. 2.33, the exhaust steam from the turbine is utilized for process heating, the process heater replacing the condenser of the ordinary Rankine cycle. The pressure at exhaust from the turbine is the saturation pressure corresponding to the temperature desired in the process heater. Such a turbine is called a *back pressure turbine*. A plant producing both electrical power and process heat simultaneously is called a *cogeneration plant*. When the process steam is the basic need, and the

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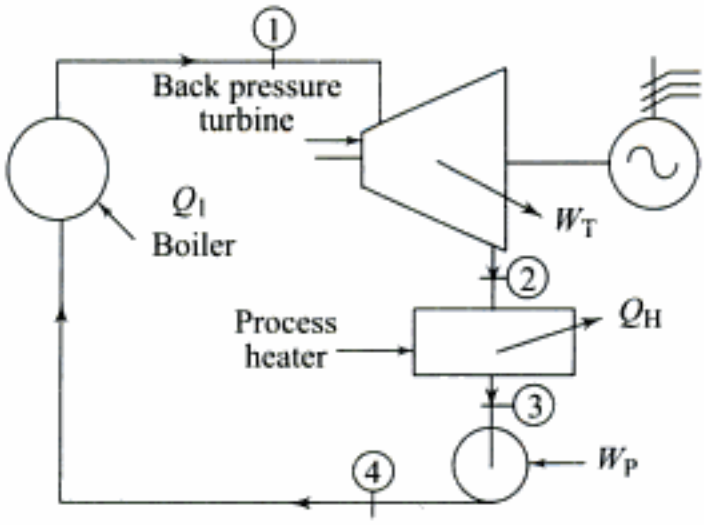

Fig. 2.33 *Cogeneration plant with a back pressure turbine*

power is produced incidentally as a by-product, the cycle is often called a by-product power cycle. Figure 2.34 shows the T–s plot for such a cycle. If W_T is the turbine output in kW, Q_H is the process heat required in kJ/h, and w_s is the steam flow rate in kg/h.

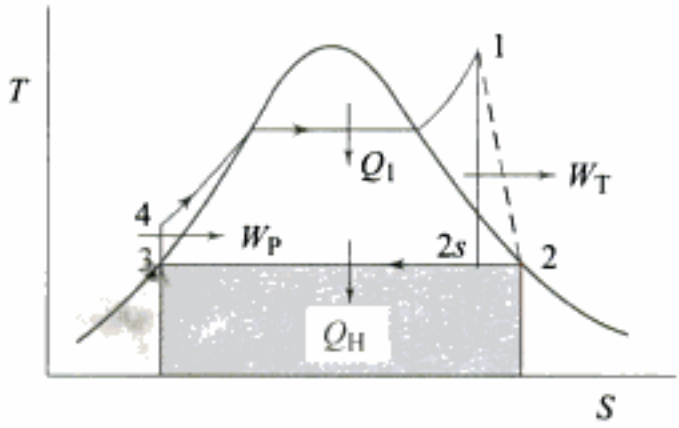

Fig. 2.34 *By-product power cycle with a back pressure turbine*

$$W_T \times 3600 = w_s(h_1 - h_2) \qquad \text{and} \qquad Q_H = w_s(h_2 - h_3)$$

$$\therefore \qquad W_T \times 3600 = \frac{Q_H}{h_2 - h_3}(h_1 - h_2)$$

or,

$$Q_H = \frac{W_T \times 3600 \times (h_2 - h_3)}{h_1 - h_2} \text{ kJ/h} \tag{2.51}$$

Of the total energy input Q_I (as heat) to the cogeneration plant, W_T part of it only is converted into shaft work or electricity. The remaining energy $(Q_I - W_T)$, which would otherwise have been a waste, as in the Rankine cycle, by second law, is utilized as process heat.

The cogeneration plant efficiency η_{co} is given by

$$\eta_{co} = \frac{W_T + Q_H}{Q_I} \tag{2.52}$$

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For separate generation of electricity and steam, the heat added per unit total energy output is

$$\frac{1}{\eta_e} + \frac{1-e}{\eta_h}$$

where e = electricity fraction of total energy output

$$= \frac{W_T}{W_T + Q_H}$$

η_e = electric plant efficiency

η_h = steam (or process heat) generator efficiency

The combined efficiency η_c for separate generation is therefore given by

$$\eta_c = \frac{1}{\dfrac{e}{\eta_e} + \dfrac{1-e}{\eta_h}} \tag{2.53}$$

Cogeneration is beneficial if the efficiency of the cogeneration plant, Eq. (2.52), is greater than that of separate generation, Eq. (2.53).

Back pressure turbines are quite small with respect to their power output because they have no great volume of exhaust to cope with, the density being high. They are usually single cylinder and hence, cheap in terms of cost per MW compared to condensing sets of the same power. Besides their use in process industries and petrochemical installations, back pressure turbines are used for desalination of sea-water, district heating, and also for driving compressors and feed pumps.

2.17.2 Pass-Out Turbine

In many cases, the power available from the back pressure turbine through which the whole of the heating steam flows is appreciably less than that required in a factory. This may be due to relatively high back pressure, or small heating requirement or both. Pass-out turbines are employed in these cases, where a certain quantity of steam is continuously extracted from the turbine at an intermediate stage for heating purposes at the desired temperature and pressure (Fig. 2.35).

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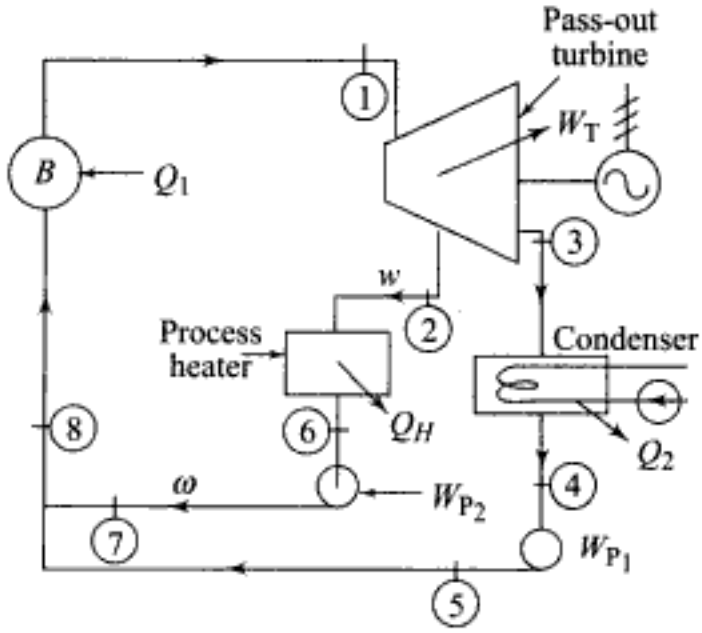
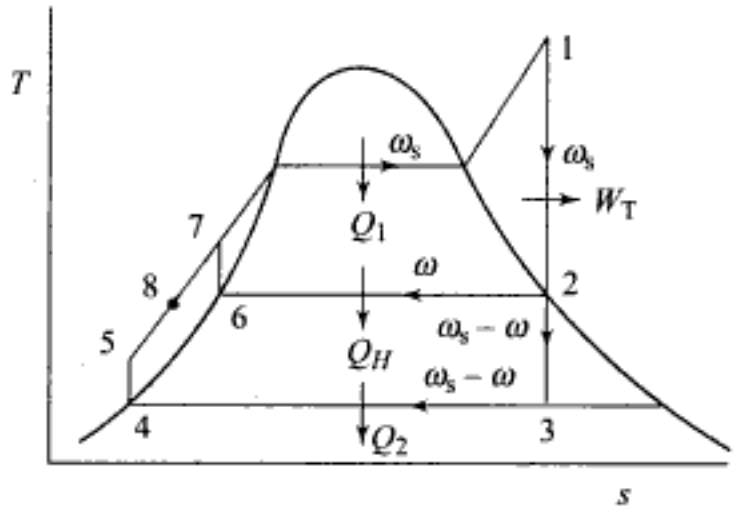


(a)



(b)

Fig. 2.35 *Cogeneration plant with a pass-out turbine*

$$\left.\begin{aligned} &Q_1 = w_s(h_1 - h_8); \qquad Q_2 = (w_s - w)(h_3 - h_4) \\ &Q_H = w(h_2 - h_6); \qquad W_T = w_s(h_1 - h_2) + (w_s - w)(h_2 - h_3) \\ &W_p = (w_s - w)(h_5 - h_4) + w(h_7 - h_6); \qquad wh_7 + (w_s - w)h_5 = w_s h_8 \end{aligned}\right\} \quad (2.54)$$

where w_s is the boiler capacity and w is the steam flow rate required at the desired temperature for process heating.

2.17.3 Process Heat Unit Topping the Power Cycle

In some plants, such as a high temperature cement kiln, the primary heat is used directly for process requirements. The low grade waste heat from the process heater is then used to generate electricity, obviously at low efficiency. Such a cycle has a combined efficiency lying below that given by Eq. (2.53) and therefore, is of less thermodynamic or economic interest.

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A cogeneration plant is only advisable from an economic viewpoint if the cost of electricity generated by it is less than that purchased from a utility system. If a utility is not available cogeneration becomes necessary irrespective of the cost of generation. In general, very low fraction of electric to total energy (e) are not considered economical for cogeneration.

Example 2.1 Steam at 40 bar, 500 °C flowing at the rate of 5500 kg/h expands in a h.p. turbine to 2 bar with an isentropic efficiency of 83%. A continuous supply of steam at 2 bar, 0.87 quality and a flow rate of 2700 kg/h is available from a geothermal energy source. This steam is mixed adiabatically with the h.p. turbine exhaust steam and the combined flow then expands in a l.p. turbine to 0.1 bar with an isentropic efficiency of 78%. Determine the power output and the thermal efficiency of the plant. Assume that 5500 kg/h of steam is generated in the boiler at 40 bar, 500 °C from the saturated feedwater at 0.1 bar.

Had the geothermal steam not been added, what would have been the power output and efficiency of the plant? Neglect pump work.

Solution With reference to Fig. E2.1

$$h_1 = 3445.3 \text{ kJ/kg}, s_1 = 7.0901 \text{ kJ/kg } K = 1.5301 + x_{2s} \times 5.5970$$

$$x_{2s} = \frac{5.5600}{5.5970} = 0.9934$$

$$h_{2s} = 504.7 + 0.9934 \times 2201.9 = 2692.04 \text{ kJ/kg K}$$

$$h_1 - h_2 = 0.83(3445.3 - 2692.04) = 625.21 \text{ kJ/kg}$$

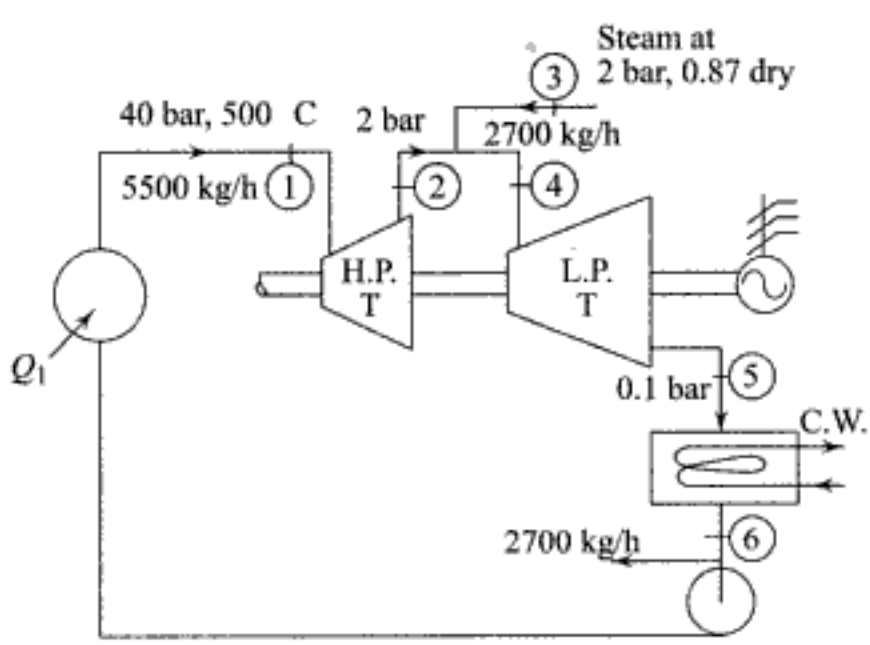
$$h_2 = 3445.3 - 625.21 = 2820.09 \text{ kJ/kg}$$

$$h_3 = 504.7 + 0.87 \times 2201.9 = 2420.4 \text{ kJ/kg}$$

$$2700h_3 + 5500h_2 = (2700 + 5500)h_4$$

$$h_4 = \frac{2700 \times 2420.4 + 5500 \times 2820.09}{8200} = 796.96 + 1891.52$$

$$= 2688.48 \text{ kJ/kg}$$



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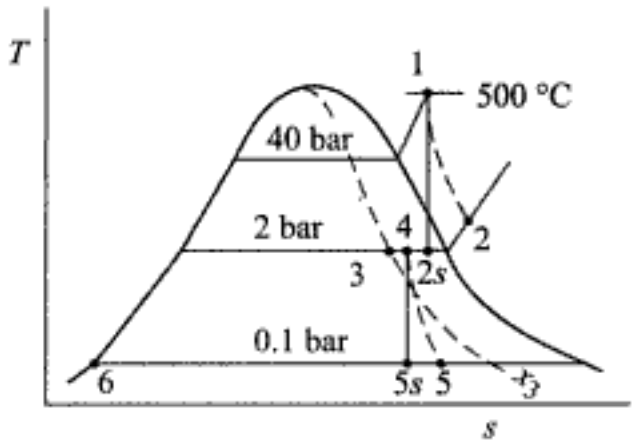

(b)

Fig. E2.1

$$h_4 = 504.7 + x_4 \times 2201.9 = 2688.48;\ x_4 = \frac{2183.78}{2201.9} = 0.9917$$

$$s_4 = 1.5301 + 0.9917 \times 5.5970 = 7.0806 \text{ kJ/kg–K}$$

$$= s_{5s} = 0.6493 + x_{5s} \times 7.5009$$

$$x_{5s} = 0.8574; \quad h_{5s} = 191.84 + 0.8574 \times 2392.8 = 2243.44 \text{ kJ/kg}$$

$$h_4 - h_5 = 0.78(2688.48 - 2243.44) = 347.1 \text{ kJ/kg}$$

$$h_6 = 191.83 \text{ kJ/kg}$$

$$\dot{W} = 5500(h_1 - h_2) + 8200\,(h_4 - h_5) = 5500 \times 625.21 + 8200 \times 347.1$$

$$= 6284875 \text{ kJ/h} = 1745.8 \text{ kW} \qquad (Ans.)$$

$$Q_1 = 5500(h_1 - h_6) = 5500\,(3445.3 - 191.8) \times \frac{1}{3600} = 4970.63 \text{ kW}$$

$$\eta_{cycle} = \frac{1745.8}{4970.63} = 0.353 \qquad \text{or} \qquad 35.3\%$$

Without geothermal heat supply:

$$W_T = 5500\,(h_1 - h_2) = 955.18 \text{ kW} \qquad (Ans.)$$

$$Q_1 = 5500\,(h_1 - h_6) = 4970.63 \text{ kW}$$

$$\eta_{cycle} = \frac{955.18}{4970.63} = 0.1922 \quad \text{or} \quad 19.22\% \qquad (Ans.)$$

Example 2.2 A steam power plant with inlet steam to the h.p. turbine at 90 bar and 500 °C, and condensation at 40 °C produces 500 MW. It has one stage of reheat optimally placed which raises the steam temperature back to 500 °C. One closed feedwater heater with drains cascaded back to the condenser receives bled steam at the reheat pressure, and the remaining steam is reheated and then expanded in the l.p. turbine. The h.p. and l.p. turbines have isentropic efficiences of 92% and 90%, respectively. The isentropic efficiency of the pump is 75%. Calculate (a) the mass flow rate of steam at turbine inlet in kg/s, (b) the cycle efficiency, and (c) the cycle work ratio. Use TTD = – 1.6 °C.

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Solution The flow and T–s diagrams are shown in Fig. E2.2. The optimum reheat pressure is taken to be 20% of the boiler pressure, which becomes $0.2 \times 90 = 18$ bar. Now, $h_1 = 3386.1$ kJ/kg, $s_1 = 6.6576$ kJ/kg–K $= s_{2s}$, $h_{2s} = 2915$ kJ/kg. $h_3 = 3469.8$ kJ/kg.

$$s_3 = 7.4825 = s_{4s} = s_f + x_{4s}\, s_{fg}$$

$$\therefore \qquad x_{4s} = \frac{7.4825 - 0.5725}{7.6845} = 0.8992$$

$$h_{4s} = 167.57 + 0.8992 \times 2406.7 = 2331.7 \text{ kJ/kg}$$

$$h_5 = 167.57 \text{ kJ/kg}, \quad h_7 = 883.42 \text{ kJ/kg}$$

$$W_{Ps} = \int v\, dp = 0.001008 \times 90 \times 10 = 9.072 \text{ kJ/kg}$$

$$h_{6s} = 176.64 \text{ kJ/kg}$$

$$h_1 - h_2 = 0.92\,(3386 - 2915) = 433.3 \text{ kJ/kg}, \quad \text{or} \quad h_2 = 3386.1 - 433.3$$

$$= 2952.8 \text{ kJ/kg}$$

$$h_3 - h_4 = 0.9(3469.8 - 2331.7) = 1024.29 \text{ kJ/kg}$$

$$h_4 = 3469.8 - 1024 = 2445.5 \text{ kJ/kg}$$

$$W_P = h_6 - h_5 = \frac{9{\cdot}072}{0.75} = 12.1 \text{ kJ/kg}, \quad \text{or} \quad h_6 = 167.57 + 12.1$$

$$= 179.67 \text{ kJ/kg}$$

$$t_{sat} \text{ at 18 bar} = 207.15\ ^\circ\text{C} \qquad \therefore\ t_9 = 207.15 + 1.6 = 208.75\ ^\circ\text{C}$$

$$h_9 = 875 \text{ kJ/kg}$$

$$1(h_9 - h_6) = m(h_2 - h_7), \quad m = \frac{875 - 179.67}{2952.8 - 883.4} = 0.336 \text{ kg}$$

$$W_T = (h_1 - h_2) + (1 - m)(h_3 - h_4) = 433.3 + 0.664 \times 1024.3$$

$$= 1113.435 \text{ kJ/kg}$$

$$W_{net} = W_T - W_P = 1113.435 - 12.1 = 1101.335 \text{ kJ/kg}$$

$$w_s = \frac{500 \times 10^3}{1101.335} = 454 \text{ kg/s} \qquad \textbf{Ans. (a)}$$

$$Q_1 = h_1 - h_9 + (1 - m)(h_3 - h_2)$$

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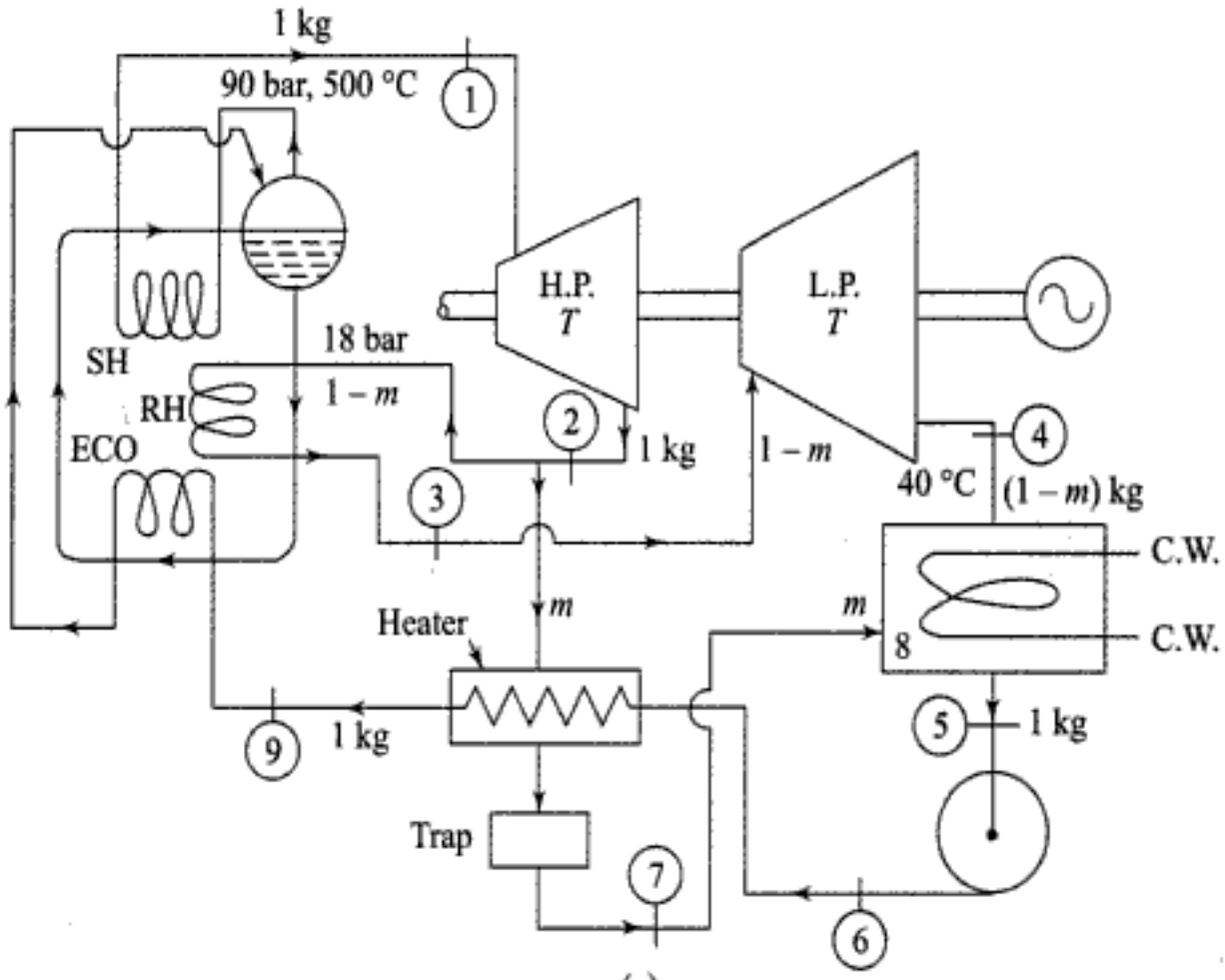

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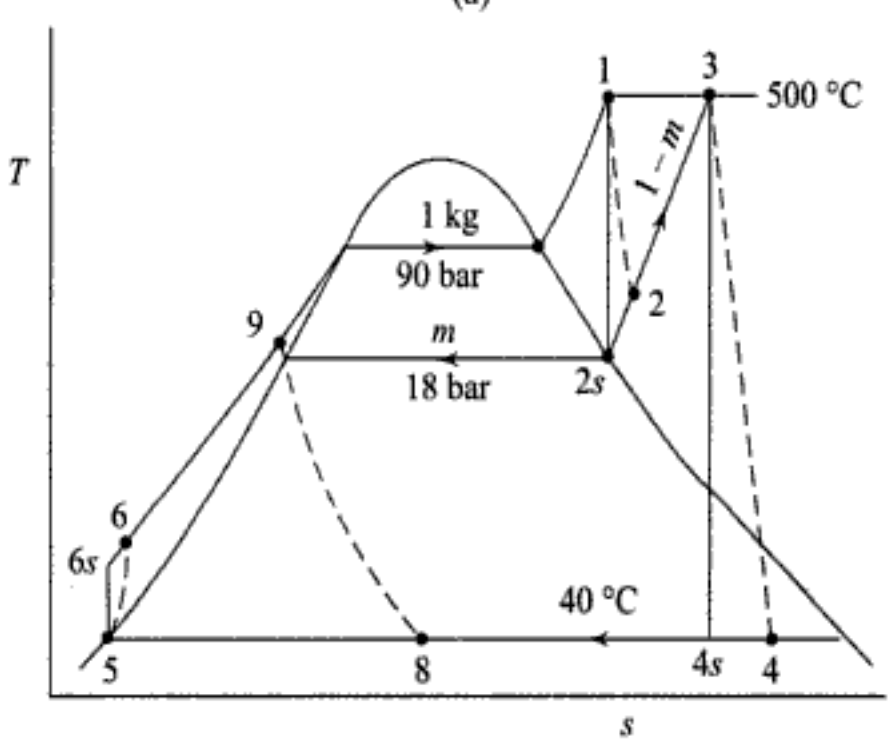

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Fig. E2.2

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$$= 3386.1 - 875 + 0.664(3469.8 - 2952.8) = 2854.3 \text{ kJ/kg}$$

$$\eta_{cycle} = \frac{1101.335}{2854.3} = 0.3858, \quad \text{or} \quad 38.58\% \qquad \textit{Ans.}(b)$$

$$\text{Work ratio} = \frac{W_{net}}{W_T} = \frac{1101.335}{1113.435} = 0.989 \qquad \textit{Ans.}\ (c)$$

Example 2.3 An ideal steam power plant operates between 70 bar, 550 °C and 0.075 bar. It has seven feedwater heaters. Find the optimum pressure and temperature at which the heaters operate.

Solution With reference to Fig. E2.3,

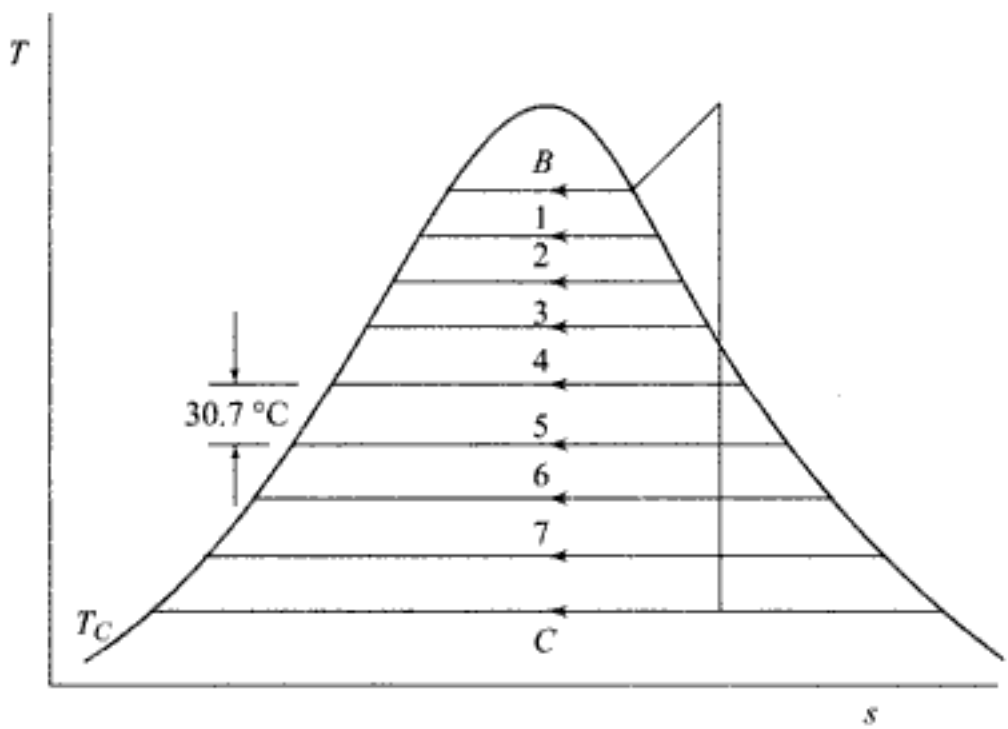

Fig. E2.3

T_B = saturation temperature at 70 bar = 285.9 °C

T_C = saturation temperature at 0.075 bar = 40.3 °C

The temperature rise per heater for maximum cycle efficiency

$$= \frac{285.9 - 40.3}{7+1} = 30.7 \text{ °C}$$

Heater 1: $t_1 = 285.9 - 30.7 = 255.2$ °C

$p_1 = 4.33$ MPa (from steam table)

Heater 2: $t_2 = 255.2 - 30.7 = 224.5$ °C; $p_2 = 2.5318$ MPa

Heater 3: $t_3 = 193.8$ °C; $p_3 = 1.367$ MPa

Heater 4: $t_4 = 163.1$ °C; $p_4 = 0.6714$ MPa

Heater 5: $t_5 = 132.4$ °C; $p_5 = 0.2906$ MPa

Heater 6: $t_6 = 101.7$ °C; $p_6 = 0.108$ MPa

Heater 7: $t_7 = 71.0$ °C; $p_7 = 32.65$ kPa

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Example 2.4 In a power plant, the efficiencies of the electric generator, turbine (mechanical), boiler, cycle and the overall plant are 0.97, 0.95, 0,92, 0.42 and 0.33, respectively. What percentage of the total electricity generated is consumed in running the auxiliaries?

Solution

$$\eta_{\text{plant}} = \eta_{\text{Boiler}} \times \eta_{\text{Turbine(mech)}} \times \eta_{\text{Generator}} \times \eta_{\text{Cycle}} \times \eta_{\text{Auxiliaries}}$$

$$\therefore \quad \eta_{\text{Auxiliaries}} = \frac{0.33}{0.97 \times 0.95 \times 0.92 \times 0.42} = 0.9268$$

$$1 - 0.9268 = 0.0732$$

or 7.32% of total electricity generated is consumed by the auxiliaries.

Example 2.5 A steam generator comprises a boiler, a superheater, an economiser and an air preheater. The feedwater enters the economiser at 140 °C and leaves as saturated liquid. Air is preheated from a temperature of 25 °C to 250 °C. Steam leaves the boiler drum at 60 bar, 0.98 dry and leaves the superheater at 450 °C. When using coal with a calorific value of 25.2 MJ/kg, the rate of evaporation is 8.5 kg steam per kg coal and the air fuel ratio is 15 : 1 by mass. Neglecting heat losses and pressure drops, estimate the heat transfer per kg fuel in each component and the efficiency of the steam generator. What are the percentages of the total heat absorption taking place in the economiser, boiler and the superheater, respectively? Assume c_p of air and water as 1.005 and 4.2 kJ/kg K, respectively.

Solution With reference to Fig. E2.5, $h_1 = 140 \times 4.2 = 588$ kJ/kg

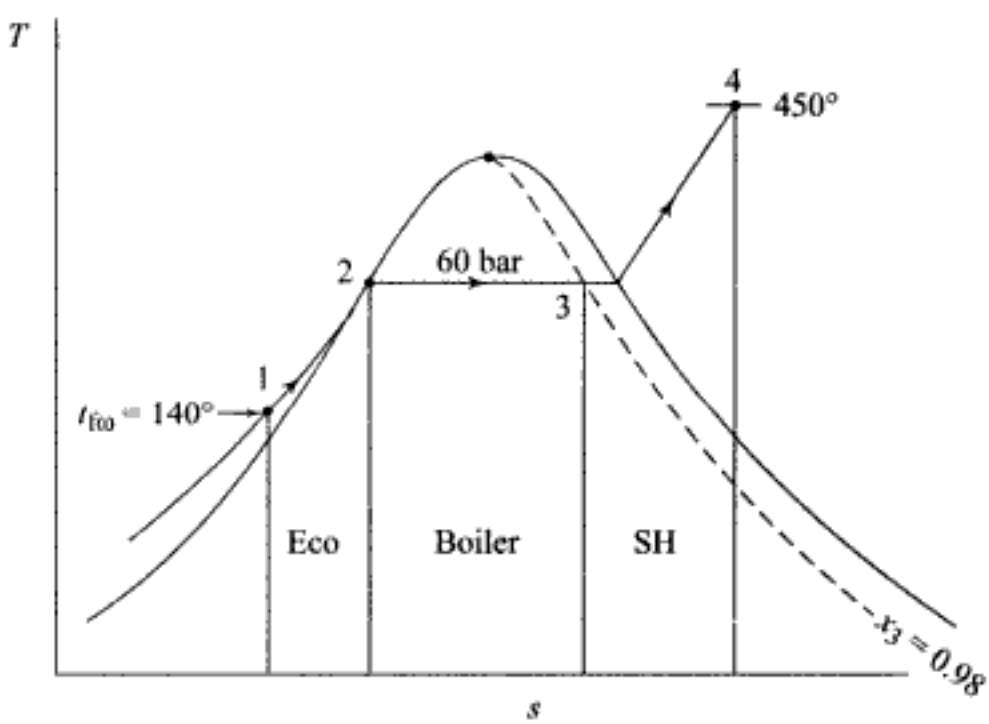

Fig. E2.5

$h_f = 1213.35$ kJ/kg, $h_{fg} = 1571.0$ (at 60 bar)
$h_4 = 3301.8$ kJ/kg,

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$$h_3 = 1213.35 + 0.98 \times 1571.0 = 2752.93 \text{ kJ/kg}$$

$$\eta_{\text{St.gen}} = \frac{w_s (h_4 - h_1)}{w_f \times \text{C.V.}} = \frac{8.5 \times 2713.8}{25.2 \times 1000} = 0.9154 \quad \text{or} \quad 91.54\%$$

Heat transfer in the economiser

$$= \frac{w_s (h_2 - h_1)}{w_f} = 8.5 \times 625.35 \times 10^{-3} = 5.3155 \text{ MJ/kg}$$

Heat transfer in the boiler

$$= \frac{w_s (h_3 - h_2)}{w_f} = 8.5 \times 1539.58 \times 10^{-3} = 13.086 \text{ MJ/kg}$$

Heat transfer in the superheater

$$= \frac{w_s (h_4 - h_3)}{w_f} = 8.5 \times 548.87 \times 10^{-3} = 4.665 \text{ MJ/kg}$$

Heat transfer in the air pre-heater

$$= \frac{w_a c_{p_a} (t_2 - t_1)}{w_f} = 15 \times 1.005 \times (250 - 25) \times 10^{-3} = 3.392 \text{ MJ/kg}$$

Percentage of total heat absorbed in the economiser

$$= \frac{h_2 - h_1}{h_4 - h_1} \times 100 = \frac{625.35}{2713.8} \times 100 = 23.04\%$$

Percentage of total heat absorbed in the boiler

$$= \frac{h_3 - h_2}{h_4 - h_1} \times 100 = \frac{1539.58}{2713.8} \times 100 = 56.73\%$$

Percentage of total heat absorbed in the superheater

$$= \frac{h_4 - h_3}{h_4 - h_1} \times 100 = \frac{548.87}{2713.8} \times 100 = 20.23\%$$

Eample 2.6 Steam at 150 bar, 550 °C is expanded in an h.p. turbine to 20 bar when it is reheated to 500 °C and expanded in i.p. and l.p. turbines to condenser pressure of 0.075 bar. There are five feedwater heaters, one extraction from h.p. turbine at 50 bar, 3 from i.p. turbine at 10 bar, 5 bar and 3 bar, and one from l.p. turbine at 1.5 bar. The middle heater is the deaerator and all others are closed heaters. Assuming ideal conditions, determine (a) the cycle efficiency, (b) the feedwater temperature at inlet to the steam generator, (c) the steam rate, (d) the heat rate, (e) the quality of steam at turbine exhaust, and (f) the power output if the steam flow rate is 300 t/h. Take TTD = 0 for all the heaters.

Solution With reference to Fig. E2.6,

$$h_1 = 3448.6 \text{ kJ/kg}, \quad s_1 = s_2 = s_3 = 6.5199 \text{ kJ/kg K}$$

$$h_4 = 3467.6 \text{ kJ/kg}, \quad s_4 = s_5 = s_6 = s_7 = s_8 = s_9 = 7.4317 \text{ kJ/kg K}$$

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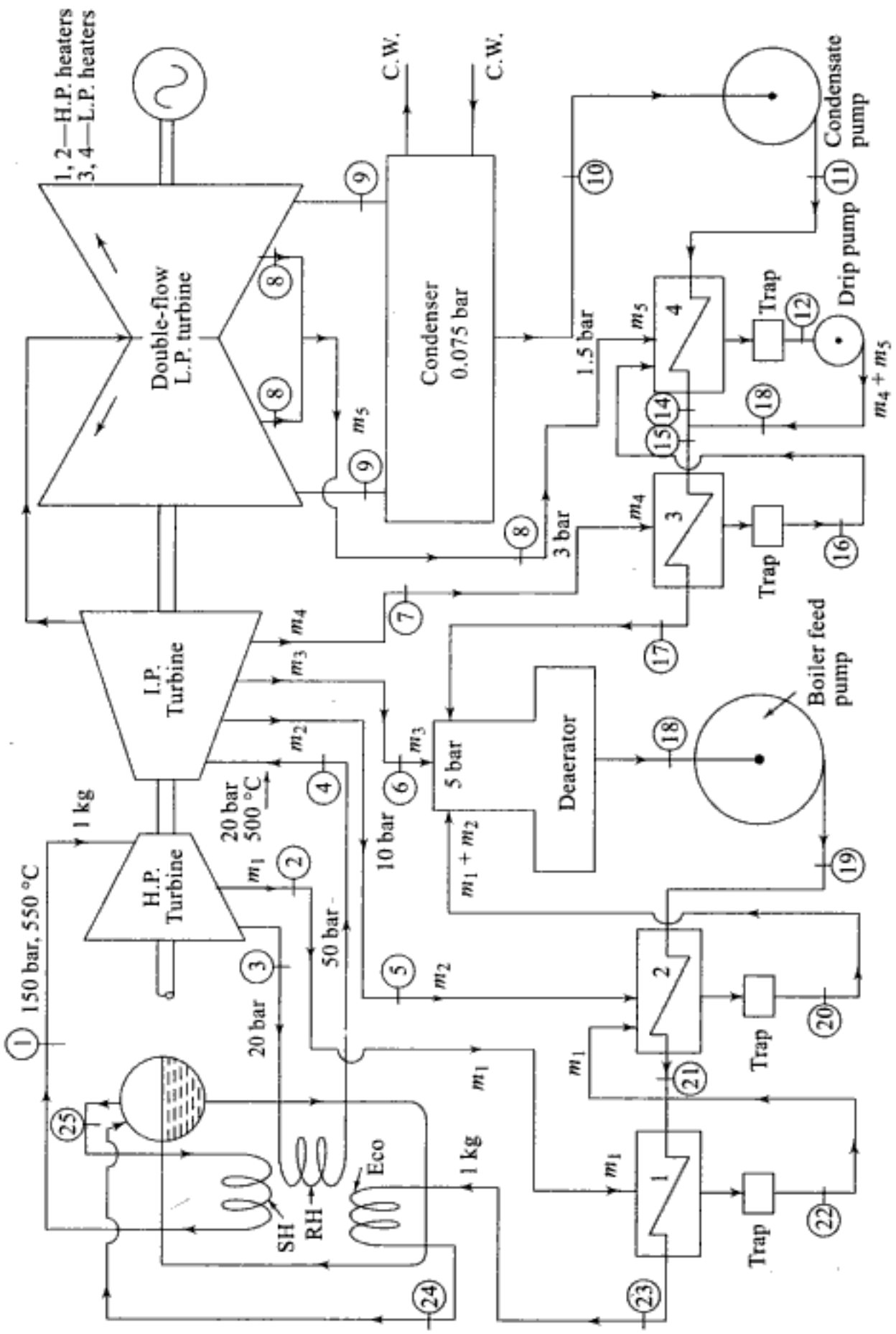

Fig. E2.6 (a)

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$t_2 = 370\ °C, \qquad h_2 = 3112\ kJ/kg$

$t_3 = 245\ °C, \qquad h_3 = 2890\ kJ/kg$

$t_5 = 400\ °C, \qquad h_5 = 3250\ kJ/kg$

$t_6 = 300\ °C, \qquad h_6 = 3050\ kJ/kg$

$t_7 = 225\ °C, \qquad h_7 = 2930\ kJ/kg$

$t_8 = 160\ °C, \qquad h_8 = 2790\ kJ/kg$

$$7.4317 = 0.5764 + x_9 \times 7.6751$$

$x_9 = 0.8932$

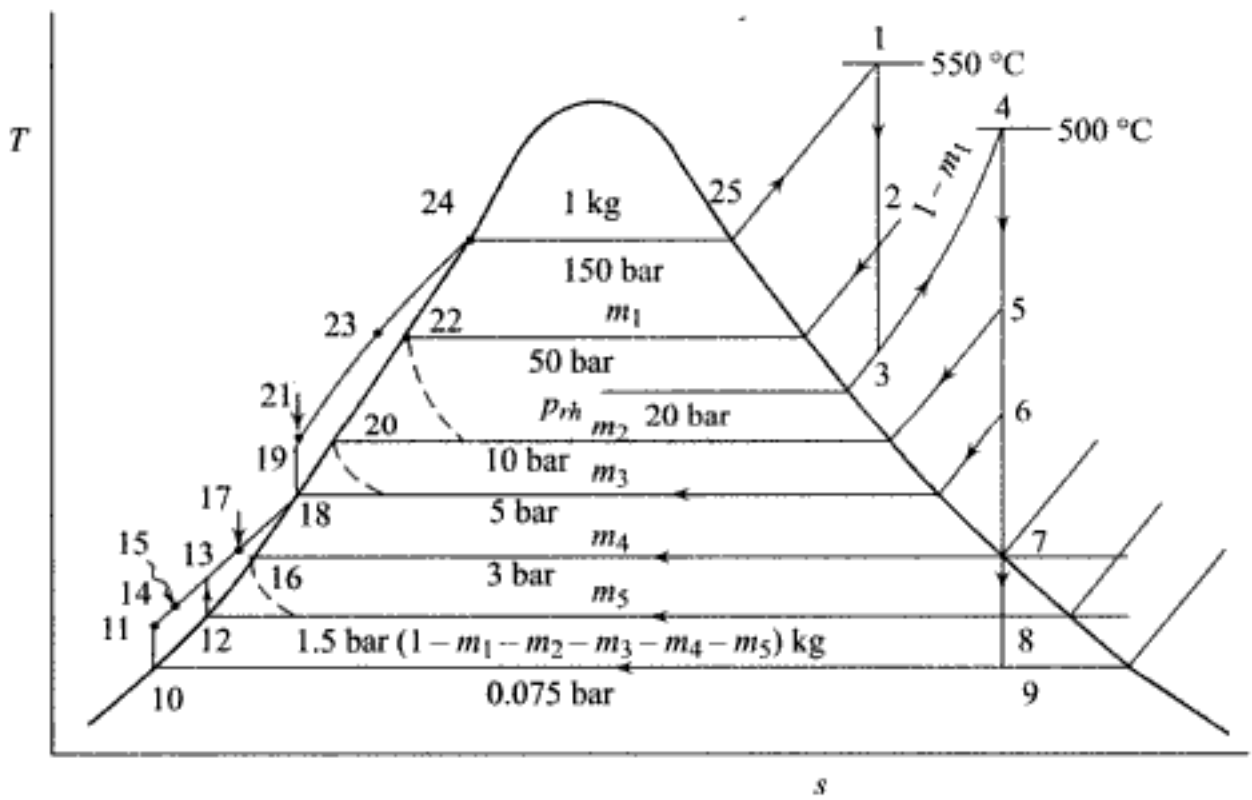

Fig. E2.6 (b)

$h_9 = 168.79 + 0.8932 \times 2406.0 = 2317.83\ kJ/kg$

$h_{10} = 168.79\ kJ/kg$

$h_{11} = 168.79 + 0.001 \times 5 \times 100 = 169.29\ kJ/kg$

$h_{12} = 467.11\ kJ/kg, \quad t_{14} = 111.37, \quad h_{14} = 467\ kJ/kg$

$h_{12} = h_{13} = h_{14} = h_{15} = 467\ kJ/kg$

$h_{16} = h_{17} = 561.47\ kJ/kg, \quad h_{18} = 640.23\ kJ/kg$

$h_{19} = 640.23 + 0.001 \times 145 \times 100 = 654.63\ kJ/kg$

$h_{20} = 762.8\ kJ/kg = h_{21}, \quad h_{22} = h_{22} = h_{23} = 1154.23\ kJ/kg$

Heater 1

$$m_1(h_2 - h_{22}) = 1(h_{23} - h_{21})$$

$$m_1 = \frac{1154.23 - 762.8}{3112 - 1154.23} = \frac{391.43}{1957.77} = 0.2\ kg$$

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Heater 2

$$m_2(h_5 - h_{20}) + m_1(h_{22} - h_{20}) = 1(h_{21} - h_{19})$$

$$m_2(3250 - 762.8) + 0.2(1154.23 - 762.8) = (762.8 - 654.6)$$

$$m_2 \times 2487.2 + 78.29 = 108.2$$

$$m_2 = 0.012 \text{ kg}$$

Heater 3

$$m_3(h_6 - h_{18}) + (m_1 + m_2)(h_{20} - h_{18}) = (1 - m_1 - m_2 - m_3)(h_{18} - h_{17})$$

$$m_3(3050 - 640.2) + 0.212(762.8 - 640.2) = (0.788 - m_3)(640.2 - 561.5)$$

$$m_3 \times 2409.8 + 26.00 = 62.02 - 78.7\text{m}_3$$

$$m_3 = 36.02/2488.5 = 0.0145 \text{ kg}$$

Heater 4

$$m_4(h_7 - h_{16}) = (1 - m_1 - m_2 - m_3)(h_{17} - h_{15})$$

$$m_4(2930 - 561.5) = 0.7735(561.5 - 467) = 73.096$$

$$m_4 = 0.031 \text{ kg}$$

Heater 5

$$m_5(h_8 - h_{12}) + m_4(h_{16} - h_{12}) = (1 - m_1 - m_2 - m_3 - m_4 - m_5)(h_{14} - h_{11})$$

$$m_5(2790 - 467) + 0.031(561.5 - 467) = (0.7425 - m_5)(467 - 169.3)$$

$$m_5 \times 2323 + 2.93 = 221.04 - 297.7 \text{ m}_5$$

$$m_5 = \frac{218.11}{2620.7} = 0.0832 \text{ kg}$$

$$\begin{aligned} W_T &= 1(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1)(h_4 - h_5) \\ &\quad + (1 - m_1 - m_2)(h_5 - h_6) + (1 - m_1 - m_2 - m_3)(h_6 - h_7) \\ &\quad + (1 - m_1 - m_2 - m_3 - m_4)(h_7 - h_8) \\ &\quad + (1 - m_1 - m_2 - m_3 - m_4 - m_5)(h_8 - h_9) \\ &= (3448.6 - 3112) + 0.8(3112 - 2890) + 0.8(3467.6 - 3250) \\ &\quad + 0.788(3250 - 3050) + 0.7735(3050 - 2930) \\ &\quad + 0.7425(2930 - 2790) + 0.6593(2790 - 2317.8) \\ &= 336.6 + 177.6 + 174.1 + 157.6 + 92.8 + 104.0 + 311.3 \\ &= 1354.0 \text{ kJ/kg} \end{aligned}$$

$$W_p = 0.5 + 14.5 + 0.15 = 15.15 \text{ kg}$$

$$W_{net} = 1354 - 15.15 = 1338.85 \text{ kJ/kg}$$

$$\begin{aligned} Q_1 &= 1(h_1 - h_{23}) + (1 - m_1)(h_4 - h_3) \\ &= 3448.6 - 1154.2 + 0.8(3467.6 - 2890) = 2756.48 \text{ kJ/kg} \end{aligned}$$

$$\therefore \quad n_{cycle} = \frac{1338.85}{2756.48} = 0.4857, \quad \text{or} \quad 48.57\% \quad \textit{Ans.} \text{ (a)}$$

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t_{23} = feedwater temperature at inlet to the steam generator

= 264 °C *Ans.*(b)

$$\text{Steam rate} = \frac{3600}{W_{net}} = \frac{3600}{1338.85} = 2.69 \text{ kJ/kWh} \qquad \textit{Ans. (c)}$$

$$\text{Heat rate} = \frac{Q_1}{W_{net}} \times 3600 = \frac{3600}{0.4857} = 7412 \text{ kJ/kWh} \qquad \textit{Ans. (d)}$$

Quality of steam at turbine exhaust = x_9 = 0.8932 *Ans.* (e)

$$\text{Power output} = \frac{1338.85 \times 300 \times 10^3}{3600} \text{ kW} = 111.57 \text{ MW} \qquad \textit{Ans. (f)}$$

Eample 2.7 A textile factory requires 10 t/h of steam for process heating at 3 bar saturated and 1000 kW of power, for which a back pressure turbine of 70% internal efficiency is to be used. Find the steam condition required at inlet of the turbine.

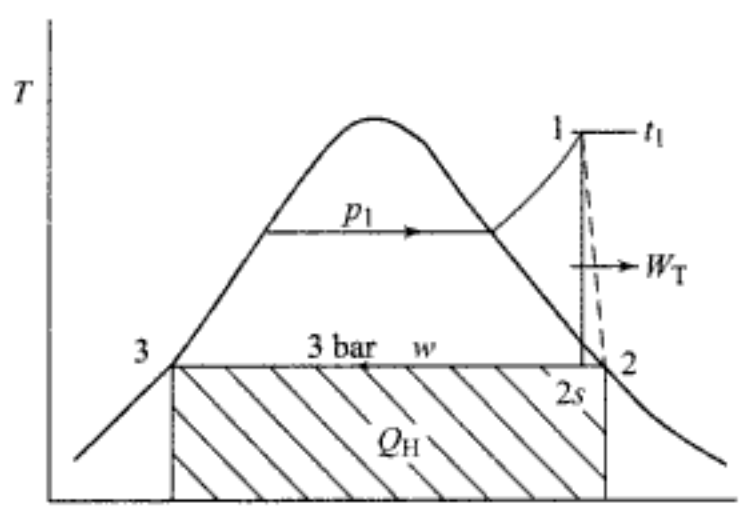

Fig. E2.7

Solution If w = mass flow rate of steam,

$$w(h_1 - h_2) = 1000 \text{ kW}$$

$$\therefore \qquad h_1 - h_2 = \frac{1000 \times 3600}{10{,}000} = 360 \text{ kJ/kg}$$

With reference to Fig. E2.7, $h_2 = 2725.3$

$$\therefore \qquad h_1 = 3085.3 \text{ kJ/kg}$$

Now, $$h_1 - h_{2s} = \frac{360}{0.7} = 514.286 \text{ kJ/kg}$$

$$\therefore \qquad h_{2s} = 3085.3 - 514.286 = 2571.014 \text{ kJ/kg}$$

$$h_{2s} = h_f + x_{2s}\, h_{fg} = 561.47 + x_{2s} \times 2163.8 = 2571.014$$

$$\therefore \qquad x_{2s} = 0.9287$$

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$$s_{2s} = 1.6718 + 0.9287(6.9919 - 1.6718) = 6.6125 \text{ kJ/kg K}$$
$$= s_1$$

Corresponding to $h_1 = 3085.3$ kJ/kg and

$s_1 = 6.6125$ kJ/kg K, from the Mollier diagram,

$p_1 = 37.3$ bar, $t_1 = 344$ °C, which is the state of steam at turbine inlet.

Eample 2.8 In a cogeneration plant, the power load is 5.6 MW and the heating load is 1.163 MW. Steam is generated at 40 bar and 500 °C and is expanded isentropically through a turbine to a condenser at 0.06 bar. The heating load is supplied by extracting steam from the turbine at 2 bar, which condensed in the process heater to saturated liquid at 2 bar and then pumped back to the boiler. Compute (a) the steam generation capacity of the boiler in t/h, (b) the heat input to the boiler in kW, (c) the fuel burning rate of the boiler in t/h if a coal of calorific value 25 MJ/kg is burned and the boiler efficiency is 88%, (d) the heat rejected to the condenser, (e) the rate of flow of cooling water in the condenser if the temperature rise of water is 6°C. Neglect pump work.

Solution With reference to Fig. E2.8,

$$h_1 = 3445.3 \text{ kJ/kg}, \quad s_1 = 7.0901 = s_2 = s_3$$

$$7.0901 = 1.5301 + x_2 \times 5.5970 \qquad \text{or} \qquad x_2 = \frac{5.56}{5.597} = 1.0$$

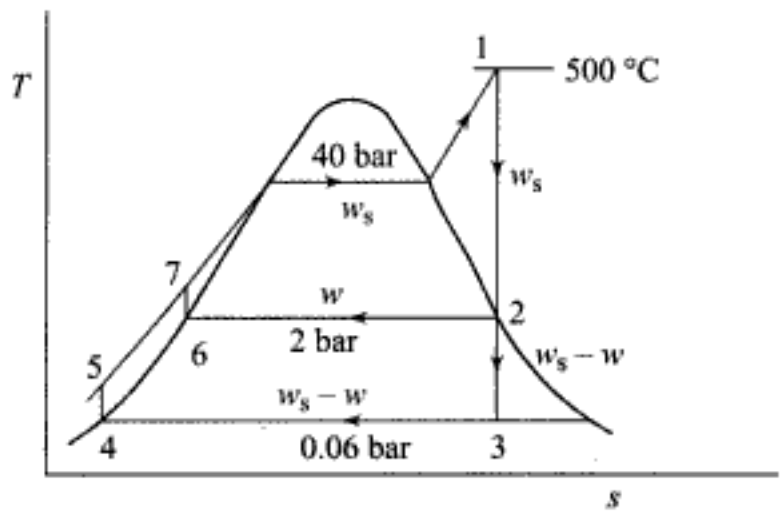

Fig. E2.8

$\therefore$ $$h_2 = h_g = 2706.7 \text{ kJ/kg}, \; h_2 - h_6 = h_{fg} = 2201.9 \text{ kJ/kg}$$

If w is the rate of steam extraction for process heating,

$$w(h_2 - h_6) = 1.163 \times 10^3$$

$$w = \frac{1.163 \times 10^3}{2201.9} = 0.528 \text{ kg/s} = 1901.4 \text{ kg/h}$$

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$$s_1 = 7.0901 = s_f + x_3 s_{fg} = 0.52 + x_3 \times 7.815$$

$$x_3 = 0.84$$

$$h_3 = 149.79 + 0.84 \times 2416 = 2180.59 \text{ kJ/kg}$$

Total work output, $W_T = w_s(h_1 - h_2) + (w_s - w)(h_2 - h_3)$

$$5.6 \times 10^3 = w_s \times 738.6 + w_s \times 526.11 - 277.8$$

$$w_s = \frac{5877.8}{1264.7} = 4.648 \text{ kg/s} = 16731 \text{ kg/h} = 16.73 \text{ t/h} \qquad \textit{Ans.}(a)$$

$$h_7 = 504.7 + 1.061 \times 10^{-3}(40 - 2) \times 100 = 508.73 \text{ kJ/kg}$$

$$h_5 = 149.79 + 1.006 \times 100 \times 40 \times 10^{-3} = 153.8 \text{ kJ/kg}$$

$$Q_1 = (w_s - w)(h_1 - h_5) + w(h_1 - h_7)$$

$$= (4.648 - 0.528)(3445.3 - 153.8) + 0.528(3445.3 - 508.73)$$

$$= 4.120 \times 3291.5 + 0.528 \times 2936.57 = 15111.5 \text{ kJ/s}$$

$$= 15.111 \text{ MW} \qquad \textit{Ans.}(b)$$

$$h_{boiler} = \frac{Q_1}{w_f \times \text{C.V.}} = \frac{15.111}{w_f \times 25} = 0.88$$

$$w_f = 0.687 \text{ kg/s} = 2473.2 \text{ kg/h} = 2.473 \text{ t/h} \qquad \textit{Ans.}(c)$$

$$Q_2 = (w_s - w)(h_3 - h_4) = 4.12 \times 2030.8 = 8367 \text{ kW}$$

$$= 8.367 \text{ MW} \qquad \textit{Ans.}\ (d)$$

If w_c = water flow rate in the condenser,

$$Q_2 = w_c\, c_p(t_2 - t_1)$$

$$\therefore \quad w_c = \frac{8367}{4.187 \times 6} = 333.05 \text{ kg/s} = 0.333 \text{ m}^3/\text{s} \qquad \textit{Ans.}\ (e)$$

Example 2.9 In a combined power and process plant the boiler generates 21,000 kg/h of steam at a pressure of 17 bar and temperature 230 °C. A part of the steam goes to a process heater which consumes 132.56 kW, the steam leaving the process heater 0.957 dry at 17 bar being throttled to 3.5 bar. The remaining steam flows through an h.p. turbine which exhausts at a pressure of 3.5 bar. The exhaust steam mixes with the process steam before entering the l.p. turbine which develops 1337.5 kW. At the exhaust, the pressure is 0.3 bar and the steam is 0.912 dry. Draw the line and *T–s* diagrams of the plant and determine (a) the steam quality at the exhaust of the h.p. turbine, (b) the power developed by the h.p. turbine, and (c) the isentropic efficiency of the h.p. turbine.

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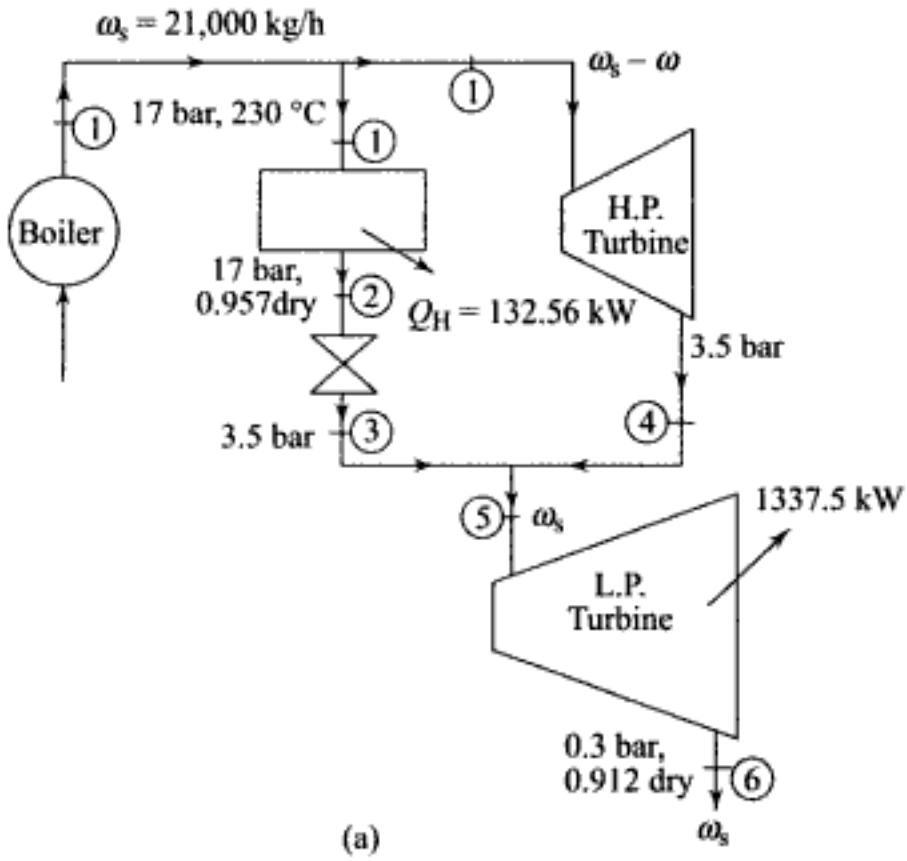
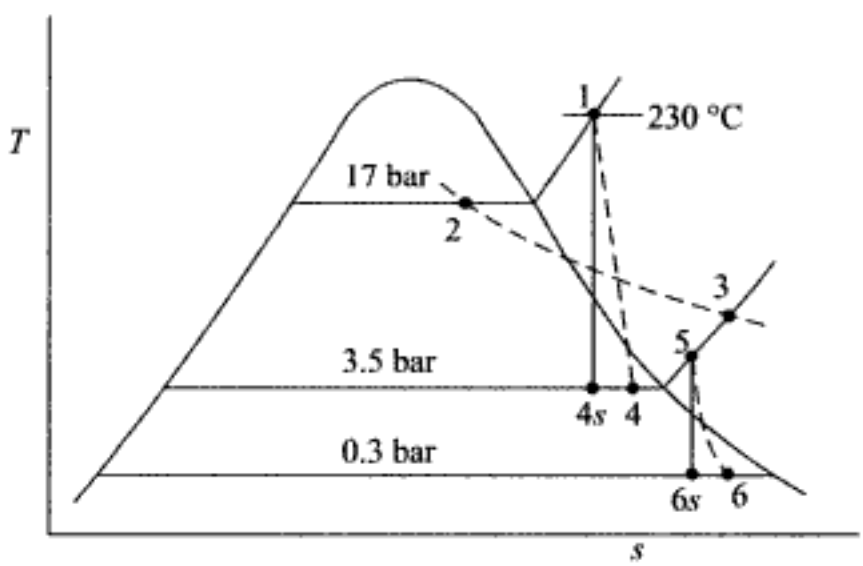
Solution The flow and T–s diagrams of the plant are shown in Fig. E2.9.



(a)



(b)

Fig. E2.9

$$h_1 = 2869.7 \text{ kJ/kg}, s_1 = 6.5408 \text{ kJ/kg K}$$

$$h_2 = 870.44 + 0.957 \times 1924.7 = 2712.38 \text{ kJ/kg}$$

$$h_2 = h_3$$

$$w_s (h_5 - h_6) = 1337.5 \text{ kW}$$

$$\therefore \quad h_5 - h_6 = \frac{1337.5 \times 3600}{21000} = 229.29 \text{ kJ/kg}$$

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$$h_6 = 289.23 + 0.912 \times 2336.1 = 2419.75 \text{ kJ/kg}$$

$$\therefore \quad h_5 = 2649.04 \text{ kJ/kg}$$

$$s_1 = s_{4s} = 6.5408 = 1.7275 + x_{4s} \times 5.2130$$

$$x_{4s} = 0.923$$

$$h_{4s} = 584.33 + 0.923 \times 2148.1 = 2567.03 \text{ kJ/kg}$$

$$w(h_1 - h_2) = 132.56 \text{ kW}$$

$$w = \frac{132.56}{2867.7 - 2712.38} = 0.843 \text{ kg/s}$$

$$w_s = \frac{21000}{3600} = 5.833 \text{ kg/s}, \quad w_s - w = 4.990 \text{ kg/s}$$

$$wh_3 + (w_s - w)\, h_4 = w_s h_5$$

$$0.843 \times 2712.38 + 4.99 \times h_4 = 5.833 \times 2649.04$$

$$2286.54 + 4.99\, h_4 = 15451.85$$

$$h_4 = 2638.33 \text{ kJ/kg} = 584.33 + x_4 \times 2148.1$$

$$x_4 = 0.956 \qquad \textit{Ans.} \text{ (a)}$$

$$(W_T)_{HP} = (w_s - w)\,(h_1 - h_4) = 4.99 \times 231.37 = 1154.54 \text{ kW} \quad \textit{Ans.} \text{ (b)}$$

$$(h_{is})_{HP} = \frac{h_1 - h_4}{h_1 - h_{4s}} = \frac{2869.7 - 2638.33}{2869.7 - 2567.03} = \frac{231.37}{302.67}$$

$$= 0.7644 \quad \text{or,} \quad 76.44\% \qquad \textit{Ans.} \text{ (c)}$$

SHORT-ANSWER QUESTIONS

2.1 What do you understand by externally irreversible and internally irreversible Rankine cycle?

2.2 What is a pinch point? What is its effect on the size and efficiency of the steam generator?

2.3 What is the mean temperature of heat addition? What is its effect on cycle efficiency?

2.4 Explain the different methods by which the mean temperature of heat addition can be increased.

2.5 How is the maximum pressure of a steam cycle fixed up?

2.6 When does reheating of steam become necessary? Explain the effect of reheat on cycle output and efficiency.

2.7 How is the optimum reheat pressure arrived at?

2.8 Why are more than two reheats not used in practice?

2.9 Why is the thermal efficiency of a condensing steam power plant less in a warm region than in a cold region?

2.10 Why is the ideal regenerative cycle not practicable?

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2.11 Explain the effect of regeneration on steam cycle output and efficiency.
2.12 The use of regnerative feedwater heating increases the capital cost but reduces the operating cost of a steam power plant. Explain.
2.13 Give a comparative estimate of open and closed feedwater heaters.
2.14 What is heater drip?
2.15 What is the function of a steam trap?
2.16 What do you understand by terminal temperature difference? Can it be negative? Explain.
2.17 What is the effect of TTD on heater size and cycle efficiency?
2.18 What is a drain cooler?
2.19 Why is one of the feedwater heaters always an open heater? What is it called?
2.20 Explain the function of the deaerator. Why is deaerator installed at a large height from the basement?
2.21 Why is hydrazine injected at the suction of the boiler feed pump?
2.22 Explain the optimum degree of regeneration. How is it arrived at?
2.23 Show that regenerative feedwater heating improves the cycle efficiency.
2.24 What is a supercritical steam cycle?
2.25 What is the common unit size of a steam power plant? What will be its projected size in the next decade?
2.26 What is boiler efficiency?
2.27 Show that the overall plant efficiency is a product of five component efficiencies.
2.28 What do you understand by the efficiency of the auxiliaries, n_{aux}?
2.29 What is heat rate? What is the difference between net cycle heat rate and gross cycle heat rate?
2.30 What do you understand by cogeneration of power and process heat? Explain its thermodynamic advantage.
2.31 What is a back pressure turbine? What are its applications?
2.32 Explain cogeneration plant efficiency.
2.33 What is a pass-out turbine and when is it used?

PROBLEMS

2.1 A cyclic steam power plant is to be designed for a steam temperature at turbine inlet of 360 °C and an exhaust pressure of 0.08 bar. After isentropic expansion of steam in the turbine, the moisture content at the turbine exhaust is not to exceed 15%. Determine the maximum allowable steam pressure at the turbine inlet, and calculate the Rankine cycle efficiency for these steam conditions. Estimate also the mean temperature of heat addition.

[*Ans*. 16.83 bar, 31.7%, 187.5 °C]

2.2 A steam power station uses the following cycle: Steam at boiler outlet: 150 bar, 550 °C. Reheat at 40 bar to 550 °C. Condenser at 0.1 bar. Assuming ideal processes find (a) the quality at turbine exhaust, (b) the cycle efficiency, and (c) the steam rate.

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[*Ans.* (a) 0.88, (b) 43.9%, (c) 2.18 kg/kWh]

2.3 In a single-heater regenerative cycle the steam enters the turbine at 30 bar, 400 °C and the exhaust pressure is 0.10 bar. The feedwater heater is a direct contact type which operates at 5 bar. Find (a) the efficiency and the steam rate of the cycle, and (b) the increase in mean temperature of heat addition, efficiency and steam rate, as compared to the Rankine cycle (without regeneration). Neglect pump work.

[*Ans.* (a) 35.36%, 3.93 kg/kWh, (b) 27.4 °C, 1.18%, 0.47 kg/kWh]

2.4 A simple steam power cycle uses solar energy for the heat input. Water in the cycle enters the pump as a saturated liquid at 40 °C, and is pumped to 2 bar. It then evaporates in the boiler at this pressure, and enters the turbine as saturated vapour. At the turbine exhaust the conditions are 40 °C and 10% moisture. The flow rate is 150 kg/h. Determine (a) the turbine isentropic efficiency, (b) the net work output, (c) the cycle efficiency, and (d) the area of the solar collector needed if the collectors pick up 0.58 kW/m^2.

[*Ans.* (a) 76.7%, (b) 15.5 kW, (c) 12.78%, (d) 182 m^2]

2.5 In a nuclear power plant heat is transferred in the reactor to liquid sodium. The liquid sodium is then pumped to a heat exchanger where heat is transferred to steam. The steam leaves this heat exchanger as saturated vapour at 50 bar, and is then superheated in an external gas fired superheater to 600 °C. The steam enters the turbine, which has one extraction point at 4 bar, where steam flows to an open feedwater heater. The turbine efficiency is 75% and the condenser temperature is 40 °C. Determine the heat transfer in the reactor and in the superheater to produce a power output of 80 MW, and the thermal efficiency.

[*Ans.* 178.5 MW, 71.1 MW, 0.32]

2.6 In a reheat cycle, the initial steam pressure and the maximum temperature are 150 bar and 550 °C respectively. If the condenser pressure is 0.1 bar and the moisture at the condenser inlet is 15%, and assuming ideal processes, determine (a) the reheat pressure, (b) the cycle efficiency, and (c) the steam rate.

[*Ans.* (a) 13.5 bar, (b) 43.6%, (c) 2.05 kg/kWh]

2.7 In a reheat steam cycle, the maximum steam temperature is limited to 500 °C. The condenser pressure is 0.1 bar and the quality at turbine exhaust is 0.8778. Had there been no reheat the exhaust quality would have been 0.7592. Assuming ideal processes, determine (a) the reheat pressure, (b) the boiler pressure, (c) the cycle efficiency, and (d) the steam rate.

[*Ans.* (a) 30 bar, (b) 150 bar, (c) 50.51%, (d) 1.9412 kg/kWh]

2.8 In a reheat cycle steam at 500 °C expands in an h.p. turbine till it is saturated vapour. It is reheated at constant pressure to 400 °C and then expands in an l.p. turbine to 40 °C. If the maximum moisture content at the turbine exhaust is limited to 15%, find (a) the reheat pressure, (b) the boiler pressure, (c) the net specific work output, (d) the cycle efficiency, and (e) the steam rate. Assume all processes ideal.

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What would have been the quality at turbine exhaust, the net work output and the cycle efficiency, without the reheating of steam?

2.9 A regenerative cycle operates with steam supplied at 30 bar and 300 °C, and the condenser pressure is 0.08 bar. The extraction points for two heaters (one closed and one open) are 3.5 bar and 0.7 bar, respectively. Calculate the thermal efficiency of the plant neglecting pump work.

[*Ans*. 36%]

2.10 The net power output of an ideal reheat regenerative steam cycle is 80 MW. Steam enters the h.p. turbine at 80 bar, 500 °C and expands till it becomes saturated vapour. Some of the steam then goes to an open feedwater heater and the balance is reheated to 400 °C, after which it expands in an l.p. turbine to 0.07 bar. Compute (a) the reheat pressure, (b) the steam flow rate to the h.p. turbine, (c) the cycle efficiency, and (d) the rate of flow of cooling water in the condenser if the temperature rise of water is 8 °C, (e) If the velocity of steam flowing from the turbine to the condenser is limited to 130 m/s, find the diameter of the connecting pipe.

[*Ans*. (a) 6.5 bar, (b) 58.4 kg/s, (c) 43.7% (d) 3146.5 kg/s, (e) 2.97 m]

2.11 Steam is generated at 70 bar, 500 °C and expands in a turbine to 30 bar with an isentropic efficiency of 77%. At this condition, it is mixed with steam twice its mass at 30 bar, 400 °C. The mixture then expands with an isentropic efficiency of 80% to 0.06 bar. At a point in the expansion where the pressure is 5 bar, steam is bled for feedwater heating in a direct contact heater, which raises the feedwater to the saturation temperature of the bled steam. Calculate the mass of steam bled per kg of high pressure steam and the cycle efficiency. Assume that the L.P. expansion condition line is straight.

[*Ans*. 0.53 kg, 31.9%]

2.12 A certain chemical plant requires heat from process steam at 120 °C at the rate of 5.83 MW and power at the rate of 1 MW from the generator terminals. Both the heat and power requirements are met by a back pressure turbine of 80% brake efficiency (brake output/isentropic output) and 85% internal efficiency, which exhausts steam at 120 °C, dry and saturated. All the latent heat released during condensation is utilized in the process heater. Find the pressure and temperature of steam at inlet to the turbine. Assume 90% efficiency for the electric generator.

[*Ans*. 22.5 bar. 360 °C]

2.13 A 10 MW steam turbine operates with steam at 40 bar, 400 °C at the inlet and exhausts at 0.1 bar. 10,000 kg/h of steam at 3 bar are to be extracted for process work. The turbine has 75% isentropic efficiency throughout. Find the boiler capacity required.

[*Ans*. 13.74 kg/s = 49.46 t/h]

2.14 A pass-out two-stage turbine receives steam at 50 bar, 350 °C. At 1.5 bar, the h.p. steam exhausts and 12,000 kg of steam per hour are taken at this stage for process purposes. The remainder is reheated at 1.5 bar to 250 °C and then

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expanded through the l.p. turbine to a condenser pressure of 0.05 bar. The power output from the turbine unit is 3750 kW. Take isentropic efficiency for h.p. and i.p. stages as 0.84 and 0.81 respectively. Calculate the boiler capacity required.

[*Ans*. 18.5 t/h]

2.15 A 850 MW steam power plant operates with turbine inlet at 100 bar, 550 °C and condenser pressure at 0.05 bar. There are three feedwater heaters placed optimally as follows: (i) the h.p. heater is of the closed type with drains cascaded backward, (ii) the i.p heater is of the open type, and (iii) the l.p. heater is of the closed type with drains pumped forward. Each of the turbine sections has the same isentropic efficiency of 90%. The pumps have isentropic efficiencies of 80%. Calculate (a) the mass flow rate of steam at turbine inlet, (b) the mass flow rate of cooling water in the condenser, if it undergoes a 10 °C temperature rise, (c) the cycle efficiency, and (e) the cycle heat rate.

2.16 A coal-fired power plant has a turbine generator rated at 1000 MW gross. The plant requires 9% of this power for its internal operations. It uses 9800 tonnes of coal of heating value 26 MJ/kg per day. The steam generator efficiency is 86%. Calculate the gross and net station heat rates and efficiencies.

[*Ans*. 10.616 MJ/kWh, 11.667 MJ/kWh, 33.9%, 30.86%]

2.17 Steam is supplied to a pass-out turbine at 40 bar, 400 °C and dry saturated process steam is required at 4 bar. The l.p. stage exhausts at 0.1 bar and the condition line for the turbine expansion may be assumed to be straight. If the power load is 1000 kW and the maximum process load is 1.4 kW, estimate the maximum steam flow through the high and low pressure stages. Assume that the steam just condenses in the process plant.

[*Ans*. 3837.6 kg/h, 3635.3 kg/h]

2.18 A steam power plant has the boiler efficiency of 92%, turbine (mechanical) efficiency of 96%, and electric generator efficiency of 97%. If 7% of the power generated is consumed in running the auxiliaries of the plant and the overall plant efficiency is 34%, find the percentage of energy lost in the condenser.

[*Ans*. 57.4%]

2.19 An ideal steam power plant operates between 150 bar, 550 °C and 0.075 bar. It has seven feedwater heaters. Find the optimum pressures and temperatures at which the heaters operate.

2.20 In a cogeneration plant, 10^6 kg/h of steam at 80 bar, 480 °C expands in the h.p. turbine to 10 bar. From the exhaust 4×10^5 kg/h of steam is extracted for process heating. The remaining steam expands in the l.p. turbine to 0.08 bar. Saturated liquid at 0.08 bar leaving the condenser is pumped to 9.5 bar where it mixes with the condensate from the process heater leaving at 9.5 bar, 120°C. The entire flow is then pumped to 80 bar. The isentropic efficiencies of the turbines and the pumps are 86% and 80%, respectively. Determine (a)

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the heating load, in kJ/h, (b) the power developed by the turbines, in kW, and (c) the rate of heat transfer in the steam generator, in kJ/h.

[*Ans.* (a) 9.53 kJ/h, (b) 236500 kW, (c) 3.032×10^9 kJ/h]

2.21 In a reheat-regenerative steam cycle, steam enters the h.p. turbine at 80 bar, 480 °C and expands to 7 bar. The steam is then reheated to 440 °C before entering the l.p. turbine, where it expands to the condenser pressure of 0.08 bar. Steam extracted from the h.p. turbine at 20 bar is fed to a closed feedwater heater, from which the drain at 205 °C is cascaded to an open feedwater heater through a trap. Steam extracted from the l.p. turbine at 3 bar is also fed into the open feedwater heater, from which the total flow is pumped into the steam generator. The net power output of the cycle is 100 MW. Assuming ideal processes, determine (a) the cycle efficiency, and (b) the rate of steam generation in kg/h.

[*Ans.* (a) 43%, (b) 2.8×10^5 kg/h]

2.22 A pressurized water reactor power plant operates at pressures and temperatures as shown in Fig. P2.22. The steam leaving the heat exchanger is at 50 bar, dry saturated and the condensate is saturated to the saturation temperature of the bled steam at 5 bar in an open feed heater. Assuming all processes to be ideal and neglecting pump work estimate (a) the ratio of the working fluids in the two circuits, (b) the fraction of the steam supply which is bled from the turbine, and (c) the cycle efficiency.

[*Ans.* (a) 10 :1, (b) 0.22, (c) 39%]

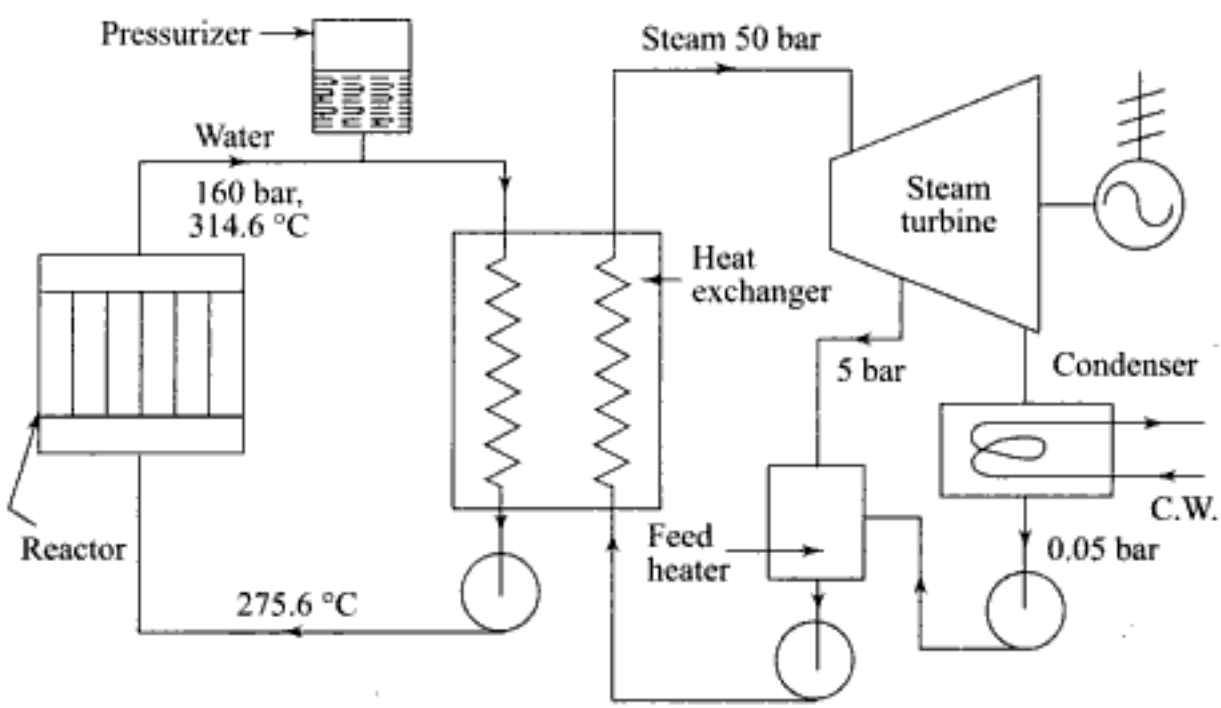

Fig. P2.22

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3

Combined Cycle Power Generation

3.1 FLAWS OF STEAM AS WORKING FLUID IN POWER CYCLE

There are certain drawbacks with steam as the working substance in a power cycle. The maximum temperature gained in steam cycles using the best available material is about 600 °C, while the critical temperature of steam is 375°C, which necessitates large superheating and permits the addition of only an infinitesimal amount of heat at the highest temperature.

High moisture content is involved during expansion of steam in going to higher steam pressures in order to obtain higher mean temperature of heat addition (T_{m_1}). The use of reheat thus becomes necessary. Since reheater tubes are costly and the steam plant layout becomes complex, the use of more than two reheats is hardly recommended. Also, as pressure increases, the metal stresses increase, and the increase in thicknesses of the walls of boiler drums, tubes, pipe-lines and so on is not in proportion to the pressure increase, but much faster because of the prevalence of high temperature.

It may be noted that high T_{m_1} is desired for high cycle efficiency. The need for high pressures is only forced due to weak characteristics of steam.

If the lower limit is now considered, it is seen that at the heat rejection temperature of 40 °C, the saturation pressure of steam is 0.075 bar, which is considerably lower than atmospheric pressure. The temperature of heat rejection can still be lowered using some refrigerant as a coolant in the condenser. The corresponding vacuum will be still higher, and to maintain such high vacuum in the condenser is a big problem.

It is the low temperature of heat rejection which is of real interest. The necessity for vacuum is a disagreeable characteristic of steam.

The saturated vapour line in the T–s diagram of steam is sufficiently inclined, so that when steam is expanded to lower pressures (for higher turbine

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output as well as cycle efficiency), it involves more moisture content, which is not desired from the consideration of the erosion of turbine blades in later stages.

3.2 CHARACTERISTICS OF IDEAL WORKING FLUID FOR VAPOUR POWER CYCLE

The desirable characteristics of the working fluid in a vapour power cycle to obtain the best thermal efficiency are given below:

(a) The fluid should have a high critical temperature so that the saturation pressure at the maximum permissible temperature (metallurgical limit) is relatively low. It should have a large enthalpy of vaporization at that pressure.
(b) The saturation pressure at the temperature of heat rejection should be above the atmospheric pressure so as to avoid the necessity of maintaining vacuum in the condenser.
(c) The specific heat of liquid should be small so that little heat transfer is required to raise the liquid to the boiling point.
(d) The saturated vapour line of *T–s* diagram should be steep, very close to the turbine expansion process so that excessive moisture does not appear during expansion.
(e) The freezing point of the fluid should be below the room temperature, so that it does not get solidified while flowing through the pipelines.
(f) The fluid should be chemically stable and should not contaminate the materials of construction at any temperature.
(g) The fluid should be non-toxic, non-corrosive, not excessively viscous, and low in cost.

The characteristics of such a hypothetical working fluid are approximated in the *T–s* diagram as shown in Fig. 3.1. Some superheat is desired to reduce piping losses and improve turbine efficiency. The bounded area of the cycle is almost a rectangle, and its thermal efficiency is very close to the Carnot efficiency.

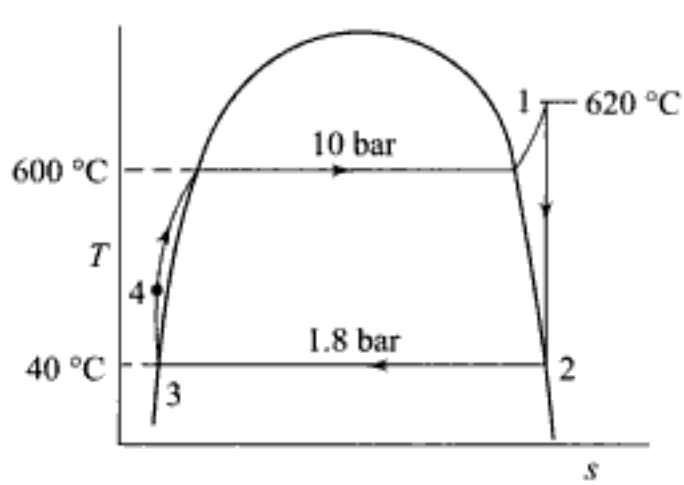

Fig. 3.1 *T-s diagram of an ideal working fluid for a vapour power cycle*

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3.3 BINARY VAPOUR CYCLES

No single fluid can meet all the requirements as mentioned above. Although in the overall evaluation, water is better than any other working fluid, at high temperatures, however, there are a few better fluids, and notable among them are: (a) diphenyl ether. $(C_6H_5)_2O$, (b) aluminium bromide $AlBr_3$ and (c) liquid metals like mercury, sodium, potassium and so on. Among these, only mercury has actually been used in practice. Diphenyl ether could be considered but it has not yet been used because like most organic substances, it decomposes gradually at high temperatures. Aluminium bromide is a possibility and yet to be considered.

As at pressure of 12 bar, the saturation temperatures for water, aluminium bromide, and mercury are 187 °C, 482.5 °C and 560 °C, respectively. The highest cyclic temperature consistent with the best available material for use in power plant is about 560 °C. Therefore, mercury is a better working fluid in the high temperature range because at 560 °C, its vaporization pressure is relatively low. Its critical pressure and temperature are 1080 bar and 1460 °C, respectively.

But in the low temperature range, mercury is unsuitable because its saturation pressure becomes exceedingly low, and it would be impractical to maintain such a high vacuum in the condenser. At 30 °C, the saturation pressure of mercury is only 2.7×10^{-4} cm Hg. Its specific volume at such a low pressure is very large, and it would be difficult to accommodate such a large volume flow.

For this reason, to take advantage of the beneficial features of mercury in the high temperature range and to get rid of its deleterious effects in the low temperature range, mercury vapour leaving the mercury turbine is condensed at a higher temperature and pressure, and the heat released during the condensation of mercury is utilized in evaporating water to form steam to operate on a conventional turbine.

Thus, in the binary (or two fluid) cycle, two cycles with different working fluids are coupled in series, the heat rejected by one being utilized in the other.

The flow diagram of mercury-steam binary cycle and the corresponding *T–s* diagram are given in Figs 3.2 and 3.3 respectively. The mercury cycle *a–b–c–d* is a simple Rankine cycle using saturated vapour. The heat rejected by mercury during condensation (process *b–c*) is transferred to boil water and form saturated vapour (process 5–6). The saturated vapour is heated from the external source (furnace) in the superheater (process 6–1). Superheated steam expands in the turbine and is then condensed. The condensate is then pumped to the economiser where it is heated till it becomes saturated liquid by the outgoing flue gases (process 4–5). The saturated liquid then goes to the mercury condenser–steam boiler, where the latent heat is absorbed. In an actual plant, the steam cycle is always a regenerative cycle with feedwater heating, but for the sake of simplicity, this complication has been omitted.

Let m represent the flow rate of mercury in the mercury cycle per kg of steam circulating in the steam cycle. Then, for 1 kg of steam,

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$$\left.\begin{aligned} Q_1 &= m(h_a - h_d) + (h_1 - h_6) + (h_5 - h_4), \qquad Q_2 = h_2 - h_3 \\ W_T &= m(h_a - h_b) + (h_1 - h_2), \\ W_P &= m(h_d - h_c) + (h_4 - h_3) \end{aligned}\right\} \tag{3.1}$$

$$h_{cycle} = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_P}{Q_1}$$

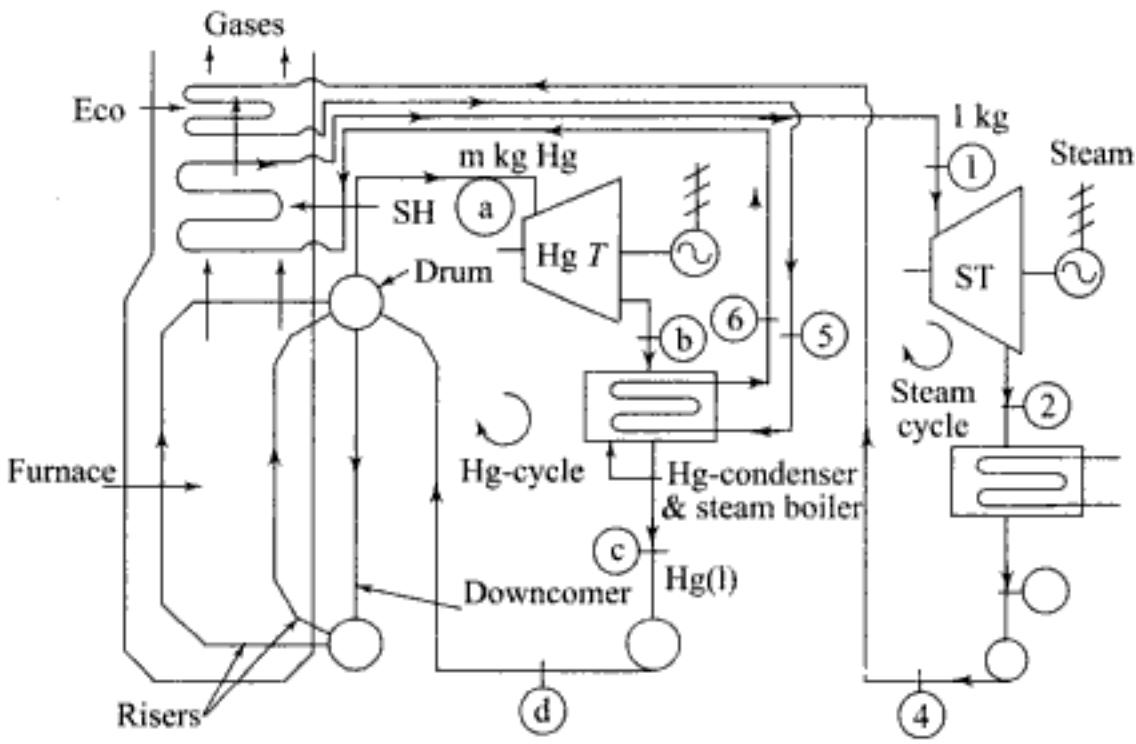

Fig. 3.2 *Flow diagram of mercury-steam binary cycle*

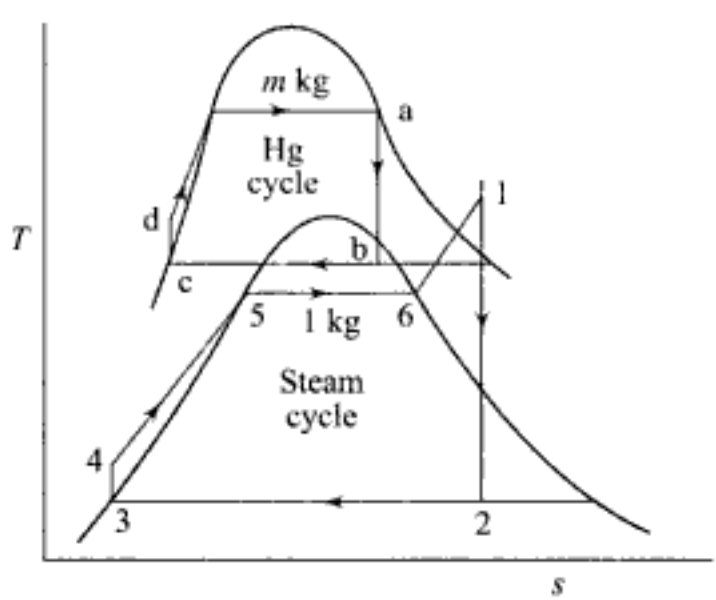

Fig. 3.3 *T–s diagram of mercury-steam binary cycle*

and steam rate (s.s.c) = $\dfrac{3600}{W_T - W_P}$ kg/kWh

The energy balance of the mercury condenser-steam boiler gives

$$m(h_b - h_c) = (h_6 - h_5)$$

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$$\therefore \qquad m = \frac{h_6 - h_5}{h_b - h_c} \text{ kg Hg/kg } H_2O \tag{3.2}$$

To vaporize one kg of water, 7 to 8 kg of mercury must condense.

The mercury-steam cycle first proposed by Emmet in 1925 [1] was in actual commercial use for more than four decades. A series of such plants were developed by the General Electric of the USA in the forties culminating in the installation of the Schiller 40 MW plant at Portsmouth. New Hampshire, which went on load in 1950. The thermal efficiencies of these plants are high (above 50%) and mercury makes up the necessary thermodynamic constraints. However, such plants have never attained wide commercial acceptance because there has always been the possibility of improving steam cycles by increasing pressure and temperature, and by using reheat and regeneration. Over and above, mercury is expensive, limited in supply and highly toxic. Because of the low latent heat of mercury, high mercury flow rates (about 8 times the steam flow rate) are needed. Special precautions are necessary to prevent leakage of mercury. Thus, the capital costs are high. Although the New Hampshire plant has now been dismantled due to higher overall cost per unit of electricity generated, it was the first major combined cycle plant and represented a great engineering achievement.

3.4 COUPLED CYCLES

The mercury steam cycle represents the two-fluid cycles where two Rankine cycles have been coupled in series. The mercury cycle is called the topping cycle and the steam cycle is called the bottoming cycle. If a sulphur dioxide cycle is added to it in the low temperature range, so that the heat released during the condensation of steam is utilized in forming sulphur dioxide vapour which expands in another turbine, then the mercury–steam–sulphur dioxide cycle is a three fluid or tertiary cycle. Similarly, other liquid metals, apart from mercury, like sodium or potassium, may be considered for a working fluid in the topping cycle. Apart from SO_2, other refrigerants (ammonia, freons etc.) may be considered as working fluids for the bottoming cycle. Figure 3.4 shows a sodium–mercury–steam tertiary cycle on T–s coordinates, having thus three cycles coupled in series.

3.4.1 Thermodynamics of Rankine–Rankine Coupled Cycles

If two Rankine cycles with two different working fluids are coupled in series, the heat lost by one is absorbed by the other (Fig. 3.5), as in the mercury–steam binary cycle. Let η_1 and η_2 be the efficiencies of the topping and bottom cycles, respectively, and η be the overall efficiency of the combined cycle.

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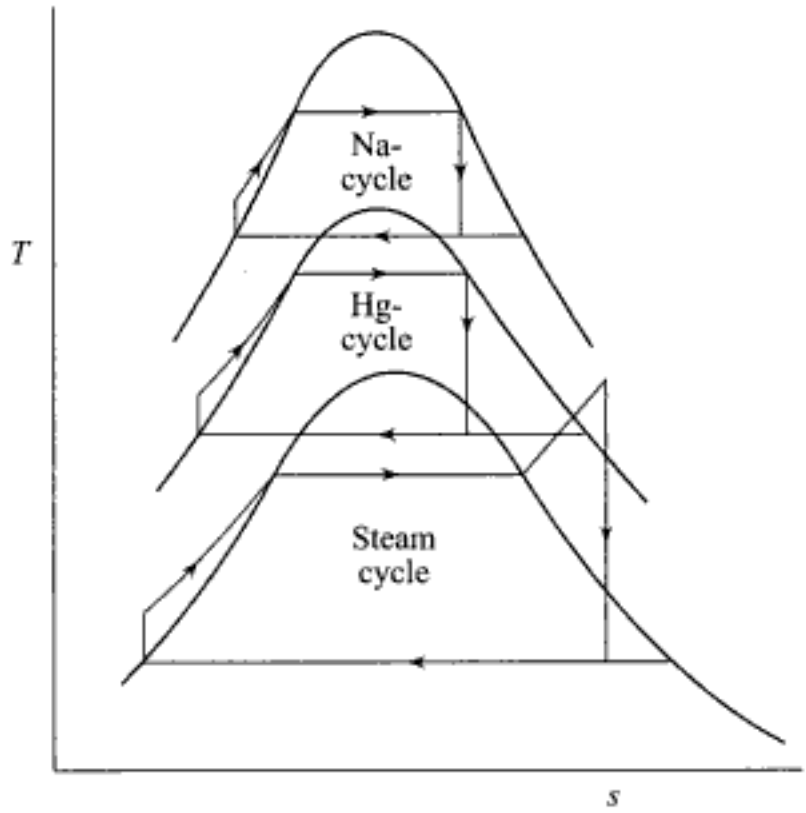

Fig. 3.4 *Sodium–mercury–steam tertiary cycle*

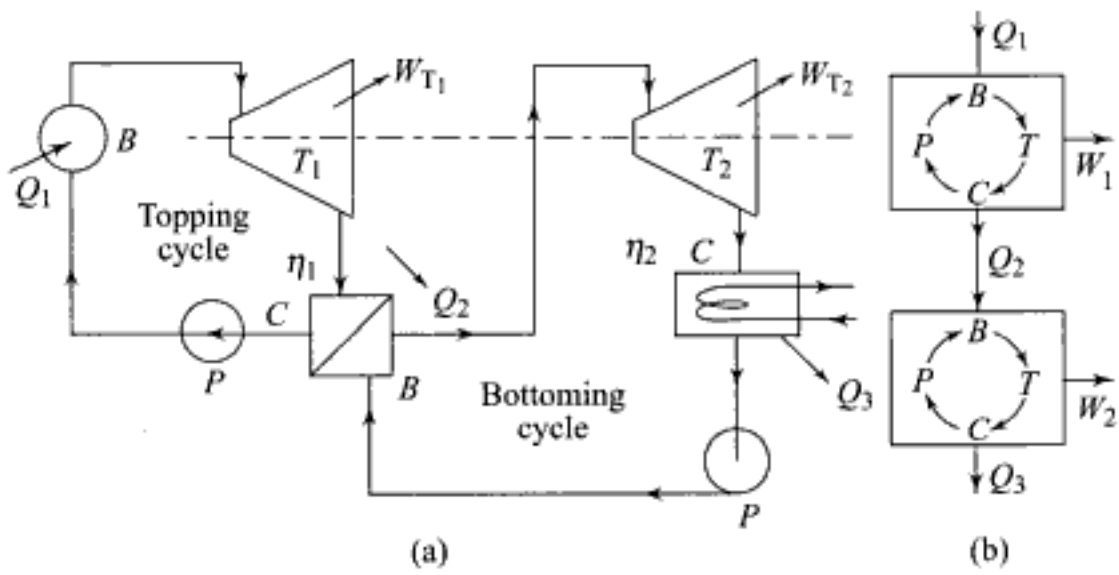

Fig. 3.5 *Two vapour power cycles coupled in series*

$$\eta_1 = 1 - \frac{Q_2}{Q_1} \quad \text{and} \quad \eta_2 = 1 - \frac{Q_3}{Q_2}$$

or $$Q_2 = (1 - \eta_1)Q_1 \quad \text{and} \quad Q_3 = Q_2(1 - \eta_2)$$

Now, $$\eta = 1 - \frac{Q_3}{Q_1} = 1 - \frac{Q_2(1-\eta_2)}{Q_1} = 1 - \frac{Q_1(1-\eta_1)(1-\eta_2)}{Q_1}$$

$\therefore$ $$\eta = 1 - (1 - \eta_1)(1 - \eta_2) \tag{3.3}$$

or $$1 - \eta = (1 - \eta_1)(1 - \eta_2) \tag{3.4}$$

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For n cycles coupled in series, the overall efficiency would be given by

$$1 - \eta = (1 - \eta_1)(1 - \eta_2) \ldots (1 - \eta_n)$$

or

$$1 - \eta = \prod_{i=1}^{n} (1 - \eta_i) \tag{3.5}$$

∴ Total loss = product of losses in all the cycles in series

For two-cycles coupled in series, from Eq. (3.3),

$$\eta = 1 - (1 - \eta_1 - \eta_2 + \eta_1 \eta_2)$$

∴

$$\eta = \eta_1 + \eta_2 - \eta_1 \eta_1 \tag{3.6}$$

Thus, the overall efficiency of two cycles coupled in series equals the sum of the individual efficiencies minus their product.

By combining two cycles in series, even if individual efficiencies are low, it is possible to achieve a fairly high combined efficiency, which cannot be attained by a single cycle. For example, if $\eta_1 = 0.50$ and $\eta_2 = 0.40$,

$$\eta = 0.5 + 0.4 - 0.5 \times 0.4 = 0.70$$

It is almost impossible to achieve such a high efficiency in a single cycle.

For sodium–mercury–steam tertiary cycle, if $\eta_1 = 0.5$. $\eta_2 = 0.4$ and $\eta_3 = 0.4$, from Eq. (3.3),

$$\eta = 1 - (1 - \eta_1)(1 - \eta_2)(1 - \eta_3) = 1 - 0.5 \times 0.6 \times 0.6 = 0.82$$

Of course, there are other losses which have not been considered.

There is thus great scope of improving cycle efficiency and achieving fuel economy using multifluid coupled cycles and utilizing the favourable characteristics of the working fluids in different temperature ranges for bulk power generation.

3.5 COMBINED CYCLE PLANTS

The maximum steam temperature in a power cycle does not exceed 600 °C, although the temperature in a dry bottom pulverized coal furnace is about 1300 °C. Therefore, there is a great thermal irreversibility and a decrease of availability because of heat transfer from combustion gases to steam through such a large temperature difference. By superposing a high temperature power plant as a topping unit to the steam plant, a higher energy conversion efficiency from fuel to electricity can be achieved, since the combined plant operates through a higher temperature range. Combined plants may be of the following types:

(i) Gas turbine–steam turbine plant
(ii) MHD–steam plant
(iii) Thermionic–steam plant
(iv) Thermoelectric–steam plant

MHD-steam, thermionic-steam and thermoelectric-steam power plants have been discussed in Chapter 13.

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3.6 GAS TURBINE—STEAM TURBINE POWER PLANT

The air standard cycle for a gas turbine power plant is the Brayton cycle which, like Rankine cycle, also consists of two reversible adiabatics and two reversible isobars, but unlike Rankine cycle the working fluid does not undergo phase change. The efficiency of the cycle is given by,

$$\eta_{\text{Brayton}} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}} \tag{3.7}$$

where r_p is the pressure ratio and γ is the specific heat ratio.

A gas turbine plant can be either open or closed. Simple, open gas turbine plant is shown in Fig. 3.6. Since the product of combustion is the working fluid which produces power by doing work on the blades of the gas turbine, it is an internal combustion plant. However, unlike the reciprocating internal combustion engine, the gas turbine is a steady flow device and the blades are always subjected to the highest gas temperature. To limit the maximum gas temperature to about 1200 K at inlet to the turbine consistent with the material used, a high air-fuel ratio is used. The disadvantages of a gas turbine power plant in a utility system are the following:

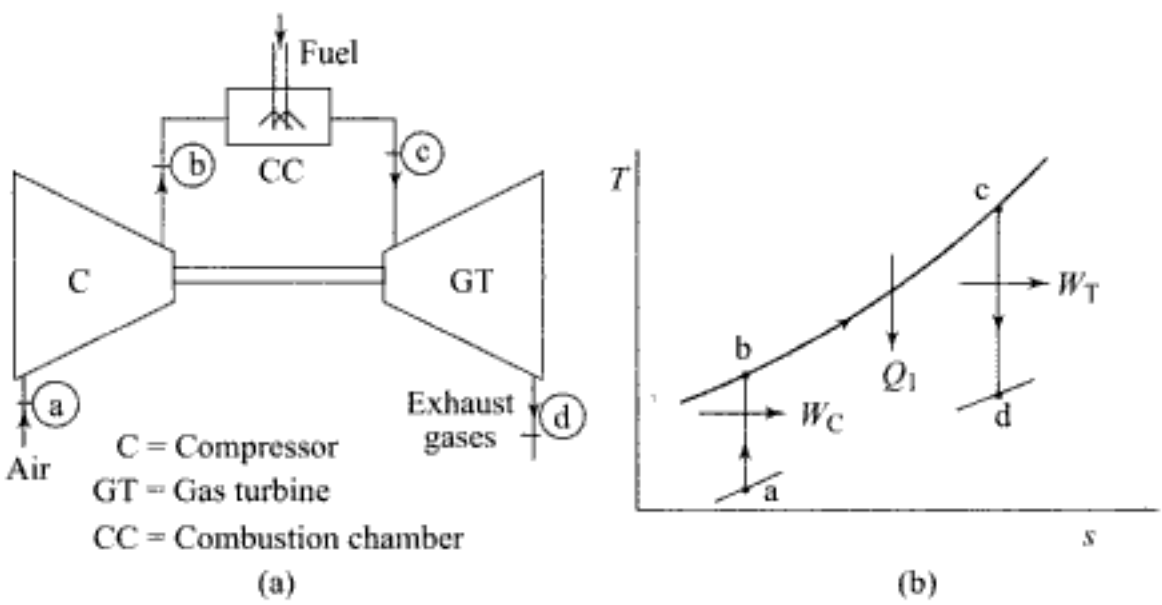

Fig. 3.6 *Open cycle gas turbine plant*

1. Large compressor work input, since the power required to drive the compressor is considerably higher than that required by a pump for the same pressure rise. The compressor thus consumes a large part of the work produced by the turbine.
2. Large exhaust loss, since the exhaust gas temperature is quite high and also the mass flow rate of gas is large due to high air–fuel ratio used.
3. Machine inefficiencies, since with the decrease in compressor efficiency (η_c) the work input to the compressor increases and with the decrease in

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turbine efficiency (η_T) the work output from the turbine decreases. At certain values of η_c and η_T, a situation may arise when the compressor consumes more power than what the turbine develops. So, the machine efficiencies of the compressor and the turbine have to be high enough to yield justifiable net work output.

4. Low cycle efficiency, due to the large exhaust loss, large compressor work and machine inefficiencies.
5. Costly fuel, since the cost of kerosene and other fuels used is much higher than that of coal. Its availability is also not always guaranteed.

Due to the above factors, the cost of power generated by a stationary gas turbine plant for a utility system is high. However, a gas turbine plant offers certain advantages also, as given below:

1. Less installation cost
2. Less installation time
3. Quick starting and stopping
4. Fast response to load changes

So, a gas turbine plant is often used as a peaking unit for certain hours of the day when the energy demand is high. A large steam plant designed to meet peak loads would operate at an uneconomical load factor during most of the year.

To overcome its low cycle efficiency, a gas turbine may be used in conjunction with a steam turbine plant in an utility base load station, to offer the utilities the gas turbine advantages of quick starting and stopping and permit flexible operation of the combined plant over a wide range of loads.

3.6.1 Thermodynamics of Brayton–Rankine Combined Cycle Plant

Let us consider two cyclic power plants coupled in series, the topping plant operating on Brayton cycle and the bottoming one operating on Rankine cycle (Fig. 3.7). Helium gas may be the working fluid in the topping plant and water in the bottoming plant. As shown in Section 3.4.1, the overall efficiency of the combined plant is given by Eq. (3.6)

$$\eta = \eta_1 + \eta_2 - \eta_1 \eta_2$$

where η_1 and η_2 are the thermal efficiencies of the Brayton cycle and the Rankine cycle, respectively.

1. Heat loss between two plants in series In the previous section, it was assumed that all the heat rejected by the topping plant is absorbed by the bottoming one. However, there is always some heat loss and the heat absorbed is always less than the heat rejected.

Let Q_L be the heat loss between the two plants (Fig. 3.8). The overall plant efficiency is

$$\eta = \frac{W_1 + W_2}{Q_1}$$

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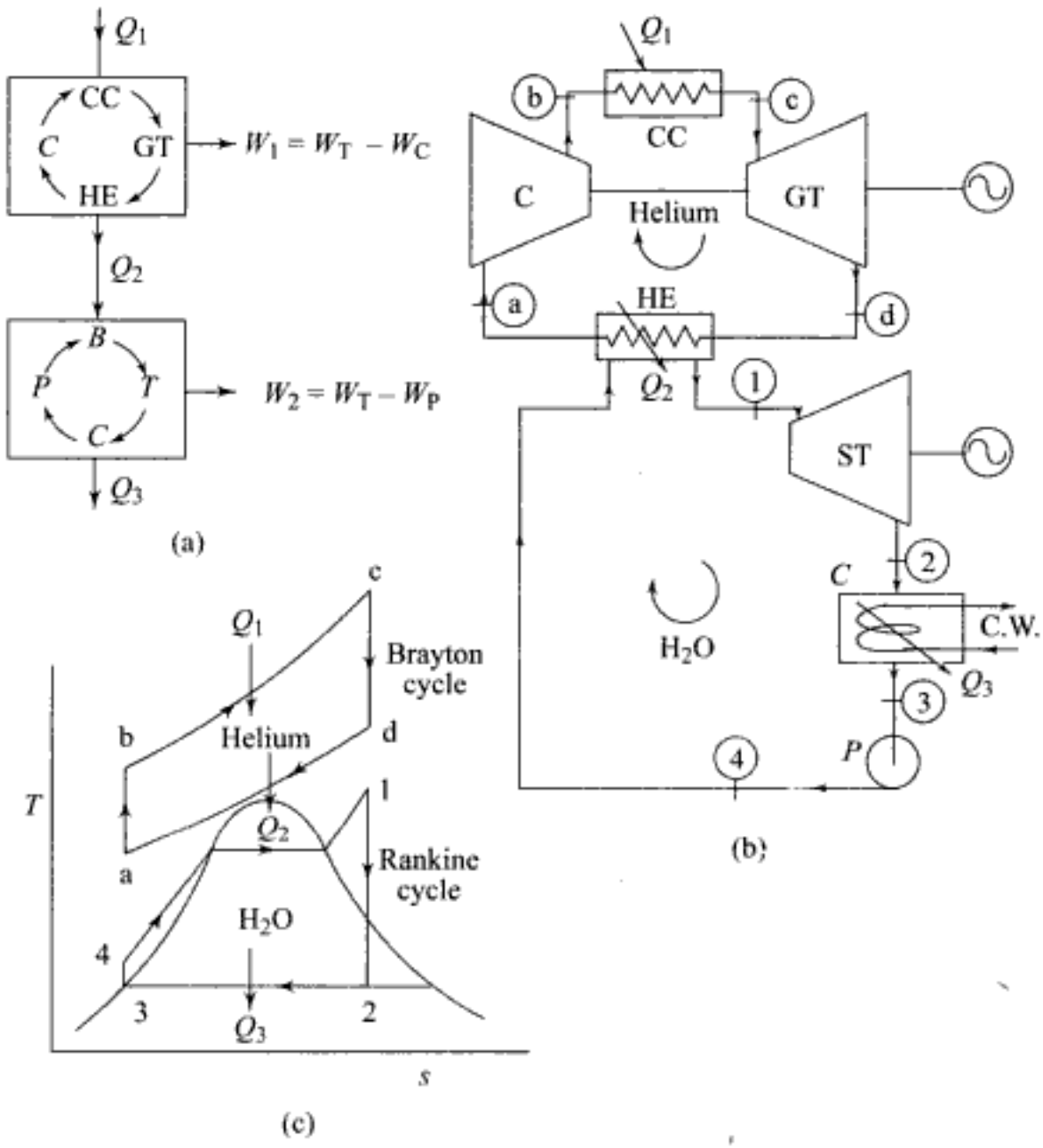

Fig. 3.7 *Brayton/Rankine combined cycle plant*

Now, $\eta_1 = \dfrac{W_1}{Q_1}$, $\eta_2 = \dfrac{W_2}{Q_3}$,

$$Q_3 = Q_2 - Q_L = Q_1(1 - \eta_1) - Q_L$$

Therefore,

$$\eta = \eta_1 + \eta_2 \frac{Q_3}{Q_1} = \eta_1 + \eta_2 [(1 - \eta_1) - Q_L/Q_1]$$

$$= \eta_1 + \eta_2 - \eta_1\eta_2 - \eta_2 x_L \qquad (3.8)$$

where x_L is the fraction of the heat supplied which is lost (Q_L/Q_1). The overall efficiency can also be expressed in a different way. The ratio of Q_3/Q_2 is the efficiency of the boiler in the bottoming plant, so that

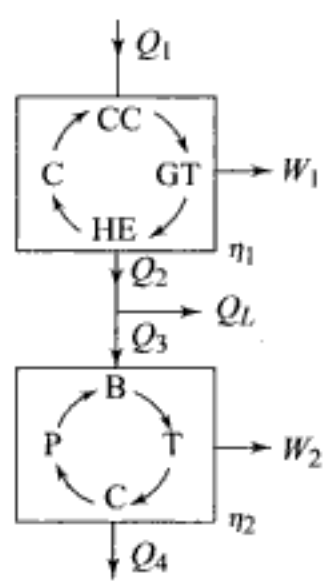

Fig. 3.8 *Heat loss in between two plants in series*

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$$\eta_B = \frac{Q_3}{Q_2} = 1 - \frac{Q_L}{Q_2}$$

Now,
$$\eta = \frac{W_1 + W_2}{Q_1} = \eta_1 + \frac{\eta_2 Q_3}{Q_1} = \eta_1 + \eta_2 \eta_B \frac{Q_2}{Q_1}$$
$$= \eta_1 + \eta_2 \eta_B (1 - \eta_1) = \eta_1 + \eta_2 \eta_B - \eta_1 \eta_2 \eta_B$$
$$= \eta_1 + (\eta_o)_2 - h_1 (\eta_o)_2 \tag{3.9}$$

where $(\eta_o)_2 = \eta_B$. η_2 is the overall efficiency of the bottoming plant.

2. Two cyclic plants operating in parallel Let us consider two cyclic plants operating in parallel, one on Brayton cycle and the other on Rankine cycle (Fig. 3.9). The total heat supplied Q_1 is divided between the two plants Q_2 and Q_4, so that $x_1 = Q_2/Q_1 = Q_2/(Q_2 + Q_4)$ is the fraction of the total heat taken by the plant operating on Brayton cycle. The net works done by the two plants are

$$W_1 = \eta_1 Q_2 \quad \text{and} \quad W_2 = \eta_2 Q_4$$

The overall efficiency of the combined plant is

$$\eta = \frac{W_1 + W_2}{Q_1} = \frac{W_1 + W_2}{Q_2 + Q_4} = \frac{\eta_1 Q_2 + \eta_2 Q_4}{Q_2 + Q_4}$$

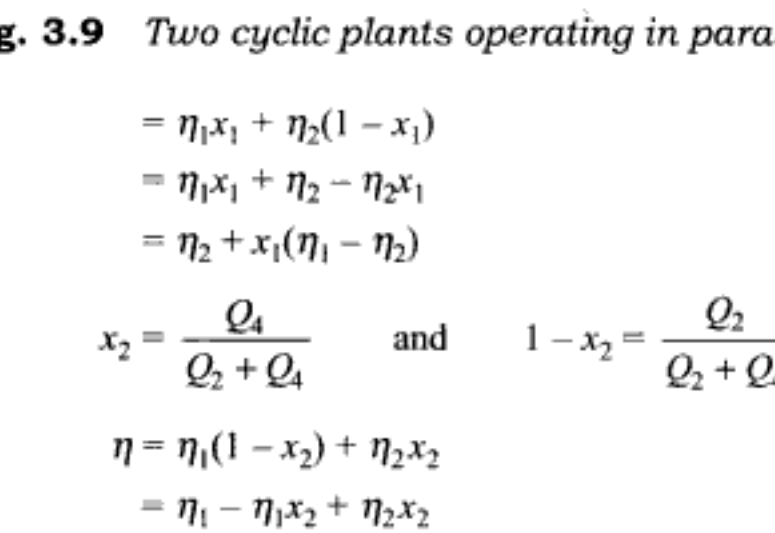

Fig. 3.9 *Two cyclic plants operating in parallel*

$$= \eta_1 x_1 + \eta_2 (1 - x_1)$$
$$= \eta_1 x_1 + \eta_2 - \eta_2 x_1 \tag{3.10}$$
$$= \eta_2 + x_1 (\eta_1 - \eta_2)$$

If
$$x_2 = \frac{Q_4}{Q_2 + Q_4} \quad \text{and} \quad 1 - x_2 = \frac{Q_2}{Q_2 + Q_4}$$

then
$$\eta = \eta_1 (1 - x_2) + \eta_2 x_2$$
$$= \eta_1 - \eta_1 x_2 + \eta_2 x_2 \tag{3.11}$$
$$= \eta_1 - x_2 (\eta_1 - \eta_2)$$

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If $\eta_1 > \eta_2$, then from Eq. (3.10), $\eta > \eta_2$, but from Eq. (3.11), $\eta < \eta_1$. Therefore, η lies between η_1 and η_2. There is, thus, no advantage to the parallel system. If the cyclic plant 1 operating on Brayton cycle could absorb more heat, say, equal to $(Q_2 + Q_4)$, then it would be advantageous to use that plant alone.

3. Series-parallel plants with two cyclic plants in series having supplementary firing Let the fraction of the total heat supplied that is used for supplementary heating (Fig. 3.10) be

$$x_2 = \frac{Q_4}{Q_1} \quad \text{so that} \quad 1 - x_2 = \frac{Q_2}{Q_1} = \frac{Q_2}{Q_2 + Q_4}$$

Now, $$W_1 = \eta_1 Q_2, \qquad Q_3 = (1 - \eta_1)Q_2, \qquad W_2 = \eta_2 Q_5$$

Therefore, the overall plant efficiency is given by

$$\begin{aligned}
\eta &= \frac{W_1 + W_2}{Q_1} = \frac{\eta_1 Q_2 + \eta_2 Q_5}{Q_2 + Q_4} \\
&= \eta_1(1 - x_2) + \eta_2 \frac{Q_4 + Q_3}{Q_2 + Q_4} \\
&= \eta_1(1 - x_2) + \eta_2\left[x_2 + (1 - \eta_1)\frac{Q_2}{Q_1}\right] \\
&= \eta_1(1 - x_2) + \eta_2[x_2 + (1 - \eta_1)(1 - x_2)] \\
&= \eta_1 + \eta_2 - \eta_1\eta_2 - x_2\eta_1(1 - \eta_2) \qquad (3.12)
\end{aligned}$$

Therefore, the overall efficiency of a series–parallel plant is less than that of two coupled cycles in series since the last term in Eq. (3.12) is positive. In absence of supplementary heating, i.e. when $x_2 = 0$, the overall efficiency reduces to that of an ideal series plant.

4. Series-parallel plants with two plants in series having supplementary heat supply and heat loss in between the two plants Let x_2 and x_L be the fractions of heat supply (Q_1) which represent supplementary heat supply (Q_4) and heat loss to the surroundings (Q_L), respectively (Fig. 3.10).

$$x_2 = \frac{Q_4}{Q_1} = \frac{Q_4}{Q_2 + Q_4}, \quad x_L = \frac{Q_L}{Q_1} = \frac{Q_L}{Q_2 + Q_4}, \quad \eta_1 = 1 - \frac{Q_3}{Q_2}, \quad W_1 = \eta_1 Q_2$$

$$W_2 = \eta_2 Q_5, \quad Q_5 = Q_4 + Q_3 - Q_L, \quad Q_3 = (1 - x_2)(1 - \eta_1)Q_1$$

So, the overall efficiency of the combined plant becomes

$$\begin{aligned}
\eta &= \frac{W_1 + W_2}{Q_1} = \frac{\eta_1 Q_2 + \eta_2 Q_5}{Q_2 + Q_4} \\
&= \eta_1(1 - x_2) + \eta_2 \frac{Q_4 + Q_3 - Q_L}{Q_2 + Q_4}
\end{aligned}$$

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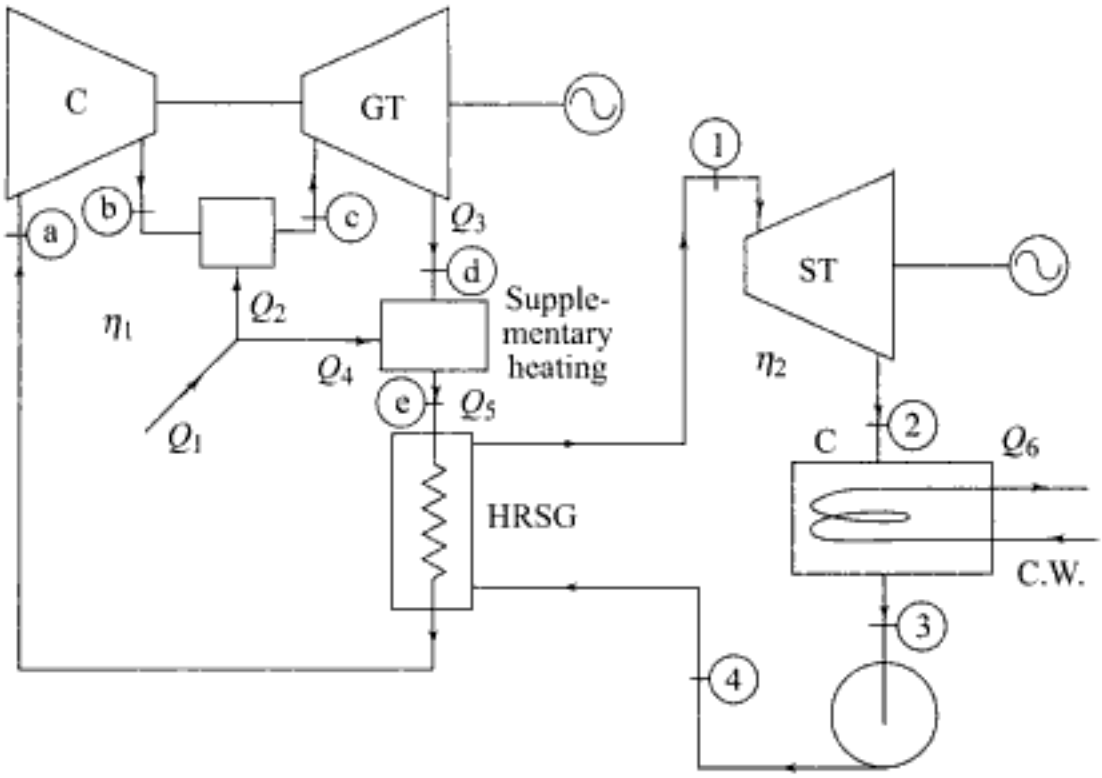

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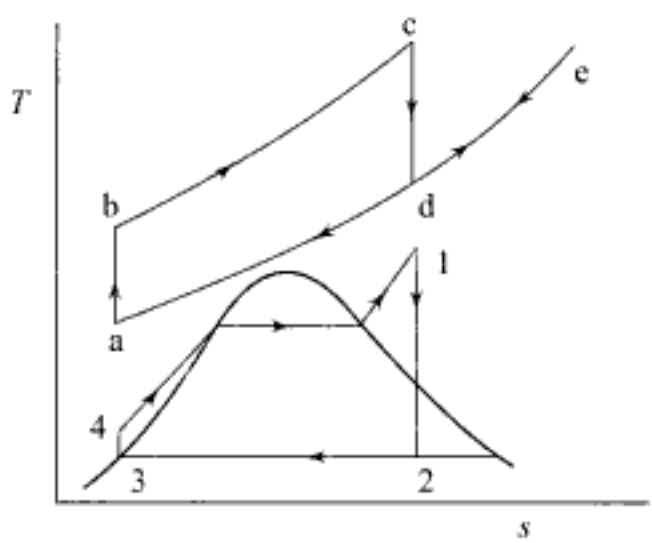

(b)

Fig. 3.10 *Brayton/Rankine cyclic plants with supplementary heating*

$$= \eta_1(1 - x_2) + \eta_2 \left[x_2 + \frac{(1-\eta_1)Q_2}{Q_2 + Q_4} - x_L \right]$$

$$= \eta_1(1 - x_2) + \eta_2[x_2 + (1 - \eta_1)(1 - x_2) - x_L]$$

$$= \eta_1 + h_2 - \eta_1\eta_2 - x_L\eta_2 - \eta_1 x_2(1 - \eta_2) \tag{3.13}$$

If $\quad x_2 = 0,\ \eta = \eta_1 + \eta_2 - \eta_1\eta_2 - x_L\eta_2$

which is the same as Eq. (3.8) derived earlier. It can also be expressed in a different way by defining the boiler efficiency

$$\eta_B = \frac{Q_5}{Q_5 + Q_1} = 1 - \frac{Q_L}{Q_5 + Q_1} = 1 - \frac{x_L Q_1}{Q_4 + Q_3 - Q_L + Q_1}$$

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$$= 1 - \frac{x_L}{x_2 + (1 - x_2)(1 - \eta_1)}$$

or
$$x_L = (1 - \eta_B)[x_2 + (1 - x_2)(1 - \eta_1)] \tag{3.14}$$

Therefore, from Eq. (3.13) the overall plant efficiency of the combined plant becomes

$$\eta = \eta_1 + \eta_2 - \eta_1\eta_2 - \eta_2(1 - \eta_B)[x_2 + (1 - x_2)(1 - \eta_1)] - \eta_1 x_2(1 - \eta_2)$$
$$= \eta_1(1 - x_2) + (\eta_0)_2[x_2 + (1 - x_2)(1 - \eta_1)] \tag{3.15}$$

where $(\eta_0)_2 = \eta_B\eta_2$ is the overall efficiency of the bottom cycle. In absence of supplementary heating, $x_2 = 0$ and

$$\eta = \eta_1 + (\eta_0)_2(1 - \eta_1) = \eta_1 + (\eta_o)_2 - \eta_1(\eta_o)_2$$

which is the same as Eq. (3.9) derived earlier.

3.6.2 Effect of Supplementary Heating on the Overall Efficiency of the Combined Gas Turbine–Steam Turbine Power Plant

Some thermodynamic concepts of different arrangements of combined cycles have been discussed above. We would now consider a practical gas turbine–steam turbine (GT–ST) power plant where the exhaust from the gas turbine is utilized to generate steam and operate a conventional steam power plant. We will examine how the supplementary heating, i.e. burning additional fuel into the oxygen rich exhaust of the gas turbine, affects the overall efficiency of the combined plant.

Fuel is burnt partially first in the combustion chamber (CC) of the gas turbine plant (Q_1) and then heat recovery steam generator (HRSG) (Q_2). If the net work outputs of the GT and the ST plants be W_1 and W_2, respectively, and η_1 and η_2 are their corresponding efficiencies, then the overall efficiency is

$$\eta = \frac{W_1 + W_2}{Q_1 + Q_2} = \frac{\eta_1 Q_1 + \eta_2[(1 - \eta_1)Q_1 + Q_4]}{Q_1 + Q_2}$$

Now,

$$\frac{\partial\eta}{\partial Q_2} = \frac{1}{(Q_1 + Q_2)^2}\left[(Q_1 + Q_2)\left\{(1 - \eta_1)Q_1\frac{\partial\eta_2}{\partial Q_2} + Q_2\frac{\partial\eta_2}{\partial Q_2} + \eta_2\right\}\right.$$
$$\left. - \{\eta_1 Q_1 + \eta_2[(1 - \eta_1)Q_1 + Q_4]\}\right]$$

$$= \frac{(1 - \eta_1)Q_1\dfrac{\partial\eta_2}{\partial Q_2} + Q_2\dfrac{\partial\eta_2}{\partial Q_2} + \eta_2}{Q_1 + Q_2} - \frac{\eta_1 Q_1 + \eta_2[(1 - \eta_1]Q_1 + Q_4]}{(Q_1 + Q_2)^2}$$

Additional firing of fuel in the HRSG improves the overall efficiency if $\partial\eta/\partial Q_2 > 0$,

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or $$\{(1-\eta_1)Q_1 + Q_2\}\frac{\partial\eta_2}{\partial Q_2} + \eta_2 > \frac{\eta_1 Q_1 + \eta_2[(1-\eta_1)Q_1 + Q_4]}{(Q_1 + Q_2)}$$

Since the right hand expression is the overall efficiency of the plant,

$$\{Q_1(1-\eta_1) + Q_2\}\frac{\partial\eta_2}{\partial Q_2} > (\eta - \eta_2)$$

Now, the left hand expression inside the bracket represents the total heat input to the HRSG, which is equal to W_2/η_2, and the inequality becomes

$$\frac{W_2}{\eta_2}\frac{\partial\eta_2}{\partial Q_2} > (\eta - \eta_2)$$

or $$\frac{\partial \ln \eta_2}{\partial Q_2} > \frac{\eta - \eta_2}{W_2} \tag{3.16}$$

With additional firing of the fuel in the HRSG, the power output of the steam cycle (W_2), as well as its efficiency (η_2) increase and so the increase in the overall efficiency diminishes. This was also confirmed in Section 3.6.1 for a coupled Brayton–Rankine cycle plant. Therefore, *supplementary firing is becoming less and less attractive*. Generally it is more profitable to burn the fuel in the combustor of the gas turbine plant itself since the heat is supplied to the system at a temperature higher than that in the steam cycle.

For the compression process of air, (Fig. 3.11),

$$\frac{T_b}{T_a} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} \qquad \text{where } \gamma = 1.4$$

For the expansion of combustion gases in the gas turbine

$$\frac{T_c}{T_d} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} \qquad \text{where } \gamma = 1.3 \text{ (may be assumed)}$$

$$W_1 = (W_{net})_{GT} = w_a\,[c_{pg}\,(T_c - T_d) - c_{pa}(T_b - T_a)] \tag{3.17}$$

neglecting the mass of fuel (for a high air–fuel ratio), and w_a being the mass flow of air.

$$(W_T)_{ST} = w_s(h_1 - h_2) \tag{3.18}$$

where w_s is the steam flow rate.
By energy balance,

$$w_a c_{pg}\,(T_e - T_f) = w_s(h_1 - h_4) \tag{3.19}$$

Now,

$$Q_1 = w_a\,c_{pg}\,(T_c - T_b + T_e - T_d) \tag{3.20}$$

and the overall efficiency is

$$\eta = \frac{(W_{GT})_{net} + W_{ST}}{Q_1}$$

neglecting the pump work.

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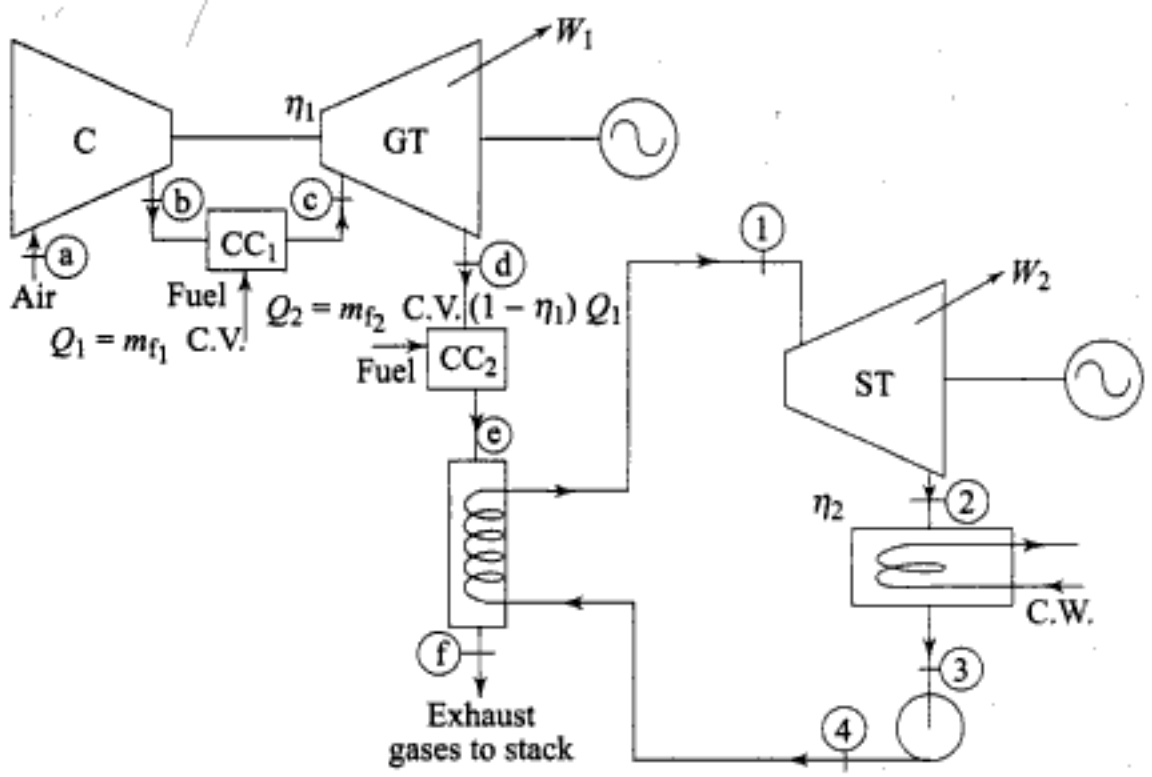

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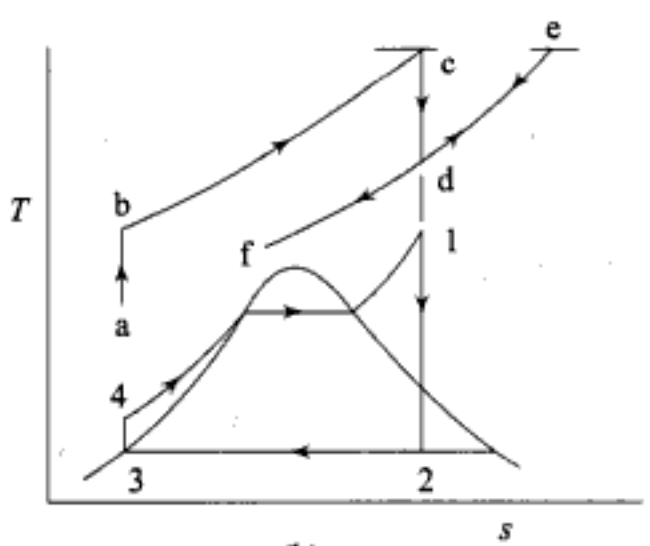

(b)

Fig. 3.11 *GT–ST combined plant with supplementary combustion of fuel*

As inlet temperatures to gas turbine keep increasing (due to the use of better material and blade cooling), the importance of supplementary firing diminishes further. However, supplementary firing may provide increased operating and fuel flexibilities in CC plants, which may fall into the following two categories.

1. Combined cycle plants with limited supplementary firing Supplementary firing raises the temperature of the exhaust gas to 800 to 900 °C. Relatively high flue gas temperature raises the condition of steam (84 bar, 525 °C), thereby improving the efficiency of the steam cycle. Following is given an example (Kehlhoffer 1991):

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Fuel	:	Natural gas
GT output	:	69.1 MW
ST output	:	78.3 MW
Station service power	:	2.1 MW
Heat input to GT (LHV)	:	230.0 MW
Heat input to the supplementary firing unit (LHV)	:	79.6 MW
Efficiency of the GT cycle	:	30.0%
Energy in the exhausts	:	159.3 MW
Efficiency of the steam cycle	:	32.9%
Net efficiency of the plant	:	46.9%

2. CC plants with maximum supplementary firing

Maximum supplementary firing refers to the maximum fuel that can be fired with the oxygen available in the GT exhaust. The steam cycle is conventional with reheat and regeneration. Hot GT exhaust is used as the combustion air. Regenerative air preheater is not required. However, a fresh air fan, which makes it possible to operate the steam plant even when the GT is not in operation, increases the availability of the unit.

The use of large supplementary firing in CC systems with high GT inlet temperatures causes the efficiency to drop. For this reason, CC plants with maximum supplementary firing are only of minimal importance today, in comparison to simple CC installations. They have, however, two advantages: (a) Coal can be burnt in the steam generator as supplementary fuel, (b) It has very good part-load efficiency.

3.6.3 Advantages of Combined Cycle Power Generation

The worldwide demand for combined cycle power plants is growing dramatically, with some experts forecasting explosive growth over the next decade [2]. In its basic form, a gas turbine exhausting into a heat recovery steam generator (HRSG) that supplies steam to a steam turbine cycle is the most efficient system of generating electricity today. Thanks to defence and aircraft research and development programs, the gas turbine technology has forged ahead considerably during the last two decades with unit plant efficiencies of 30 to 35%, comparable to the state-of-the-art fossil-fired power stations. The advantages of combined gas-steam cycles may be summarized as follows:

1. High overall plant efficiency Efficiencies exceeding 50% can be attained.

2. Low investment costs Because 2/3 of the output is produced in a GT and only 1/3 in the simple ST, the investment costs required are approximately 30% less than those for a conventional steam power plant.

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3. Small amount of water required The amount of cooling water required is only about 40 to 50% as much as for a steam plant.

4. Great operating flexibility The simple steam cycle makes it possible to start-up and shut-down the plants quickly, which also affects efficiency in a positive direction (reducing start-up losses).

5. Phased installation Because the gas turbines can go into operation much sooner than the steam plant, installation in stages is possible. The gas turbine plant can keep on generating power as the steam plant is under construction. This makes it possible to adjust the growth in demand for energy in a grid. Later, a coal gasification unit can be installed if there is too sharp an increase in the price of oil or gas.

6. Simplicity of operation It is simpler to run than a conventional steam power plant. Moreover, because combined cycle plants are generally operated fully automatically, they are especially suitable for use where operating staff is less experienced.

7. Low environmental impact Gas burning combined cycle (CC) plants in particular are ideally suitable for use in heavily populated regions because of their high efficiency and low emission levels of pollutants. In particular, very low NOx levels of clean CC plants are one of their most attractive features. Furthermore, gas-fired CC plants produce per kWh only 40% of the CO_2 produced by a coal-fired plant [25].

8. Advantages for cogeneration of heat and electricity

The good thermodynamic properties of CC plants are highly suitable for cogeneration of heat and electricity. Electrical yields of more than 40% are quite common in heating or industrial power plants with a back pressure turbine.

Large output combined with high cycle efficiency, low emission level and low investment cost are the main attractive features of the CC power generation.

By dividing the expansion process into two temperature ranges, 1100–550 °C in the gas turbine and 550 °C to ambient temperature in the steam turbine, high overall efficiencies exceeding 50% are achieved.

3.6.4 Dual Pressure Steam Cycle

Rankine cycle is required to be split in order to decrease the irreversibility involved in the heat transfer between the hot exhaust gas source and the water/steam. Figure 3.12(a) shows a t, q diagram indicating heat flow from the flue gas to the water steam cycle. It can be seen that more heat can be extracted from the exhaust gas if the steam pressure is lower (p_2). The integral $\int(1 - T_o/T)\,dq$ describes the maximum exergy, where the ratio T_o/T is a measure of the exergy loss. In Fig. 3.12(b), T_o/T is plotted as a function of q. It is seen that at high steam pressure, the temperature difference or the irreversibility is reduced in the high temperature region (area C), and the losses are dominant in the low temperature region (area A). At low steam pressure, the losses in the high

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temperature range (area C) prevail. Thus, there is an optimum steam pressure when the sum of areas $A + B + C$ is at a minimum, thus minimizing the exergy loss.

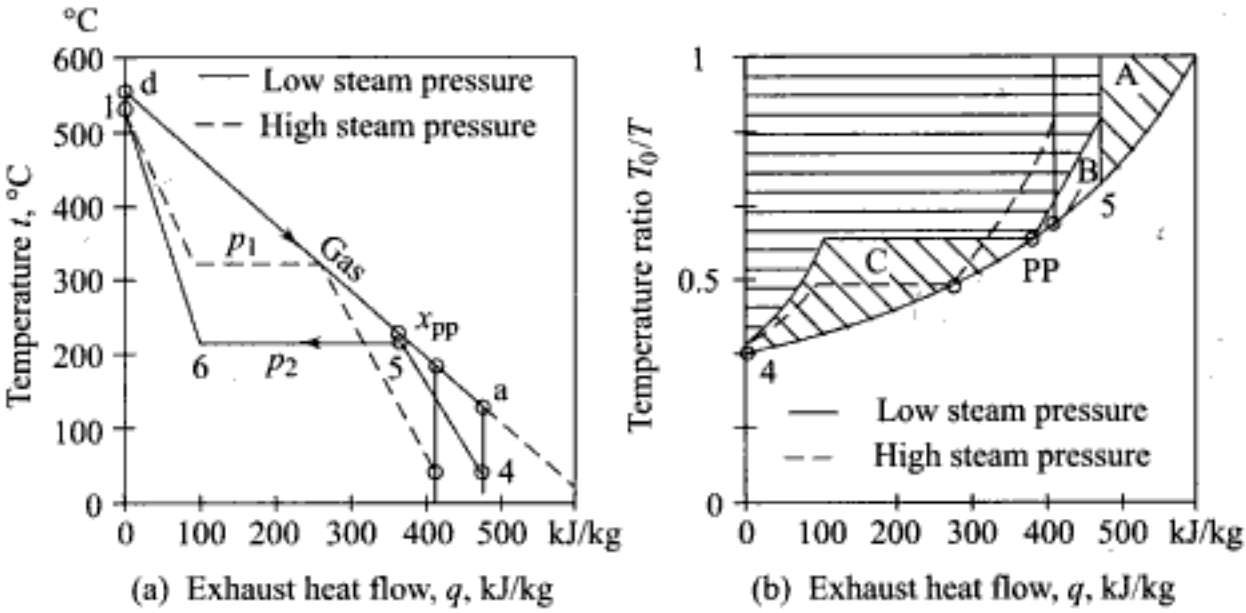

Fig. 3.12 *Effect of live steam pressure on energy utilization and overall energy loss*

In Fig. 3.12(a), the temperature of the exhaust gases cools from d to a as the temperature of the water–steam rises from 4 to 1. The smallest temperature differences between the two streams occur at the "*pinch points*" x–5 and d–1, sometimes called "*the temperature approach*". Normally, the minimum allowable value of the temperature approach is 20 K (°C) for reasons of satisfactory control, more commonly 30 K.

Figure 3.13 shows a schematic diagram of a dual pressure steam generator which is used to supply a two-stage steam turbine (HP and LP stages). There are two separate boilers, within the boiler casing, namely, the high pressure (HP) boiler which supplies the HP turbine, the exhaust steam from the HP stage of the turbine is then directed to the LP turbine. Steam from the low pressure (LP) boiler is directed to the LP stage of the turbine. Each boiler comprises the usual economiser, evaporator and superheater sections. Figure 3.13(c) is a repetition of Fig. 3.12(a) but is applied to a dual pressure steam cycle. By comparing Figs 3.12(a) and 3.13(c), it will be seen that the temperature difference between the gas side and steam side in the dual pressure cycle is considerably smaller than in the single pressure cycle. This gives the dual pressure cycle a considerable thermodynamic advantage since the smaller the temperature difference between the two streams for a given heat trasfer rate, the smaller the degree of irreversibility external to the steam cycle, hence the greater the amount of work which may be able to be produced. However, the smaller the mean temperature difference, the larger the amount of heat transfer surface that would be required to effect the heat transfer rate.

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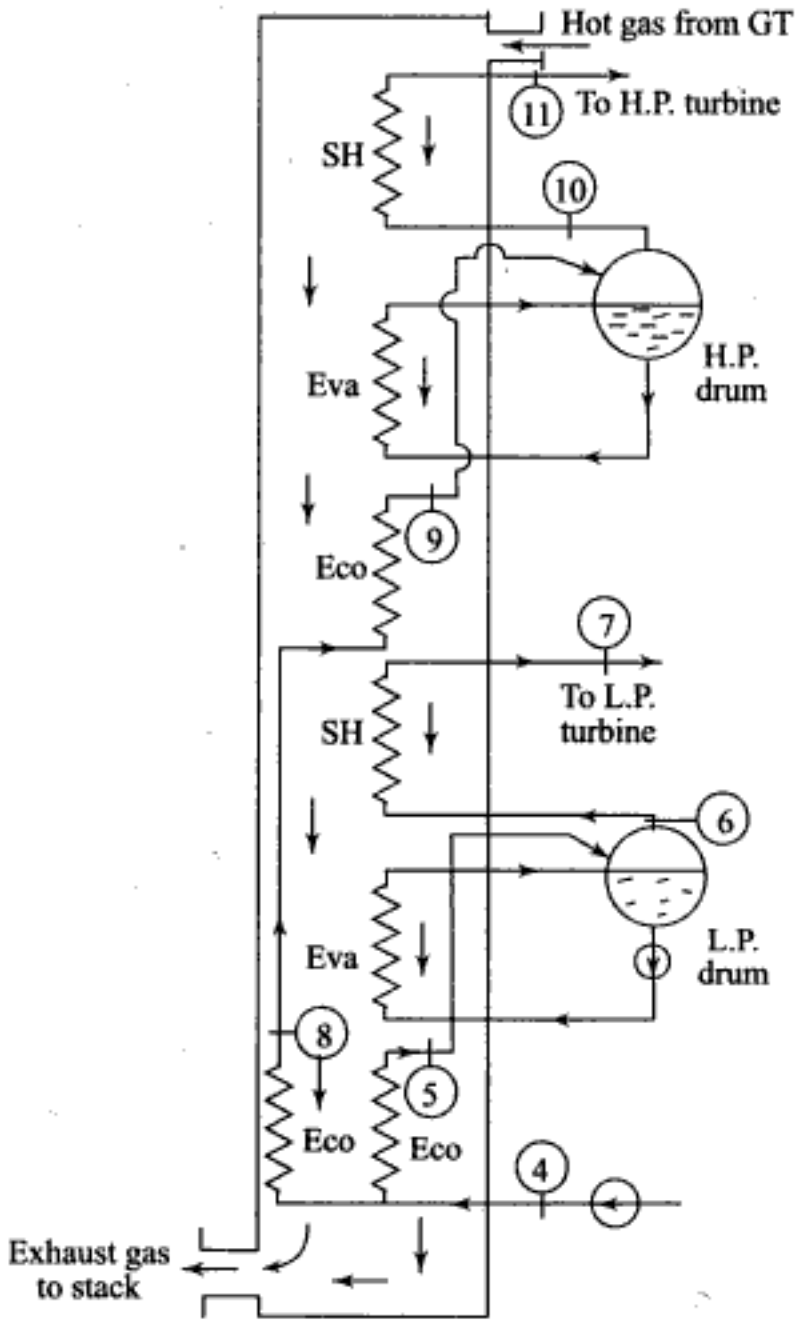

(a)

Fig. 3.13 *Contd*

3.6.5 Example of GT–ST Combined Cycle Plant

For simplicity, let us consider a modern gas turbine burning natural gas whose exhaust gases are used to raise steam, which is then used to drive a suitable turbine which drives an alternator. The relevant data for the gas turbine [6] is given in Table 3.1.

If the fuel gas has a calorific value of 48 MJ/kg, the fuel-to-electricity efficiency of the gas turbine is about 33%.

Figures 3.14.1(a) and 3.14.2(a) show the panoramic views of Doswell and Teesside combined cycle power plants, while Figs. 3.14.1(b) and 3.14.2(b) give the details of their respective steam generators (HRSG).

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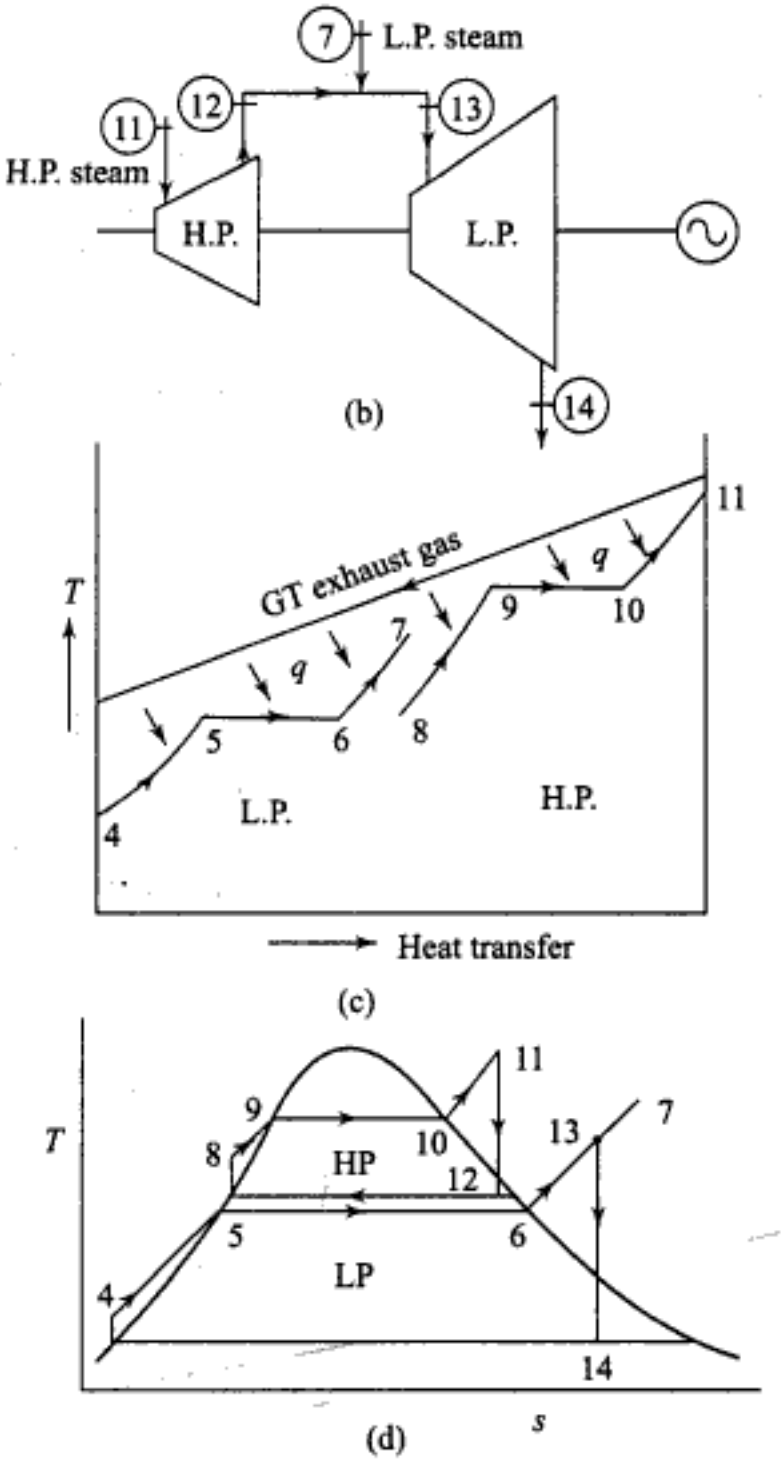

Fig. 3.13 *Dual pressure steam cycle (a) HRSG, (b) H.P. and L.P. turbines, (c) T–q diagram, (d) T–s diagram*

Table 3.1 *Basic parameters for gas turbine of combined cycle plant*

1.	Gas inlet temperature to turbine	1060 °C
2.	Pressure ratio	10.7
3.	Mass flow of air	495 kg/s
4.	Mass flow of fuel/gas	9.2 kg/s
5.	Gas outlet temperature from turbine	533 °C
6.	Electrical power output	144 MW

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Single pressure cycle (Fig. 3.12) Assuming c_p of the exhaust gases from the gas turbine to be 1.114 kJ/kg K and that the gases are cooled to 107 °C (to give a sufficient working margin above dew point temperature), the allowable temperature drop of the exhaust gases is (553–107) or 446 K, and the rate of heat transfer from the exhaust gases to the steam generator.

$$Q_1 = w\, c_p\, (T_1 - T_2) = (495 + 9.2) \times 1.114 \times 446 = 251 \text{ kW}$$

Assuming a temperature approach (at pinch point) as 20 K, the temperature of the emerging steam at point 1 is thus limited to 553–20 = 533°C. The problem then is to estimate the largest pressure and mass flow rate of steam which gives a sufficiently large temperature approach at the pinch point $x - 5$ [Fig. 3.12(a)].

Now

$$w_s(h_1 - h_4) = w_g\, c_{pg}\, (T_d - T_a) \tag{3.21}$$

$$w_s(h_5 - h_4) = w_g\, c_{pg}\, (T_x - T_a) \tag{3.22}$$

By trial-and-error estimation from the above two Eq. (3.21) and (3.22) and from steam properties, we find T_1 = 522 °C. The feedwater temperature, T_4 = 38 °C, which shows that at a pressure as low as 5 bar the temperature approach, $T_x - T_5$, is less than 20 K and the corresponding mass flow rate of steam is 74 kg/s.

Dual pressure cycle The HP and LP steam(Fig.3.13) are to be calculated by trial-and-error as before, but with added complexity. The detailed treatment is given by Horlock [11]. Employing some regenerative feedheating, the steam conditions yielded are: 6.5 bar, 200 °C and a flow of about 13.5 kg/s for the LP boiler and 80 bar, 520 °C and a flow of about 66 kg/s for the HP boiler.

Assuming a condenser pressure of 0.068 bar, the specific enthalpy drops in the HP and LP turbines are 670 and 570 kJ/kg, respectively. Therefore, the steam turbine power output

$$W_{st} = (13.5 + 66) \times 570 + 66 \times 670$$
$$= 45.3 + 44.2 = 89.5 \text{ MW}$$

Thus we have approximately equal power output from both LP and HP stages and sufficient for an alternator output of 85 MW. Since the heat input to steam plant from the gas turbine exhaust gases has been shown to be 251 MW, the thermal efficiency of the steam cycle is thus 89.5/251 = 0.357 or 35.7%.

Since the power output of the gas turbine was about 144 MW, the output from the combined plant is about 144 + 85 = 229 MW electrical. If the calorific value of fuel is about 48 MJ/kg and the mass flow of fuel consumption is 9.2 kg/s, the fuel-to-electricity efficiency of the combined plant is 229/(9.2 × 48) = 0.496 or 49.6%.

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Fig. 3.14.1 (a) *Combined cycle power plant at Doswell, Virginia, USA (663 MW)*

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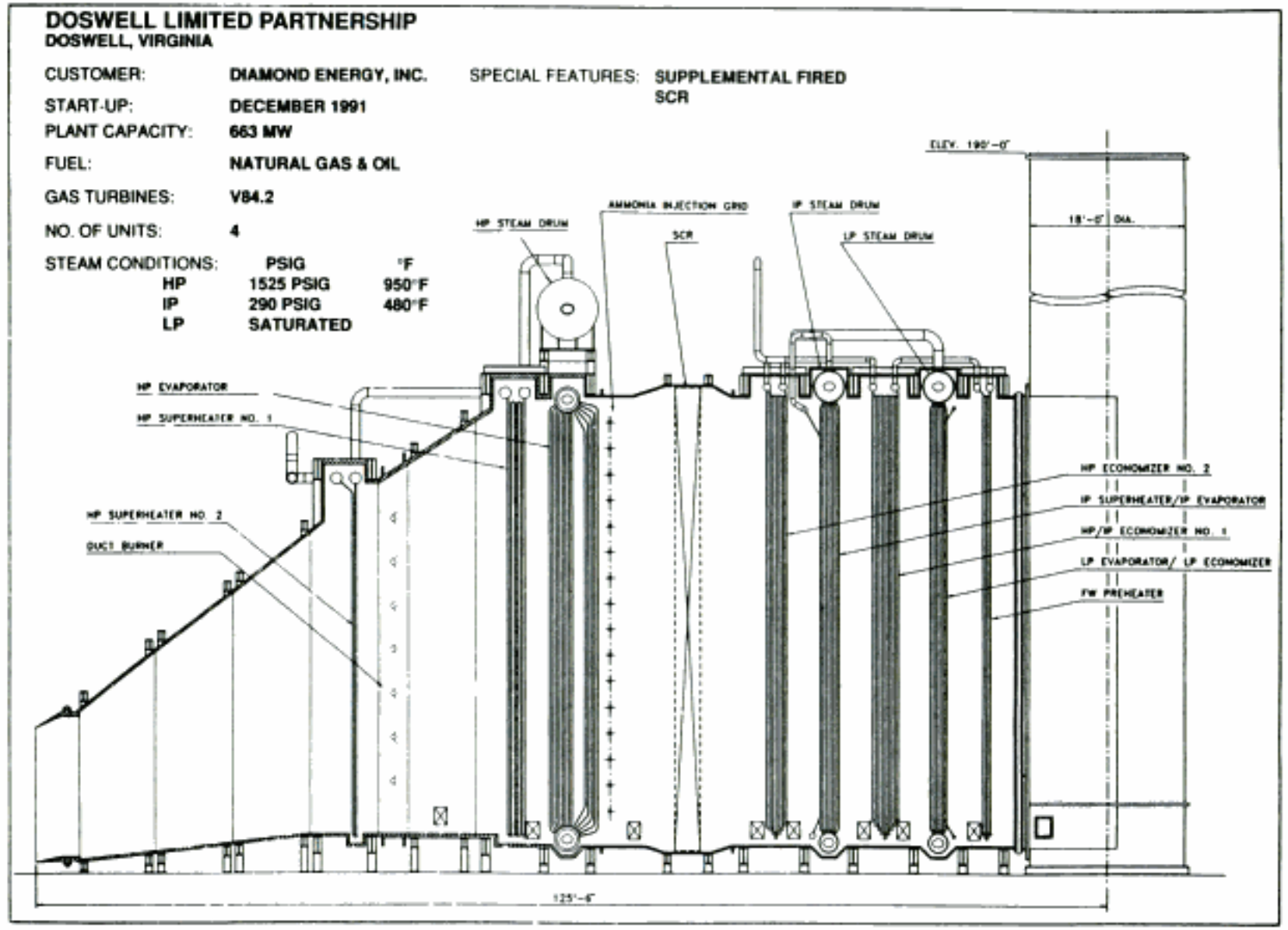

Fig. 3.14.1 (b) *Heat recovery steam generator of Doswell 663 MW c.c. power plant*

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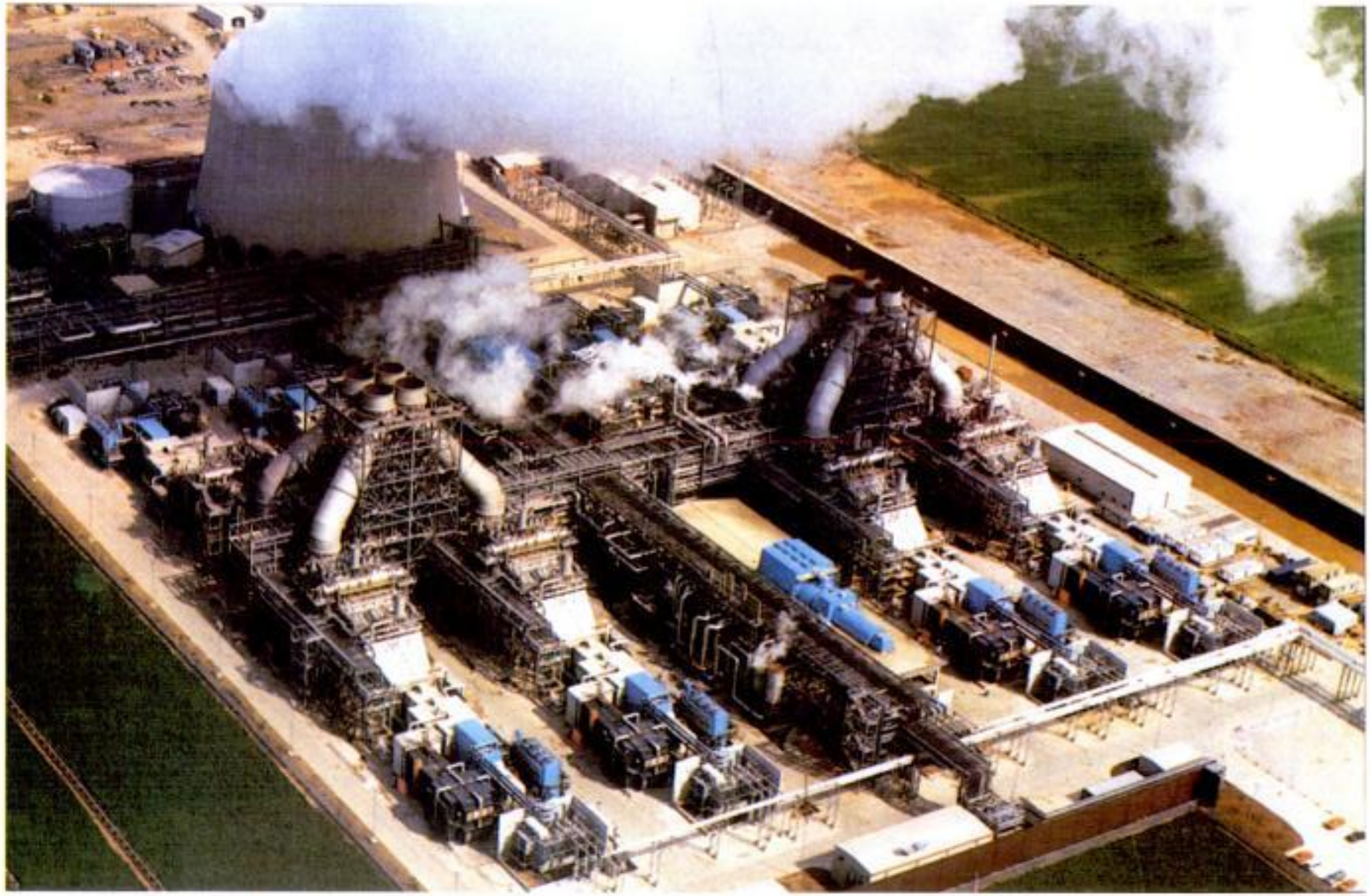

Fig. 3.14.2 (a) *Combined cycle power plant at Teesside, England (1875 MW)*

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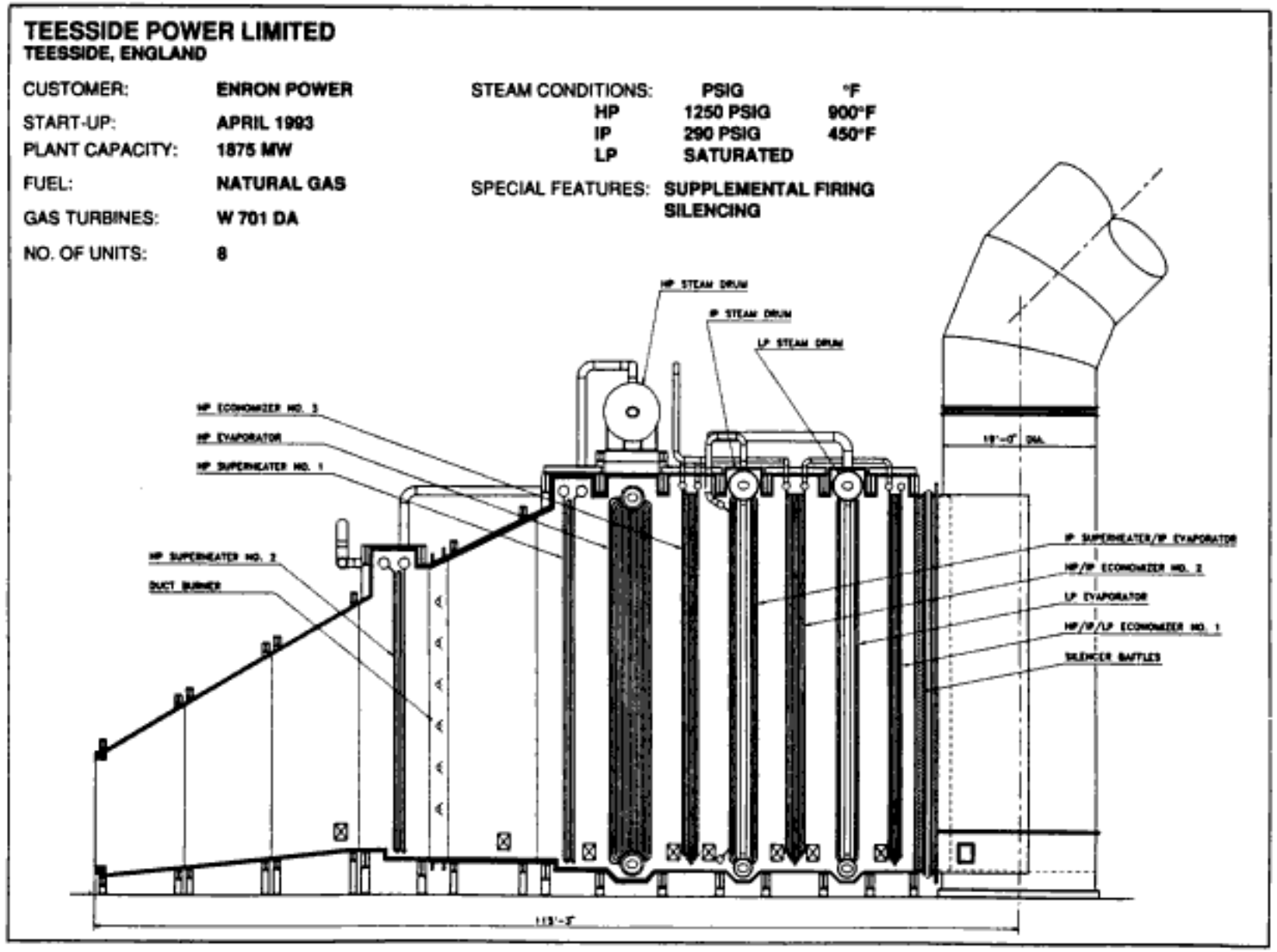

Fig. 3.14.2 (b) *Heat recovery steam generator of Teesside, 1875 MW c.c. power plant*

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3.6.6 Exergy Analysis of Combined Cycles

The exergy of a system is the maximum work obtainable as the system comes to equilibrium with the surroundings (Bejan, 1992). The higher the value of exergy, more is the work obtainable from the system. The First Law of Thermodynamics makes only an energy balance of a system or a control volume. It does not make any distinction of different forms of energy, particularly between work and heat, or heat (internal energy) available at different temperatures. It is the Second Law which asserts that from engineering viewpoint, all forms of energy are not of the same quality. Energies of two systems may be quantitatively equal, but qualitatively they may be different. When steam is adiabatically throttled, its energy does not change, but its quality degrades. Exergy is a measure of energy quality and exergetic (or second law) efficiency is a measure of the perfectness of a thermal system. While energy of a system in any process remains constant, a part of its exergy is always destroyed.

For dry solid fossil fuels the ratio (ϕ) of chemical exergy ε_o to the net calorific value $(NCV)_o$ with a mass ratio of oxygen to carbon (o/c) less than 0.667 is given by Kotas, 1985, p.267.

$$\frac{\varepsilon_o}{(NCV)_o} = \phi = 1.0437 + 0.1882\left(\frac{h}{c}\right) + 0.0610\left(\frac{o}{c}\right) + 0.0404\left(\frac{n}{c}\right) \tag{3.23}$$

where c, h, o and n are the mass fractions of carbon, hydrogen, oxygen and nitrogen, respectively.

For solid fuels with the mass ratio $2.67 > o/c > 0.667$,

$$\phi = \frac{1.0438 + 0.1882\left(\frac{h}{c}\right) - 0.2509\left\{1 + 0.7256\left(\frac{h}{c}\right)\right\} + 0.0383\left(\frac{n}{c}\right)}{1 - 0.3035\left(\frac{o}{c}\right)} \tag{3.24}$$

For liquid fuels the effect of sulphur was included in the correlation

$$\phi = 1.0401 + 0.1728\left(\frac{h}{c}\right) + 0.0432\left(\frac{o}{c}\right) + 0.2169\left(\frac{s}{c}\right)\left\{1 - 2.0628\left(\frac{h}{c}\right)\right\} \tag{3.25}$$

For gaseous fuels of a given composition, the chemical exergy can be calculated from

$$\varepsilon_o = -\Delta h_o + T_o \Delta s_o + RT_o\left[x_{o_2}\ln\frac{p_{o_2}}{p_o} - \Sigma x_k \ln\frac{p_k}{p_o}\right] \tag{3.26}$$

where the subscript k refers to the components of the products of composition. Typical values of ϕ for some industrial fuels are given in Table 3.2.

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Table 3.2 *Typical values of ϕ for some industrial fuels*

Fuel	ϕ
1. Coke	1.05
2. Coal	1.06 – 1.10
3. Wood	1.15 – 1.30
4. Fuel oils and petrol	1.04 – 1.08
5. Natural gas	1.04 ± 0.5%
6. Coal gas	1.00 ± 1%
7. Blast furnace gas	0.98 ± 1%
8. Hydrogen	0.985
9. Carbon monoxide	0.973

Now, for a chemically reactive system
$\varepsilon_o = -\Delta g_o$ = the change in the standard Gibbs function change (kJ/k mol)
and $\Delta h = -(\text{NCV})_o$ = the change in the enthalpy of formation (kJ/k mol). Now, from Eq. (3.23)

$$\phi = \frac{-\Delta G_o}{-\Delta H_o} \tag{3.27}$$

and
$$\Delta H_o = -w_f \times (\text{NCV})_o \text{ (kW)} \tag{3.28}$$

where w_f is the fuel burning rate (kg/s).
Therefore, ΔG_o can be estimated.
Again,

$$\Delta G_o = \Delta H_o - T_o \Delta S_o$$

$$\therefore \quad T_o \Delta S_o = \text{rate of exergy loss in combustion or reaction}$$
$$= -w_f \times (\text{NCV})_o + \phi [w_f \times (\text{NCV})_o]$$
$$= w_f x (\text{NCV})_o (\phi - 1) \tag{3.29}$$

Various amounts of lost work or exergy loss due to irreversibility can be estimated for each component in the plant as given below. The combined cycle can be assumed as given in Fig. 3.15.

Compressor: Rate of energy dissipation or irreversibility in the compressor,

$$I^o_{\text{comp}} = w_a T_o (s_2 - s_1)$$
$$= w_a T_o \left[c_{p_a} \ln \frac{T_2}{T_1} - R_a \ln \frac{p_2}{p_1} \right] \tag{3.30}$$

where $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1} \right)^{(\gamma - 1)/(\gamma \eta_c)}$, η_c being the isentropic efficiency of the compressor.

Combustor: The rate of energy dissipation due to combustion,

$$I_{\text{comb}} = T_o [(S_p)_3 - (S_R)_2] \tag{3.31}$$

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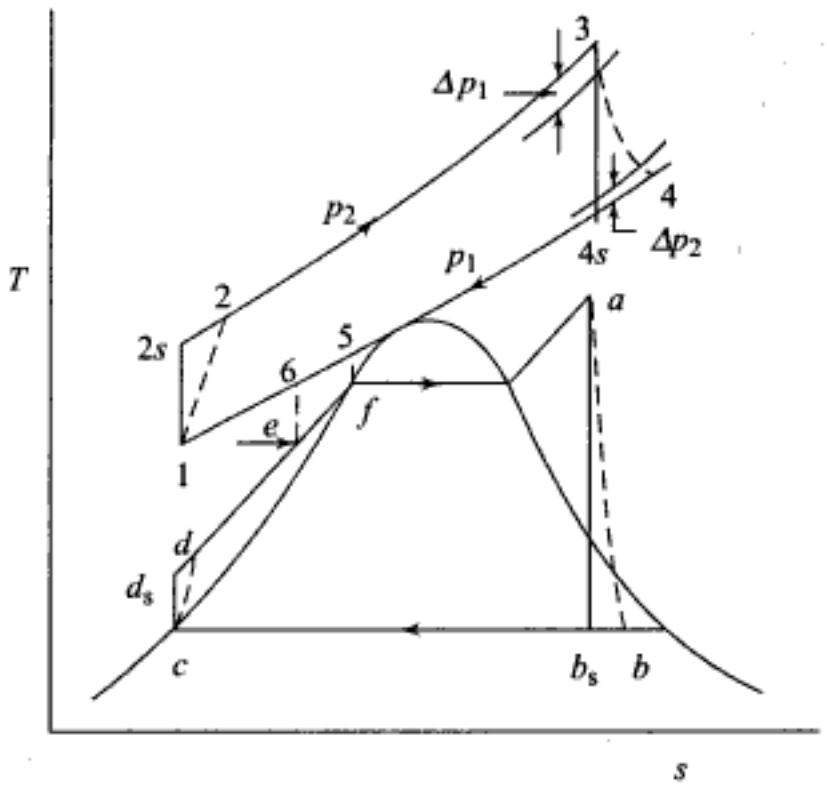

Fig. 3.15 *Combined GT–ST cycle for exergy analysis*

where $(S_R)_2 = (S_A)_2 + (S_F)_2$ and the subscripts P, R, A and F represent products, reactants, air and fuel, respectively.

$$\therefore\ I_{comb} = T_o\{[(S_P)_3 - (S_P)_o] + (S_P)_o - [(S_A)_2 - (S_A)_o + (S_F)_o + (S_A)_o]\}$$

Now,

$$\Delta S_o = (S_P)_o - [(S_F)_o + (S_A)_o]$$

$$I_{comb} = T_o\ \{[(S_P)_3 - (S_P)_o] - [(S_A)_2 - (S_A)_o] + \dot{\Delta} S_o\}$$

$$= T_o \left\{\left[w_g\, c_{p_g} \ln\frac{T_3}{T_o} - w_g\, R_g \ln\frac{p_3}{p_o}\right] - w_a\, c_{p_a} \ln\frac{T_2}{T_o} - w_a\, R_a \ln\frac{p_2}{p_o} + \Delta S_o\right\} \tag{3.32}$$

where $T_o\dot{\Delta} S_o$ is given in Eq. (3.29).

Gas turbine: Rate of exergy loss in the gas turbine,

$$I_{GT} = w_g\, T_o\,(s_4 - s_3)$$

$$= w_g\, T_o \left[c_{p_g} \ln\frac{T_4}{T_3} - R_g \ln\frac{p_4}{p_3}\right] \tag{3.33}$$

where $\dfrac{T_4}{T_3} = \left(\dfrac{p_3}{p_4}\right)^{\frac{(\gamma-1)\eta_T}{\gamma}}$ and η_T is the isentropic efficiency of the gas turbine.

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HRSG: Rate of exergy loss in the heat recovery steam generator,

$$I_{\text{HRSG}} = T_o\,[w_s(s_a - s_e) + w_g(s_6 - s_4)]$$

$$= T_o\left[w_s(s_a - s_e) + w_g\left(c_{p_g}\ln\frac{T_6}{T_4} - R_g\ln\frac{p_6}{p_4}\right)\right] \tag{3.34}$$

Exhaust loss: Rate of exergy loss due to exhaust flue gases.

$$I_{\text{exhaust}} = \int_{T_6}^{T_o}\left(1 - \frac{T_o}{T}\right)\mathrm{d}Q$$

$$= w_g\,c_{pg}\left[(T_6 - T_o) - T_o\ln\frac{T_6}{T_o}\right] \tag{3.35}$$

Steam turbine: Rate of exergy loss in the steam turbine,

$$I_{\text{ST}} = w_s T_o\,(s_b - s_a) \tag{3.36}$$

Condenser: Rate of exergy loss in the condenser,

$$I_{\text{cond}} = w_s\,T_o\,(s_c - s_b) + \frac{T_o}{T_2}Q_2 + Q_o \tag{3.37}$$

where $\dot{Q}_2$ is the rate of heat rejection from the condenser at temperature T_2, and $\dot{Q}_o$ is the rate of heat loss to the environment.

Feed pump: Rate of exergy loss in the feedpump,

$$I_{\text{FP}} = w_s\,T_o\,(s_d - s_c) \tag{3.38}$$

Rate of exergy losses in the feedwater heaters, cooling towers and so on can similarly be estimated. Total rate of exergy losses in the plant,

$$\Sigma I = I_{\text{comp}} + I_{\text{comb}} + I_{\text{GT}} + I_{\text{HRSG}} + I_{\text{exhaust}}$$
$$+ I_{\text{ST}} + I_{\text{cond}} + I_{\text{FP}} + \ldots \tag{3.39}$$

Total rate of exergy input $= -\Delta G_o = -\Delta H_o \times \phi$

Total rate of power generation $= W_{\text{GT}} + W_{\text{ST}}$

$\therefore$ Total rate of exergy output (power output) + total rate of exergy loss = total rate of exergy input

Exergetic or second law efficiency

$$= \frac{\text{total rate of exergy output}}{\text{total rate of exergy input}} = \frac{W_{\text{GT}}^{\text{P}} + W_{\text{ST}}^{\text{P}}}{\Delta G_o} \tag{3.40}$$

The advantage of exergy analysis is that it provides the locations where the exergy losses or irreversibilities are more. Ways and means can then be explored, say, by thermodynamically optimizing the operating or geometrical parameters, so as to reduce these losses to the practical minimum values.

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One example illustrating the method of using exergy analysis in such a plant has been given later (Example 3.4). It is seen that the maximum exergy loss occurs in the combustion of fuel.

3.6.7 Choice of Gas Turbine

Figure 3.16 shows that as the pressure ratio $r_p \left(= \frac{p_2}{p_1}\right)$ increases the efficiency of the Brayton cycle increases, till the maximum or Carnot efficiency is reached, for which

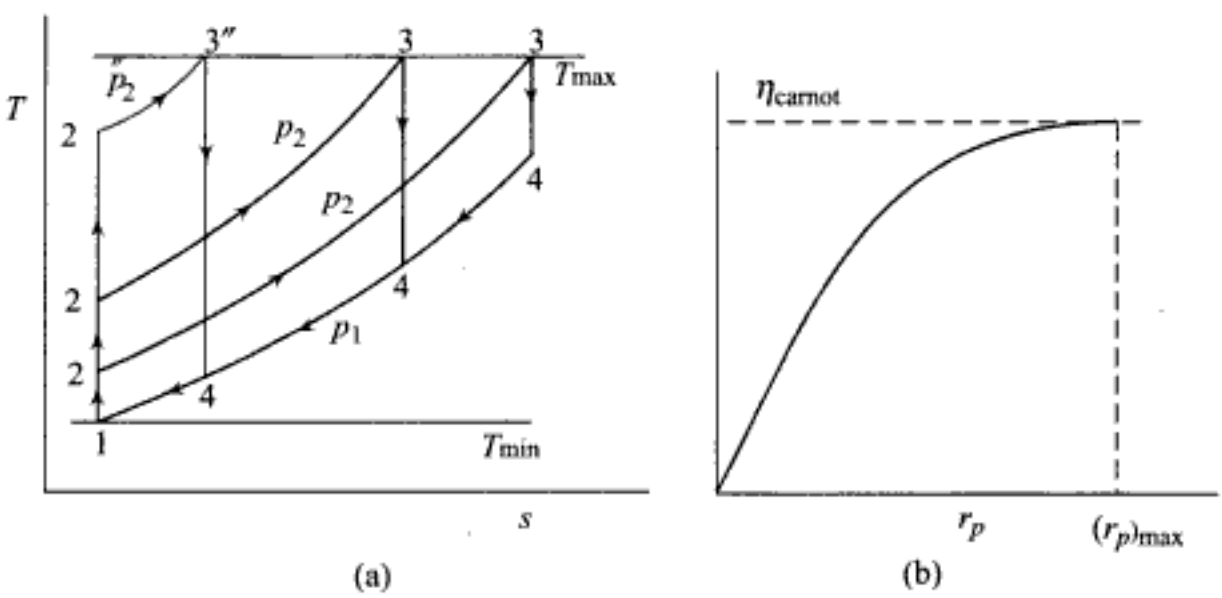

Fig. 3.16 *Effect of pressure ratio on specific output and efficiency of Brayton cycle*

$$(r_p)_{max} = \left(\frac{T_{max}}{T_{min}}\right)^{\gamma/(\gamma-1)} \tag{3.41}$$

where T_{max} is the maximum practicable inlet temperature fixed by the material and T_{min} is the ambient temperature. However, as r_p increases, the turbine exhaust temperature (T_4) decreases, which will in turn decrease the output and efficiency of the bottoming steam cycle.

Again, there is an optimum pressure ratio at which the net work output becomes a maximum (Nag, 1981), which is given by

$$(r_p)_{opt} = \left(\frac{T_{max}}{T_{min}}\right)^{\gamma/2(\gamma-1)} \tag{3.42}$$

and $$(W_{net})_{max} = c_p\,[(T_{max})^{1/2} - (T_{min})^{1/2}]^2 \tag{3.43}$$

Due to economic considerations, present day gas turbines are generally optimized with respect to not efficiency but maximum power density. Fortunately, this optimum coincides fairly accurately with the optimum

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efficiency of the combined cycle plant (r_p = 10.5 to 12). As a result, most of today's gas turbines are optimally suited for combined cycle installations.

Gas turbines of a more complicated design, i.e. with intercooling in the compressor and reheating in the turbine, are less suitable for combined cycles. The gas turbine with the highest efficiency (with intercooling, reheating and regeneration) does not necessarily produce the best overall efficiency of the combined cycle plant. The turbine inlet temperature is a far more important factor.

Gas turbines are standardized. One given machine is used even for widely different ambient conditions. The exhaust heat available for the steam cycle is thus a given quantity and the problem lies in its maximum conversion to mechanical energy. The exhaust gases cannot be cooled down below a certain temperature, the sulphuric acid dewpoint, to avoid low temperature corrosion.

3.6.8 Choice of Steam Plant

The simplest arrangement of a CC plant is a single-pressure system without any special equipment added (Fig. 3.17). The Figure shows that the exhaust gases of a 70 MW gas turbine are used to generate 35 kg/s steam at 34 bar, 475°C. The steam then drives a steam turbine with an output of 34 MW. Because of the good river-water cooling system the pressure in the condenser is 0.04 bar, resulting in a gross efficiency of 45%. The rest of fuel energy is removed in the condenser (28.3%), in the stack (25.2%) or is lost elsewhere (1.5%). The main technical data of the plant (Kehlhoffer, 1991) are given below:

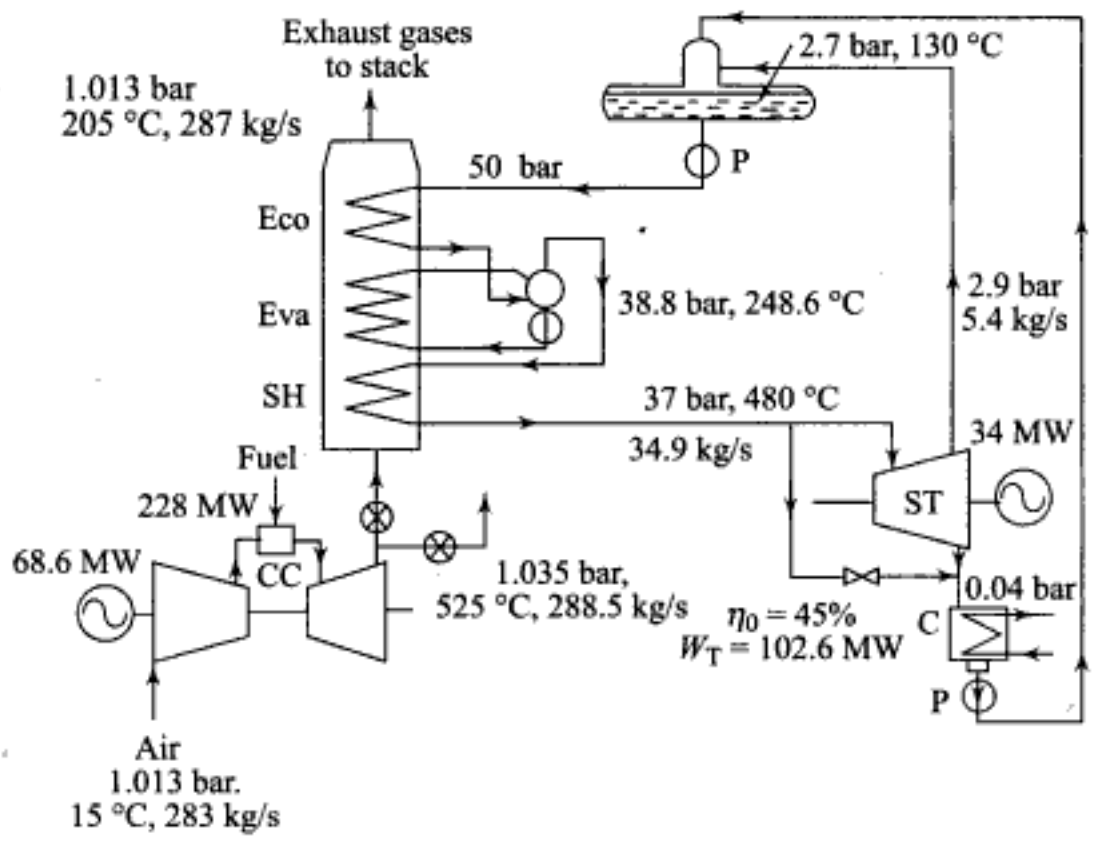

Fig. 3.17 *Combined cycle plant with single pressure steam cycle*

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Gas turbine output	68.6 MW	
Steam turbine output	34.0	MW
Station service power required	1.1	MW
Net power output of plant	101.5	MW
Thermal energy supplied (Diesel fuel)	228	MW
Efficiency of gas turbine plant	30.1%	
Energy contained in exhaust gases	157 MW	
Utilization rate for waste energy	63.3%	
Efficiency of the steam cycle	21.7%	
Gross efficiency of the plant	45.0%	
Net efficiency of the plant	44.5%	

The temperature profiles and the heat absorption rates are plotted in Fig. 3.18.

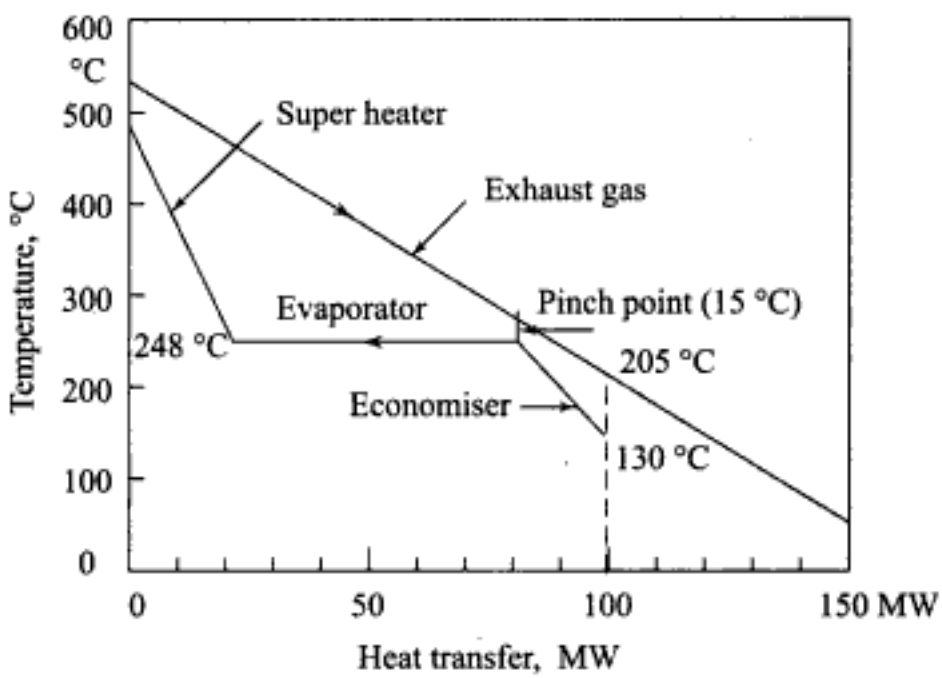

Fig. 3.18 *Temperature profiles of exhaust gas and water-steam along with heat absorption rates*

Figures 3.19 and 3.20 show the energy flow and exergy flow diagrams, respectively. The heat removed in the condenser is only about half that of a conventional steam power plant of the same size. The cooling water requirement is also halved.

A simplified flow diagram for a two-pressure steam cycle system for fuels containing sulphur is shown in Fig. 3.21. Two-pressure or dual cycle steam plant reduces the thermal irreversibility in heat transfer from exhaust gas to steam.

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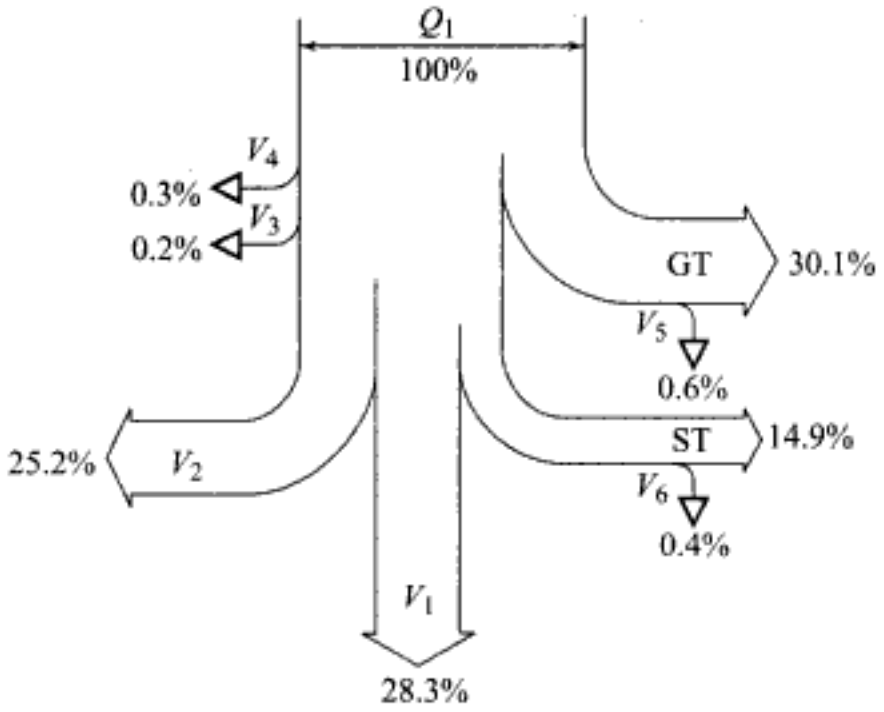

Fig. 3.19 *Energy flow diagram of the combined cycle plant with GT and ST together converting 45% of the input energy (Q_1) into electricity*

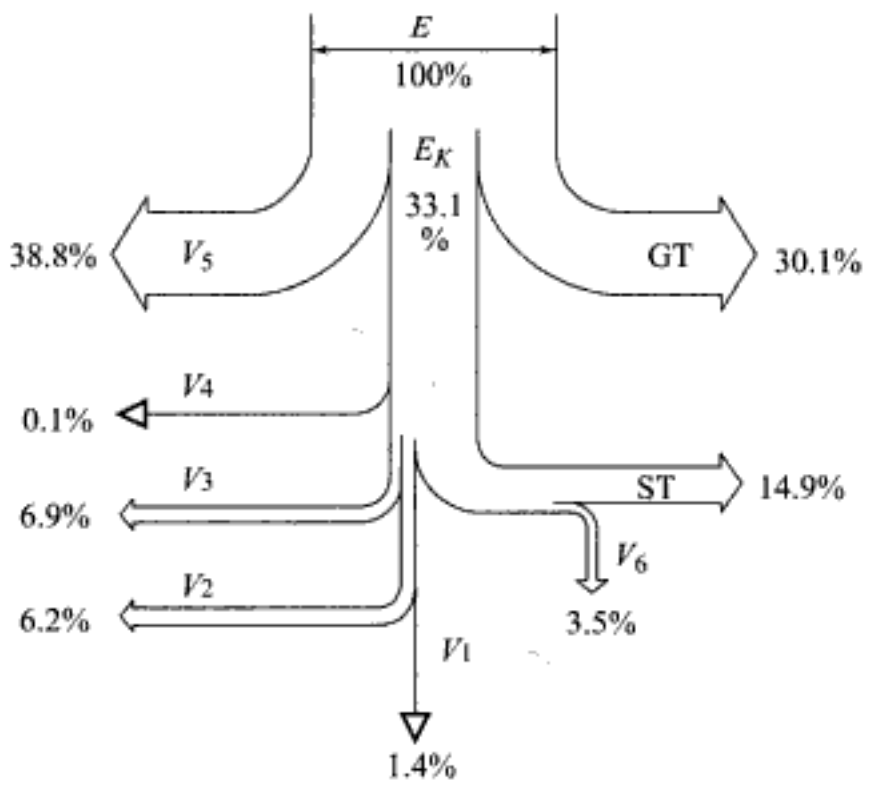

Fig. 3.20 *Exergy flow diagram of a typical combined cycle power plant*

In Fig. 3.19, the symbols used are being explained below:

Q_1 = energy input

V_1 = energy loss in condenser

V_2 = energy loss in stack

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V_3 = energy loss due to radiation in HRSG
V_4 = energy loss in flue gas bypass
V_5 = energy loss in generator and radiation, gas turbine
V_6 = energy loss in generator and radiation, steam turbine

The symbols used in Fig. 3.20 are explained below:

E = exergy input
V_1 = exergy loss in condenser
V_2 = exergy loss in stack
V_3 = exergy loss in HRSG
V_4 = exergy loss in flue gas bypass
V_5 = exergy loss in gas turbine
V_6 = exergy loss in steam turbine
E_k = exergy supplied in the HRSG

3.6.9 Coal-based Combined Cycle Plants

Coal is a lower grade fuel compared to oil and natural gas, but reserves of it are very large and much effort has been devoted to developing clean coal technologies. Successful utilisation of coals for combined cycle power generation has necessitated the development of firing systems whose products of combustion have

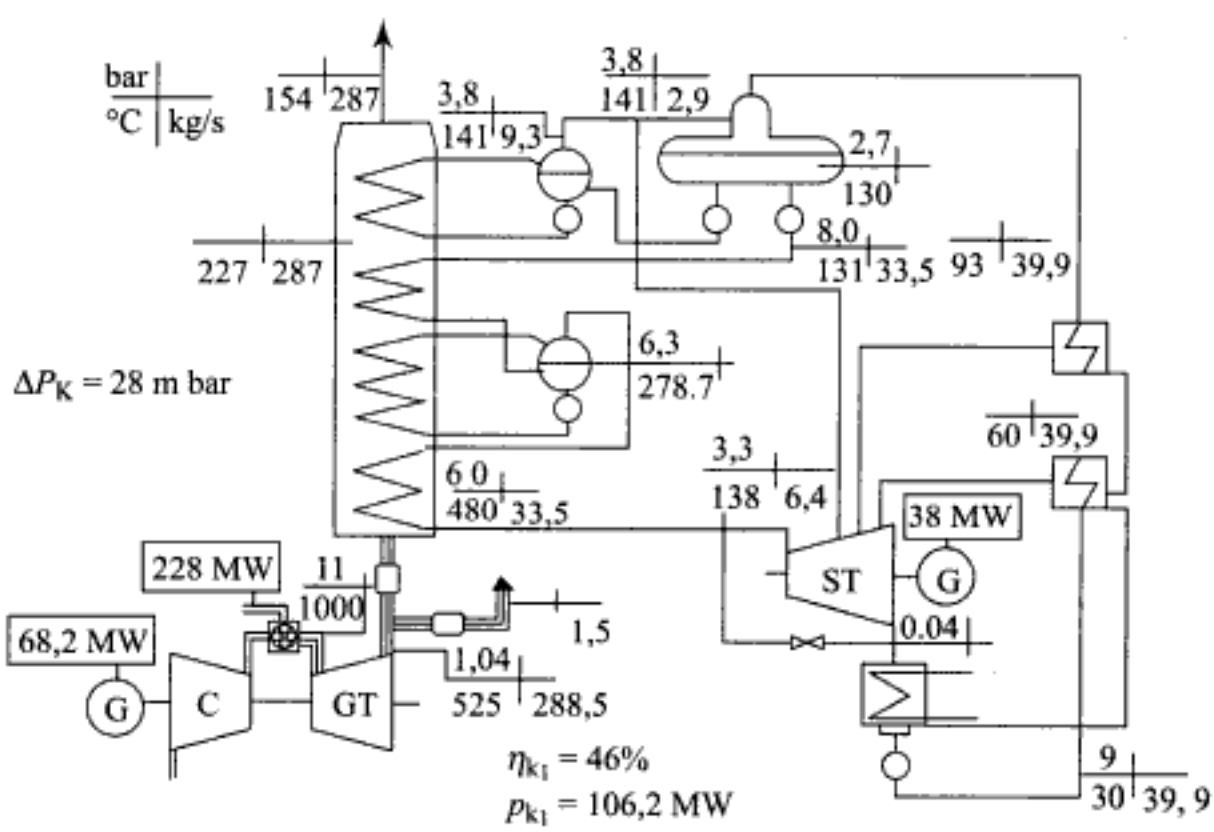

Fig. 3.21 *A simplified flow diagram for a combined cycle plant with two-pressure steam cycle*

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(a) sufficiently low concentrations of particulates to reduce erosion and ensure a satisfactory life of the gas turbine.

(b) sufficiently low concentrations of pollutant gases and particulates in the exhaust to satisfy environmental legislation relating to discharges from power plants.

To reduce the concentration of particulates in products of combustion before entering the gas turbine, hot gas clean-up systems like multi-cyclones, ceramic filters and so on have been developed. For control of emission of oxides of sulphur and nitrogen, different techniques like low NO_X burners, staged combustion, flue gas scrubbing, etc. are being put into use.

Following are the two dominant coal-based technologies.

1. Pressurized Fluidized Bed Combustion (PFBC) System, which may be either a bubbling fluidized bed or a circulating fluidized bed.
2. Integrated Gasification Combined Cycle (IGCC).

1. PFBC-based combined cycle Figure 3.22 shows a schematic of a system using a pressurized fluidized bed combustor (PFBC) to supply hot gas at elevated pressure to a gas turbine via a hot gas clean-up system. Coal and limestone are supplied to the pressurized combustor. The limestone used as the bed material absorbs sulphur (see Chapter 6). Cooling tubes immersed in the fluidized bed are used to generate steam which is supplied to the steam turbine. The combustion products leaving the combustor are passed through a clean-up system before being expanded in the gas turbine. The exhaust gases are then passed through a heat exchanger (economiser) which heats the feedwater before being discharged.

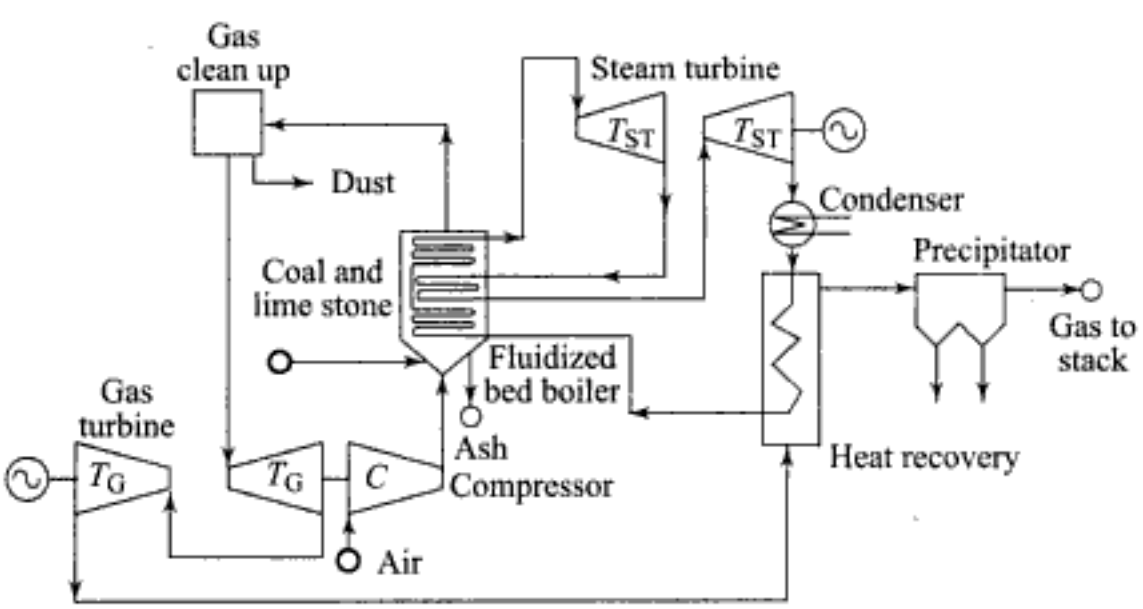

Fig. 3.22 *A PFBC combined cycle*

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Some typical data of a PFBC-based combined cycle plant (Pillai, 1991) is

Steam conditions	:	180 bar, 540 °C
Combustor exit temperature	:	850 °C
Heat transfer from GT exhaust to steam plant	:	10–15% of the total
Ratio of GT power to ST power	:	~ 0.25
Net efficiency	:	42–44%
Combustor pressure	:	10–16 bar

The exit temperature of the exhaust gases may be estimated from (Nag, 1995),

$$T_c - T_d = \eta_T T_c \left[1 - \frac{1}{r^{(\gamma-1)/\gamma}}\right] \tag{3.44}$$

where

T_c = gas temperature at inlet to the gas turbine.
T_d = gas exit temperature,
η_T = isentropic efficiency of the gas expansion,
r = pressure ratio of the expansion, and
γ = ratio of specific heats of gas.

Taking $\gamma = 1.3$, $\eta_T = 0.85$ and $T_c = 850$ °C = 1123 K; for $r = 10$, $T_d = 705$ K (432 °C); and for $r = 16$, $T_d = 646$ K (373 °C). Thus, as the pressure ratio is raised, there is less energy available in the exhaust gases from the gas turbine as a result of which the output of the steam cycle decreases. The gas turbine efficiency is, however, raised when the pressure ratio is increased.

The temperature in the PFBC is limited to about 850 °C because this is the most favourable temperature of sulphur retention and is below the ash fusion temperature of most coals.

2. Integrated gasification combined cycle (IGCC) Coal is gasified, either partially or wholly, and the synthetic gas produced after clean-up is burnt in the combustion chamber of the gas turbine. It is called an integrated gasification combined cycle (IGCC). The main features of the cycle are shown in Fig. 3.23. Coal and limestone are fed to a pressure vessel, the coal being gasified by oxygen and steam. The ash and limestone form a molten slag which is discharged and the synthetic gas produced is cooled. The use of air instead of oxygen produces a gas of lower calorific value. The exhaust gases from the GT raise steam in the HRSG. Several different forms of gasifier have been developed, e.g. Texaco, Shell, Dow, Lurgi and so on. The Texaco process has been used in the Cool Water demonstration plant in California as given in Table 3.3 (Plumley, 1985).

The thermodynamic performance of an IGCC power plant studied by Nag and De (1998) shows that there is an optimum pressure ratio for the gas cycle at a given temperature ratio (T_3/T_0) for maximum overall cycle efficiency.

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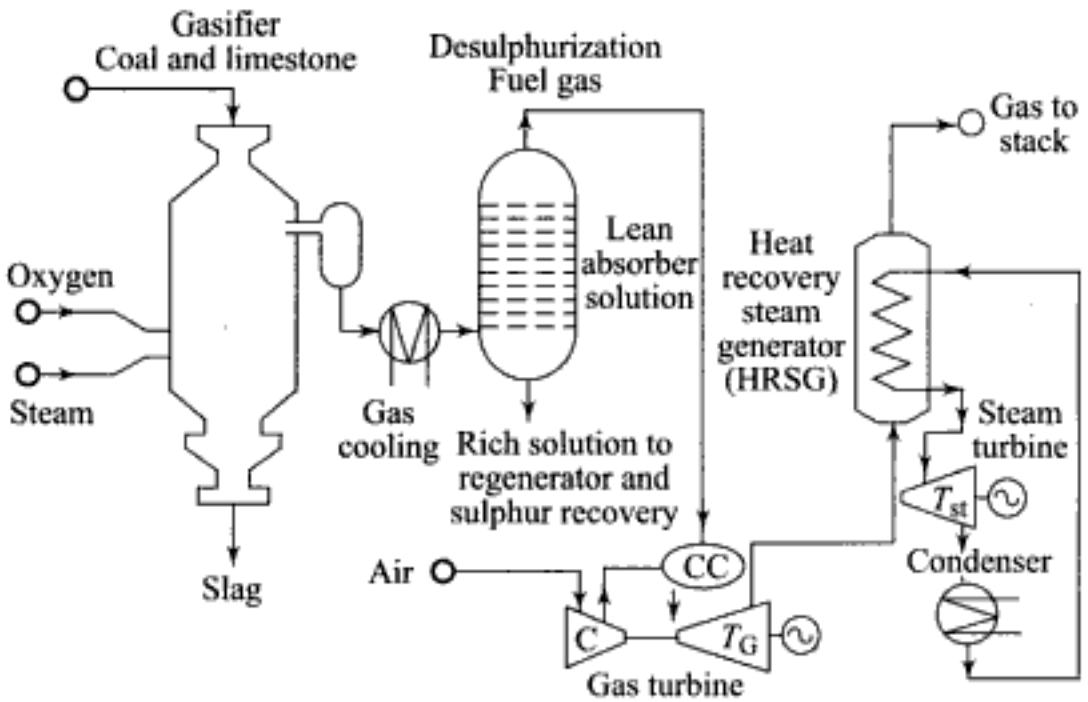

Fig. 3.23 *Gasifer combined plant (IGCC)*

Table 3.3 *Cool Water IGCC Plant*

	Design operating conditions
1. Gasifier pressure	4.137 MN/m^2
Gasifier temperature	1200 °C – 1500 °C
2. HCV of syngas	9700–10,400 kJ/m^3
3. Gas turbine power output	65 MW
4. Steam turbine inlet pressure	9.3 MN/m^2
Steam turbine inlet temperature	510 °C
Steam turbine output	55 MW
5. Coal supply	100 t/d
6. Gross efficiency	0.38
7. Net output	92 MW
8. Net heat rate	12,112 kJ/kWh
9. Net efficiency	0.297

The system shown in Fig. 3.24 is known as the British Coal "Topping Cycle" and it allows the gas turbine inlet temperature to be raised considerably higher than 850 °C by burning fuel gas produced by partial gasification of the coal. Char from the gasifier is burnt in the PFBC to raise hot gas and generate steam. A gas turbine inlet temperature of 1200 °C yields a significant increase in the efficiency of the combined cycle. The temperature of exhaust gases from the gas turbine can be estimated as before.

For $T_c = 1200\ °C = 1473\ K$ and $r = 16$,

$$1473 - T_d = 0.85 \times 1473 \left[1 - \left(\frac{1}{16}\right)^{0.25}\right] = 626$$

or $T_d = 847\ K$ or $574°C$.

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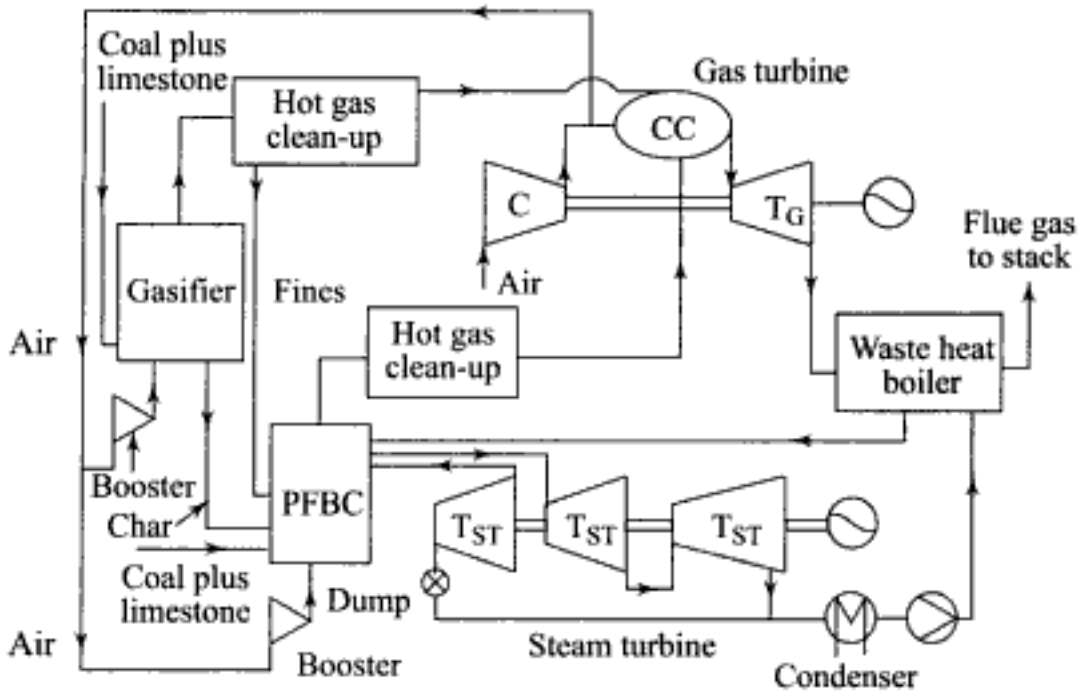

Fig. 3.24 *Outline of PFBC topping cycle (single shaft gas turbine)*

Compared with 373 °C as found earlier, for T_c = 800 °C. The higher gas turbine exit temperature (574 °C) allows more heat recovery from the exhaust gases and hence greater heat supply to the steam cycle.

Figure 3.25 shows a circulating fluidized bed (CFB) being used for burning char and fines from the pressurized gasifier to generate steam. The low pressure turbine exhaust gases as well as the gases exiting the CFBC are used for raising steam in the clean gas boiler (Dawes *et al.*, 1991).

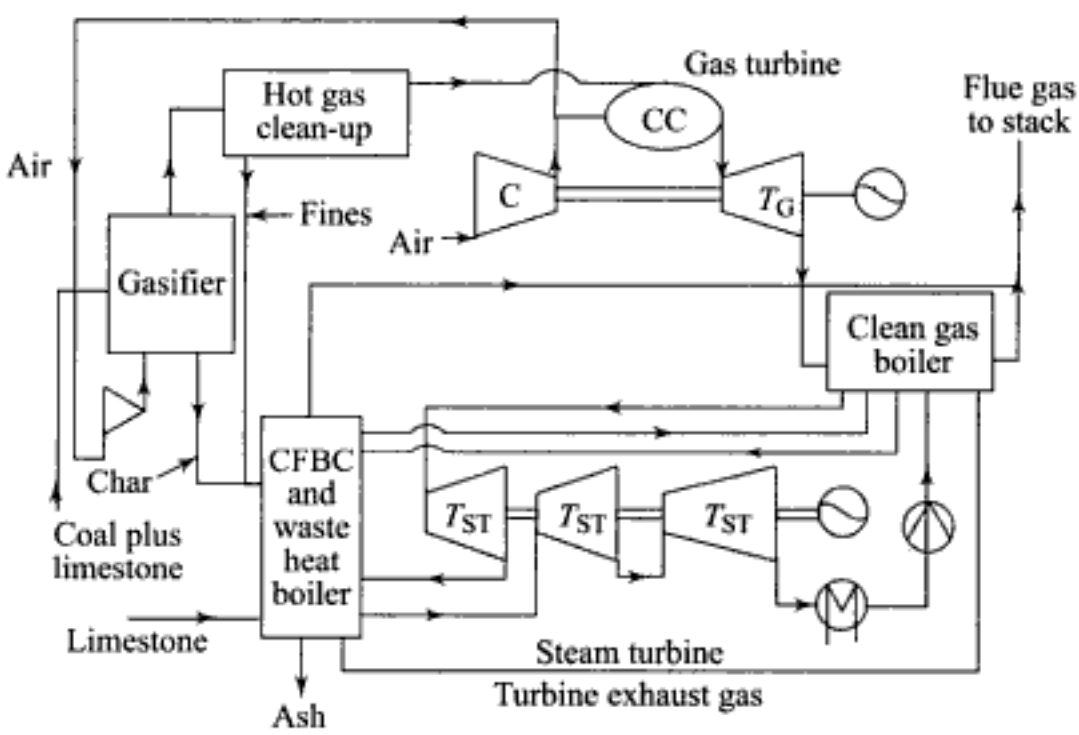

Fig. 3.25 *Outline of circulating fluidized bed combustion (CFBC) topping cycle*

An interesting development of the bottoming cycle is the one as proposed by Kalina (1994) in which a mixture of ammonia and water is used as the working

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fluid. When the binary aqua-ammonia mixture is heated, the ammonia begins to boil first, since its boiling point is lower. As it boils off, the concentration of ammonia in the mixture decreases and the boiling point of the mixture increases. This leads to a better match for the temperature profiles of the exhaust of the topping GT plant and the boiling binary mixture in the bottoming cycle, thus reducing thermal irreversibility. An exergy analysis of the Kalina cycle was made by Nag and Gupta (1998) to find the optimum ammonia concentration at the turbine inlet.

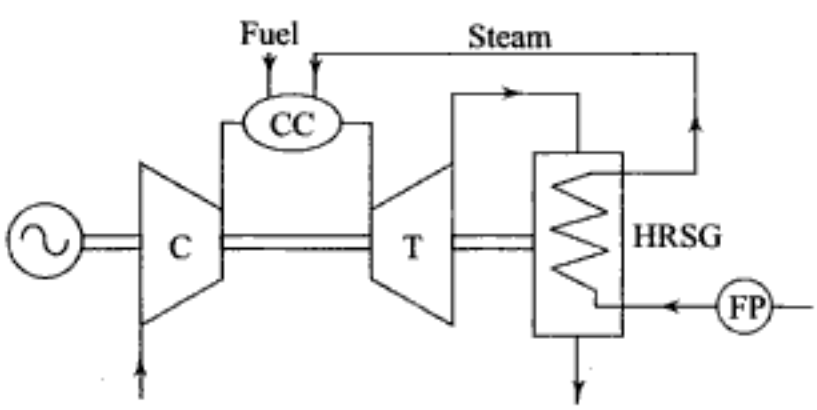
Using the latest technology, future 500 MWe IGCC stations could be designed to operate with an efficiency of 46% (LHV). This is the result of optimal integration of ASEA Brown Boveri (Sweden) type 13E gas turbine with the Shell Coal Gasification Process (USA) and an elevated pressure air separation unit (for oxygen) from Air Products (USA) (Demkolec, 1990).

Based on studies made by Westinghouse (USA), conceptual designs and planning for development of a direct coal-fired advanced combined cycle have been completed, which will burn unbeneficiated low-cost utility-grade coal and meet environmental standards, using a pressurized (14 bar) slagging combustor, hot gas clean-up and a modified gas turbine exhausting into a HRSG (Bannister *et al.*, 1992).

Just as combined cycles have already replaced steam cycles when noble fuels (i.e. natural gas or distillate oils) are utilized to generate electricity, they are likely to become predominant also when coal is the fuel because integrated gasification into a combined cycle yields higher operating efficiency as well as the most environmentally benign conversion of coal to electricity.

3.6.10 Steam Injection into Gas Turbine (STIG)

Steam injection into gas turbines for power augmentation represents a combined Brayton and Rankine cycle in which steam generated in the HRSG by the exhaust gases from the gas turbine is injected into the combustion chamber of the gas turbine (Fig. 3.26). Conceptually, it is equivalent to a combined cycle except that the steam is expanded together with the gas in the same turbine instead of a separate steam turbine.



C—Compressor T—Turbine FP—Feed pump
HRSG—Heat recovery steam generator
CC—Combustion chamber

Fig. 3.26 *Basic steam injection gas turbine (STIG)*

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One way to reduce NO_x formation during combustion is by lowering the temperature of the flame, since the speed of reaction producing NO_x is noticeably rapid only at very high temperatures. Injecting water or steam into the combustor can produce the temperature reduction desired. However, it has been observed that there is a drop in temperature of more than 2% with steam injection and almost 5% with water injection (Kehlhoffer, 1991). The status of steam injected gas turbines (STIG) has been discussed by Tuzson (1992). A recent EPRI (USA) study reports that there is no special economic advantage of STIG (Finckh and Pfost, 1992).

3.6.11 Repowering

The conversion of older power plants into combined cycle units—known as *repowering*—is one interesting way to continue using at least parts of older steam power plants which have become uneconomical. In this procedure, the boilers are normally replaced with high output modern gas turbines and waste heat boilers. Steam turbines of older power stations with relatively low steam inlet pressure and temperature are well adapted for combined cycle operation. These 20–25 years old steam turbines still have a considerable service life left, but their boilers are often ready for scrapping.

Figures 3.27 and 3.28 show the example of a conventional steam turbine plant, before and after such a conversion (repowering). It was possible to reuse the following components:

1. building
2. steam turbine and generator
3. condenser
4. main cooling system
5. main transformer
6. high voltage equipment

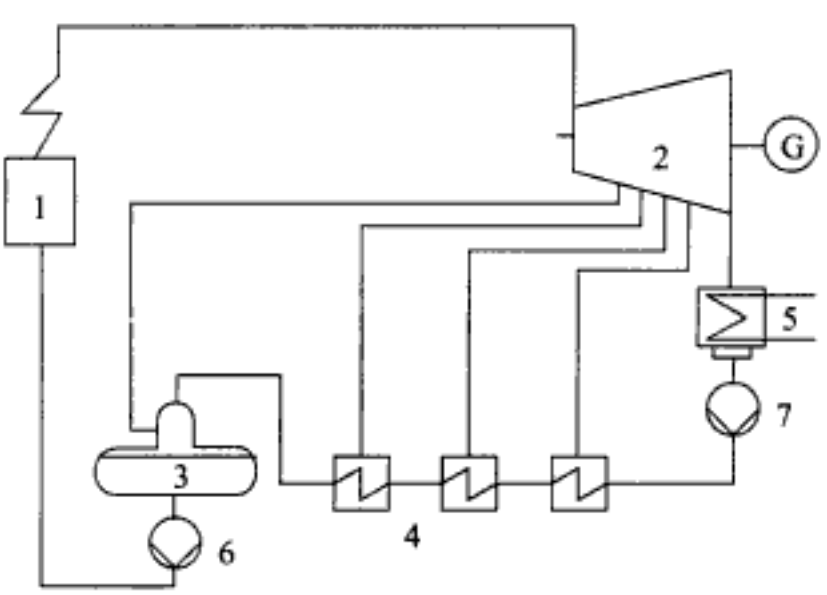

Fig. 3.27 *Steam Power plant before conversion to a combined cycle plant*

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1. Steam generator
2. Steam turbine
3. Feedwater tank, deaerator
4. Low pressure preheater
5. Condensor
6. Feed pump
7. Condensate pump

The following components were dismantled:

1. boiler
2. piping and fittings
3. feedwater heaters
4. condensate pumps
5. boiler feed pumps
6. control equipment

The list may vary from one case to another. Feedwater preheating system for a CC plant is different in design from that for a conventional steam power plant because of thermodynamic considerations. In most cases, no extraction points are needed any more on the steam turbine (Fig. 3.28), which increases the power output from the turbine.

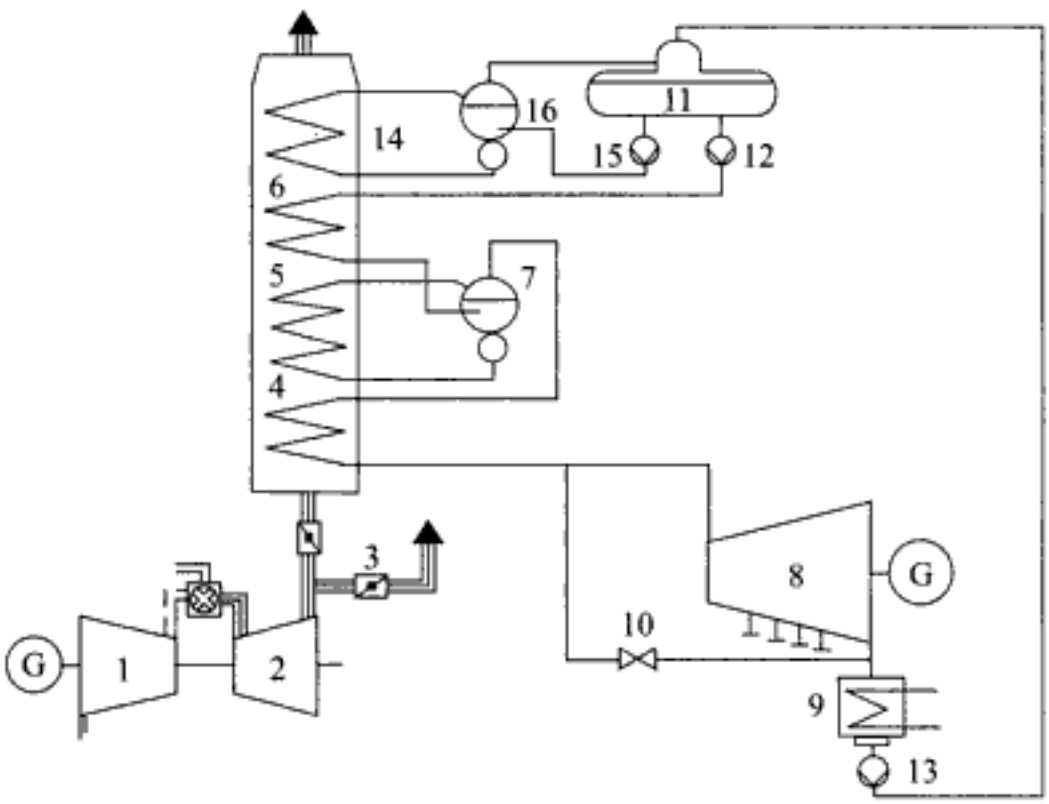

Fig. 3.28 *Combined-cycle plant with existing steam turbine*

1. Gas Turbine
2. Compressor
3. Flue gas bypass
4. Superheater
5. Evaporator
6. Economizer
7. Drum
8. Steam turbine
9. Condenser
10. Steam bypass
11. Feedwater tank/deaerator
12. HP Feedwater pump
13. Condensate pump
14. LP evaporator
15. LP feedwater pump
16. LP drum

Table 3.4 shows the significant gain in efficiency that can be attained with repowering. The power output of the plant has actually been tripled. Since the design of the gas turbine is standardized, there are fixed limits on the amount of

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steam that can be produced, and these do not necessarily lie close to the design values for the steam turbine. One advantage of supplementary firing in repowering is the possibility of closely adapting the steam condition according to the requirement of the steam turbine.

Table 3.4 *Comparison of steam power plant before and after repowering to a combined cycle plant*

	Before repowering	*After repowering*
Net output from power plant	30.3 MW	102.2 MW
Power output from ST	32.0 MW	35.0 MW
Power output from GT	–	68.4 MW
Station service power	1.7 MW	1.2 MW
Heat input	107 MW	228 MW
Net efficiency	28.3%	44.8%

3.6.12 Combined Cycle Plants for Cogeneration

Thermodynamic superiority of the combined cycle plant over a conventional power plant is even more pronounced in cogeneration plants than it is in plants used only to generate electricity. Whenever both electrical power and process steam are needed, it is thermodynamically and economically better to produce both products in a single plant.

Figure 3.29 shows a combined cycle cogeneration plant where a back pressure steam turbine has been used. Table 3.5 gives the relevant operating data for such a plant (Kehlhoffer, 1991).

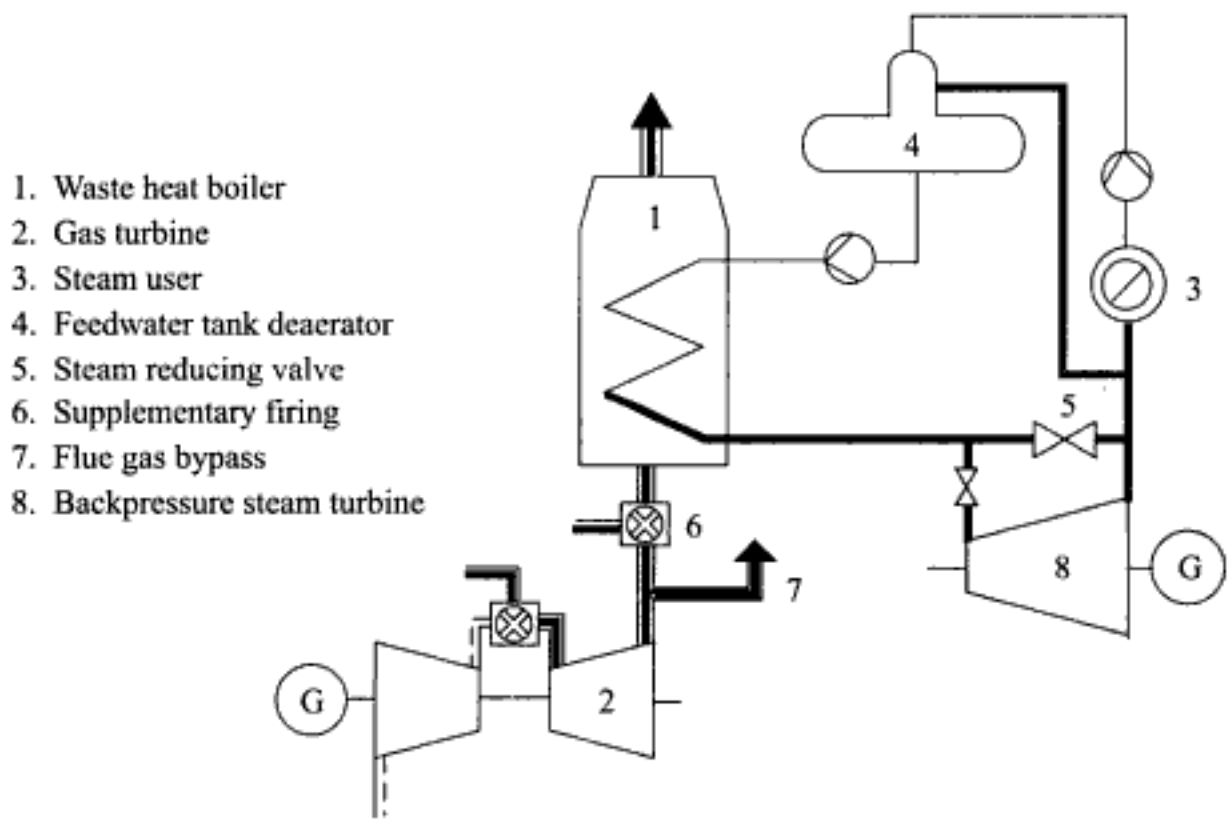

Fig. 3.29 *Diagram of the principle of a combined-cycle plant used for cogeneration*

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The pressure of process steam and the power coefficient, defined as the ratio of electricity generated to the process heat supplied, are important for design. The higher is the pressure of process steam, the less is the electricity produced. If it is very high, the use of a steam turbine becomes questionable since the pressure difference is then too small. In that case, the steam process reduces to a waste heat boiler.

Supplementary firing makes it possible to lower the power coefficient. However, this capability is limited because lowering it too much would reduce the thermodynamic advantage of the combined cycle plant. Combined cycle plants should be employed only where the power coefficient is high.

Table 3.5 *Operating data of a combined cycle cogeneration plant*

Fuel	*Natural gas*
Gas turbine power output	69.1 MW
Back pressure steam turbine output	44.7 MW
Station service power	1.4 MW
Net power output of the plant	112.4 MW
Heat input to the gas turbine (LHV)	230.0 MW
Heat input in supplementary firing (LHV)	79.6 MW
Process steam flow	65.3 kg/s
Process steam pressure	3.5 bar
Thermal energy of process steam	152 MW
Rate of fuel utilization	85.4%
Power coefficient $\left[\frac{\text{electricity generated}}{\text{process heat supplied}}\right]$	0.74
Electrical yield	36.8%
Efficiency of power production	79.9%

An extraction or pass-but steam turbine offers considerable design and operating flexibility to achieve high power coefficients for combined cycle cogeneration plants for district heating, desalination of sea-water and so on.

3.6.13 Environmental Impact

Very efficient desulphurization and clean-up of the coal gas in the gas clean-up system in a combined cycle plant produce a very low environmental impact. Because of the low combustion temperature, only small amounts of NO_x are formed. Due to the long residence time and low combustion temperature the amount of CO formed is not significant. The overall efficiency of a combined cycle plant is high and so the amount of CO_2 discharged into the atmosphere per unit mass of fuel burnt is less, thus mitigating the green-house effect.

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3.6.14 Scope of GT-ST Combined Cycle

Coal gasification combined cycle technology found niche markets in areas where coal is readily available. Several large scale IGCC plants are now being developed worldwide. Three similar PFBC facilities are now operating in Spain, Sweden and the USA (Collins, 1993). The Sweden plant commissioned in 1990 delivers 135 MW to the electric grid and 225 MW to the district heating system.

Of the 113 GW (i.e. 113,000 MW) new generating capacity to be installed in the USA during the 90s, it has been estimated that 51.9 GW will be combined cycle based facilities (Collins, 1993).

A significant number of more than 3500 utilities of 20 GW capacity in the USA completing 30 years service would be candidates for repowering, where the aging steam generator is likely to be replaced by a gas turbine/generator and an HRSG (Collins, 1993). The giant combined cycle power station at Seoinchon, South Korea, commissioned in 1993 and fuelled by natural gas has a total electric generating capacity of 1887 MW and an efficiency exceeding 55% which is the world's top overall efficiency (Power, 1993).

In India, natural gas-based combined cycle power plants under the NTPC are operating at (i) Auraiya (near Kanpur, 652 MW), (ii) Anta (Rajasthan, 430 MW), (iii) Kawas (M.P., 650 MW), (iv) Dadri (near Delhi, 817 MW) and (v) Gandhar (Surat, 650 MW).

The technology of combined cycle power generation:

(a) is well validated, even as a base-load power station,
(b) is more efficient compared to highly developed coal fired steam power plants,
(c) has low levels of pollutant emission,
(d) can be built quickly and relatively cheaply, and
(e) can be profitably employed for repowering an old steam power plant.

It bids fair to become the dominant technology for power production in the twenty-first century.

Example 3.1 A mercury cycle is superposed on the steam cycle operating between the boiler outlet condition at 40 bar, 400°C and the condenser temperature of 40°C. The heat released by mercury condensing at 0.2 bar is used to impart the latent heat of vaporization to the water in the steam cycle. Mercury enters the mercury turbine as saturated vapour at 10 bar. Compute (a) kg of mercury circulated per kg of water, and (b) the efficiency of the combined cycle.

The property values of saturated mercury are given below:

p(bar)	t(°C)	h_f	h_g	s_f	s_g	v_f	v_g
		(kJ/kg)		(k/kg K)		(m^3/kg)	
10	515.5	72.23	363.0	0.1478	0.5167	80.9×10^{-6}	0.0333
0.2	277.3	38.35	336.55	0.0967	0.6385	77.4×10^{-6}	1.163

Solution With reference to Fig. E3.1 for the steam cycle,

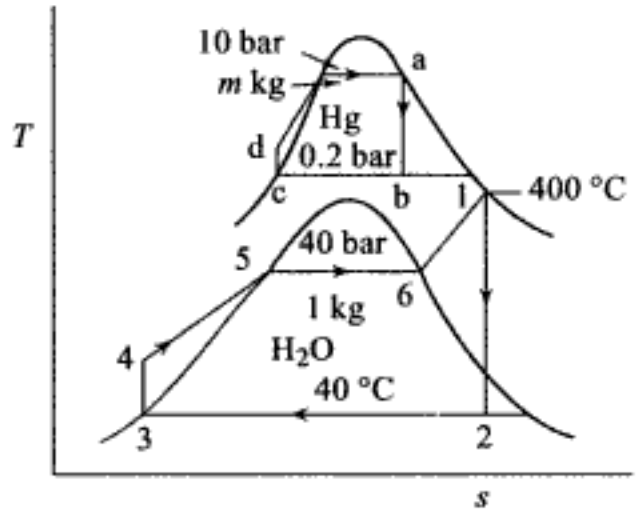

Fig. E3.1

$h_1 = 3213.6$ kJ/kg

$s_1 = 6.7690$ kJ/kg K $= s_2 = 0.5725 + x_2(8.2570 - 0.5725)$

$x_2 = 0.8064$

$h_2 = 167.57 + 0.8064 \times 2406.7 = 2074.8$ kJ/kg

$h_3 = 167.57$ kJ/kg

$h_4 = 167.57 + 40 \times 100 \times 1.008 \times 10^{-3} = 171.6$ kJ/kg

$h_5 = 1087.31$, $h_6 = 2801.4$ kJ/kg

For the mercury cycle,

$h_a = 363$ kJ/kg

$s_a = 0.5167$ kJ/kg K $= s_b = 0.0967 + x_b\,(0.6385 - 0.0967)$

$x_b = 0.7751$

$h_b = 38.35 + 0.7751\,(336.55 - 38.35) = 269.48$ kJ/kg

$h_c = 38.35 = h_d$

Let m = mass of mercury circulated per kg of steam. Then, from energy balance,

$$m = \frac{h_6 - h_5}{h_b - h_c} = \frac{2801.4 - 1087.31}{269.48 - 38.35} = 7.4159 \text{ kg Hg/kg } H_2O$$

Ans. (a)

$$Q_1 = m(h_a - h_d) + 1(h_1 - h_6) + 1(h_5 - h_4)$$

$$= 7.4159(363 - 38.35) + (3213.6 - 2801.4) + (1087.31 - 171.6)$$

$$= 3733.01 \text{ kJ/kg}$$

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$$Q_2 = h_2 - h_3 = 2074.8 - 167.57 = 1907.23 \text{ kJ/kg}$$

$$h_{\text{combined cycle}} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{1907.23}{3733.01} = 0.489 \quad \text{or,} \quad 48.9\% \qquad \textit{Ans.}(b)$$

Example 3.2 In a cogeneration binary cycle, superheated steam enters the turbine with a mass flow rate of 5 kg/s at 40 bar, 440°C and expands isentropically to 1.5 bar. Half of the flow is extracted at 1.5 bar and used for industrial process heating. The remaining steam passes through a heat exchanger which serves as the boiler of the Refrigerant-12 cycle and the condenser for the steam cycle. The condensate leaves the heat exchanger as saturated liquid at 1 bar, where it is combined with the return flow from the industrial process at 60 °C and 1 bar, before being pumped isentropically to the steam generator. The Refrigerant-12 cycle is an ideal Rankine cycle with refrigerant entering the turbine at 16 bar, 100 °C and saturated liquid leaving the condenser at 9 bar. Determine (a) the rate of heat transfer in the steam generator, (b) the net power output of the binary cycle, (c) the rate of heat transfer to the industrial process.

Solution For the steam cycle (Fig. E3.2),

$$h_1 = 3307.1 \text{ kJ/kg}, \; s_1 = 6.9041 \text{ kJ/kg K} = s_2$$

$$h_2 = 2570.8 \text{ kJ/kg}, \; h_3 = 417.46 \text{ kJ/kg}$$

$$T_6 = 60°\text{C, compressed liquid}$$

$$h_6 = h_f(60) + v_f(60)(p - p_{sat}.60°\text{C})$$

$$= 251.13 + (1.0172 \times 10^{-3})(1 - 0.1994) \times 100 = 251.2 \text{ kJ/kg}$$

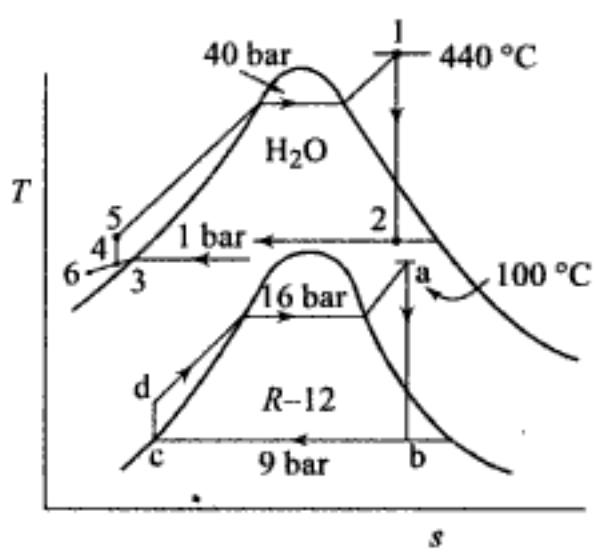

Fig. E3.2

Energy balance for adiabatic mixing of the streams 3 and 6 gives

$$\dot{m}_3 h_3 + \dot{m}_6 h_6 = (\dot{m}_3 + \dot{m}_6) h_4$$

where $\dot{m}_3 = \dot{m}_6 = 2.5$ kg/s

$$h_4 = \frac{2.5 \times 417.46 + 2.5 \times 251.2}{5} = 334.33 \text{ kJ/kg}$$

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$T_4 = 80°C, h_5 = \quad h_4 + v_4(p_5 - p_4)$

or $h_5 = 334.33 + (1.0291 \times 10^{-3})\,(40 - 1)\,100 = 338.34$ kJ/kg

For the R-12 cycle,

At 16 bar, 100 °C, $h_a = 241.58$ kJ/kg

$$s_a = 0.7656 \text{ kJ/kg K} = s_b$$

$$h_b = 229.43 \text{ kJ/kg}$$

$$h_c = 71.93 \text{ kJ/kg}$$

$$h_d = 71.93 + 0.7914 \times 10^{-3}\,(16 - 9)\,100 = 72.48 \text{ kJ/kg}$$

$$\dot{Q}_1 = \dot{m}_1\,(h_1 - h_5) = 5(3307.1 - 338.34) = 14.844 \text{ kW}$$

Ans. (a)

$(\dot{W}_{net})_{steam} = W_{T1} - W_{P1} = 5[(3307.1 - 2570.8) - (338.34 - 334.33)] = 3661$ kW

$\dot{m}_3(h_2 - h_3) = \dot{m}_{R-12}\,(h_a - h_d)$

$2.5(2570.8 - 417.46) = \dot{m}_{R-12}\,(241.58 - 72.48)$

$\therefore \quad \dot{m}_{R-12} = 31.83$ kg/s

$(\dot{W}_{net})_{R-12} = \dot{m}_{R-12}\,[(h_a - h_b) - (h_d - h_c)]$

$= 31.83\,[(241.58 - 229.43) - (72.48 - 71.93)] = 368.9$ kW

Total power output = 3661 + 368.9 = 4030 kW *Ans.* (b)

For the industrial process,

$$\dot{Q}_H = \dot{m}_{process}\,(h_2 - h_6) = 2.5\,(2570.8 - 251.2) = 5799 \text{ kW} \qquad \textit{Ans. (c)}$$

Example 3.3 In a combined gas turbine-steam turbine power plant, the exhaust gas from the open cycle gas turbine is the supply gas to the steam generator of the steam cycle at which additional fuel is burned in the gas. The pressure ratio for the gas turbine is 7.5, the air inlet temperature is 15 °C and the maximum cycle temperature is 750°C.

Combustion of additional fuel raises the gas temperature to 750 °C and the gas leaves the steam generator at 100°C. The steam is supplied to the turbine at 50 bar, 600 °C and the condenser pressure is 0.1 bar. The total power output of the plant is 200 MW. The calorific value of the fuel burned is 43.3 MJ/kg. Neglecting the effect of the mass flow rate of fuel on the air flow, determine (a) the flow rates of air and steam required, (b) the power outputs of the gas turbine and steam turbine, (c) the thermal efficiency of the combined plant, (d) the air fuel ratio. Take c_p = 1.11 kJ/kg K and $\gamma = 1.33$ for combustion gases, and $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for air. Neglect pump work.

Solution For the gas turbine cycle (Fig. E3.3)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = (7.5)^{0.4/1.4}$$

$$T_2 = 288 \times (7.5)^{2/7} = 512.165 \text{ K}$$

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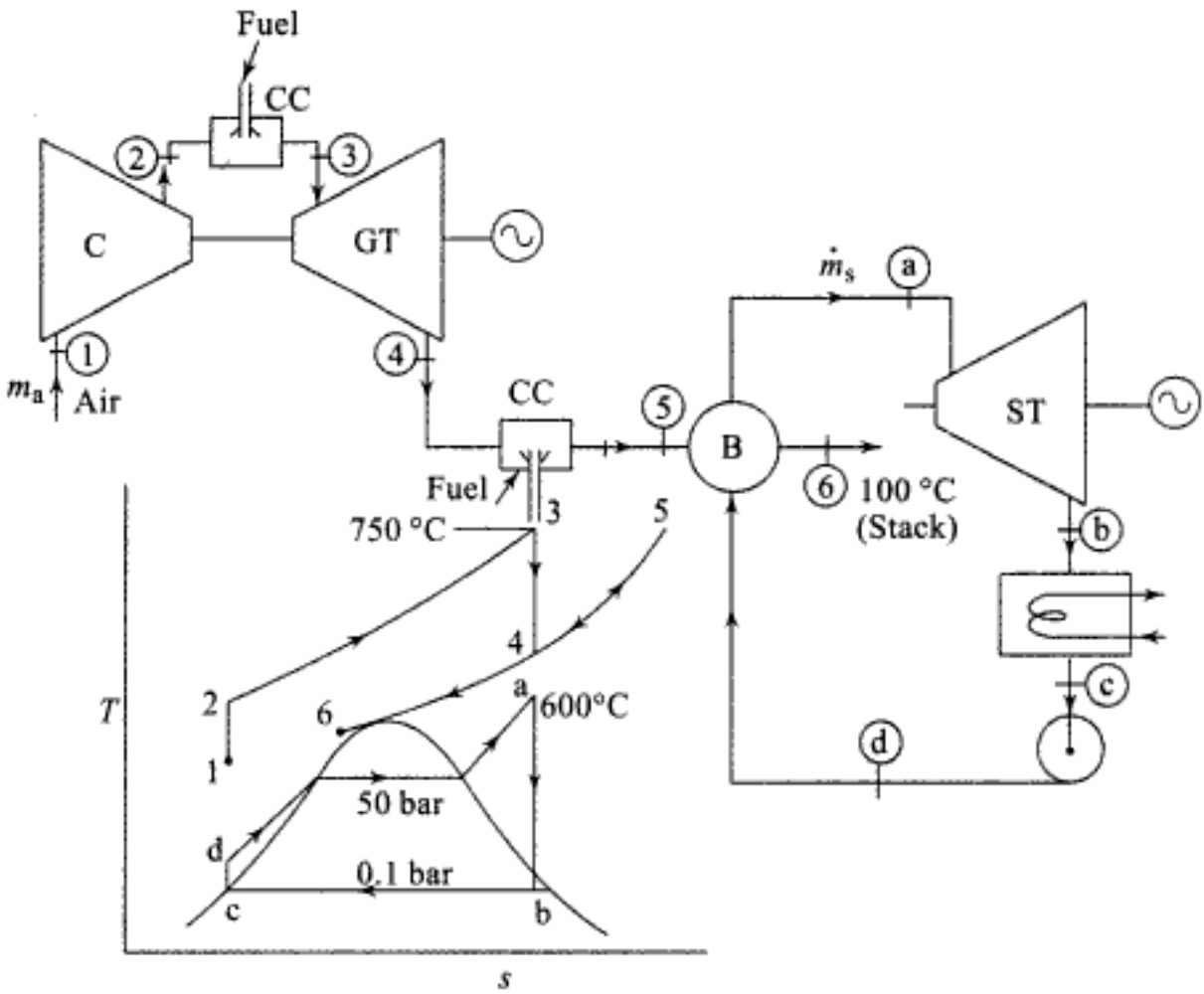

Fig. E3.3

$$T_4 = T_3 \left(\frac{p_4}{p_3}\right)^{(\gamma-1)/\gamma} = \frac{1023}{(7.5)^{0.33/1.33}} = 620.52 \text{ K}$$

For the steam cycle, from Mollier chart,

$$h_a = 3670,\ h_b = 2305,\ h_c = h_d = 192 \text{ kJ/kg}$$

$$(W_{net})_{GT} = \dot{m}_a\ c_{p_g}\ (T_3 - T_4) - \dot{m}_a\ c_{p_a}\ (T_2 - T_1)$$

$$= \dot{m}_a\ 1.11\ (1023 - 620.52) - \dot{m}_a\ 1.005\ (512.65 - 288)$$

$$= 221.47\ \dot{m}_a$$

$$(W_T)_{ST} = \dot{m}_s\ (h_a - h_b) = \dot{m}_s(3670 - 2305) = 1365\ \dot{m}_s$$

$$1365\ \dot{m}_s + 221.47\ \dot{m}_a = 200 \times 10^3 \qquad (1)$$

Now, $\dot{m}_a\ c_{p_g}\ (T_5 - T_6) = \dot{m}_s\ (h_a - h_d)$

$$\dot{m}_a\ 1.11\ (1023 - 373) = \dot{m}_s\ (3670 - 192)$$

$$\therefore \qquad \frac{\dot{m}_s}{\dot{m}_a} = \frac{715}{3478} = 0.2056 \qquad (2)$$

Substituting in Eq. (1),

$$1365 \times 0.2056\ \dot{m}_a + 221.47\ \dot{m}_a = 200 \times 10^3$$

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$$\therefore \quad \dot{m}_a = \frac{200 \times 10^3}{500.26} = 398.34 \text{ kg/s}$$

$$\dot{m}_s = 0.2056 \times 398.34 = 81.9 \text{ kg/s} \qquad \textit{Ans. (a)}$$

$$(W_{net})_{GT} = 221.47 \times 398.34 \times 10^{-3} = 88.22 \text{ MW}$$

$$(W_T)_{ST} = 200 - 88.22 = 111.78 \text{ MW} \qquad \textit{Ans. (b)}$$

$$Q_1 = \dot{m}_a \; c_{p_g} \; (T_3 - T_2 + T_5 - T_4)$$

$$= 398.34 \times 1.11[1023 - 512.165 + 1023 - 620.52]$$

$$= 400.19 \text{ MW}$$

$$h_{thermal} = \frac{W_{net}}{Q_1} = \frac{200}{400.19} = 50\% \qquad \textit{Ans. (c)}$$

$$Q_1 = \dot{m}_f \times 43.3 \times 10^3 \text{ kJ/kg}$$

where $\dot{m}_f$ is the fuel burning rate.

$$\dot{m}_f \times 43.3 \times 10^3 = \dot{m}_a \times 1.11 \times 913.31$$

$$\frac{\dot{m}_a}{\dot{m}_f} = \frac{43300}{1013.77} = 42.7 \qquad \textit{Ans. (d)}$$

Example 3.4 The following data refer to a combined cycle power plant:

Inlet condition of air to the compressor	:	1 bar, 25 °C
Pressure ratio of the compressor	:	8
Maximum gas temperature at inlet to the gas turbine	:	900 °C
Pressure drop in the combustion chamber	:	3%
Efficiency of the compressor	:	0.88
Efficiency of the gas turbine	:	0.88
Calorific value of liquid octane (C_8H_{18}) used as fuel	:	44.43 MJ/kg
Specific heat of air	:	1.006 kJ/kg K
Specific heat of gas	:	1.148 kJ/kg K
Specific heat ratio of gas	:	1.333
Specific heat ratio of air	:	1.4
Condition of steam at inlet to the steam turbine	:	40 bar, 425 °C
Condenser pressure	:	0.04 bar
Feed water temperature to the HRSG	:	170.4 °C
Efficiency of the steam turbine	:	0.82
Pressure drop of gas in the HRSG	:	5 kPa
Steam flow rate	:	29.235 kg/s

Assume $\psi = (\Delta G_0)/(\Delta H_o) = 1.0401 + 0.1728 \left(\frac{h}{c}\right)$

where (h/c) is the mass ratio of hydrogen to carbon in the fuel.

(a) Determine the total power output and the overall efficiency of the combined cycle plant and estimate the "lost heat" coefficient x_L. (b) Give an energy balance of the plant and estimate the exergetic efficiency.

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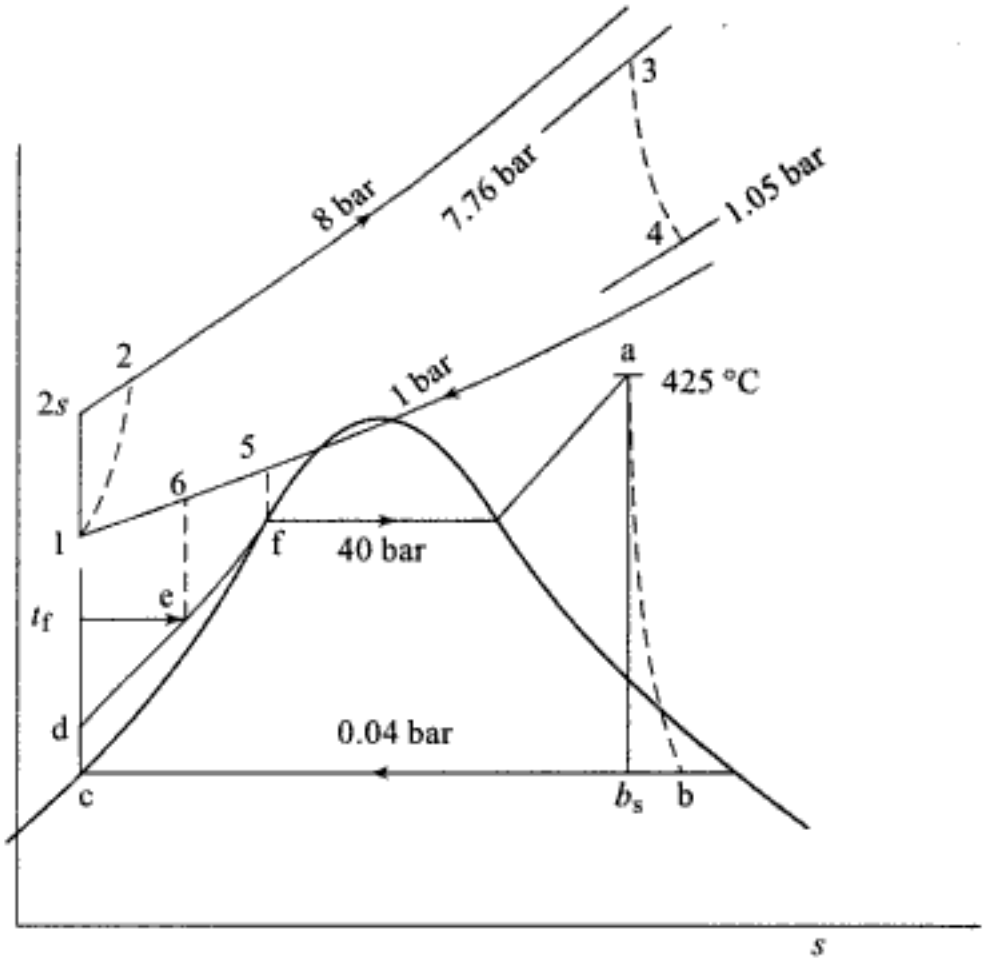

Fig. E3.4

Solution Gas turbine plant:

$$p_1 = 1 \text{ bar}, p_2 = 8 \text{ bar}, T_1 = 298 \text{ K}, \eta_c = 0.88$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/(\gamma\eta_c)} = 8^{0.4/(1.4 \times 0.88)} = 8^{0.325} = 1.965$$

$\therefore$ $$T_2 = 298 \times 1.965 = 586 \text{ K} = 313 \text{ °C}$$

Combustor:

Pressure loss = 0.03 × 8 = 0.24 bar

$\therefore$ $p_3 = 8 - 0.24 = 7.76$ bar

Let the flow rate of combustion gas be 1 kg/s and that of fuel f kg/s.

$\therefore$ The flow of air = $(1 - f)$ kg/s

Therefore,

$$f \times CV = 1.c_{p_g}(T_3 - T_4) - (1 - f)c_{p_a}(T_2 - T_o)$$

$$f \times 44{,}430 = 1 \times 1.148\,(900 - 25) - (1 - f)\,(1.006)\,(313 - 25)$$

$$= 1004.5 - 289.7\,(1 - f)$$

$$f = \frac{714.8}{44140.7} = 0.0162 \text{ kg/s}$$

$$\text{Air fuel ratio} = \frac{1 - f}{f} = \frac{0.9838}{0.0162} = 60.73$$

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Now, $C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$

Air fuel ratio for stoichiometric combustion $= \dfrac{12.5 \times 32}{0.232 \times 114} = 15.12$

Excess air $= \dfrac{60.73 - 15.12}{15.12} = 3.02$ or 302%

Gas turbine: $p_4 = 1 + 0.05 = 1.05$ bar, $T_3 = 1173$ K

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{(\gamma-1)\eta_T/\gamma} = \left(\frac{7.76}{1.05}\right)^{(0.333 \times 0.88)/(1.333)}$$

$$= (7.39)^{0.22} = 1.553$$

$$T_4 = \frac{1173}{1.553} = 755 \text{ K} = 482 \text{ °C}$$

This is the turbine exhaust gas temperature.

HRSG:

Let the pinch point temperature difference $(T_5 - T_f)$ be 30 °C,

$$T_f = (T_{sat})_{40 \text{ bar}} = 250.4 \text{ °C}$$

$$\therefore \quad T_5 = 250.4 + 30 = 280 \text{ °C}$$

Now, from steam tables, $h_a = 3272$ kJ/kg and $h_f = 1087$ kJ/kg. By energy balance,

$$w_g \; c_{p_g} \; (T_4 - T_5) = w_s(h_a - h_f)$$

$$1 \times 1.148(482 - 280) = w_s(3272 - 1087.31)$$

$$\therefore \quad w_s = 0.106 \text{ kg/s}$$

The total heat transfer in the HRSG yields the stack temperature T_6.

At 170.4 °C, $h_e = 721.1$ kJ/kg

$$1.148\,(482 - T_6) = 0.106(3272 - 721)$$

$$\therefore \quad T_6 = 247 \text{ °C}$$

Power output:

$$h_a = 3272 \text{ kJ/kg}$$

$$s_a = 6.853 \text{ kJ/kg} = s_{bs} = 0.4226 + x_{bs} \times 8.052$$

$$x_{bs} = 0.7986$$

$$h_{bs} = 121.46 + 0.7986 \times 2432.9 = 2064.37$$

Neglecting pump work, the steam turbine output is

$$W_{ST} = w_s(h_a - h_{bs})\eta_T = 29.235(3272 - 2064.37) \times 0.82 = 28{,}863 \text{ kW}$$

Mass flow rate of gas in gas turbine

$$w_g = \frac{29.235}{0.106} = 275.8 \text{ kg/s}$$

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Air flow rate entering the compressor

$$w_a = (1 - f)275.8 = 0.9838 \times 275.8 = 271.3 \text{ kg/s}$$

Power output from the gas turbine

$$W_{GT} = 275.8 \times 1.148(900 - 482) - 271.3 \times 1.006(313 - 25)$$
$$= 132{,}346.5 - 78{,}603.2 = 53{,}744 \text{ kW}$$

$\therefore$ Total power output $= W_{ST} + W_{GT} = 28{,}863 + 53{,}744$

$= 82{,}607$ kW *Ans.*

Fuel mass flow rate, $w_f = 0.0162 \times 275.8 = 4.466$ kg/s

The overall efficiency of the combined plant

$$\eta_o = \frac{82{,}607}{4.466 \times 44{,}430} = 0.4163, \text{ or } 41.63\% \qquad \textit{Ans.}$$

Efficiency of the steam plant $= \dfrac{h_a - h_b}{h_a - h_e} = \dfrac{(3272 - 2064)\,0.82}{3272 - 721}$

$$= \frac{990.6}{2551} = 0.388$$

The efficiency of the GT plant $= \dfrac{53{,}744}{4{,}466 \times 44{,}430} = 0.2709$

"Lost heat" coefficient in the exhaust stack

$$x_L = \frac{275.8 \times 1.148\,(247 - 25)}{4.466 \times 44{,}430} = 0.354$$

From Eq. (3.8), the overall efficiency is

$$\eta_o = \eta_1 + \eta_2 - \eta_1\eta_2 - \eta_2 x_L$$
$$= 0.271 + 0.388 - 0.271 \times 0.388 - 0.388 \times 0.354$$
$$= 0.659 - 0.105 - 0.137 = 0.417$$

Thus, it is close to the value of 0.4163 obtained earlier.

Exergy fluxes and irreversibilities:

Given $$\psi = \frac{-\Delta G_o}{-\Delta H_o} = 1.0401 + 0.1728\left(\frac{h}{c}\right)$$

where (h/c) is the mass ratio of hydrogen to carbon in the fuel.

For octane (C_8H_{18}), $\psi = 1.0401 + 0.1729\left(\dfrac{18 \times 1}{8 \times 12}\right) = 1.0725$

$$\Delta H_o = w_f \times (CV)_o = 4.466 \times 44{,}430 = 198.424 \text{ kW}$$
$$\Delta G_o = \psi \times \Delta H_o = 1.0725 \times 198{,}424 = 212{,}810 \text{ kW}$$

$\therefore$ $$T_o\,\Delta S_o = \Delta G_o - \Delta H_o = 212.810 - 198{,}424 = 14{,}386 \text{ kW}$$

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Various amounts of lost work or exergy destruction due to irreversibility:

Compressor: Lost work or rate of energy dissipation or irreversibility

$$I_c = w_a T_o (s_2 - s_1)$$

where $$s_2 - s_1 = c_{p_a} \ln \frac{T_2}{T_1} - R_a \ln \frac{p_2}{p_1} \text{ where } R_a = \frac{c_p(\gamma - 1)}{\gamma}$$

$$= 1.006 \ln \frac{313 + 273}{298} - \frac{0.4 \times 1.006}{1.4} \ln 8$$

$$= 0.6795 - 0.5977 = 0.0818 \text{ kJ/kg K}$$

$\therefore$ $$I_c = 271.3 \times 298 \times 0.0818 = 6613 \text{ kW}$$

Combustor: Rate of energy dissipation or lost work in the combustor is

$$\dot{I}_{comb} = T_o [(\dot{S}_p)_3 - (\dot{S}_R)_2]$$

where $(\dot{S}_R)_2 = (\dot{S}_A)_2 + (\dot{S}_F)_o$ and the subscripts *P.R.A* and *F* represent products, reactants, air and fuel respectively.

or $$\dot{I}_{comb} = T_o [(\dot{S}_P)_3 - (\dot{S}_P)_o + (\dot{S}_P)_o - \{(\dot{S}_A)_2 - (\dot{S}_A)_o + (\dot{S}_F)_o + (\dot{S}_A)_o\}]$$

Now, $$\Delta\dot{S}_o = (\dot{S}_P)_o - [(\dot{S}_F)_o + (\dot{S}_A)_o]$$

$\therefore$ $$\dot{I}_{comb} = T_o [\{(\dot{S}_P)_3 - (\dot{S}_P)_o\} + \{(\dot{S}_A)_2 - (\dot{S}_A)_o\} + \Delta\dot{S}_o]$$

$$= T_o \left[\left\{ w_g c_{p_g} \ln \frac{T_3}{T_o} - w_g R_g \ln \frac{p_3}{p_o} \right\} - \left\{ w_g' c_{p_g} \ln \frac{T_2}{T_o} - w_a' R_a \ln \frac{p_2}{p_o} \right\} + \Delta\dot{S}_o \right]$$

$$= 275.8 \times 298 \left[1.148 \ln \frac{1173}{298} - \frac{1.148}{4} \ln 7.76 \right]$$

$$- 271.4 \times 298 \left[1.006 \ln \frac{586}{298} - \frac{1.006}{3.5} \ln 8 \right] + 14.386$$

$$= 80{,}947 - 6672 + 14{,}386 = 88{,}661 \text{ kW}$$

Gas turbine: Rate of energy dissipation in the gas turbine or lost work,

$$\dot{I}_{GT} = w_g T_o (s_4 - s_3)$$

where, $$s_4 - s_3 = c_{p_g} \ln \frac{T_4}{T_3} - R_g \ln \frac{p_4}{p_3}$$

$$= -1.148 \ln (1.553) - \frac{1.148}{4} \ln \frac{1.05}{7.76}$$

$$= -0.5053 + 0.5740 = 0.0687 \text{ kJ/kg K}$$

$\therefore$ $$\dot{I}_{GT} = 275.8 \times 298(0.0687) = 5646 \text{ kW}$$

HRSG: Rate of energy dissipation or lost work in the heat recovery steam generator,

$$I_{HRSG} = T_o \Delta\dot{S} = T_o[w_s (s_a - s_e) + w_g (s_6 - s_4)]$$

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where $s_a - s_e = 6.853 - 2.046 = 4.807$ kJ/kg K

$$s_6 - s_4 = c_{p_g} \ln \frac{T_6}{T_4} - R_g \ln \frac{p_6}{p_4}$$

$$= 1.148 \ln \frac{755}{520} + \frac{1.148}{4} \ln \frac{105}{100}$$

$$= -0.428 + 0.014 = -0.414 \text{ kJ/kg K}$$

$\therefore$ $\dot{I}_{HRSG} = 298 \times 29.235 \times 4.807 - 275.8 \times 298 \times 0.414$

$= 41{,}879 - 34{,}026 = 7853$ kW

Steam turbine: Rate of energy dissipation in the steam turbine or lost work

$$\dot{I}_{ST} = w_s (s_b - s_a) T_o$$

Now, $s_a = 6.853$ kJ/kg K, $h_a = 3272$ kJ/kg, $h_{bs} = 2064.37$ kJ/kg

$h_a - h_b = \eta_T (h_a - h_{bs}) = 0.82 \times 1207.63 = 990.26$ kJ/kg

$\therefore$ $h_b = 2281.74$ kJ/kg

$h_b = 121.46 + x_b \times 2432.9 = 2281.74$ kJ/kg

$\therefore$ $x_b = 0.89$

$\therefore$ $s_b = 0.4226 + 0.89 \times 8.052 = 7.589$ kJ/kg K

$\therefore$ $\dot{I}_{ST} = 29.235\,(7.589 - 6.853) \times 298 = 6412$ kW

Exhaust loss: Rate of exergy loss due to exhaust flue gases,

$$\dot{I}_{exh} = \int_{T_6}^{T_o} \left(1 - \frac{T_o}{T}\right) dQ = w_g\, c_{p_g} \left[(T_6 - T_o) - T_o \ln \frac{T_6}{T_o}\right]$$

$$= 275.8 \times 1.148 \left[(247 - 25) - 298 \ln \frac{520}{298}\right] = 17{,}760 \text{ kW}$$

Exergy balance:

Exergy input (kW)	*Power output (kW)*	*Exergy losses (kW)*	
$-\Delta\dot{G}_o = 212{,}810$	$W^o_{GT} = 53{,}744$	Compressor	: 6613
	$W^o_{ST} = 28{,}863$	Combustor	: 88661
		Gas turbine	: 5646
		HRSG	: 7853
		Steam turbine	: 6412
		Exhaust gases	: 17760
Input = 212,810 kW	Total output = 82607 kW	Total losses = 1,32,945 kW	

Exergy output + exergy destruction = 82,607 + 132945 = 215,552 kW which is 1.3% greater than the exergy input.

Exergetic or second law efficiency $= \dfrac{82607}{212810} = 0.388$ or 38.8%

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Example 3.5 For a mercury-steam-sulphur dioxide cycle, the heat rejected in the mercury cycle is given to the steam cycle and the heat rejected in the steam cycle is utilized in the SO_2 cycle. If the efficiencies of the mercury, steam and SO_2 cycles are 0.5, 0.4 and 0.25, respectively, find the overall efficiency of the composite cycle.

Solution For 3 cycles coupled in series the overall efficiency of the combined cycle is given by

$$\eta = 1 - (1 - \eta_1)(1 - \eta_2)(1 - \eta_3)$$
$$= 1 - (1 - 0.50)(1 - 0.40)(1 - 0.25)$$
$$= 1 - 0.5 \times 0.6 \times 0.75$$
$$= 0.775 \quad \text{or} \quad 77.5\%$$

SHORT-ANSWER QUESTIONS

3.1 What are the flaws of steam as the working fluid in a power cycle?
3.2 Explain the characteristics of an ideal working fluid in a power cycle.
3.3 What are the advantages of liquid metals as the working fluid in a topping cycle?
3.4 What are the drawbacks of liquid metals as the working fluid in the low temperature region of power cycle?
3.5 Why was the mercury-steam cycle abandoned commercially?
3.6 What is a tertiary cycle? What are the probable working fluids in such a cycle?
3.7 Show that the overall efficiency of a two-fluid coupled cycle is equal to the sum of the individual efficiencies of the two cycles minus their product.
3.8 What are the merits of multifluid coupled cycles?
3.9 What are the various types of combined cycle plants? What are the inherent advantages of a such a plant?
3.10 Explain the drawbacks of a stationary gas turbine power plant for generation of electricity?
3.11 What are the advantages of a stationary gas turbine for a utility system?
3.12 What do you understand by a peaking unit? How does a gas turbine adapt to the need of a peaking unit?
3.13 When there is heat loss between two cyclic plants coupled in series, how is the overall efficiency of the plant affected?
3.14 Show that when two cyclic plants operate in parallel, the overall efficiency lies between the efficiencies of these plants.
3.15 Show that the overall efficiency of a series-parallel plant is less than that of two cycles coupled in series.
3.16 Derive an expression for the overall efficiency of series-parallel plants with two plants in series having supplementary heat supply and heat loss in between the two plants.

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3.17 Discuss the effect of supplementary heat supply on the overall efficiency of a combined cycle plant.
3.18 What do you understand by limited supplementary firing and maximum supplementary firing in a combined cycle plant?
3.19 Discuss the advantages of combined cycle power generation. Why is it so important in the present day energy scenario?
3.20 Explain the dual-pressure steam cycle in a combined cycle plant. What is its thermodynamic advantage?
3.21 What is the need of an exergy analysis for a combined plant? Give briefly an exergy analysis of a simple GT-ST combined plant.
3.22 What do you understand by exergetic efficiency?
3.23 Show that the optimum pressure ratio of a gas turbine plant for maximum specific work is

$$(r_p)_{opt} = \left(\eta_T \eta_C \frac{T_{max}}{T_{min}}\right)^{\gamma/\{2(\gamma-1)\}}$$

3.24 Give the energy flow diagram (Sankey diagram) of a combined cycle plant.
3.25 Give the exergy flow diagram (Grassmann diagram) of a combined cycle plant. How does it differ from the energy flow diagram?
3.26 Explain the use of coal in a combined cycle plant.
3.27 What is integrated gasification combined cycle?
3.28 What is a gas-cleanup system? Why is it needed?
3.29 What is a PFBC? Explain its operation.
3.30 What is a PCFB? Explain its operation.
3.31 What is a STIG? Explain its merits and demerits.
3.32 What do you understand by repowering? Explain its importance.
3.33 Give an example with sketches of a conventional steam power plant before and after repowering.
3.34 How can a combined cycle plant be used for cogeneration? What is its thermodynamic advantage?
3.35 What is power coefficient? How does it affect the performance of a cogeneration plant?
3.36 What is the environmental impact of a combined cycle plant?
3.37 Discuss the scope of GT-ST combined cycle plant in the coming decades.

PROBLEMS

3.1 A binary vapour cycle operates on mercury and steam. Saturated mercury vapour at 4.5 bar is supplied to the mercury turbine, from which it exhausts at 0.04 bar. The mercury condenser generates saturated steam at 15 bar which is expanded in a steam turbine to 0.04 bar. (a) Find the overall efficiency of the cycle. (b) If 50,000 kg/h of steam flows through the steam turbine, what is the flow through the mercury turbine? (c) Assuming that all processes are reversible, what is the useful work done in the binary vapour cycle for the specified steam flow? (d) If the steam leaving the mercury condenser is

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superheated to a temperature of 300 °C in a superheater located in the mercury boiler, and if the internal efficiencies of the mercury and steam turbines are 0.85 and 0.87, respectively, calculate the overall efficiency of the cycle. The properties of saturated mercury are given below:

p(bar)	t(°C)	h_f	h_g	s_f	s_g	v_f	v_g
		(kJ/kg)		(k/kg K)		(m^3/kg)	
4.5	450	62.93	355.98	0.1352	0.5397	79.9×10^{-6}	0.068
0.04	216.9	29.98	329.85	0.0808	0.6925	76.5×10^{-6}	5.178

[*Ans.* (a) 53%, (b) 59.35×10^4 kg/h, (c) 28.5 MW, (d) 46.2%]

3.2 Show that the efficiency of Brayton cycle can be expressed as

$$\eta = \frac{\left(\frac{r-1}{r}\right)(\theta \eta_T \eta_C - r)}{(\theta - 1)\eta_C - (r-1)}$$

where $\theta = T_3/T_1 = T_{max}/T_{min}$, $r = (r_p)$ and r_p = pressure ratio.
Also, show that the optimum pressure ratio for maximum work output is given by

$$(r_p)_{opt} = (\eta_T \, \eta_c \, \theta)^{\gamma/2(\gamma-1)}$$

3.3 In an electric generating station, using a binary vapour cycle with mercury in the upper cycle and steam in the lower, the ratio of mercury flow to steam flow is 10 : 1 on a mass basis. At an evaporation rate of 10,00,000 kg/h for the mercury, its specific enthalpy rises by 356 kJ/kg while passing through the boiler. Superheating steam in the boiler furnace adds 586 kJ to the steam specific enthalpy. Mercury gives up 251.2 kJ/kg during condensation, and steam gives up 2003 kJ/kg in its condenser. The overall boiler efficiency is 85%. The combined turbine mechanical and generator efficiencies are each 95% for the mercury and steam units. The steam auxiliaries need 5% of the energy generated by the units. Find the overall efficiency of the plant.

3.4 A sodium-mercury-steam cycle operates between 1000 °C and 40 °C. Sodium rejects heat at 670 °C to mercury. Mercury boils at 24.6 bar and rejects heat at 0.141 bar. Both sodium and mercury cycles are saturated. Steam is formed at 30 bar and is superheated in the sodium boiler to 350 °C. It rejects heat at 0.08 bar. Assume isentropic expansions, no heat losses, no regeneration and neglect pumping work. Find (a) the amounts of sodium and mercury used per kg of steam, (b) the heat added and rejected in the composite cycle per kg steam, (c) the total work done per kg steam, (d) the efficiency of the composite cycle, (e) the efficiency of the corresponding Carnot cycle, and (f) the work, heat added, and efficiency of a supercritical steam (single fluid) cycle operating at 250 bar and between the same temperature limits.
For mercury, at 24.6 bar, h_g = 366.78 kJ/kg, s_g = 0.48 kJ/kg K, and at 0.0141 bar, s_f = 0.09 and s_g = 0.64 kJ/kg K, h_f = 36.01 and h_g = 330.77 kJ/kg.

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For sodium, at 1000 °C, h_g = 4982.53 kJ/kg
At turbine exhaust, h = 3914.85 kJ/kg
At 670 °C, h_f = 745.29 kJ/kg
For a supercritical steam cycle, the specific enthalpy and entropy at the turbine inlet may be computed by linear extrapolation from the steam tables.

3.5 In a combined gas turbine-steam turbine plant, the exhaust gas from the GT being the supply gas to the steam generator at which a further supply of fuel is burned in the gas. The pressure ratio for the GT is 8, the inlet air temperature is 15 °C and the maximum cycle temperature is 800 °C.
Combustion in the steam generator raises the inlet temperature of the exhaust gas from the GT to 800 °C and the gas leaves the steam generator at 100°C. The condition of steam at ST inlet is 60 bar, 600 °C and the condenser pressure is 0.05 bar, Calculate the flow rates of air and steam required for the total power output of 190 MW and the overall efficiency of the combined plant. Assume ideal processes. What would be the overall air fuel ratio? Take c_p of combustion gases and air as 1.11 and 1.005 kJ/kg K, and γ for gases and air as 1.33 and 1.4, respectively. Neglect the effect of mass flow rate of fuel on the air flow. Take the calorific value of the fuel as 43.3 MJ/kg.

[*Ans.* 4.54, 0.54 and 39.6]

3.6 A combined power plant consisting of a closed cycle GT unit (Brayton cycle), using air as the working fluid and a ST unit (Rankine cycle) is to be designed such that the heat rejected at the GT unit is to be utilized to produce steam at the generator for the ST. The air leaving the generator is at 200 °C and it is cooled to the compressor inlet temperature by a second cooler which rejects the heat to waste.
Draw the flow and *T-s* diagram of the plant and calculate on the basis of ideal cycles, neglecting the feed pump work of the Rankine cycle, (a) the mass flow rate of steam per kg/s of air flow, (b) the total power output per kg/s of air flow, and (c) the overall efficiency of the plant. Take c_p of air as 1.005 kJ/kg K and γ= 1.4. The particulars of each cycle are as follows:

		GT cycle (Brayton)	*ST cycle (Rankine)*
Pressure (bar)	min	1	0.07
	max	5	30
Temperature (°C)	min	20	—
	max	830	300

[*Ans.* (a) 0.0762 kg/s, (b) 314.78 kJ/kg, (c) 49%]

3.7 A combined cycle power plant has a total power output of 300 MW. The gas turbine operates with a pressure ratio of 10, air inlet temperature of 30 °C and the maximum gas temperature of 1000 °C. There is the provision for supplementary firing in which the combustion of additional fuel raises the gas temperature to 900 °C. The exhaust gas from the GT flows to a HRSG from which the gas leaves at 110 °C. In the bottoming steam plant the steam is supplied to the turbine at 80 bar, 500 °C and the condenser pressure is 0.1 bar. The calorific value of fuel burned is 43.2 MJ/kg. Neglect the effect of

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the mass flow rate of fuel on the air flow and take c_p = 1.11 kJ/kg K and γ = 1.33 for combustion gases and c_p = 1.005 kJ/kg K and γ = 1.4 for air. Neglect pump work. Determine (a) the flow rates of air and steam required, (b) the power outputs of the gas turbine and steam turbine, (c) the overall efficiency of the combined plant, and (d) the air-fuel ratio. Draw the flow and T-s diagrams.

[*Ans.* (a) 442.62 kg/s, 121.056 kg/s, (b) 146.47 MW, 153.53 MW, (c) 53.5%, (d) 34.1]

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4

Fuels and Combustion

The primary fuels which are burned to release heat and generate steam in boilers are the fossil fuels in the form of coal, fuel oil and natural gas, which represent the remains of plant and animal life that are preserved in the sedimentary rocks. Besides these, industrial wastes like blast furnace gas, coke oven gas, refinery gas, sugar factory refuse (bagasse), saw mill wood dust, rice husk, etc. are also used as boiler fuels, often to boost one of the primary fossil fuels. When more than one type of fuel is simultaneously burned to meet the total heating requirement, the boiler is said to have a combination firing.

4.1 COAL

Coal is the principal energy source, particularly in India because of its large deposits and availability. Coal originated from vegetable matter which grew millions of years ago. Trees and plants falling into water decayed and later produced peat bogs. Huge geological upheavals buried these bogs under layers of silt. Subterranean heat, soil pressure and movement of earth's crust distilled off some of the bog's moisture and hardened it to form brown coal or lignite. Continuing subterranean activity and metamorphosis produced higher grades of coal. According to geological order of formation, coal may be of the following types: (1) Peat, (2) Lignite, (3) Subbituminous, (4) Bituminous, (5) Subanthracite, and (6) Anthracite, with increasing percentages of carbon. After anthracite, graphite is formed. Anthracite contains more than 86% fixed carbon (in amorphous form) and less volatile matter. Volatile matter helps in the ignition of coal. So, it is often difficult to burn anthracite. Bituminous coal is the largest group containing 46–86% of fixed carbon and 20–40% of volatile matter. It can be low-volatile, medium-volatile and high-volatile. The lower the volatility, the higher the heating value. Lignite is the lowest grade of coal containing moisture as high as 30% and high volatile matter. According to ASTM (American Society of Testing and Materials), peat is not regarded as a rank of coal. Peat contains up to 90% moisture and is not attractive as a utility fuel. Rank carries the meaning

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of degree of maturation (carbonisation) and is a measure of carbon content in coal. Lignite is considered to be low rank and anthracite to be high rank.

4.2 COAL ANALYSIS

There are two types of coal analysis: proximate and ultimate, both done on a mass per cent basis. Both these types may be based on: (a) as-received basis, useful for combustion calculations, (b) dry or moisture free basis, (c) dry mineral-matter-free or combustible basis.

4.2.1 Proximate Analysis

The proximate analysis indicates the behaviour of coal when it is heated. When 1 g sample of coal is subjected to a temperature of about 105 °C for a period of 1 hour, the loss in weight of the sample gives the moisture content of the coal.

When 1 g sample of coal is placed in a covered platinum crucible and heated to 950 °C and maintained at that temperature for about 7 min, there is a loss in weight due to the elimination of moisture and volatile matter. The latter may now be determined since moisture has been calculated from the previous test. Volatile matter consists of hydrogen and certain hydrogen–carbon compounds which can be removed from the coal simply by heating it.

By subjecting 1 g sample of coal in an uncovered crucible to a temperature of about 720 °C until the coal is completely burned, a constant weight is reached, which indicates that there is only ash remaining in the crucible. Complete combustion of coal is determined by repeated weighing of the sample.

Fixed carbon is the difference between 100% and the sum of the percentages of moisture, ash and volatile matter. However, this difference does not represent all the carbon that was in the coal. Some of the carbon may have been in the form of hydrocarbons which may have been distilled off while determining the volatile matter. It is also possible that some of this fixed carbon may include sulphur, nitrogen and oxygen. So, the proximate analysis of coal gives

$$FC + VM + M + A = 100\% \text{ by mass} \tag{4.1}$$

The amount of VM indicates whether the coal will burn with a short or long flame and whether it will tend to produce smoke. The more volatile the coal, the more it will smoke.

Figure 4.1 shows the trend in moisture, violatile matter and fixed carbon when expressed on a dry ash-free basis. The general trend with increasing rank is an increase in the heating value and fixed carbon and a corresponding decrease in moisture and VM. This trend is so pronounced that a classification system based on the fuel ratio (ratio of fixed carbon to volatile matter) has been used as a rough indicator of a coal's rank.

Lower rank coals (lower fuel ratio) are characterised by a greater oxygen content, that aids ignition and enhances combustibility and flame stability. High combustibility improves carbon burnout (reduces carbon carryover) and hence

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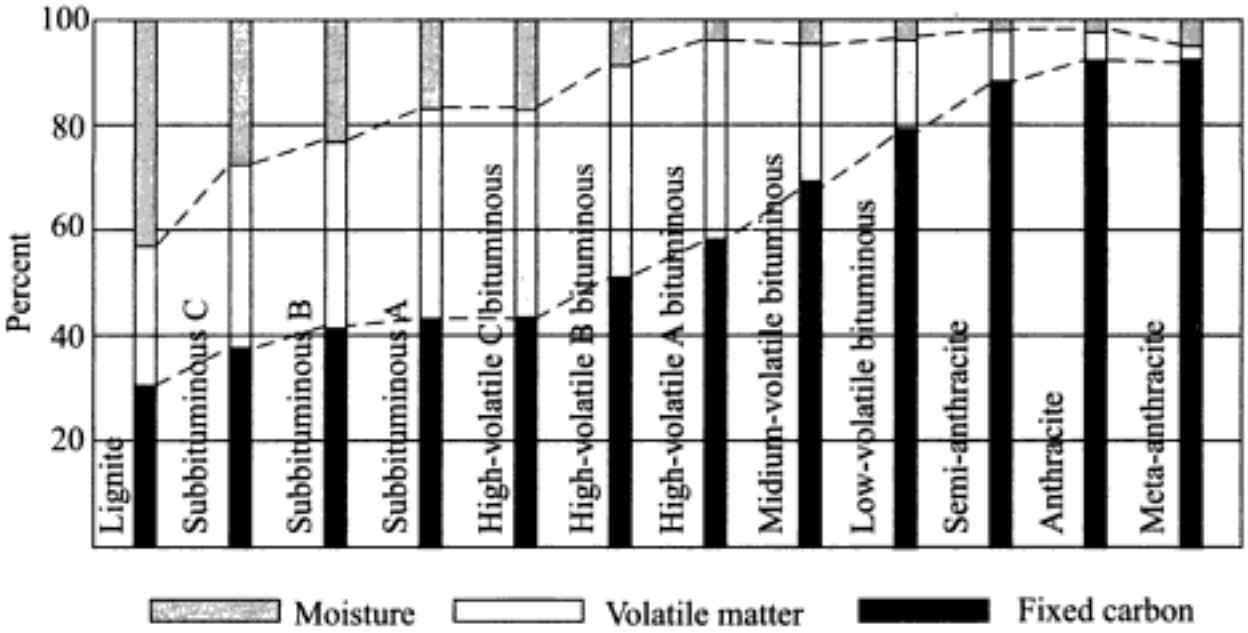

Fig 4.1 *Coal rank compared with proximate analysis*

boiler efficiency and for pulverized coal-fired units, this allows the coal to be ground to a coarser size. Low rank coals (high moisture content) produce a "self-pulverization" of the coal particles during combustion. As the inherent moisture in the pore structure of the coal is heated and expands rapidly, its volume increases (as water flashes to steam at atmospheric pressure, the volume expansion is 1600 to 1), thus fragmenting the coal particles. This exposes more surface area for combustion.

4.2.2 Ultimate Analysis

The ultimate analysis gives the chemical elements that comprise the coal substance, together with ash and moisture. The coal substance consists of organic compounds of carbon, hydrogen, and oxygen derived from the original vegetable matter. The analysis shows the following components on mass basis: carbon (C), hydrogen (H), oxygen (O), nitrogen (N), sulphur (S), moisture (M) and ash (A). Therefore,

$$C + H + O + N + S + M + A = 100\% \text{ by mass} \tag{4.2}$$

The dry and ash free analysis on combustible basis is obtained on dividing C, H, O, N and S by the fraction $\left(1 - \frac{M + A}{100}\right)$.

4.2.3 Coal Properties

There are certain properties of coal which are important in power plant applications. They are swelling index, grindability, weatherability, sulphur content, heating value and ash softening temperature.

Swelling index Some types of coal during and after release of volatile matter become soft and pasty and form agglomerates. These are called *caking*

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coal. In a fixed bed, such as a travelling grate stoker, the coal must not cake as it burns. The consequent agglomeration disturbs greatly the availability of air and so the coal does not completely burn yielding low combustion efficiency. Coal that does not cake is called *free-burning coal*. It breaks apart during combustion exposing large surface area to the air, thus enhancing the combustion process. Caking coals are used to produce coke by heating in a coke oven in the absence of air, with the volatile matter driven off. Coal devoid of volatile matter is called coke, which is largely needed in steel plants. A qualitative evaluation method, called the *swelling index*, has been devised to determine the extent of caking of a coal. A free-burning coal has a high value of swelling index, which indicates that it somewhat expands in volume during combustion. When modern pulverized coal burners are used, the swelling property of coal is, however, of less importance.

Grindability Grindability is often an important criterion for selecting a coal. This property of coal is measured by the standard *grindability index*, which is inversely proportional to the power required to grind the coal to a specified particle size for burning. Grindability of a standard coal is defined as 100. If the coal selected for use at a power plant has a grindability index of 50, it would require twice the grinding power of the standard coal to produce a specified particle size.

Weatherability It is a measure of how well coal can be stockpiled for long periods of time without crumbling to pieces. Modern power plants normally stockpile 60 to 90 days' supply of coal in a large pile near the power plant. The coal unloaded from wagons is packed in a long trapezoidal pile. Excessive crumbling or weathering of the coal due to climatic conditions may result in small particles of coal which can be dispersed by wind or rain.

Sulphur content Sulphur content in coal is combustible and generates some energy by its oxidation to SO_2. Sulphur dioxide is a major source of atmospheric pollution. Their is an environmental regulation on SO_2 emission. The operating cost of SO_2 removal equipment need be considered while selecting a coal with high sulphur content.

Heating value The heating value or calorific value of coal is a property of fundamental importance. It may be determined on as-received, dry, or dry-and-ash-free basis. It is the heat transferred when the products of complete combustion of a sample of coal (or other fuel) are cooled to the initial temperature of air and fuel. It is normally determined in a standard test in a bomb calorimeter, where a coal sample of known mass is burnt with pure oxygen supply completely in a stainless steel bomb or vessel surrounded by a known mass of water, and the rise in water temperature is noted. Two different heating values are cited for coal. The higher heating value (HHV) assumes that the water vapour in the products condenses and thus includes the latent heat of vaporization of the water vapour formed by combustion. The lower heating value (LHV) assumes that the water vapour formed by combustion leaves as vapour itself. Therefore,

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$$\text{LHV} = \text{HHV} - m_w h_{fg} \tag{4.3}$$

where m_w is the mass of water vapour formed given by

$$m_w = M + 9H + \gamma_A w_A \tag{4.4}$$

where M and H are the mass fractions of moisture and hydrogen in the coal, γ_A is the specific humidity of atmospheric air and W_A is the actual amount of air supplied per kg of coal. For energy balance and efficiency calculations of steam generators, HHV of fuel is considered in the USA, whereas LHV is the standard used in European practice.

If the ultimate analysis is known, the HHV of anthracite and bituminous coals can be determined approximately by using Dulong and Petit formula as given below:

$$\text{HHV} = 33.83\ \text{C} + 144.45\left(\text{H} - \frac{0}{8}\right) + 9.38\ \text{S}, \quad \text{in MJ/kg} \tag{4.5}$$

where C, H, O and S are mass fractions of carbon, hydrogen, oxygen and sulphur in coal. Assuming the latent heat of vaporization h_{fg} at the partial pressure of water vapour in the combustion products as 2.395 MJ/kg, the lower heating value of coal from Eq. (4.3) is given by

$$\text{LHV} = \text{HHV} - 2.395\ m_w, \qquad \text{in MJ/kg} \tag{4.6}$$

For lower-rank fuels, Eq. (4.5) usually underestimates the HHV.

Ash softening temperature The ash softening temperature is the temperature at which the ash softens and becomes plastic. This is somewhat below the melting point of ash. The design of the steam generator greatly depends on the ash softening temperature (a.s.t.) of the coal. If the furnace temperature is higher than the a.s.t., all the ash will melt and would come out of the furnace bottom continuously as molten slag. For a furnace that would discharge ash in the solid form, a high ash softening temperature would be required. A stoker furnace must use coal with a high a.s.t., otherwise clinkers would be formed. Clinkers, which are large masses of fused ash, cause troubles in discharge and also make combustion inefficient.

Spontaneous combustion Combustion (oxidation) of coal can take place rapidly as in a furnace or slowly on a stockpile. If it takes place slowly, there is a degradation or loss of energy content and hence in the value of fuel. The factors which influence spontaneous combustion and which can lead to a big fire, are the following.

(a) Rank of coal, low rank coals are more susceptible because of their higher porosity.
(b) Amount of surface area exposed to air.
(c) Ambient temperature, with high solar insolation aiding it.
(d) Oxygen content of coal.
(e) Free moisture in coal
(f) Configuration of the coal stockpile: steep conical piles with coarse coal at the edges and fines near the top are more susceptible because they

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promote natural convection (chimney effect) and good air flow through the pile to support combustion as it develops.

To prevent spontaneous combustion, it is important to maintain a dry pile and compaction at regular intervals.

4.3 FUEL OIL

Petroleum is believed to have been formed during past geological ages from decayed marine life, both vegetable and animal. Dead marine animals and vegetable matter accumulated for millions of years ultimately got transformed into oil, mainly in sedimentary rocks, by pressure and heat. Oil deposits accumulated in the rocks and sands below the earth's crust. Oil generally has a body of water below and pressurized natural gas above. Fairly thick and dense earth strata (caprock) cover most deposits preventing seepage (Fig. 4.1a). Oil wells drilled through this layer penetrate the deposits. The pressure forces the gas and oil to the surface. After the pressure has diminished, the oil must be pumped.

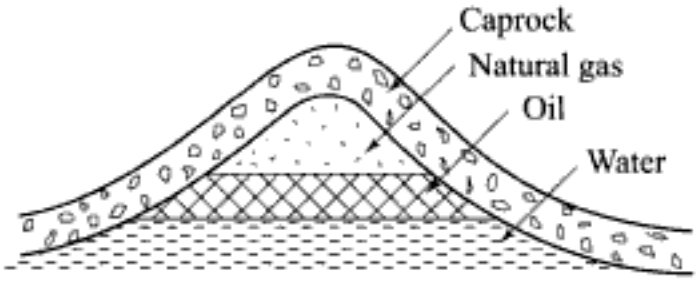

Fig. 4.1(a) *An oil deposit*

Liquid fuels are an excellent energy source. They are easy to handle, easy to store and easy to burn. They have nearly constant heating values. They are primarily a mixture of hydrocarbon compounds, which may also contain nitrogen, oxygen and sulphur. The bulk of the hydrocarbons belong to the paraffin series, C_nH_{2n+2}, like methane (CH_4), ethane (C_2H_6), propane (C_3H_8) and butane (C_4H_{10}) which are gaseous, and pentane (C_5H_{12}), hexane (C_6H_{14}) and octane (C_8H_{18}) which are liquid at STP. In addition, there can be isoparaffins, cycloparaffins and aromatic compounds. The proportion of various hydrocarbon groups varies widely with geographic location. The ultimate analysis of oil indicates the mass fractions of carbon, hydrogen, oxygen, nitrogen and sulphur. Regardless of the crude oil source, its composition is fairly uniform within close limits, as given below

Carbon 83–87%, Hydrogen 11–16%,

Oxygen + Nitrogen 0–7%, Sulphur 0–4%

There can also be some moisture and sediment.

Crude oil is seldom used as such. In the refining process it is distilled into a number of fractions. The lighter fractions (having lower boiling point) like

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gasoline, aviation fuel, kerosene, light diesel oil, heavy diesel oil, lubrication oil and so on are principally transportation and machine fuels. The heavier fractions are used for boiler fuels and chemical production.

The required physical properties of fuel oil are specific gravity, viscosity, pour point, flash point and heating value. The *pour point* is the lowest temperature at which the oil will flow under standard pressure conditions. The *flash point* is the minimum temperature at which the oil may be ignited.

4.4 NATURAL AND PETROLEUM GAS

Natural gas was formed millions of years ago from decaying vegetable matter generally along with petroleum. Oil wells drilled into the geologic formation containing the gas and trapped oil release gas predominantly. In some fields, the natural gas is flared or burned at the wellhead because of lack of facilities to transport the gas. In regions close to the source, natural gas has been used as a power plant fuel.

Transportation of natural gas is made through pipelines. In India there is the HBJ (Hazira-Bijaipur-Jagdishpur) pipeline which transports natural gas from Hazira in Gujarat to Jagdishpur in U.P. Natural gas is the cleanest of all fossil fuels. It is free from ash and mixes well with air to undergo complete combustion producing very little smoke. It consists of a mixture of the most volatile paraffins-methane to pentane. It has a high hydrogen content and produces a considerable amount of water vapour when burned. The heat of combustion varies from 33.5 to 40 MJ/m^3. The specific gravity is 0.63 relative to air.

Since the major constituent of all natural gases is methane (critical temperature – 83 °C), cryogenic temperatures are required to maintain the gas as a liquid at moderate pressures (e.g. – 100 °C at 36 bar). Liquid natural gas (LNG) is transported by special tankers and stored in spherical pressure vessels to be used when needed, particularly during peak load.

Compressed natural gas (CNG) is now being considered as the alternative fuel for automobiles. Liquid petroleum gas (LPG) refers to hydrocarbons, such as propane, propylene, butane, butylene and so on, which are liquefied under moderate pressures and at normal temperatures. It is used widely as domestic fuel and also to supplement natural gas flow.

4.5 EMULSION FIRING

A suspension of a finely divided fluid in another is called an *emulsion*. An emulsion of water in heavy oil has been tried for boiler firing. When atomized, the drops of such an emulsion undergo microexplosions of the entrained water as they enter the hot combustion chamber. This causes additional atomization and enhances fuel surface-to-volume ratio. This promotes combustion, reduces carbon loss by way of soot etc. It also reduces excess air requirement and improves combustion efficiency. It can help conserve fuel oil.

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4.6 COAL–OIL AND COAL–WATER MIXTURES

Liquid fuels using mixtures of fine coal in oil have been known as colloidal fuel, coal–in–oil and more recently, coal–oil mixtures (COM). This technology has been of interest since the late nineteenth century and tried on a limited scale since 1930, with promising results. In the fifties and sixties, oil and natural gas became progressively cheaper so that economic balance was shifted decidedly in favour of oil and gas–fired plants. After 1973, the price of oil and gas rose sufficiently and the interest in the use of COM was revived again. The main aspects of its preparation and burning are: (1) Grinding of coal to the desired size, (2) Mixing of coal in oil, (3) Flow of COM in pipes, (4) Stability of the colloidal fuel in storage, (5) Combustion of the colloidal fuel, and (6) Economics of its manufacture and use. For coal weight concentrations up to 40% the colloidal mixture obeys the fundamental law for viscous fluid flow in laminar range. It may be transported in pipes without clogging and fed through oil burner nozzles for combustion. For stabilizing the mixture, aluminium stearate and spent alkylation acid could be used (Lord *et al.*, 1982). It has been found that combustion of colloidal fuels up to 40% coal concentration with 99% of the coal powder passing through a 230 mesh (61 μm) was better than, or equal to that of coal or oil alone.

Advantages of COM (1) It is cheaper than oil. (2) It can be handled like oil in pipes and by pumps. (3) It has a combustion efficiency that equals or exceeds that of oil. (4) It enables better utilization of storage space than does coal. (5) It assists in conservation of petroleum resources. (6) It eliminates the danger of spontaneous combustion in stored coal. (7) It is heavier than water, so fires are extinguished more easily. (8) It has an ash and moisture content less than that of coal. (9) It better utilizes low-rank coals. (10) It can be burned in present oil-burning equipment with efficiencies exceeding 80%.

Coal–water mixtures (CWM) having greater ability than COM to replace oil are often preferred. The CWM contains 70–80% coal mass, the rest water, plus a fraction of a percent of a stabilizer. Fine coal is suspended in the water and with the injection of a suitable stabilizer it is agitated to form a uniform slurry. The slurry can be handled. stored, transported, and fired like oil. There is a reduction in the heating value, since the water evaportating absorbs the latent heat from the energy released by combustion. So, the steam generator may require to be de-rated.

4.7 INDUSTRIAL WASTES AND BYPRODUCTS

Industrial wastes which are combustible are receiving increased attention as fuels for steam generators. They, thus, serve the twin objective, one is to dispose of them and the other is to reduce the use of oil.

Gaseous byproducts are very attractive, particularly the refinery gas and the coke oven gas. *Refinery gas* is generated during the conversion of crude oil to gasoline and other refinery products. It has a high heating value and is often

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blended with lower heating value gas byproducts from the refinery prior to combustion. *Coke-oven* gas is generated during the production of coke from raw coal in a coke oven where the volatile matter is distilled off and the coke-oven gas is separated from the volatile matter by cooling and extraction. It consists of about 50% hydrogen, 33% methane and the remaining 17% of various other gases (by volume). Its heating value varies from 14.2 to 21.3 MJ/m^3.

Regenerator gas produced in refinery by catalytic-cracking processes has about 10% CO and high inert gas and solid contents available at a high temperature of more than 540 °C, and can be burned at the site. *Blast furnace* gas generated during reduction processes of iron oxide to iron in the blast furnace contains about 30% CO, but has a high dust loading. It requires cleaning prior to burning, otherwise, it would cause plugging of burners and fouling of heat transfer surfaces.

The liquid wastes include solvents, waste oil and oil sludges, oil-water emulsions, polymers, resins, chlorinated hydrocarbons, phenols, tars, combustible chemicals, greases and fats. The main disadvantage of such liquids is that they vary considerably in heating value, flash and fire points, viscosity and moisture content.

There are some solid wastes which are available as fuels, the notable ones being wood waste and sugar factory refuse. *Wood waste* is found in saw mills and in paper, pulp, furniture, plywood and similar industries. Most woods have about the same chemical composition having a proximate analysis: fixed carbon 25%, volatile matter 70–75%, and ash 0.5–5%, on a dry basis. They vary in density and moisture content. Their heating value ranges between 19.3 to 21.0 MJ/kg. To burn wood efficiently, it must be cut down to chip size to permit continuous feeding and should not have a moisture content exceeding 60 percent. Wood burns more cleanly than oil or coal and causes much less pollution. A few wood-burning power plants are operating in the USA.

Sugarcane waste, also called *bagasse*, is that portion of sugarcane which remains after the sugar juice is extracted. Consisting mainly of cellulose fibres, its proximate analysis is about 12.5% fixed carbon, 84% volatile matter and 3.5% ash, on a dry basis. It contains more thatn 50% moisture. To burn it, it is cut to short fibres and fines. Its heating value ranges between 8.4 to 9.77 MJ/kg. Sugar mills generate enough bagasse to meet the demand for cogeneration of both process steam and electricity. Numerous bagasse fired steam and electricity generating plants are in operation near sugar mills throughout the world for decades.

Solid wastes generated by industrial and domestic processes as garbage are often used as boiler fuel in incinerators, particularly in cities, for generation of electricity or process steam.

Apart from municipal wastes and refuse-derived fuels (RDF), another waste that has become popular in recent years is used tyres because of its high heating value (about 32 MJ/kg). Some power plants in the USA burn automobile and truck tyres as their sole fuel. Used tyres are also being burned at coal-fired stations as supplemental fuel similar to municipal refuse.

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4.8 SYNTHETIC FUELS

Synthetic fuels, also called *synfuels*, are gaseous and liquid fuels produced largely from coal in an economical and environmentally acceptable manner. These fuels cannot only be used for steam generation in power plants but also for domestic, industrial and transportation purposes.

4.8.1 Coal Gasification

In the ninteenth century and the first part of the twentieth century there was widespread use of coal gas produced by destructive distillation of coal for illuminating and cooking purposes. On heating coal in absence of air, coal was carbonized to coke by removing its volatiles and using them as a byproduct gas. This gas was distributed in urban areas as *town gas*. The coke thus produced was burned in big beds with less than the stoichiometric quantity of air to yield the *producer gas*:

$$\underset{\text{Coke}}{2C} + \underset{\text{Air}}{(O_2 + 3.76N_2)} \longrightarrow \underset{\text{Producer gas}}{2CO + 3.76\,N_2} \tag{4.7}$$

For each mole of oxygen in air there are 3.76 moles of nitrogen. When the bed was heated to a high temperature, the flow of air was replaced by a flow of steam and water gas was produced:

$$\underset{\text{Coke}}{C} + \underset{\text{Steam}}{H_2O} \longrightarrow \underset{\text{Water gas}}{CO + H_2} \tag{4.8}$$

Water gas reaction is endothermic and the bed gets cooled. The steam flow is then replaced by air flow and exothermic partial combustion, Eq. (4.7), reheated the bed. Air flow is again replaced by steam flow. The alternate production of water gas and producer gas continued till the coke bed was exhausted.

Modern gasification techniques are based on the producer and water gas reactions. However, they are more efficient and the production processes are continuous.

The basic process involves several steps. The coal is first ground into a sand-like powder and then preheated and dried to reduce caking during conversion. For a caking coal a high-temperature pretreatment is often used to give the coal particles a thin coating of oxygen to prevent sticking. Figure 4.2 illustrates the basic steps in the gasification process.

Coal gasification results in three gas mixtures, classified according to their heating value. They are called low-heating value, medium-heating value, and high-heating value gas. The process illustrated by the lower flow sheet (Fig. 4.2) yields a *low-heating value gas* (which is a mixture of water gas and producer gas), called *synthesis gas*. The process illustrated by the upper flow sheet yields a *high-heating value gas*, called *pipeline gas*, with properties close to that of natural gas.

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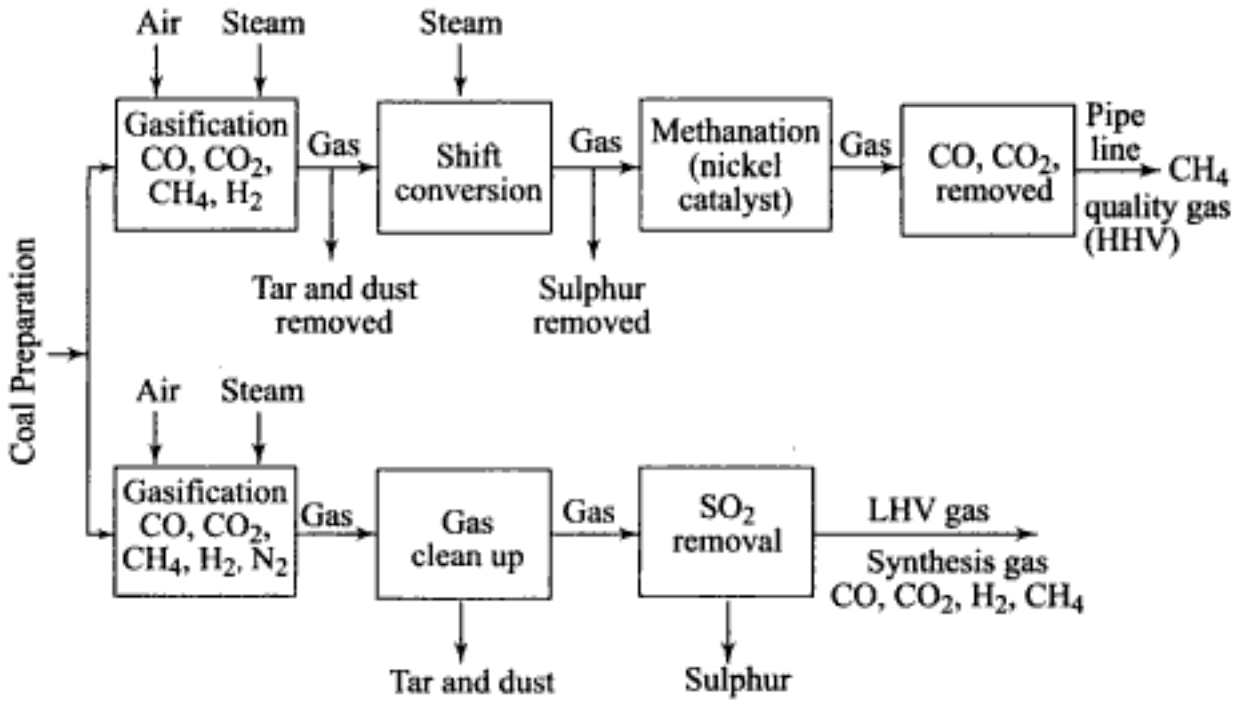

Fig. 4.2 *Coal gasification processes*

1. The feedstock is reacted with air and steam. The air quantity is less than stoichiometric.

$$C + O_2 + 3.76\,N_2 \longrightarrow CO_2 + 3.76\,N_2 \tag{4.9}$$

CO_2 from this reaction reacts further with additional carbon in the rich mixture to give

$$C + CO_2 + 3.76\,N_2 \longrightarrow 2CO + 3.76\,N_2 \tag{4.10}$$

In steam, $$C + H_2O \longrightarrow CO + H_2 \tag{4.11}$$

The result is low-heating value syngas containing CO, H_2, N_2 and some CO_2. It may also contain some CH_4.

2. To obtain *medium-heating value gas, a shift reaction* is used to produce additional hydrogen, and nitrogen is removed.

$$CO + H_2O \longrightarrow CO_2 + H_2 \tag{4.12}$$

3. To produce pipeline gas, a catalytic methanation is carried out in which the products of water gas reaction are reacted over a nickel catalyst at a temperature of about 1100°C and a pressure of 6.8 bar.

$$CO + 3H_2 \longrightarrow CH_4 + H_2O \tag{4.13}$$

The product gas is high-quality gas having a heating value of about 38 MJ/m^3 and can directly substitute for natural gas.

There is another method of classification, called hydrogasification in which fluidized coal is gasified directly with hydrogen-rich steam to a methane-rich gas that requires very little additional shifting. The overall reaction is of the form

$$CH_{0.8} + 0.55H_2O + 1.15H_2 \longrightarrow 0.575CH_4 + 0.425\,CO_2 \tag{4.14}$$

The overall efficiency of conversion by this method is higher than the earlier method.

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An attractive application of low-heating value gas in electric generation is as a fuel for a combined cycle power plant employing gas turbine and steam turbine as discussed in Chapter 3.

Notable commercial gasification systems are the Lurgi process, Winkler gasifier and Koppers-Totzek process. In the synthane process developed by US Burease of Mines, dry crushed coal is fed from a pressurized hopper into a pretreater (Fig. 4.3). The coal is reacted with oxygen and steam at 400 °C, 7 MPa to devolatilize it and prevent caking. The coal mixture flows to the gasifier where it is partially gasified in a dense phase at 800 °C, and later at 1000°C, 7 MPa. Unreacted char and ash are removed from the bottom of the vessel. The synthesis gas passes through a cyclone separator which removes dust and tar. After further cleaning and sulphur removal, the gas can be used as a low-heating value gas. It can be further reacted through a catalytic methanation process to produce pipeline quality gas. Some other commercial gasifiers have been discussed in Chapter 5.

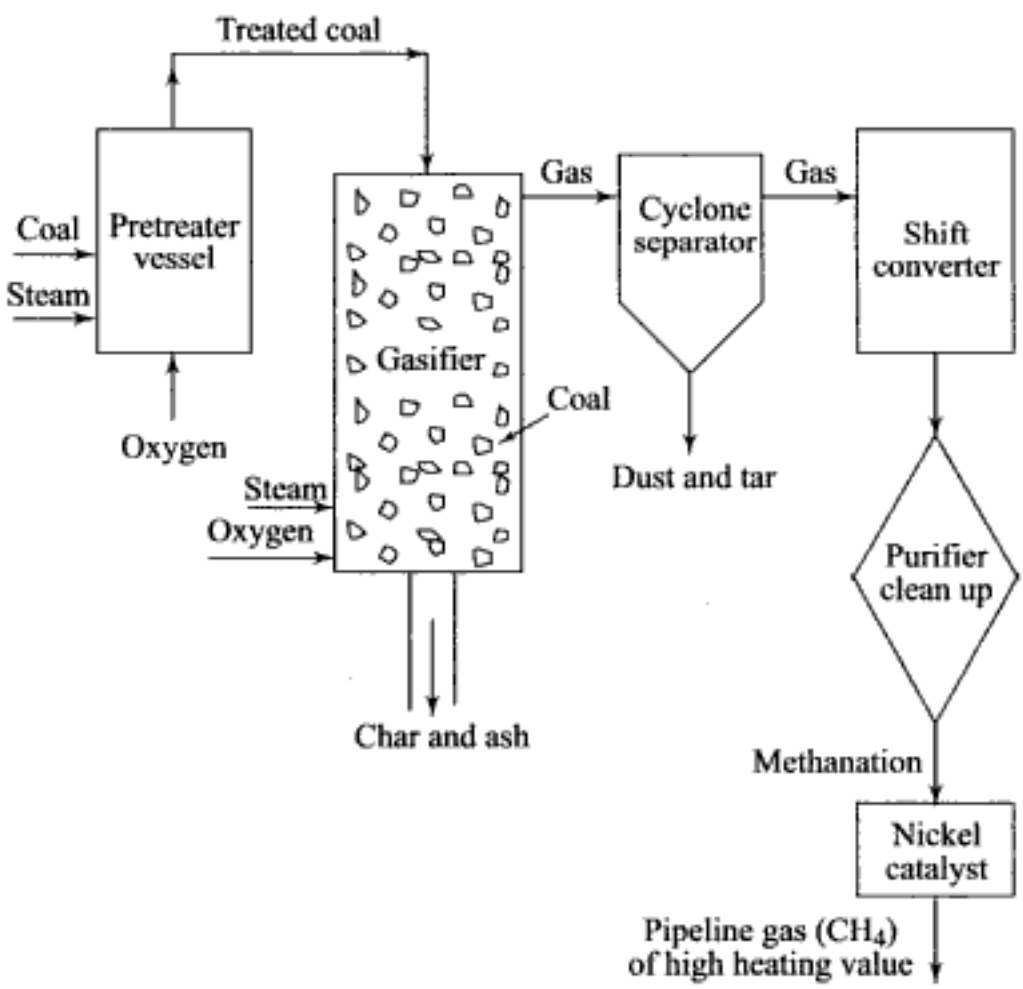

Fig. 4.3 *Synthane coal gasification process*

4.8.2 Underground Coal Gasification

Deposits of coal that could not be mined competitively by other processes might be economically utilized by underground gasification. Coal is gasified *in situ* and the gas produced is conveyed to the surface and utilized to meet various needs. *In situ* coal gasification became attractive due to the following advantages:

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1. It can extract energy from inaccessible reserves of coal that cannot be mined by conventional techniques, e.g. steeply inclined seams.
2. When the mines become very deep, it becomes uneconomical to raise coal from such deep mines. *In situ* gasification can help raise energy from such abandoned mines.
3. It reduces mining personnel and equipment and needs less coal-handling and transportation facilities.
4. It is much safer and minimizes occupational hazards.
5. Fuel gas produced by *in situ* gasification is cheaper than other energy forms.
6. When coal seams are thin, a higher recovery of the coal via *in situ* gasification is possible.
7. From the environmental standpoint, hydrogen sulphide (H_2S) rather than SO_2 is the predominant form of sulphur produced in underground coal gasification. It can be economically treated by hot carbonate scrubbing (Lamb, 1977).

Two deep drilled holes are made, one for air and the other for product gas (Fig. 4.4). After initial ignition, combustion takes place at the bottom of the air hole and the combustion zone proceeds toward the product gas hole. In the combustion zone, carbon dioxide is formed in the reaction:

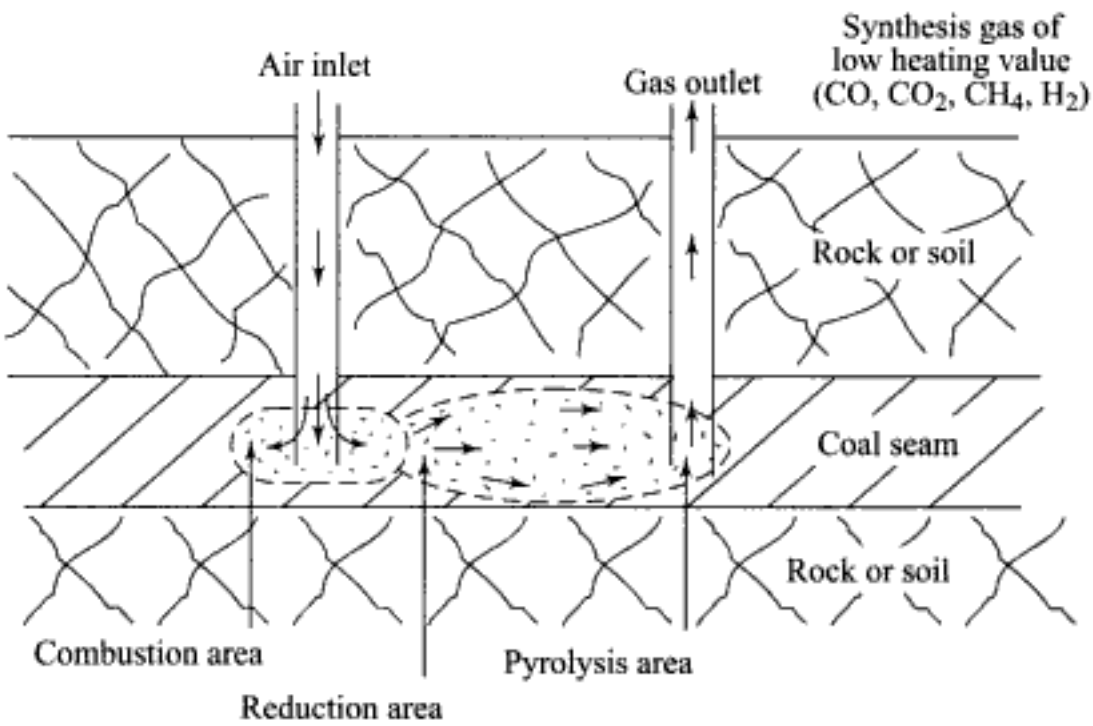

Fig. 4.4 *In-situ coal gasification*

$$C + O_2 \longrightarrow CO_2 \tag{4.15}$$

Ahead of the combustion zone is a reduction zone where carbon monoxide is formed.

$$CO_2 + C \longrightarrow 2CO \tag{4.16}$$

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From moisture in the coal, water gas reaction can take place

$$C + H_2O \longrightarrow CO + H_2 \tag{4.17}$$

Volatile matter is also released from the coal as the process proceeds. The product gases are very similar to the coal gas or water gas, as described earlier. Thus the heating value is low, about 5.5 MJ/m^3. The reaction is difficult to control. Considerable development work is required before this method is commercialized.

4.8.3 Coal Liquefaction

The growing shortage of world petroleum supplies and the rapidly increasing cost of the oil has revived interest in producing a liquid fuel from coal. Coal liquefaction technology was stimulated in both Germany and Japan by the World War-II. Japan produced aviation gasoline in a large plant in North Korea (it was under Japan during that time) which converted coke made from coal into calcium carbide in electric furnaces, then to acetylene, acetaldehyde, butyraldehide, octanol, and finally octane. In a plant in Taiwan (also under Japanese occupation), the Japanese used starch from root vegetables which they fermented to butanol, which was then converted to butyraldehyde, octanol and octane. The most important German process was the *Fischer-Tropsch process*, which is still used commercially by the SASOL plant in South Africa. South Africa has no indigenous oil of its own but has considerable coal deposit. By coal liquefaction it meets all its oil requirement and is thus independent of foreign oil.

The conversion of coal into a liquid fuel requires the addition of hydrogen to the coal. Coal has a ratio of hydrogen atoms to carbon atoms of only 0.8 to 1, while in petroleum this ratio is 1.75 to 1. There are three basic modes that have been used to liquefy coal. These are: (1) hydrogenation, (2) catalytic conversion, and (3) hydropyrolysis.

In the *hydrogenation process*, coal and catalyst are suspended as a slurry, which is reacted with hydrogen at high pressure and moderate temperature to form liquid hydrocarbons. In the *catalytic conversion process*, a synthesis gas is produced from the coal as described earlier. The hydrogen and carbon monoxide in the gas are then combined in the presence of a catalyst to form a liquid hydrocarbon fuel. In *hydropyrolysis*, coal is heated beyond 450 °C, the fraction of coal volatilized greatly exceeds the volatile matter in coal. The hydrogen entrained pulverized coal is flash pyrolyzed. Up to 50% of the coal can thus be liquefied.

The Fischer–Tropsch process first produces a mixture of CO and H_2 from coal and steam. This is followed by catalytic reactions at about 150 °C and 150 bar, which yield a range of hydrocarbons from gaseous methane to higher liquid hydrocarbons. These are then separated with methane going as pipeline gas and the rest going to different liquid fuels.

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4.9 BIOMASS

Biomass is organic matter produced by plants. both in land and water. It includes forest crops, the crops which are grown in energy farms, and animal manure. While fossil fuels take millions of years to form, the biomass is an alternate fuel, the source of which may be considered renewable, since plant life renews and adds to itself every year. It is the solar energy stored by way of photosynthesis,

$$6CO_2 + 12H_2O \xrightarrow[\text{Chlorophyll}]{\text{Sunlight}} C_6H_{12}O_6 + 6H_2O + 6O_2 \qquad (4.18)$$

air soil (glucose or starch) ejected by transpiration air

The products of photosynthesis (or biomass), also called biofuels, are bulky and contain large amounts of water. So, it is not economical to transport them over long distances. The stored energy in them must be utilized by direct burning in regions close to the source. The biomass can also be converted to a variety of gaseous, liquid or solid fuels—all biomass-derived and therefore all biofuels.

Land crops may be of the following types:

1. Sugar crops, such as sugarcane refuse or bagasse
2. Herbaceous crops, such as non-woody plants
3. Forest crops, such as cultured hybrid poplar, eucalyptus, sycamore, sweetgum, alder and other hard woods.

Animal and human wastes are an indirect land crop from which methane and ethylene can be produced while retaining the fertilizer value of the manure.

Aquatic crops grown in fresh, sea and brackish waters include seaweeds, marine algae and various kelps.

There are three forms of bioconversion routes:

1. Direct combustion, such as wood waste and bagasse
2. Thermochemical conversion
3. Biochemical conversion

Thermochemical conversion can take two routes, viz., gasification and liquefaction. Gasification is done by heating the biomass with limited oxygen to produce low heating value gas or by making the biomass react with steam and oxygen at a high pressure and temperature to produce medium heating value gas. The latter may also be subjected to liquefaction by converting it to methanol and ethanol.

Biochemical conversion can take two routes: anaerobic digestion and fermentation. *Anaerobic digestion* is the bacterial decomposition of organic matter (biomass) in the absence of air or oxygen to ultimately produce a gaseous mixture (biogas) of methane and carbon dioxide in a roughly 2 : 1 volume ratio. An anaerobe is a micro-organism that can live and grow without air or oxygen. It gets its oxygen by the decomposition of matter containing it.

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The CO_2 gas produced by anaerobic digestion can be removed to produce synthetic natural gas (SNG) of pipeline quality, while the remaining sludge retains its nitrogen to produce a good fertilizer.

Fermentation is the breakdown of complex molecules in organic compounds with the help of a ferment such as yeast, bacteria, enzymes, etc. Grains and sugar crops are converted by fermentation into ethanol. Ethanol thus produced can be mixed with gasoline to produce gasohol (90% gasoline, 10% ethanol), which can be used as an automobile fuel.

In areas not suitable for food production, large rapidly growing trees are being cultivated for the production of biofuels. Sycamore and eucalyptus are such promising trees that yield up to 16 tonnes/acre per year. It is estimated that an *energy farm* of area about 900 km^2 could produce 400 MW of electricity. *Aquatic farms* growing algae, tropical grasses, floating kelp, water hyacinth and so on can yield several hundred tonnes/acre in a year. One interesting concept is to use the warm condenser cooling water from power plants to grow large quantities of algae, kelps etc. Research is in progress to increase the photosynthetic efficiency in the conversion of solar energy to biocrops from the present low value of about 5% to around 10%.

Table 4.1 shows the various bioconversion routes for the production of biofuels, and Table 4.2 gives the heating value of some waste bioproducts.

Table 4.1 *Bioconversion process and products*

Conversion	*Process*	*Initial product*	*Final product*
Aqueous	1. Anaerobic digestion 2. Alcoholic fermentation 3. Chemical reduction	Biogas—$CH_4 : CO_2$ = 2 : 1 (by vol) (22.28 MJ/m^3)	1. Methane (38 MJ/m^3) 2. Ethanol (19 MJ/l) 3. Oils (35–40 MJ/kg)
Dry Thermo-chemical	Pyrolysis		Pyrolytic oils (23–30 MJ/kg) Gas (8–15 MJ/m^3) Char (19–32 MJ/kg)
	Hydrogasification	Low and medium energy gas (7–15 MJ/m^3)	Methane (38 MJ/m^3) Methanol (16.9 MJ/l) Ammonia, Electricity (3.6 MJ/kWh) Ethane (70.5 MJ/m^3) Char (19–32 MJ/kg)
Biophotolysis			Hydrogen (12.7 MJ/^{3})

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Table 4.2 *Heating value of some waste bioproducts*

Waste bioproducts	*Heating value*
Wood chips	18.6–20.9 MJ/kg dry weight
Sugarcane bagasse	9.5 MJ/kg
Cereal straw	16–17 MJ/kg
Organic refuse	13.2 MJ/kg

4.10 THERMODYNAMIC VIEW

The steam generator system comprises the fuel preparation and firing system, the steam generator itself and the air heaters. Figure 4.5 illustrates how a thermodynamic control volume can be drawn around this equipment. Heat released by the burning of fuel is transferred to the two steam circuits for main steam and reheat steam. Any energy remaining in the flue gas at the exit of the air heater is lost.

An important measure of performance for a power plant is its net plant heat rate (NPHR) defined as the amount of fuel energy or boiler heat input required to generate one kWh and deliver it to the transmission lines. For a coal-fueled steam electric generating unit, the net plant heat rate is given by

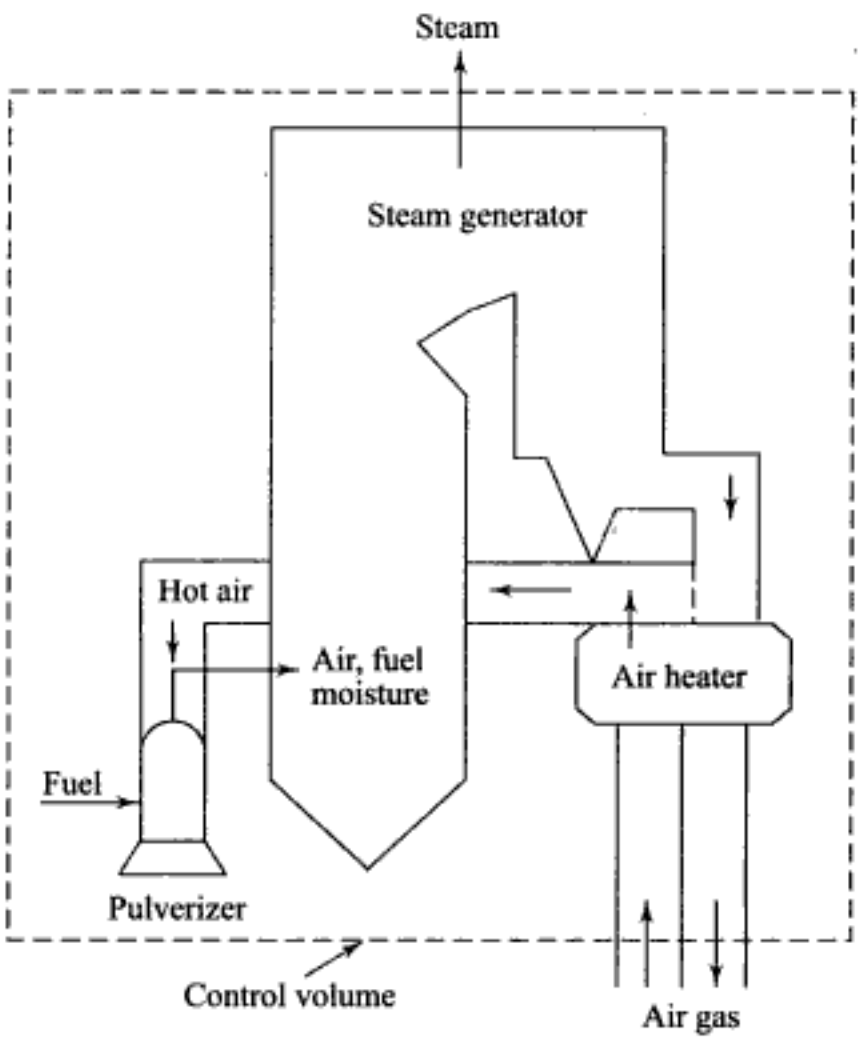

Fig. 4.5 *Thermodynanic control volume drawn around equipment*

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$$\text{NPHR} = \frac{Q_B}{(W_{net})_{plant}} = \frac{\omega_f \times \text{HHV}}{W_T - W_P - AP} \tag{4.19}$$

where Q_B = heat input to boiler from fuel (kW), ω_f = fuel burning rate (kg/s), W_T = turbine output (kW), W_P = pump input (kW), AP = auxiliary power (kW), HHV = higher heating value (kJ/kg).

The boiler efficiency, $\eta_B = \dfrac{\omega_s (h_1 - h_4)}{\omega_f \times \text{HHV}}$

The turbine heat rate (NTHR) = $\dfrac{\omega_s (h_1 - h_4)}{W_T - W_P} = \dfrac{1}{\eta_{cycle}}$

$$\therefore \qquad \omega_f \times \text{HHV} = \frac{\text{NTHR} \times \text{NTO}}{n_B}$$

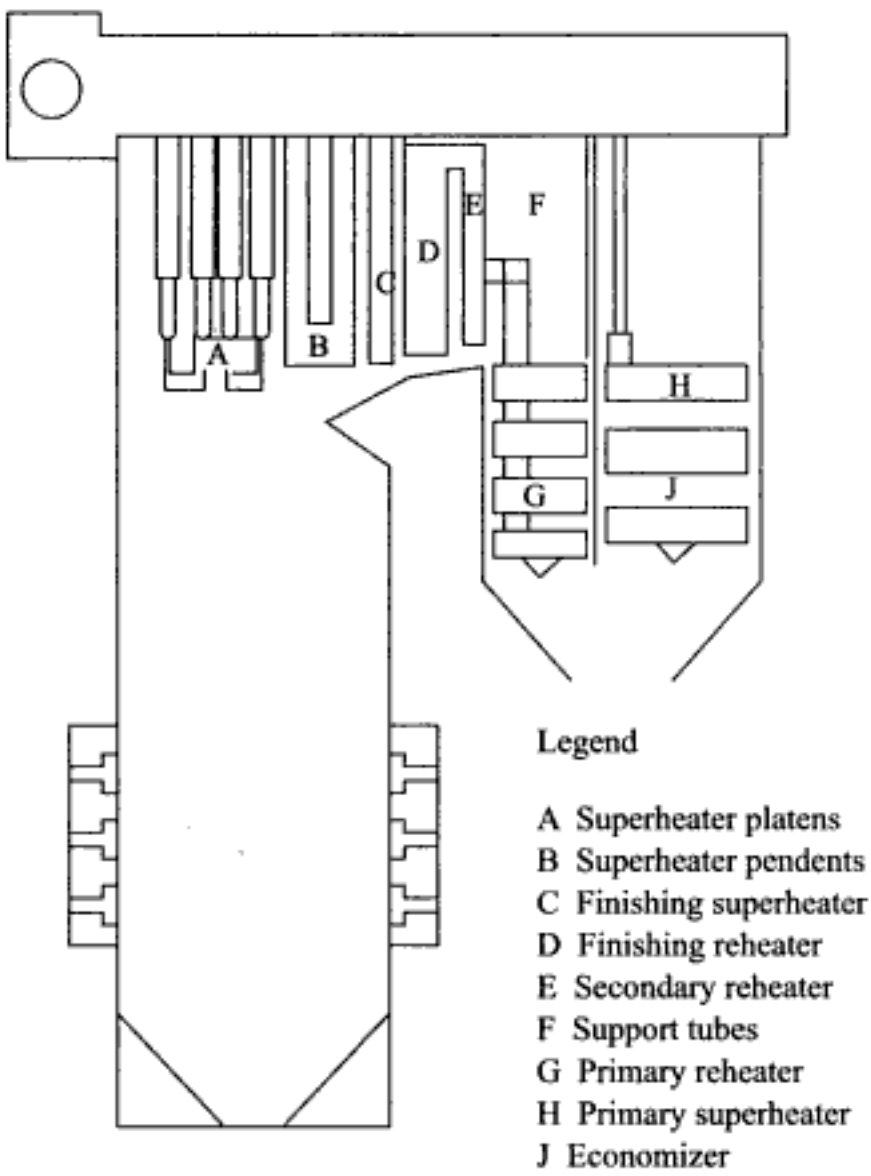

Fig. 4.6 *Example of steam generator surface arrangement*

where NTO = net turbine output = $W_T - W_p$ and $(h_1 - h_4)$ enthalpy rise of water in the boiler.

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$$\text{NPHR} = \frac{\text{NTHR} \times \text{NTO}}{\eta_B \left[1 - \frac{\text{AP}}{\text{NTO}}\right]} \tag{4.20}$$

Equation 4.20 may be used to estimate the net plant heat rate (kJ/kWh). The overall efficiency η_0 of the plant is then

$$\eta_0 = \frac{3600}{\text{NPHR}} \tag{4.21}$$

For a typical net plant heat rate of approximately 10.542 MJ/kWh, the overall plant efficiency would be 34.13 per cent.

Heat transferred within the steam generator must meet three discrete turbine cycle requirements: heat must convert feedwater to steam, superheat this steam and reheat the steam returning from the h.p. turbine. Therefore, the design of a steam generator must focus on arranging appropriate heat transfer surfaces throughout the unit to ensure that the correct quantity of heat is transferred for each requirement. Waterwall surface within the furnace collects heat to boil the feedwater. Economizer surface located at the exit of the convective pass is used to preheat the feedwater prior to feeding it into the waterwalls. Superheater surface is normally located above the furnace and in the backpass. Suspended reheat surface is located in a similar manner; radiant reheat may also be located on front or side walls of the upper furnace. The actual location of surface varies depending on boiler manufacturer, type of fuel and steam pressure. Figure 4.6 illustrates a potential surface arrangement.

4.11 COMBUSTION REACTIONS

Combustion is the high temperature oxidation of the combustible elements of a fuel with heat release. The combustible elements in coal and fuel oil are carbon, hydrogen and sulphur. The basic chemical equations for complete combustion are

$$\begin{aligned} C + O_2 &\longrightarrow CO_2 \\ 2H_2 + O_2 &\longrightarrow 2H_2O \\ S + O_2 &\longrightarrow SO_2 \end{aligned} \tag{4.19}$$

When insufficient oxygen is present, the carbon will be burned incompletely with the formation of carbon monoxide.

$$2C + O_2 \longrightarrow 2\,CO \tag{4.20}$$

In order to burn a fuel completely, four basic conditions must be fulfilled:

1. Supply *enough* air for complete combustion of fuel.
2. Secure enough *turbulence* for thorough mixing of fuel and air.
3. Maintain a furnace *temperature* high enough to ignite the incoming fuel air mixture.
4. Provide a furnace volume large enough to allow *time* for combustion to be completed.

Apart from adequate air supply, the three T's, *viz.*, *time*, temperature and turbulence have to be kept in mind while designing a furnace. Combustion may

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be said to hinge upon the word MATT—M is for mixture (turbulence), so that the fuel molecules meet the oxygen molecules, A is for the proper air–fuel ratio in order to support combustion, one T is for temperature, and the other T is for time. Since the complete mixing of the fuel and air is virtually impossible, excess air must be supplied to ensure complete combustion. The greater is the rate of mixing or turbulence, the lower would be the excess air required.

4.11.1 Stoichiometric Air

The ultimate analysis of the fuel is given by

$$C + H + O + N + S + M + A = 1.0$$

Oxygen needed for the oxidation processes can be calculated as follows:

$$\begin{array}{ccccc} C & + & O_2 & = & CO_2 \\ 12\text{ kg} & & 32\text{ kg} & & 44\text{ kg} \\ 1\text{ kg} & & 2.67\text{ kg} & & 3.67\text{ kg} \\ C\text{ kg} & & 2.67C\text{ kg} & & 3.67C\text{ kg} \\ 2H_2 & + & O_2 & = & 2H_2O \\ 4\text{ kg} & & 32\text{ kg} & & 36\text{ kg} \\ 1\text{ kg} & & 8\text{ kg} & & 9\text{ kg} \\ H\text{ kg} & & 8\text{ H kg} & & 9H\text{ kg} \\ S & + & O_2 & = & SO_2 \\ 32\text{ kg} & & 32\text{ kg} & & 64\text{ kg} \\ 1\text{ kg} & & 1\text{ kg} & & 2\text{ kg} \\ S\text{ kg} & & S\text{ kg} & & 2S\text{ kg} \end{array} \tag{4.21}$$

Oxygen required for complete combustion of 1 kg fuel is

$$W_{O_2} = 2.67C + 8H + S - O$$

where O is the oxygen in the fuel.

Air contains 23.2% oxygen by mass. Therefore, theoretically air required for complete combustion of 1 kg fuel is

$$W_T = \frac{W_{O_2}}{0.232} = \frac{2.67}{0.232}C + \frac{8}{0.232}\left(H - \frac{0}{8}\right) + \frac{1}{0.232}S$$

or,

$$W_T = 11.5C + 34.5\left(H - \frac{0}{8}\right) + 4.3S \tag{4.22}$$

where C, H, O and S are the mass fractions of carbon, hydrogen, oxygen and sulphur in the fuel as given by the ultimate analysis.

As stated earlier, complete combustion of fuel cannot be achieved if only the theoretical or stoichiometric air is supplied. Excess air is always needed for complete combustion. It is expressed as a percentage or by the use of a dilution coefficient. The per cent excess air supplied is

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$$\% \text{ excess air} = \frac{W_A - W_T}{W_T} \times 100 \tag{4.23}$$

where W_A is the actual amount of air supplied for complete combustion of 1 kg fuel. The dilution coefficient, d, is given by,

$$d = \frac{W_A}{W_T} \tag{4.24}$$

The percentage of excess air varies between 15 and 30% for most large utility boilers.

In the combustion of methane,

$$\underset{\text{reactants}}{CH_4 + 2O_2} \longrightarrow \underset{\text{products}}{CO_2 + 2H_2O} \tag{4.25}$$

Atmospheric air contains 21% oxygen, 78% nitrogen, and 1% argon by volume. In combustion calculations, however, the argon is usually neglected, and air is assumed to consist of 21% oxygen and 79% nitrogen by volume (or molar basis). On a mass basis, air containes 23.2% oxygen and 76.8% nitrogen.

For each mole of oxygen taking part in a combustion reaction, there are 0.79/0.21 or 3.76 moles of nitrogen. So, for the combustion of methane, the reaction can be written as

$$CH_4 + 2O_2 + 2(3.76)N_2 \longrightarrow CO_2 + 2H_2O + 7.52N_2 \tag{4.26}$$

With 150% theoretical air or 50% excess air, the methane combustion reaction can be written as

$$CH_4 + 2(1.5)O_2 + 2(3.76)\,(1.5)N_2 \longrightarrow CO_2 + 2H_2O + O_2 + 11.28N_2 \tag{4.27}$$

With less than needed excess air there may be a small amount of CO present in the products, called the flue gases, depending on mixing and turbulence during combustion, e.g. with 15% excess air,

$$CH_4 + 2(1.15)O_2 + 2(1.15)\,(3.76)N_2 \longrightarrow 0.95CO_2 + 0.,05CO + 2H_2O + 0.325O_2 + 8.65N_2 \tag{4.28}$$

4.11.2 Actual Air–Fuel Ratio

The actual amount of air suppled per kg fuel can be ascertained from the measured volumetric composition of combustion products. There are several methods to perform these measurements: (a) Orsat analyser, (b) Haldane apparatus, (c) Infra-red gas analyzer, and (d) Gas chromatograph. Of these, Orsat gas analyzer is very simple, portable and most frequently used. It measures the volume or mole fractions of CO_2, CO and O_2 in the dry flue gas.

Samples of flue gas are first collected by suction in sampling bottles. An Orsat analyzer contains three pipettes containing chemical solutions (Fig. 4.7). The reagents normally used are a KOH solution to absorb the CO_2 gas, pyrogallol solution to absorb the O_2 gas, and a cuprous chloride mixture ($CuCl_2$) to absorb the CO gas. By opening the stop valve and lowering the levelling bottle which contains water, 100 cm^3 of flue gas sample is collected over water

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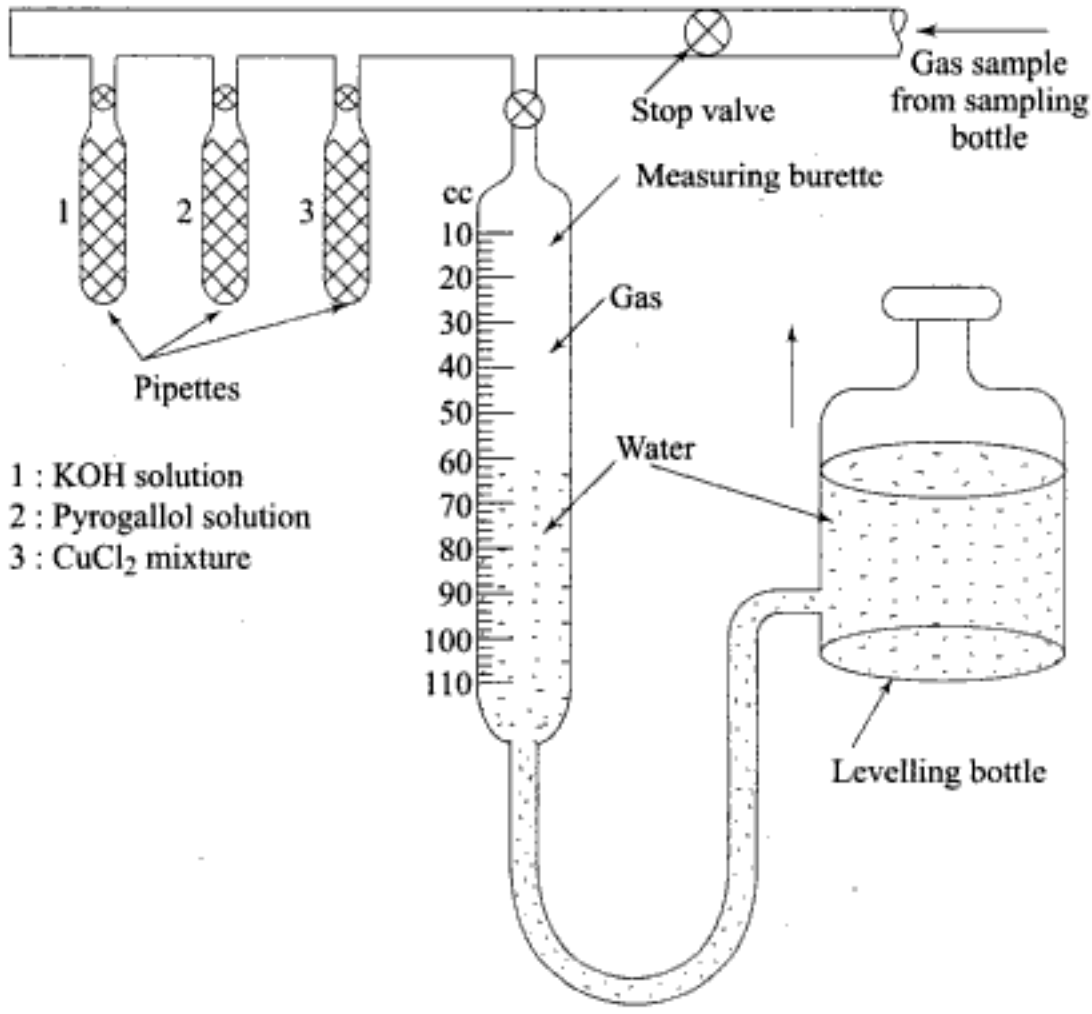

Fig. 4.7 *Orsat analyzer*

in the graduated measuring burette. The valve of pipette 1 is opened and the levelling bottle is raised, so that all the gas is passed into the pipette 1. The CO_2 content in the flue gas sample is absorbed in the KOH solution. The levelling bottle is lowered and the volume of gas is measured, say, it is 89 cm^3. That means 11% of the dry gas is CO_2 by volume. In the same manner, volumetric percentages of O_2 and CO in the flue gas are measured by opening successively the pipettes 2 and 3. The remaining unabsorbed gas is nitrogen. Since the sample is collected over water, any water vapour in the flue gas would have condensed during the collection process. The SO_2 gas will react with water in the container. So the flue gas analysis thus measured is on the dry basis:

$$CO_2 + CO + O_2 + N_2 = 100\%, \quad \text{by volume} \tag{4.29}$$

Let us assume that the dry flue gas (dfg) analysis of a gas sample is: 12% CO_2, 3% CO, 5% O_2 and 80% N_2. by volume. Therefore, 1 mole of dfg contains 0.12 mol of CO_2, 0.03 mol of CO, 0.05 mole of O_2 and 0.80 mole of N_2. The molecular weight of the dfg then becomes equal to $(0.12 \times 44 + 0.03 \times 28 + 0.05 \times 32 + 0.80 \times 28)$, or 30.12 kg/kg mol. Thus, in general, the molecular weight of dfg is given by

$$M_{dfg} = 44CO_2 + 28CO + 32O_2 + 28N_2 \tag{4.30}$$

where CO_2, CO, O_2 and N_2 are the volumetric percentages in dfg as measured, Therefore,

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$$\% \, CO_2 \text{ by mass} = \frac{44CO_2}{44CO_2 + 28CO + 32O_2 + 28N_2} \times 100$$

$$\% \, O_2 \text{ by mass} = \frac{32\,O_2}{44CO_2 + 28CO + 32O_2 + 28N_2} \times 100 \qquad (4.31)$$

$$\% \, N_2 \text{ by mass} = \frac{28N_2}{44CO_2 + 28CO + 32O_2 + 28N_2} \times 100$$

$$\% \, CO \text{ by mass} = \frac{28CO}{44CO_2 + 28CO + 32O_2 + 28N_2} \times 100$$

This is the gravimetric composition of the dry flue gas. From Eq. (4.21), it is seen that 44 kg of CO_2 gas has 12 kg carbon in it. So, ($44CO_2$) kg of CO_2 has (12 CO_2) kg of carbon. Similarly, from Eq. (4.20), (28 CO) kg of CO has (12 CO) kg of carbon. So, 1 kg mol of dfg has 12 (CO_2 + CO) kg of carbon.

$$\text{Mass of carbon per kg of dfg} = \frac{12(CO_2 + CO)}{44CO_2 + 28CO + 32O_2 + 28N_2} \qquad (4.32)$$

Let C_{ab} be the mass fraction of carbon C in the fuel which has been oxidized either to CO_2 or CO. Then $(C - C_{ab})$ is the mass fraction of unburnt carbon in the refuse. For 1 kg fuel burnt, there should be C_{ab} kg of carbon in the dfg. Therefore,

Mass of dfg produced per kg fuel

$$= \frac{C_{ab}(44CO_2 + 28CO + 32O_2 + 28N_2)}{12(CO_2 + CO)} \qquad (4.33)$$

From Eqs ((4.31) and (4.33).
Mass of N_2 in dfg per kg fuel

$$= \frac{C_{ab}(44CO_2 + 28CO + 32O_2 + 28N_2)}{12(CO_2 + CO)} \times \frac{28N_2}{44CO_2 + 28CO + 32O_2 + 28N_2}$$

$$= \frac{28N_2 C_{ab}}{12(CO_2 + CO)} \qquad (4.34)$$

This nitrogen in dfg comes from fuel as well as air. So, nitrogen coming with air per kg fuel $= \dfrac{28N_2 C_{ab}}{12(CO_2 + CO)} - N$, where N is the mass fraction of nitrogen in fuel. The actual amount of air supplied per kg fuel is

$$W_A = \frac{28N_2 C_{ab}}{12(CO_2 + CO)} \times \frac{1}{0.768} - \frac{1}{0.768} N$$

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Since N in fuel is small,

$$\therefore \qquad W_A = \frac{3.04 N_2 C_{ab}}{CO_2 + CO} \tag{4.35}$$

This is the actual air-fuel ratio used for combustion of the fuel, where C_{ab} is the fraction of carbon in fuel which has been burnt to CO_2 and CO. If W_A is measured by an air flow meter, the degree of burnout of carbon, C_{ab}, can be estimated from the relation

$$C_{ab} = \frac{(CO_2 + CO)W_A}{3.04 N_2} \tag{4.36}$$

4.11.3 Combustion Equation

Let us consider a coal having the following ultimate analysis:

C – 60%, H – 4%, S – 3.2%, O – 4.8%, N – 2%, M – 5%, and A – 21%.

The exhaust gas has the following volumetric analysis:

$CO_2 + SO_2 = 12\%$, $CO = 2\%$, $O_2 = 4\%$ and $N_2 = 82\%$

Let a moles of oxygen be supplied for 100 kg fuel. Then, the combustion equation can be written as

$$\frac{60}{12}C + \frac{4}{2}H_2 + \frac{3.2}{32}S + \frac{4.8}{32}O_2 + aO_2 + 3.76aN_2$$
$$\longrightarrow bCO_2 + dCO + eSO_2 + fO_2 + gN_2 + hH_2O$$

By equating the coefficients,

Carbon: $b + d = \frac{60}{12} = 5$ Hydrogen: $h = 2$ Sulphur: $e = \frac{3.2}{32} = 0.1$

Oxygen: $b + \frac{d}{2} + e + f + \frac{h}{2} = \frac{4.8}{32} + a$ Nitrogen: $g = 3.76a$

From the dfg analysis,

$$\frac{b+e}{b+d+e+f+g} = 0.12 \qquad \frac{d}{b+d+e+f+g} = 0.02$$

$$\frac{f}{b+d+e+f+g} = 0.04 \qquad \frac{g}{b+d+e+f+g} = 0.82$$

By solving these equations, the coefficients a, b, d, e, f, g and h are determined and given to be $d = 0.73$, $b = 4.27$, $f = 1.46$, $a = 7.045$ and $g = 26.49$.

Actual air supplied per kg coal $= \frac{32a}{0.232 \times 100} = 9.72$ kg

Let us now consider the combustion of propane gas (C_3H_8) with 80% theoretical air.

$$C_3H_8 + 5O_2 + 5(3.76)N_2 \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$

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With 80% theoretical air, the combustion equation becomes

$$C_3H_8 + 5(0.8))O_2 + 5(3.76)0.8)N_2 \longrightarrow aCO + bCO_2 + 4H_2O + 15.04N_2$$

Carbon balance : $3 = a + b$
Oxygen balance: $8 = a + 2b + 4$
By solving: $a = 2$, $b = 1$
The combustion equation can thus be written as:

$$C_3H_8 + 4O_2 + 15.04N_2 \longrightarrow 2CO + CO_2 + 4H_2O + 15.04N_2$$

4.11.4 Dew Point Temperature

From the combustion equation, the mole fraction of water vapour (x_{H_2O}) formed can be estimated.

$$x_{H_2O} = \frac{n_{H_2O}}{n_{total}} = \frac{n_{H_2O}}{n_{CO_2} + n_{CO} + n_{O_2} + n_{SO_2} + n_{N_2} + n_{H_2O}} \tag{4.37}$$

where n represents the number of moles.

The partial pressure of water vapour in the mixture of gases constituting the flue gas is given by

$$p_{H_2O} = x_{H_2O}p = x_{H_2O} \times 1 \text{ atm} = x_{H_2O} \text{ atm} \tag{4.38}$$

where p is the total pressure of the exhaust gas mixture which is 1 atm in all utility boilers. The saturation temperature at the partial pressure of water vapour is called the *dew point temperature* (d.p.t.). The flue gases are cooled in heat exchangers like economiser and air preheater so as to minimize the exhaust losses through chimney. These gases, however, should never be cooled below the dew point temperature (Fig. 4.8), as read from the steam tables. If cooled below the d.p.t., the water vapour condenses into liquid droplets which react with SO_2 or SO_3 to form acid. This acid corrodes the metal surfaces of the ducts through which the flue gas flows.

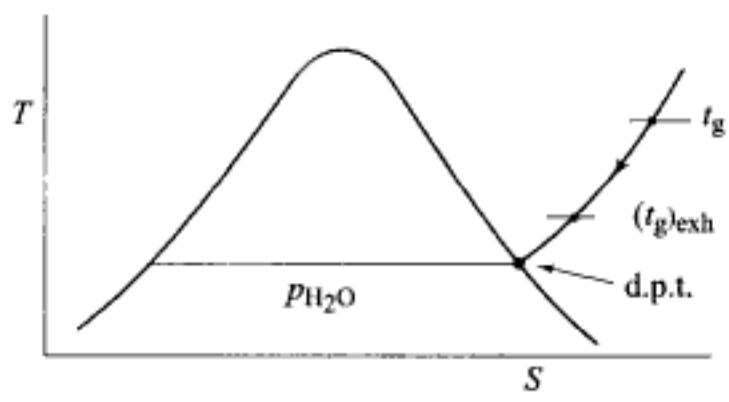

Fig. 4.8 *Dew point temperature of the exhaust flue gases*

4.11.5 Heating Value of a Fuel

Heats of reaction of coal constituents are given in Table 4.3.

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Table 4.3 *Heats of reaction of coal constituents*

	Formula and state	Product of combustion and state	Heat of reaction (kJ/kg mol)
Carbon (coke)	C(s)	CO_2(g)	– 407,000
Carbon	C(s)	CO_2(g)	– 397,000
Carbon monoxide	CO(g)	CO_2(g)	– 283,000
Hydrogen	H_2(g)	H_2O(l)	– 286,000
Sulphur	S(s)	SO_2(g)	– 291,000

For 1 kg of coal containing C kg carbon, the heat released by the carbon combustion (taking carbon to have heat of reaction of coke) at standard conditions is

Heat released by carbon combustion

$$= C \text{ kg} \times 407{,}000 \ \frac{\text{kJ}}{\text{kg-mol}} \times \frac{\text{kg-mol}}{12 \text{ kg}} = 33917 \frac{\text{kJ}}{\text{kg}}$$

Similarly, heat released by sulphur $= \dfrac{291000\text{S}}{32} = 9094\text{S kJ/kg}$

The available hydrogen is that hydrogen which is available for combustion, and is the total hydrogen less than that required to combine with the oxygen in the coal, $\left(\text{H} - \dfrac{0}{8}\right)$.

Heat released by hydrogen combustion

$$= \left(\text{H} - \frac{0}{8}\right)\frac{286000}{2} = 143{,}000\left(\text{H} - \frac{0}{8}\right) \text{ kJ/kg}$$

Therefore, the total heat released by complete combustion of 1 kg coal is

$$\text{HHV} = 33.917\text{C} + 143\left(\text{H} - \frac{0}{8}\right) + 9.094\text{S MJ/kg} \qquad (4.39)$$

The equation is very close to Dulong's formula, as given by Eq. (4.5).

4.11.6 Control of Excess Air

Proper control of the right amount of excess air maintains optimum combustion efficiency. Amounts of CO_2 and O_2 in combustion gases are indexes of excess air. The desirable CO_2 level depends on the fuel and the optimum excess air for the furnace (Fig. 4.9). Desirable O_2 values depend much less on the type of fuel (Fig. 4.10). This makes O_2 measurement the preferred method for combustion control. If the measured O_2 content is more than that desired, the air supply is to be reduced. If the O_2 measured is less than that desired, air supply is to be increased.

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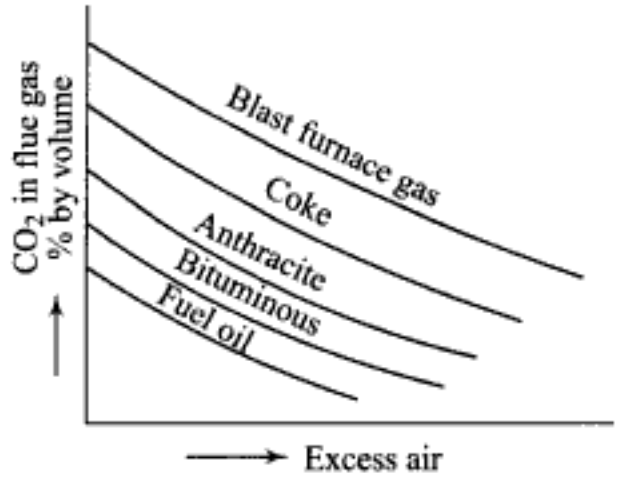

Fig. 4.9 *CO_2 variation in flue gas with fuels and excess air*

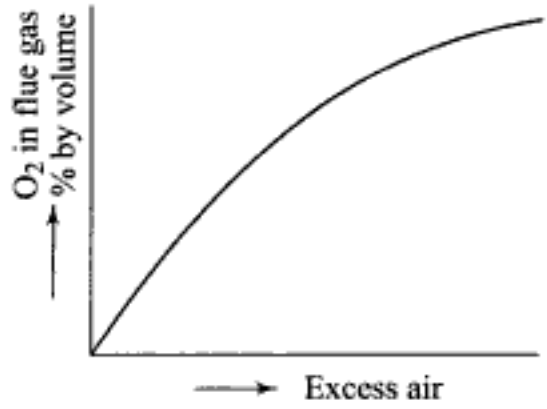

Fig. 4.10 *O_2 variation in flue gas with excess air*

Boiler losses are estimated as outlined in Section 4.13 for different steam outputs. The excess air is then adjusted by controlling air supply to show the optimum value of CO_2 or O_2. The optimum value of excess air for best combustion efficiency is then ascertained (Fig. 4.11). The excess air can also be determined using the following relation (Skrotzki and Vopat, 1960).

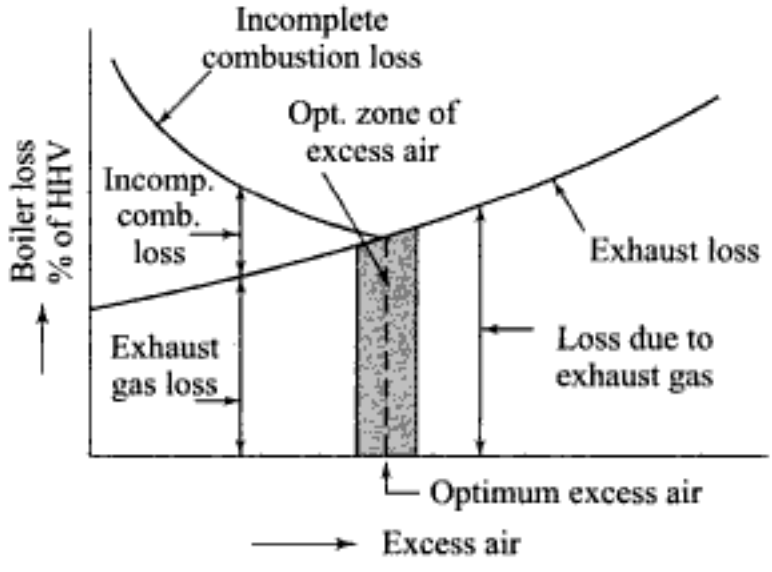

Fig. 4.11 *Optimum excess air for maximum combustion efficiency*

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$$\% \text{ excess air} = \frac{O_2 - 0.5CO}{0.264\,N_2 - (O_2 - 0.5\,CO)} \times 100 \tag{4.40}$$

where O_2, CO and N_2 are the volumetric percentages in the dry flue gas. An approximate formula for the excess air is

$$\% \text{ excess air} = 100\left[\frac{(CO_2)_0}{(CO_2 + CO)} - 1\right] = 100\left[\frac{O_2 - 0.5\,CO}{21 - O_2}\right]$$

where $(CO_2)_0$ = % CO_2 in the stoichiometric dry products, CO_2, CO, O_2 = % in the actual products. Instruments are used to regulate the air flow, gas flow and steam flow. It aids combustion control.

4.12 MASS BALANCE OF A STEAM GENERATOR

Analyses of fuel, refuse from ash pit, and flue gas, and the dry bulb and wet bulb temperatures of entering air are recorded. On the basis of 1 kg coal, Fig. 4.12 gives the material balance for a boiler furnace, where W_A is the amount of air supplied.

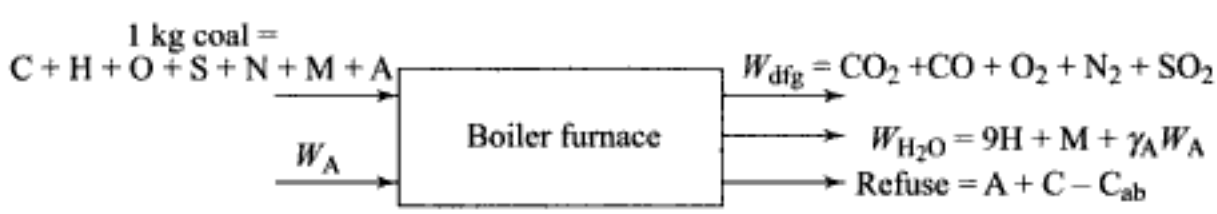

Fig. 4.12 *Material balance for a boiler furnace*

$$W_A + C + H + O + S + N + M + A = W_{dfg} + 9\,H + M + A + C - C_{ab}$$

$$W_A = W_{dfg} + 8\left(H - \frac{0}{8}\right) - C_{ab} - N - S \tag{4.41}$$

Mass of dfg produced per kg coal,

$$W_{dfg} = \frac{C_{ab}(44CO_2 + 28CO + 28N_2 + 32O_2)}{12(CO_2 + CO)}$$

$$= \frac{C_{ab}[11CO_2 + 7CO + 8O_2 + 7(100 - CO_2 - CO - O_2)]}{3(CO_2 + CO)}$$

$$= \frac{C_{ab}(4CO_2 + O_2 + 700)}{3(CO_2 + CO)} \tag{4.42}$$

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Volume of flue gases (wet) produced per kg coal

$$V_{fg} = \left[\frac{W_{dfg}}{M_{dfg}} + \frac{W_{H_2O}}{18}\right] \times 22.4 \times \frac{t_g + 273}{273} \times \frac{101.325}{p_g} \text{ m}^3 \tag{4.43}$$

where the pressure of gas p_g is in kPa and M_{dfg} is the molecular weight of dfg.

The dry refuse analysis by mass gives $A_R + C_R = 1.00$, where subscript R represents the refuse.

In 1 kg coal, $A = W_R \times A_R$, where W_R is the amount of refuse per kg coal and A_R is the mass fraction of ash in the refuse.

$$\therefore \qquad W_R = \frac{A}{A_R}$$

Mass of unburnt carbon in refuse per kg coal $= C_R W_R = C_R \dfrac{A}{A_R}$

Carbon burnout in dry gas

$$C_{ab} = C - C_R \frac{A}{A_R} \tag{4.44}$$

4.13 ENERGY BALANCE OF A STEAM GENERATOR

The fuel supplied to a furnace when completely burned releases its heating value. This energy converts the feedwater pumped to the boiler into steam (Fig. 4.13).

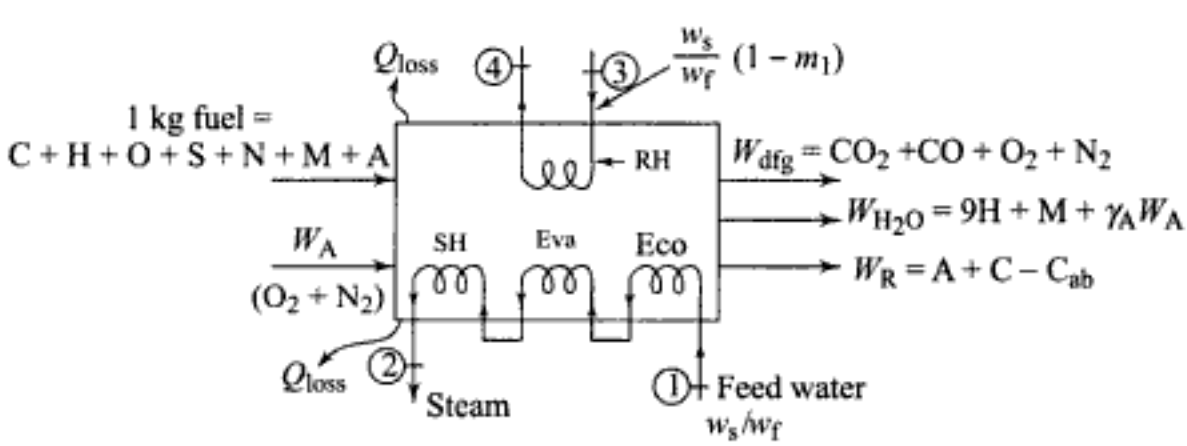

Fig. 4.13 *Energy balance of a steam generator*

Basis: 1 kg fuel

1. *Useful energy*

$$Q_1 = \frac{w_s}{w_f}\left[(h_2 - h_1)\right] + (1 - m_1)(h_4 - h_3)\right] \frac{\text{kJ}}{\text{kg}} \tag{4.45}$$

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where w_s = steam generation rate, kg/s
w_f = fuel burning rate, kg/s
m_1 = mass fraction of inlet steam bled from the h.p. turbine before reheating

2. *Energy loss due to dry exhaust gas*

$$Q_2 = W_{dfg} \cdot \bar{c}_{P_{dfg}} (t_g - t_a), \text{ kJ/kg} \tag{4.46}$$

where W_{dfg} = mass of dfg produced per kg fuel,
$\bar{c}_P$ = average specific heat of dfg, kJ/kg K
t_g = exhaust gas temperature, °C
t_a = ambient temperature, °C

3. *Energy loss due to unburnt carbon*

Q_3 = mass of unburnt carbon × its heating value

$$= (C - C_{ab}) \frac{407000}{12} = 33917\,(C - C_{ab}) \text{ kJ/kg} \tag{4.47}$$

4. *Energy loss due to incomplete combustion*
Loss of energy per kg of C oxidized to CO (from Table 4.3)

$$= \frac{283,000}{12} \text{ kJ/kg carbon} = 2358.3 \text{ kJ/kg carbon}$$

Loss of energy per kg fuel

$$Q_4 = 2358.3 \frac{\text{kJ}}{\text{kg carbon}} \times W_{dfg} \frac{\text{kg dfg}}{\text{kg fuel}}$$

$$\times \frac{28\,CO}{44CO_2 + 28CO + 32O_2 + 28N_2} \frac{\text{kg CO}}{\text{kg dfg}} \times \frac{12 \text{ kg carbon}}{28 \text{ kg CO}}$$

$$= 10100\, W_{dfg} \frac{28\,CO}{44CO_2 + 28CO + 32O_2 + 28N_2} \frac{\text{kJ}}{\text{kg}} \tag{4.48}$$

5. *Energy loss due to moisture in fuel*

$$Q_5 = M\,[4.187(100 - t_f) + 2256.8 + 2.09\,(t_g - 100)] \frac{\text{kJ}}{\text{kg}} \tag{4.49}$$

where t_f = temperature of fuel entering the furnace.

6. *Energy loss due to hydrogen in fuel*

$$Q_6 = 9H\,[4.187(100 - t_f) + 2256.8 + 2.09\,(t_g - 100) \frac{\text{kJ}}{\text{kg}} \tag{4.50}$$

7. *Energy loss due to moisture coming with air supplied*

$$Q_7 = \gamma_A W_A \times \bar{c}_P\,(t_g - t_a), \text{ kJ/kg} \tag{4.51}$$

where γ_A = specific humidity of air, (kg moisture)/(kg dry air)

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$\bar{c}_P$ = specific heat of superheated water vapour

8. *Energy loss due to ash and slag*

$$Q_8 = [(C - C_{ab}) + A]\bar{c}_P\,(t_{fu} - t_a)\ \text{kJ/kg} \tag{4.52}$$

where $\bar{c}_P$ = the average specific heat of ash, kJ/kg K

t_{fu} = the temperature of the furnace, °C

9. *Energy loss due to convection and radiation from the boiler surface.*

$$Q_9 = (h_c + h_r)\,A\,(t_w - t_a)/w_f \tag{4.53}$$

where h_c = convective heat transfer coefficient, W/m^2 K

h_r = radiative heat transfer coefficient, W/m^2 K

A = total surface area exposed to the ambient air, m^2

t_w = temperature of the wall surface of the boiler, °C

t_a = ambient temperature, °C

Energy released by complete combustion of 1 kg fuel = HHV.
Energy utilized in the heating of the working fluid,

$$Q_1 = \text{HHV} - \Sigma\ \text{energy losses} = \text{HHV} - (Q_2 + Q_3 + \cdots + Q_9)$$

Therefore, efficiency of the steam generator,

$$\eta_{\text{st. gen}} = \frac{\text{energy utilized}}{\text{energy released}} = \frac{\text{HHV} - \Sigma\ \text{energy losses}}{\text{HHV}} \tag{4.54}$$

$$= \frac{w_S[(h_2 - h_1) + (1 - m_1)(h_4 - h_3)]}{w_f \times \text{HHV}} \tag{4.55}$$

All energy losses at a particular load can be measured, and from Eq. (4.54) the efficiency of the steam generator can be estimated. Then, with the help of Eq. (4.55), the fuel consumption rate of the boiler can be determined.

4.14 DRAUGHT (OR DRAFT) SYSTEM

Large amounts of air are needed for combustion of the fuel. The gaseous combustion products in huge quantity have also to be removed continuously from the boiler furnace. To produce the required flow of either air or combustion gas, a pressure differential is needed. The term "draught" or "draft" is used to define the static pressure in the furnace, in the various ducts, and the stack. The function of the draught system is basically two-fold:

1. To supply to the furnace the required quantity of air for complete combustion of fuel.
2. To remove the gaseous products of combustion from the furnace and throw these through chimney or stack to the atmosphere.

There are two ways of producing draught: (a) natural draught and (b) mechanical draught.

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4.14.1 Natural Draught

The natural draught is produced by a chimney or a stack. It is caused by the density difference between the atmospheric air and the hot gas in the stack. For a chimney of height H metres (Fig. 4.14) the draught or pressure difference (N/m^2) produced is given by

$$\Delta p = gH(\rho_a - \bar{\rho}_g) \tag{4.56}$$

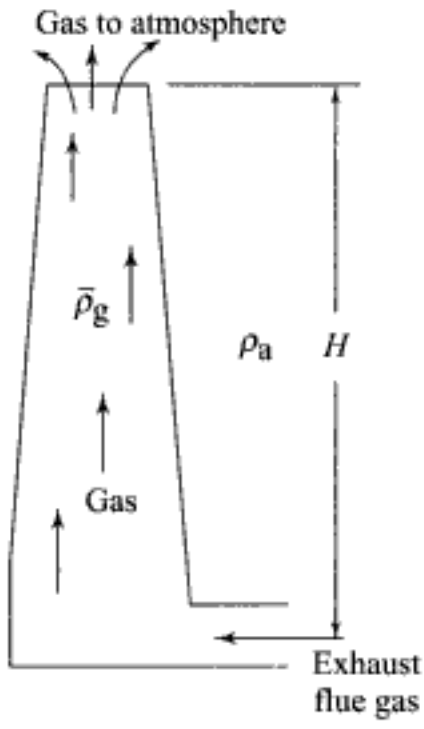

Fig. 4.14 *Draught produced by a chimney*

where ρ_a = density of atmospheric air, kg/m^3
ρ_g = average gas density in the chimney, kg/m^3
g = acceleration due to gravity, 9.81 m/s^2

Let us assume that the volume of the products of combustion is equal to the volume of air supplied for combustion, both volumes being measured at the same temperature. Thus, the volume of 1 kg of flue gas at NTP

$$v_a = \frac{0.287 \times 273}{101.325} = 0.7733 \text{ m}^3/\text{kg}$$

The volume of m kg of air per kg fuel at temperature T_a

$$V_a = 0.7733 \text{ m} \times \frac{T_a}{273}$$

$$\rho_a = \frac{1}{0.7733} \times \frac{273}{T_a} = 1.293 \left(\frac{273}{T_a}\right) \text{ kg/m}^3$$

The mass of flue gas will be $(m + 1)$ kg and its temperature is T_g.

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Density of flue gas, $\rho_g = \dfrac{m+1}{0.7733 \times m \times T_g/273}$

$$= 1.293\left(\frac{273}{T_g}\right)\left(\frac{m+1}{m}\right) = \frac{353}{T_g}\left(\frac{m+1}{m}\right)$$

∴ Draught produced (from Eq. 4.56)

$$\Delta p = 1.293 \times 273\left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right]gH$$

$$= 353\, gH\left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right] \tag{4.57}$$

If the draught is measured in terms of water column and the value is h mm,

$$\Delta p = 10^3\, gh \times 10^{-3} = 353\, gH\left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right]$$

$$\therefore \quad h = 353\, H\left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right] \tag{4.58}$$

where h is in mm and H is in m.

For given H and T_a, the natural draught depends on the average gas temperature T_g. Higher is the T_g, higher is the draught produced. But a high T_g means a large exhaust loss through chimney resulting in a lower boiler efficiency. With an optimum T_g, the amount of draught produced by density difference is thus limited.

Tall and conspicuous from a distance, chimneys or stacks are used in all power plants. In early steam generators, all the needed draught used to be produced by stacks alone to overcome all the pressure losses at the required gas flow rates. In modern boilers, the fuel burning rate is high, and so the rate of air supply as well as the rate of flue gas removal is high. There are also various heat exchangers like superheaters, reheater, economizer, and air preheater on the way to cause large pressure losses for which stacks alone are insufficient, and fans are added for producing mechanical draught. Stacks have thus two functions:

1. To assist the fans in overcoming pressure losses.
2. To help disperse the gas effluent into the atmosphere at a sufficient height to cause minimum atmospheric pollution.

Stacks introduce pressure losses of their own. These pressure losses are caused by wall friction and the pressure equivalent to the kinetic energy of the gases leaving the stack. Therefore, the actual draught produced is less than the theoretical value Δp given by Eq. (4.58), so that

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$$\Delta p_{act} = \Delta p - \rho_g \frac{\bar{V}^2}{2}\left(1 + \frac{fH}{D}\right) \tag{4.59}$$

where f = friction factor, dimensionless
H = stack height, m
D = stack inside diameter, m
V = stack exit velocity, m/s
ρ_g = average gas density in the stack, kg/m^3

Dispersion of the flue gases into the atmosphere is defined as the movement of the flue gases horizontally as well as vertically and their dilution by the atmosphere. Existing wind causes the horizontal motion of the flue gases. The vertical motion of flue gases results from the upward motion of high-velocity warm stack-exit gases to much higher elevations. The exit velocity of flue gases at stack exit results in a plume rise ΔH above the actual stack (Fig. 4.15). The gases bend in the direction of wind flow. The plume height ΔH is the height of a virtual point source above the stack obtained by extending the lines of dispersion backward. The effective stack height H_e is, thus, given by

$$H_e = H + \Delta H \tag{4.60}$$

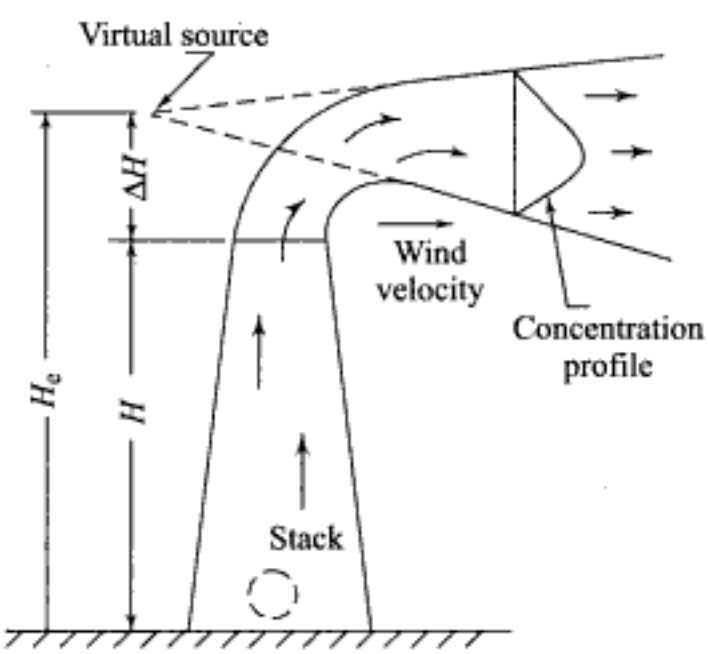

Fig. 4.15 *Dispersion of flue gases at stack exit*

Higher is the velocity of flue gases at stack exit, higher will be the plume rise ΔH and greater will be the dilution of exiting gases. There are several analytical methods of estimating ΔH. Most of these utilize a momentum term that accounts for the vertical momentum of the gas caused by the stack exit velocity and a buoyancy term that accounts for the difference between stack gas and atmospheric densities, pertinent to the prevailing situation of mixed convection in the stack.

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Correlation of Carson and Moses

$$\Delta H = 2.62\frac{Q_c^{0.5}}{V_w} - 0.029\frac{V_s D}{V_w} \tag{4.61}$$

where, V_s = stack gas exit velocity, m/s
D = stack diameter, m
V_w = wind velocity at stack exit, m/s
Q_c = heat emission, J/s = $mc_p\,(T_s - T_a)$
m = gas mass flow rate, kg/s
c_p = specific heat of gas = 1005 J/kg K for dry air at low temperature
T_s = gas temperature at stack exit, K
T_a = air temperature at stack exit, K

Correlation of Briggs

$$\Delta H = \frac{114\,CF^{1/3}}{V_w} \tag{4.62}$$

where C = dimensionless temperature gradient parameter

$$= 1.58 - 41.4\,\frac{\Delta\theta}{\Delta Z}$$

$\frac{\Delta\theta}{\Delta z}$ = air temperature gradient, K/m which is zero for neutral atmospheric stability conditions

F = buoyancy flux = $\frac{gV_s D^2 (T_s - T_a)}{4T_a}$, m^4/s^2

g = acceleration due to gravity = 9.81 m/s^2

4.14.2 Condition for Maximum Discharge through the Chimney

Mass of hot gases flowing through the chimney

$$\dot{m}_g = \rho_g A V_g$$

Since the density of hot gases is inversely proportional to its temperature, $\rho_g = C_1/T_g$, where C_1 is constant and the velocity of gases $V_g = C\sqrt{2gH_g}$, we have

$$\dot{m}_g = \frac{C_1}{T_g} \times A \times C\sqrt{2gH_g}$$

where H_g = hot gas column in m.

$$\therefore \qquad \dot{m}_g = AC_2\sqrt{2gH\left[\frac{m}{m+1}\cdot\frac{T_g}{T_a} - 1\right]} \times \frac{1}{T_g}$$

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For a given height of the chimney, H, we can write

$$\dot{m}_g = C_3 \left[\frac{m}{m+1} \cdot \frac{1}{T_a T_g} - \frac{1}{T_g^2} \right]^{1/2}$$

Thus the maximum discharge is a function of T_g,

$$\therefore \qquad \frac{d\dot{m}_g}{dT_g} = 0 = -\frac{m}{m+1} \frac{1}{T_a T_g^2} + \frac{2}{T_g^3}$$

$$\therefore \qquad T_g = 2T_a \frac{m+1}{m} \text{ and } \frac{T_g}{T_a} = 2\left(\frac{m+1}{m}\right) \tag{4.63}$$

Now,

$$\Delta p = \rho_g g H_g = 353\, gH \left(\frac{1}{T_a} - \frac{m+1}{m} \frac{1}{T_g} \right)$$

Since $\rho_g = \dfrac{353}{T_g} \dfrac{m+1}{m}$, on substitution

$$H_g = H \left(\frac{m}{m+1} \frac{T_g}{T_a} - 1 \right)$$

$$= H \left(\frac{m}{m+1} \cdot 2 \frac{m+1}{m} - 1 \right) = H$$

Thus the height of the hot gas column is equal to the chimney height for maximum discharge.

The draught in mm of water column

$$h = \frac{353H}{T_a} \left[1 - \frac{m+1}{m} \times \frac{T_a}{T_g} \right] = \frac{353H}{T_a} \left[1 - \frac{m+1}{m} \cdot \frac{m}{2(m+1)} \right]$$

$$= \frac{353H}{2T_a} = \frac{176.5H}{T_a} \tag{4.64}$$

4.14.3 Mechanical Draught

Mechanical draught is produced by fans. There are two types of fans in use today: forced draught (FD) and induced draught (ID) fans. When either one is used alone, it should overcome the total air and gas pressure losses within the steam generator.

Forced draught fans are installed at inlet to the air preheater. They handle cold air. So they have less maintenance problems, consume less power (since cold air has low specific volume, and work input per unit mass flow rate is given by $\int v dp$) and therefore, their capital and operating costs are lower. If ω_f is the fuel burning rate, v is the specific volume of inlet air and Δp_{FD} is the pressure head developed by the fan to overcome all the pressure losses, then the power required to drive the FD fan is given by

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$$\text{Power input} = \frac{W_f \times \frac{A}{F} \times v \times \Delta p}{\eta_{FD}} \tag{4.65}$$

where A/F is the air—fuel ratio and η_{FD} is the efficiency of the forced draught fan. For good reliability two forced draught fans operating in parallel are normally used, each capable of undertaking at least 60% of full load air flow when the other is out of service.

The forced draught fan if used alone, as in many large steam generators and almost all marine applications, maintains the entire system up to the stack entrance under positive gauge pressure. The furnace is then said to be pressurized. Utmost care is taken so that noxious gases do not leak out to the atmosphere. It thus needs a gas-tight furnace construction and properly designed ignition openings and inspection doors. The stack in such a situation is shorter and meant only for disposal of flue gases.

Induced draught fans are normally located at the foot of the stack. They handle hot combustion gases. Their power requirements are, therefore, greater than forced draught fans. In addition, they must cope with corrosive combustion products and fly ash. Induced draught fans are seldom used alone. They discharge essentially at atmospheric pressure and place the system upstream under negative gauge pressure. If v_g is the specific volume of flue gases handled by the ID fan and Δp_{ID} is the pressure head developed, then the power needed to drive the ID fan is given by

$$\text{Power input} = \frac{w_f\left(1 + \frac{A}{F}\right) \times \mathbf{v}_g \times \Delta p_{ID}}{\eta_{ID}} \tag{4.66}$$

When both forced and induced draught fans are used in a steam generator, the FD fans push atmospheric air through the air preheater, dampers, various air ducts, and burners into the furnace, and the ID fan sucks out the flue gases through the heat transfer surfaces in the superheaters, reheaters, economiser, gas-side air preheater and dust collectors and discharge into the stack (Fig. 4.16). The stack because of its height, adds a natural driving pressure of its own. In such a case the furnace is said to operate with *balanced* draught, meaning that the pressure in it is approximately atmospheric. Actually, it is maintained at a slightly negative gauge pressure to ensure that any leakage would be inward. Modern boilers are mostly designed with balanced draught firing.

A typical large power plant (600 MWe) may require two 1320 kW forced draught fans and two 3000 kW induced draught fans.

4.14.3 Pressure and Flow Measurement

The total pressure at a point in the system is the sum of static and dynamic pressures:

$$p_t = p_s + p_d \tag{4.67}$$

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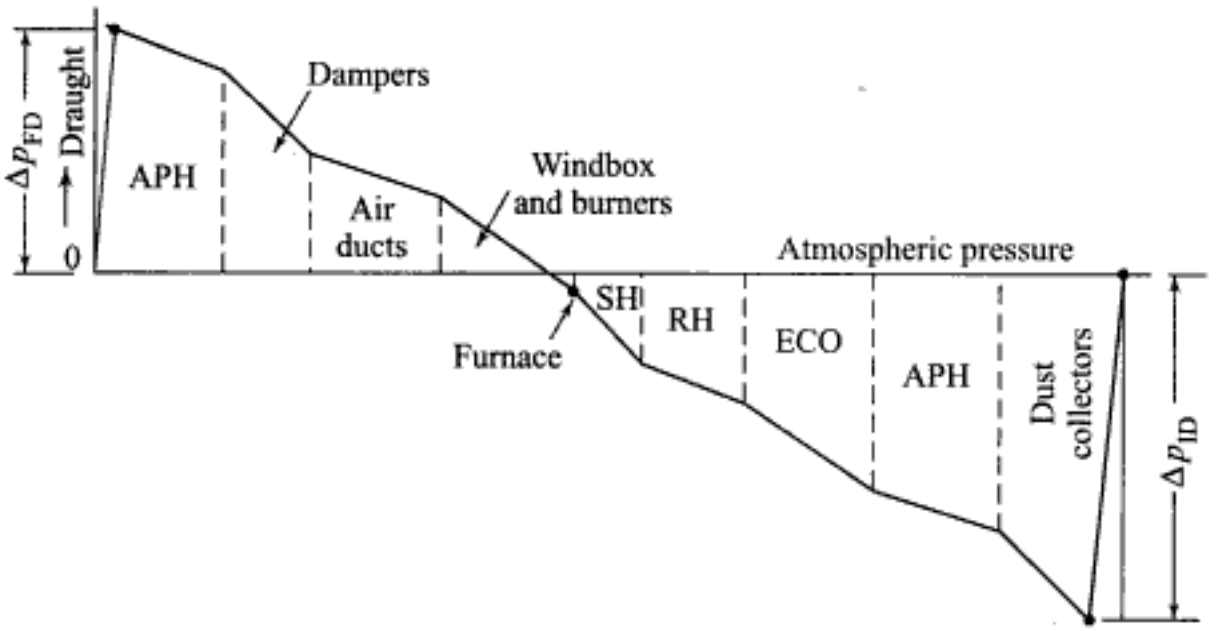

Fig. 4.16 *Balanced draught system in a modern boiler*

where p_t = total pressure, N/m^2

p_s = static pressure, N/M^2

p_d = dynamic pressure = $\rho_g \dfrac{\bar{V}^2}{2}$, N/m^2

ρ_g = gas density, kg/m^3

$\bar{V}$ = velocity of gas, m/s

Although the SI units of pressure are pascals (N/m^2), it is still customary to state the static and dynamic pressures in mm of water since measurements are commonly made using a simple, standard U-tube with water as the manometer fluid.

One metre head of water produces a pressure of

$$p = \rho g H = 1000 \frac{\text{kg}}{\text{m}^3} \times 9.81 \frac{\text{m}}{\text{s}^2} \times 1 \text{ m} = 9810 \text{ N/m}^2$$

In terms of mm of water, the dynamic pressure head, p_d, which would be measured by a pitot tube connected to a water filled manometer, becomes

$$p_d = \frac{\rho_g V^2}{9810 \times 2} \times 1000 = \frac{\rho_g V^2}{19.62}, \text{ mm } H_2O$$

where ρ_g is in kg/m^3 and V is in m/s. Then the volume flow rate of gas in a duct is given by

$$\bar{V}_o = A \left[19.62 \frac{p_d}{\rho_g} \right]^{1/2} \text{ m}^3/\text{s}$$

where A is the duct cross-sectional area in m^2.

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$$\therefore \qquad \bar{V}_o = 4.43A\sqrt{(p_d/\rho_g)}\ \text{m}^3/\text{s} \qquad (4.68)$$

where p_d is in mm of water and ρ_g is in kg/m^3.

The flue gas density depends on the moisture content and temperature. For 5% moisture and at 560 K, flue gas density is approximately 0.649 kg/m^3. Combustion air density at 560 K, is 0.636 kg/m^3.

4.15 FANS

FD and ID fans operate continuously for long periods, up to 1 or $1\frac{1}{2}$ years. So, these must be well designed, ruggedly constructed, well balanced, and highly efficient over a wide range of outputs. Typical fans have capacities of 700 m^3/s of volume flow producing 152 mm water static pressures (about 0.15 bar).

There are two types of fans, *viz.*, centrifugal and axial. In the centrifugal fan, the gases are accelerated radially through curved or flat impeller blades from rotor to a spiral or volute casing. In the axial fan, gases are accelerated parallel to the rotor axis. This is similar to a table fan, but here the fan is housed in a casing to develop static pressure. Axial fans have higher capital costs.

Centrifugal fans can have forward-curved, flat or backward curved impeller blades (Fig. 4.17). The velocity triangles at exit from the tip of the blades are shown, where the absolute velocity of gas V is the same in all the three cases. It is seen that for the same V the blade tip velocity V_b is the highest for the backward-curved blades (*c*) and the lowest for the forward-curved blades (*a*). Since $V_b = (\pi DN)/60$, for the same tip diameter D, the rpm N is the highest for the backward-curved and the lowest for the forward-curved blades. The FD fans should have high V_b so as to rotate at high speeds and handle large volume flow of air. Therefore, centrifugal fans with backward-curved blading are normally used for FD fans. The ID fans handle dust-laden flue gases and so the blades are subject to erosion by the fly ash. The erosion rate of blades is lower if the blade tip speed V_b is less and the fan rotates at lower speeds. Therefore, centrifugal fans having forward-curved or flat blading are used for ID fans. Low-speed fans with flat blades are used for particularly dirty or corrosive gases.

There are two common methods of controlling the output of fans: damper control and variable-speed control.

In damper control, a damper (butterfly valve) is used to control the flow of gas by regulating the flow area according to the load. It has the advantage of low capital cost since it needs a simple constant-speed induction ac motor. It , however, offers additional flow resistance which is overcome by increasing the power input. Dampers are usually put on the outlet side of the fan, although dampers at inlet to the fan are sometimes used. Inlet dampers consume less power than outlet dampers but are only effective for moderate load changes near full load. When used with FD fans, they are normally used in combination with outlet dampers. The use of a two-speed ac drive motor in conjuction with damper control can also reduce power consumption, since two-speed motors are less expensive than variable-speed ac drives.

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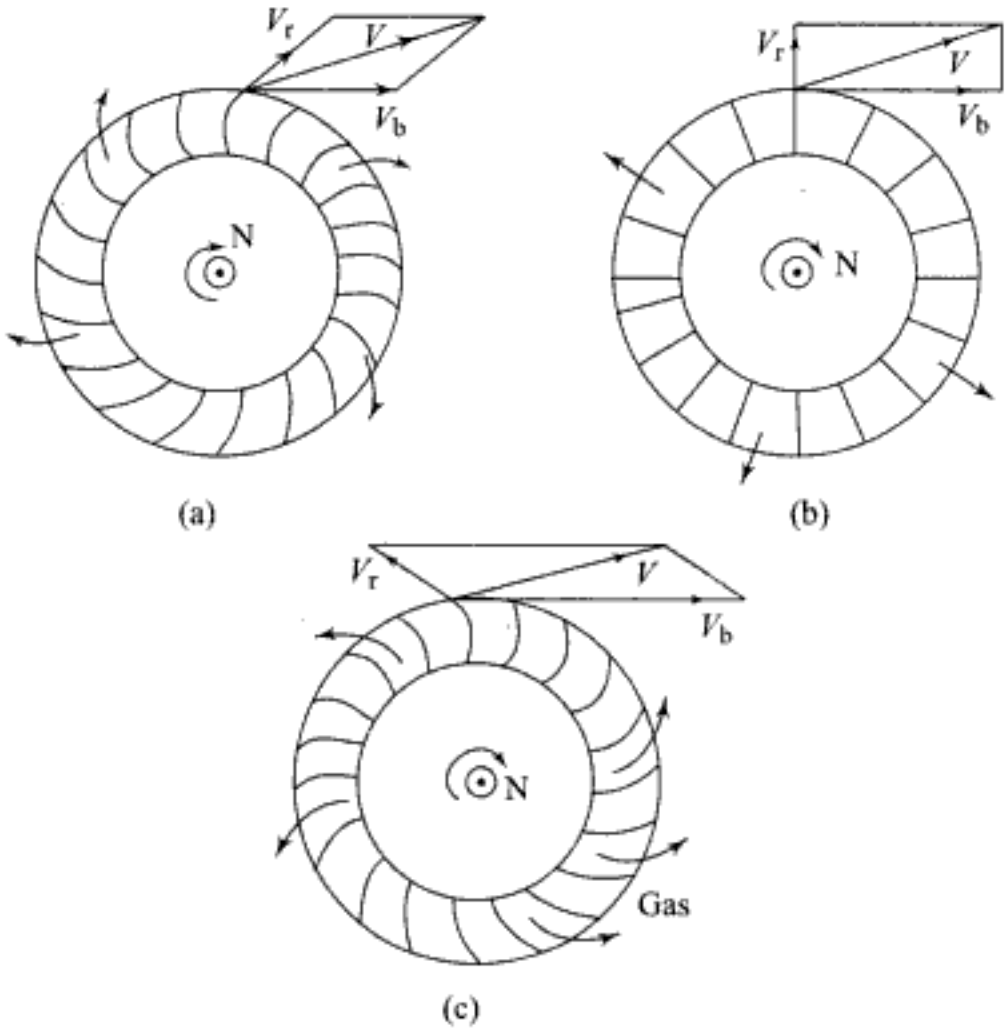

Fig. 4.17 *Centrifugal blading: (a) forward-curved, (b) flat, and (c) backward-curved. Vector diagrams show blade tip velocity V_b, gas velocity relative to blade V_r, and absolute velocity of gas leaving blade V. In all cases, V is the same*

Variable-speed control has the advantage of less power consumption and is the most efficient method for fan control. The effect of speed on fan performance is

$$\text{Volume flow} \propto N$$
$$\text{Pressure head} \propto N^2$$
$$\text{Power input} \propto N^3$$

where N is the rpm of the fan. Thus, reducing speed, say by 70%, reduces the capacity to 70%, the pressure to about 50%, and the power input to about 35%. The types of drives are

1. variable-speed steam turbine
2. hydraulic coupling
3. variable-speed dc motor
4. multiple speed ac motor
5. electronically adjustable motor drive

The major disadvantage of variable speed control is the higher capital costs than damper control.

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Two other types of fans are used in power plants: primary air fans and gas recirculation fans. Primary air fans supply air to dry and transport pulverized coal to the furnace. Gas recirculation fans are used to recirculate gas from a point between the economizer and air preheater back to the bottom of the furnace as part of a steam temperature control system.

Fans are a major source of noise in power plants. To reduce this noise, they are often housed in thick masonry acoustical enclosures or equipped with silencers, or both.

4.16 HEAT OF COMBUSTION

The first law of thermodynamics was treated so far ignoring such energy terms as chemical, electrical, and magnetic, in the energy balance for a system. When dealing with combustion systems, however, the chemical energy term cannot be ignored.

4.16.1 Open System

For a chemically reactive system (Fig. 4.18), the steady state steady flow energy equation can be written as

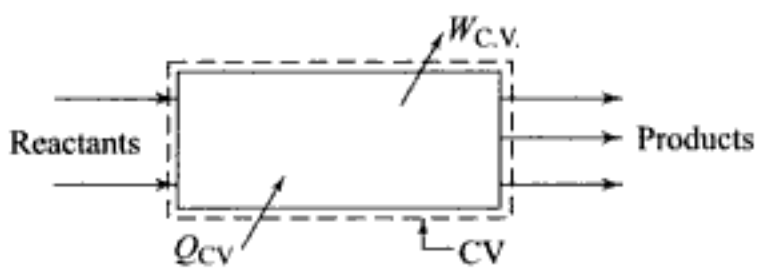

Fig. 4.18 *A steady-state steady flow chemically reactive system*

$$H_R + Q_{CV} = H_P + W_{CV} \tag{4.69}$$

ignoring the changes in kinetic and potential energies. Here, H_R and H_p are the enthalpies of the reactants and products, respectively, evaluated for their constituents at their respective pressures and temperatures; Q_{CV} is the net heat added to the system from the surroundings (which is usually negative, since in combustion heat is usually rejected to the surroundings); and W_{CV} is the net work done by the system.

Both the reactants and products are usually composed of several constituents. Therefore, Eq. (4.69) is written in the form

$$\sum_R (mh) + Q_{CV} = \sum_P (mh) + W_{CV} \tag{4.70}$$

where m is the mass and h is the specific enthalpy of each constituent. Let us consider the combustion of ethane.

$$C_2H_6 + 3.5O_2 + (3.5)(3.76)N_2 \longrightarrow 2CO_2 + 3H_2O + (3.5)(3.76)N_2 \tag{4.71}$$

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The enthalpies of the various reactants and products are those that start at the same datum of composition, temperature and pressure, which are arbitrarily chosen as the elemental substances at 25 °C and 1 atm, respectively. For example, C_2H_6 is formed from elemental carbon C and hydrogen H_2 (Fig. 4.19). When the reactants carbon and hydrogen enter the reactor at 25 °C, 1 atm and the product C_2H_6 is cooled back to 25 °C, 1 atm, 2817.3 kJ/kg C_2H_6 of heat is rejected. The reaction may be written as

$$2C + 3H_2 \longrightarrow C_2H_6 + 2817.3 \text{ kJ/kg } C_2H_6$$

Similarly, for CO_2,

$$C + O_2 \longrightarrow CO_2 + 8946.8 \text{ kJ/kg } CO_2$$

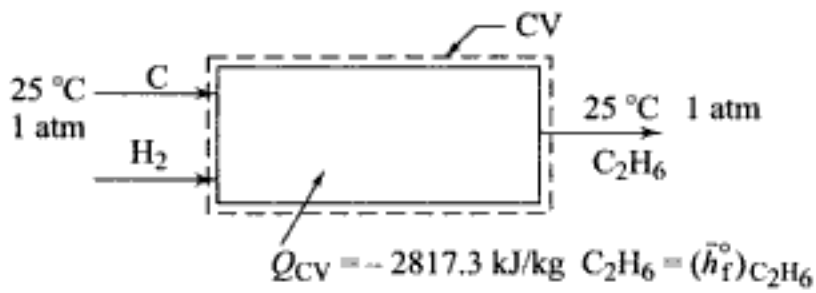

Fig. 4.19 *Formation of ethane*

Table 4.4 *Enthalpy of formation h_f^0 at 25 °C, 1 atm*

Substance	*Formula*	*Molecular weight*	*State*	*h_f^0 (kJ/kg)*
Carbon	C	12.011	solid	0
Oxygen	O_2	32.000	gas	0
Hydrogen	H_2	2.016	gas	0
Nitrogen	N_2	28.016	gas	0
Sulphur	S	32.060	solid	0
Carbon monoxide	CO	28.011	gas	– 3948.3
Carbon dioxide	CO_2	44.011	gas	– 8946.8
Water	H_2O	18.016	liquid	– 15875.5
			vapour	– 13430.8
Methane	CH_4	16.043	gas	– 4669.8
Ethane	C_2H_6	30.070	gas	– 2817.3
Propane	C_3H_8	44.097	gas	– 2356.3
Butane	C_4H_{10}	58.124	gas	– 2171.6
Octane	C_8H_{18}	114.230	liquid	– 2189.5
			vapour	– 1826.0
Nitric oxide	NO	30.008	gas	– 3020.8
Nitrogen dioxide	NO_2	46.008	gas	– 733.3
Sulphur dioxide	SO_2	64.060	gas	– 4632.8

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The quantities 2817.3 and 8946.8 leave the system and hence are negative. They are called enthalpies of formation (h^o_f). Table 4.4 gives values of h^o_f for various substances at 25 °C, 1 atm. The enthalpy of all elements at the standard reference state (25 °C, 1 atm) is assigned the value of zero. It may be noted that if the combustion equation involves a liquid, such as octane or water, there will be two enthalpies of formation, depending upon whether they start or end in a liquid or vapour state.

Because chemical reactions, such as Eq. (4.70) are balanced in terms of moles and not masses, and because $m = n\mu$, where n = number of moles and μ = molecular weight, Eq. (4.70) may be written as

$$\sum_R n_i \bar{h}_i + Q_{CV} = \sum_R n_e \bar{h}_e + W_{CV} \tag{4.72}$$

where $\bar{h}$ is in kJ/kg mol. Table 4.5 gives the values of the enthalpy of formation (h^0_f) of a number of substances in kJ/kg mol at 25 °C, 1 atm.

In most cases, however, the reactants and products are not at 25 °C, 1 atm. Therefore, the change of enthalpy (in a constant pressure of SSSF process) between 25 °C, 1 atm and the given state must be known. Thus the enthalpy at any temperature and pressure, $\bar{h}_{T,p}$ is

$$\bar{h}_{T,p} = \left(\bar{h}^0_f\right)_{298K,\ 1\ atm} + (\Delta\bar{h})_{298\ K,\ 1\ atm \longrightarrow T,p}$$

For convenience, the subscripts are usually dropped, and

$$\bar{h}_{T,p} = \bar{h}^0_f + \Delta\bar{h} \tag{4.73}$$

Table 4.5 *Enthalpy of formation, Gibbs function of formation, and absolute entropy of various substances at 25 °C, 1 atm pressure*

Substance	Molecular Weight, M	$\bar{h}^0_f$ kJ/kg mol	$\bar{g}^0_f$ kJ/kg mol	$\bar{s}_o$ kJ/kg mol K
CO(g)	28.001	– 119529	– 137150	197.653
CO_2(g)	44.001	– 393522	– 394374	213.795
H_2O(g)	18.015	– 241827	– 228583	188.883
H_2O(1)	18.015	– 285838	– 237178	70.049
CH_4(g)	16.043	– 74873	– 50751	186.256
C_2H_2(g)	26.038	+ 226731	+ 209234	200.958
C_2H_4(g)	28.054	+ 52283	+ 68207	219.548
C_2H_6(g)	30.070	– 84667	– 32777	229.602
C_3H_8(g)	44.097	– 103847	– 23316	270.019
C_4H_{10}(g)	58.124	– 126148	+ 16914	310.227
C_8H_{18}(g)	114.23	– 208447	+16859	466.835
C_8H_{18}(1)	114.23	– 249952	+ 6940	360.896

where $\Delta\bar{h}$ represents the difference in enthalpy between any given state and the enthalpy at 298 K, 1 atm.

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Table A.4 in the appendix gives the values of $\Delta\bar{h} = \bar{h}^0 - \bar{h}_{298}$ in kJ/kg mol for various substances at different temperatures.

4.16.2 Closed System

Combustion equations for fuels burning in a closed system, such as a cylinder or bomb, may be obtained by writing the energy balance for the closed system. Thus,

$$U_R + Q = U_p + W \tag{4.74}$$

where W is the non-flow work. For gases,

$$H = U + pV = U + n\bar{R}T$$

where $\bar{R}$ is the universal gas constant.

4.17 HEATING VALUES: ENTHALPY OF COMBUSTION

The enthalpy of combustion is defined as the difference between the enthalpy of products and the enthalpy of the reactants when complete combustion occurs at a given temperature and pressure. Therefore,

$$\bar{h}_{RP} = H_P - H_R$$

or

$$\bar{h}_{RP} = \sum_R n_e(\bar{h}_f^0 + \Delta\bar{h})_e - \sum_R n_i(\bar{h}_f^0 + \Delta\bar{h})_i \tag{4.75}$$

where $\bar{h}_{RP}$ is the enthalpy of combustion of the fuel. The values of the enthalpy of combustion of different hydrocarbon fuels at 25 °C, 1 atm are given in Table 4.6.

The internal energy of combustion, $\bar{u}_{RP}$ is defined in a similar way.

$$\bar{u}_{RP} = U_p - U_R$$

$$= \sum_R n_e\left(\bar{h}_f^0 + \Delta\bar{h} - p\bar{v}\right)_e - \sum_R n_i\left(\bar{h}_f^0 + \Delta\bar{h} - p\bar{v}\right)_i \tag{4.76}$$

If all the gaseous constituents are considered ideal gases and the volume of liquid and solid considered is assumed to be negligible compared to gaseous volume,

$$\bar{u}_{RP} = h_{RP} - \bar{R}T(n_{\text{gaseous products}} - n_{\text{gaseous reactants}}) \tag{4.77}$$

In the case of a constant pressure or steady flow process, the negative of the enthalpy of combustion is frequently called the heating value at constant pressure, which represents the heat transferred from the chamber during combustion at constant pressure.

The internal energy of combustion, $\bar{u}_{RP}$ is defined in a similar way.

$$\bar{u}_{RP} = U_p - U_R$$

$$= \sum_R n_e(\bar{h}_f^0 + \Delta\bar{h} - p\bar{v})_e - \sum_R n_i(\bar{h}_f^0 + \Delta\bar{h} - p\bar{v})_i \tag{4.76}$$

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Table 4.6 *Enthalpy of combustion of some hydrocarbons at 25° C*

		Liquid H_2O in Products (Negative of Higher Heating Value)		Vapour H_2O in Products (Negative of Lower Heating Value)	
Hydrocarbon	Formula	Liquid Hydrocarbon kJ/kg fuel	Gaseous Hydrocarbon kJ/kg fuel	Liquid Hydrocarbon kJ/kg fuel	Gaseous Hydrocarbon kJ/kg fuel
Paraffin family					
Methane	CH_4		−55496		−50010
Ethane	C_2H_6		−51875		−47484
Propane	C_3H_8	−49975	−50345	−45983	−46353
Butane	C_3H_{10}	−49130	−49500	−45344	−45714
Pentane	C_5H_{12}	−48643	−49011	−44983	−45351
Hexane	C_6H_{14}	−48308	−48676	−44033	−45101
Heptane	C_7H_{16}	−48071	−48436	−44557	−44922
Octane	C_8H_{18}	−47893	−48256	−44425	−44788
Decane	$C_{10}H_{22}$	−47641	−48000	−44239	−44598
Dodecane	$C_{12}H_{24}$	−47470	−47828	−44110	−44467
Olefin family					
Ethene	C_2H_4		−50296		−47158
Propene	C_3H_6		−48917		−45780
Butene	C_4H_8		−48453		−45316
Pentene	C_5H_{10}		−48134		−44996
Hexene	C_6H_{12}		−47937		−44800
Heptene	C_7H_{14}		−47800		−44662
Octene	C_8H_{16}		−47693		−44556
Nonene	C_9H_{18}		−47612		−44475
Decene	$C_{10}H_{20}$		−47547		−44410
Alkylbenzene family					
Benzene	C_6H_6	−41831	−42266	−40141	−40576
Methylbenzene	C_7H_8	−42473	−42847	−40527	−40937
Ethylbenzene	C_8H_{10}	−42997	−43395	−40924	−41322
Propylbenzene	C_9H_{12}	−43416	−43800	−41219	−41603
Butylbenzene	$C_{10}H_{14}$	−43748	−44123	−41453	−41828

If all the gaseous constituents are considered ideal gases and the volume of liquid and solid considered is assumed to be negligible compared to gaseous volume,

$$\bar{u}_{RP} = h_{RP} - \bar{R}\,T\,(n_{\text{gaseous products}} - n_{\text{gaseous reactants}}) \qquad (4.77)$$

In the case of a constant pressure or steady flow process, the negative of the enthalpy of combustion is frequently called the heating value at constant

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pressure, which represents the heat transferred from the chamber during combustion at constant pressure.

Similarly, the negative of the internal energy of combustion is sometimes designated as the heating value at constant volume in the case of combustion because it represents the amount of heat transfer in the constant volume process.

The higher heating value (HHV) or higher calorific value (HCV) is the heat transferred when H_2O in the products is in the liquid state (see Section 4.2.3). The lower heating value (LHV) or lower calorific value (LCV) is the heat transferred in the reaction when H_2O in the products is in the vapour state.

4.18 THEORETICAL FLAME TEMPERATURE

If a combustion process occurs adiabatically in the absence of work transfer or changes in KE and PE, then the steady flow energy equation becomes

$$H_R = H_P \qquad \text{or} \qquad \sum_R n_i \bar{h}_i = \sum_P n_e \bar{h}_e$$

or
$$\sum_R n_i \left(\bar{h}_f^0 + \Delta\bar{h}\right)_i = \sum_P n_e \left(\bar{h}_f^0 + \Delta\bar{h}\right)_e \tag{4.78}$$

For such a process, the temperature of the products is called the adiabatic or theoretical flame temperature, which is the maximum temperature achieved for the given reactants. The maximum theoretical flame temperature of a fuel corresponds to complete combustion with just the theoretically required quantity of pure oxygen. Similarly, the maximum flame temperature using air corresponds to complete combustion with the stoichiometric quantity of air. This temperature is lower than the maximum flame temperature in pure oxygen because of the dilution effect of nitrogen. In actual combustion chambers, flame temperature is further reduced due to incomplete combustion, excess air requirements, heat losses to the surrounding walls, and by heat loss to ash particles suspended in the combustion products.

The adiabatic flame temperature can be controlled by the amount of excess air supplied. The maximum permissible temperature in a gas turbine is fixed from metallurgical considerations (i.e., the material used for gas turbine blades). So, close control of the temperature of the products is achieved by controlling the excess air. For a given reaction, the adiabatic flame temperature can be theoretically computed by trial and error. The energy of the reactants H_R being known, a suitable temperature is chosen for the products so that the energy of the products. H_P at that temperature becomes equal to the energy of the reactants H_R. This helps in the selection of material for the combustor.

Figure 4.20 shows the variation of adiabatic flame temperature with excess air. Combustion temperatures so calculated are higher than actual because *dissociation* of some of the products takes place at high temperatures. For example, some CO_2 may dissociate to form CO and $\frac{1}{2}O_2$, which is an endothermic reaction that lowers the temperature. Chemical equilibrium occurs at a certain temperature when the reaction rate is the same in both directions, i.e., when

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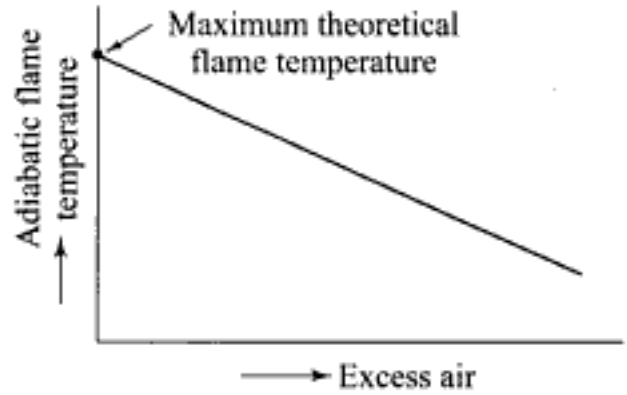

Fig. 4.20 *Variation of adiabatic flame temperature with excess air*

$$CO + \frac{1}{2}O_2 \rightleftharpoons CO_2 \quad \text{and} \quad H_2 + \frac{1}{2}O_2 \rightleftharpoons H_2O$$

The higher is the temperature, more is the dissociation. However, the effect of dissociation is lower for lean mixtures because the excess oxygen tends to drive the reactions toward completion, and for higher furnace pressure. Thus, the effect of dissociation is not significant in steam generator furnace combustion, which occurs with excess air at about 1500 °C.

It may be noted that all combustion calculations can be done if there is heat transfer to or from the environment, work done by or on the system, or even changes in kinetic or potential energy between products and reactants by including the appropriate terms in the steady state steady flow energy equation.

4.19 FREE ENERGY OF FORMATION

The thermodynamic potential which controls equilibrium in a system depends on the particular constraints imposed on the system. If the constraints are constant temperature and pressure, for any process

$$dG_{TP} < 0 \tag{4.79}$$

where G is the Gibbs function defined as $G = H - TS$, which is also called the free energy. For a process at constant T and P, G always decreases and becomes a minimum at the state of equilibrium. It is the free energy, G, which determines whether a chemical reaction will occur spontaneously. The driving force in a chemical reaction is the difference in free energy between the initial and final states, ΔG, where at constant temperature

$$\Delta G = \Delta H - T\Delta S \tag{4.80}$$

The free energy of the chemical elements at 1 atm. 25°C is arbitrarily assumed to be zero. The *standard free energy of formation of a compound*, g_f^0, is the free energy change when one mole of the compound is formed directly from its constituent elements, both the reactants and the products being at 298 K and partial pressures of 1 atm. If the reaction takes place at some higher temperature, the free energy of formation g_f^0 will have a different value. In actual

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situations, the partial pressures of the reactants and products will not be 1 atm. The effect of such pressure changes on the free energy of a substance can be evaluated from the relation at constant temperature

$$dg = v\,dp$$

where v is the molal specific volume. For an ideal gas in an isothermal process from the standard pressure p_0 to the operating pressure p_1,

$$\Delta\bar{g} = \int_{p_0}^{p_1} v dp = \bar{R}T \ln \frac{p_1}{p_0} \tag{4.81}$$

where $\bar{R}$ is the universal gas constant. The free energy change

$$\Delta G = \bar{g}_f^0 + \Delta\bar{g} = \bar{g}_f^0 + \bar{R}T \ln \frac{p_1}{p_0} \tag{4.82}$$

If p is expressed in atmospheres and $p_0 = 1$ atm.

$$\Delta G = \bar{g}_f^0 + \bar{R}T \ln p_1 \tag{4.83}$$

If the substance is not an ideal gas, we define a quantity f, called the fugacity such that

$$\lim_{p \to 0} \frac{f}{p} = 1 \tag{4.84}$$

and

$$\Delta G = \bar{g}_f^0 + \bar{R}T \ln \frac{f_1}{f_0} \tag{4.85}$$

4.20 EQUILIBRIUM CONSTANT K_P

Let us consider a reaction

$$C_2H_6(g) + 3.5\ O_2(g) \longrightarrow 2CO_2(g) + 3H_2O(g)$$

taking place at temperature T. The initial state is defined by a partial pressure of ethane, $p_{C_2H_6}$, and the partial pressure of oxygen, p_{O_2}. In the final state, CO_2 and H_2O have partial pressures of p_{CO_2} and p_{H_2O}. Using the free energies of formation, the free energy change for the system Δg can be expressed as

$$\Delta G = 2(\Delta g_f)_{CO_2} + 3(\Delta g_f)_{H_2O} - [(\Delta g_f)_{C2H6} + 3.5\ (\Delta g_f)_{O2}] \tag{4.86}$$

where from Eq. (4.83)

$$2(\Delta g_f)_{CO_2} = 2\ (\bar{g}^0{}_f)_{CO_2} + \bar{R}\ T \ln p^2{}_{CO_2}$$
$$3\ (\Delta g_f)_{H_2O} = 3\ (\bar{g}^0{}_f)_{H_2O} + \bar{R}\ T \ln p^3{}_{H_2O}$$
$$(\Delta g_f)_{C_2H_6} = (\bar{g}^0{}_f)_{C_2H6} + \bar{R}\ T \ln p_{C_2H_6}$$
$$3.5\ (\Delta g_f)_{O_2} = 3.5\ (\bar{g}^0{}_f)_{O_2} + \bar{R}\ T \ln p^{3.5}{}_{O_2}$$

Substituting in Eq. (4.86), the free energy change becomes

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$$\Delta G = 2(\bar{g}^0{}_f)_{CO_2} + 3\,(\bar{g}^0{}_f)_{H_2O} - [(\bar{g}^0{}_f)_{C_2H_6} + 3.5\,(\bar{g}^0{}_f)_{O_2}]$$
$$+ \bar{R}\ T(\ln p^2{}_{CO_2} + \ln p^3{}_{H_2O} - \ln p_{C_2H_6} - \ln p^{3.5}{}_{O_2}) \qquad (4.87)$$

The first four terms of the above equation represent ΔG^0, the change in the free energy for the reaction when the products and the reactants all have partial pressures of 1 atm, while the remaining terms represent the effect of partial pressures on the free energy change. Making the standard free energy change Δg_f^0 as

$$\Delta G^0 = n_3 g_3 + n_4 g_4 - n_1 g_1 - n_2 g_2$$

where n's are the stoichiometric coefficients and g's are the free energy of formation, Eq. (4.87) can be written as

$$\Delta G = \Delta G^0 + \bar{R}\ T\ \ln \frac{p^2_{CO_2}\ p^3_{H_2O}}{p_{C_2H_6}\ p^{3.5}_{O_2}} \qquad (4.88)$$

In generalized notations,

$$\Delta G = \Delta G^0 + \bar{R}\ T\ \ln \frac{p_3^{\nu_3}\ p_4^{\nu_4}}{p_1^{\nu_1}\ p_2^{\nu_2}} \qquad (4.89)$$

where p_3 and p_4 are the partial pressures of the products, and p_1 and p_2 are those of the reactants for the reaction

$$n_1 A_1 + n_2 A_2 \longrightarrow n_3 A_3 + n_4 A_4$$

For exothermic readings, ΔG^0 is negative and for endothermic reactions it is positive. In Eq. (4.89), ΔG^0 is normally much higher than the second term representing the effect of partial pressures. In a chemical reaction, as the reactants are consumed and the products are formed, the partial pressures of the reactants decrease and those of the products increase, which make ΔG less negative and thus, decrease the net driving force for the reaction. In the reverse reaction, A_3 may react with A_4 to form A_1 and A_2. The free energy change for this reverse reaction

$$n_3 A_3 + n_4 A_4 \longrightarrow n_1 A_1 + n_2 A_2$$

$$\Delta G = (\Delta G^0)_{reverse} + \bar{R}\ T\ \ln \frac{p_1^{\nu_1}\ p_2^{\nu_2}}{p_3^{\nu_3}\ p_4^{\nu_4}} \qquad (4.90)$$

As the forward reaction proceeds the driving force for the reverse reaction increases. Because of the competing reactions, the reactive system tends to reach an equilibrium condition in which the two driving forces are equal. When this occurs, we have

$$n_1 g_1 + n_2 g_2 = n_3 g_3 + n_4 g_4$$

so that $\Delta G = 0$ at equilibrium and from Eq. (4.87),

$$\Delta G^0 = -\ \bar{R}\ T \ln \frac{p_3^{\nu_3}\ p_4^{\nu_4}}{p_1^{\nu_1}\ p_2^{\nu_2}} \qquad (4.91)$$

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In an isothermal process, ΔG^o and T are constant, and so the logarithm of the quotient on the r.h.s. of Eq. (4.91) is constant for equilibrium at a given temperature. This constant is called the *equilibrium constant* K_p expressed in terms of partial pressures. Therefore, Eq. (4.91) can be expressed as

$$\Delta G^o = -\bar{R}\ T\ \ln K_p \tag{4.92}$$

where

$$K_p = \frac{p_3^{\nu_3}\ p_4^{\nu_4}}{p_1^{\nu_1}\ p_2^{\nu_2}} \tag{4.93}$$

When any of the reactants is not an ideal gas, the appropriate partial pressure must be replaced by a fugacity ratio. If A_2 is not an ideal gas, then

$$K_p = \frac{p_3^{\nu_3}\ p_4^{\nu_4}}{p_1^{\nu_1}\ (f_2/f_{02})^{\nu_2}} \tag{4.94}$$

If one of the compounds is a solid, the fugacity ratio for that compound is unity. If A_2 is a solid,

$$K_p = \frac{p_3^{\nu_3}\ p_4^{\nu_4}}{p_1^{\nu_1}} \tag{4.95}$$

From Eq. (4.90), the equilibrium constant at a given temperature T is given by

$$K_p = \exp\left(-\frac{\Delta G^o}{\bar{R}T}\right) \tag{4.96}$$

With data of ΔG^o as a function of T being available, the values of log K_p for typical reactions relevant to combustion are computed and plotted in Fig. 4.21.

In the combustion of coal, there are several competing reactions which determine the degree of carbon combustion:

$$C(S) + O_2(g) \rightleftharpoons CO_2(g) \tag{4.97a}$$

$$C(S) + \frac{1}{2}\ O_2(g) \rightleftharpoons CO(g) \tag{4.97b}$$

$$C(S) + CO_2(g) \rightleftharpoons 2CO(g) \tag{4.97c}$$

$$CO(g) + \frac{1}{2}\ O_2(g) \rightleftharpoons CO_2(g) \tag{4.97d}$$

The hydrogen and sulphur present are burned to H_2O and SO_2.

The equilibrium constant for the reaction (4.97a) is given by

$$K_p = \frac{p_{CO_2}}{p_{O_2}} \tag{4.98}$$

and for the reaction (4.97c), it is

$$K_p = \frac{(p_{CO})^2}{p_{CO_2}} \tag{4.99}$$

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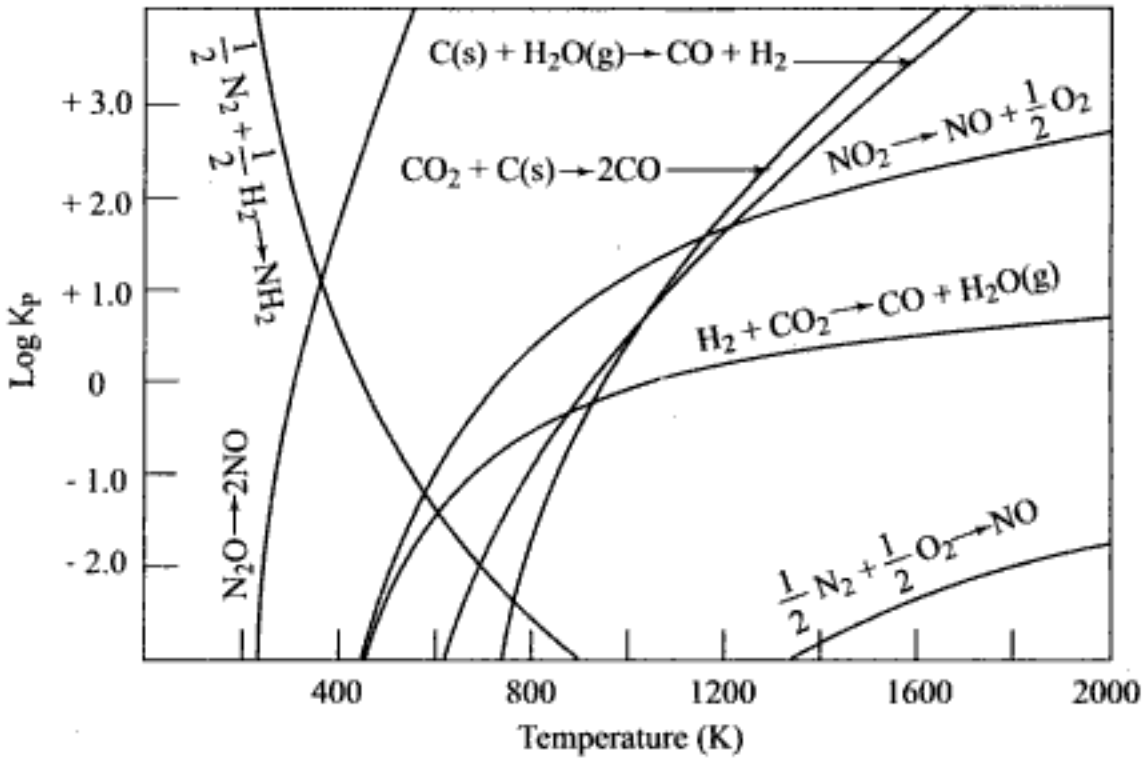

Fig. 4.21 *Equilibrium constants for selected chemical reactions*

In modern coal burning plants, the gases leaving the flame are above 450 °C and the equilibrium constant K_p is very large for reactions (4.97a), (4.97(b) and (4.97d). It is, therefore, normally assumed that these reactions are completed. Reaction (4.95c) is a slower reaction with the relative amounts of CO_2 and CO given by Eq. (4.98). The temperature dependent value of the equilibrium constant is obtained from Fig. 4.21.

4.21 EFFECT OF DISSOCIATION

It was mentioned in Section 4.16 that the effect of dissociation is to reduce the flame or products temperature since heat is absorbed (endothermic process) in the dissociation. At high temperatures of combustion, dissociation of the products becomes significant. For example, in the combustion of carbon monoxide:

$$CO + \frac{1}{2}\,O_2 \rightleftharpoons CO_2$$

The reaction proceeds in both directions, and a fraction, a, of CO_2 produced is dissociated into CO and O_2:

$$a\,CO_2 \longrightarrow a\left(CO + \frac{1}{2}O_2\right)$$

In the equilibrium condition, the products consist of

$$(1-a)\,CO_2 + a\,CO + \frac{1}{2}\,a\,O_2$$

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The degree of dissociation, a, can be determined from the total pressure of the products mixture, p, and the equilibrium constant K_p (which is a function of temperature). For the above reaction,

$$K_p = \frac{p_{CO_2}}{p_{CO}(p_{O_2})^{1/2}} \tag{4.100}$$

Under appropriate conditions carbon can reduce water vapour to produce CO and H_2, which is an important reaction for coal gasification. However, in the presence of oxygen, CO and H_2 produced will again get oxidized, and the reduction of steam need not be considered in normal combustion calculations.

In case of incomplete combustion, the amount of heat released depends upon the relative amounts of CO_2 and CO formed, which in turn depend on the chemical equilibrium achieved as given by

$$K_p = \frac{p_{CO}^2}{p_{CO_2}}$$

Let us consider the combustion process as given by the overall equation,

$$1C + 1H + bS + \left(\frac{a}{4} + b + d\right)(O_2 + 3.76N_2) \rightleftharpoons (1 - x - y)C$$
$$+ yCO_2 + xCO + \frac{a}{2}H_2O + bSO_2 + \left(d + \frac{a}{4} + b\right)(3.76\,N_2) \tag{4.101}$$

By oxygen balance,

$$\frac{a}{4} + b + d = y + \frac{x}{2} + \frac{a}{4} + b$$

$$y + \frac{x}{2} = d$$

or, $$x = 2\,(d - y)$$

The partial pressure of CO is then given by

$$p_{CO} = \frac{x}{x + y + \frac{a}{2} + b + \left(d + \frac{a}{4} + b\right)3.76} = \frac{x}{\alpha}$$

$$\therefore \qquad p_{CO} = \frac{2(d - y)}{\alpha}$$

where p_x represents the partial pressure of a component x.

Hence:

$$p_{CO_2} = \frac{1 - \alpha}{1 - \alpha + \alpha + \frac{1}{2}\alpha}\,p = \frac{1 - \alpha}{1 + \frac{1}{2}\alpha}\,p \tag{4.102}$$

$$p_{CO} = \frac{\alpha p}{1 + \frac{1}{2}\alpha} \quad \text{and} \quad p_{O_2} = \frac{\frac{1}{2}\alpha p}{1 + \frac{1}{2}\alpha}$$

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$$\therefore \qquad K_p = \frac{1-\alpha}{\alpha}\left[\frac{\alpha+2}{\alpha\, p}\right]^{1/2} \tag{4.103}$$

Example 4.1 In an oil fired boiler the fuel had an analysis by mass: carbon 84%, hydrogen 10% sulphur 3.2%, oxygen 1.6%, remainder incombustible. The analysis of dry flue gas by volume gave: combined $CO_2 + SO_2$ 15.72%, O_2 1%, there being no CO or SO_3. Calculate per kg of fuel (a) mass of air supplied, (b) percentage excess air supplied, (c) mass of dry flue gas formed, and (c) mass of water vapour formed.

Solution Let us consider 100 kg of fuel oil. Let x moles of oxygen are supplied for combustion. The combustion equation can be written as

$$\frac{84}{12}C + \frac{10}{2}H_2 + \frac{3.2}{32}S + \frac{1.6}{32}O_2 + xO_2 + 3.76\,xN_2$$

$$\longrightarrow bCO_2 + dSO_2 + eO_2 + fN_2 + gH_2O$$

Equating coefficients:

Carbon $\qquad \frac{84}{12} = b \qquad \therefore \quad b = 7$

Hydrogen $\qquad \frac{10}{2} = g \qquad \therefore \quad g = 5$

Oxygen $\qquad \frac{1.6}{32} + x = b + d + e + \frac{g}{2}$

Sulphur $\qquad \frac{3.2}{32} = d \qquad \therefore \quad d = 0.1$

Nitrogen $\qquad 3.76x = f$

Volumetric analysis:

$$\frac{b+d}{b+d+e+f} = 0.1572, \qquad \frac{e}{b+d+e+f} = 0.01$$

On dividing, $\qquad \frac{b+d}{e} = 15.72$

or $\qquad e = 7.1/15.72 = 0.452$

Substituting in O_2 – balance,

$$0.05 + x = 7 + 0.1 + 0.452 + 2.5 \qquad \text{or} \qquad x = 10.002$$

Mass of oxygen supplied = $10.002 \times 32 = 320.1$ kg

$\therefore$ Mass of air supplied for 100 kg fuel = $\frac{320.1}{0.232} = 1380$ kg

$\therefore$ Mass of air supplied per kg fuel, $W_A = 13.80$ kg *Ans.*(a)

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Theoretical air required per kg fuel

$$W_{Th} = 11.5C + 34.5\left(H - \frac{0}{8}\right) + 4.3S$$

$$= 11.5 \times 0.84 + 34.5\left(0.1 - \frac{0.016}{8}\right) + 4.3 \times 0.032$$

$$= 9.66 + 3.38 + 0.14 = 13.18 \text{ kg}$$

$$\text{Percentage excess air} = \frac{W_A - W_{Th}}{W_{Th}} \times 100 = \frac{13.80 - 13.18}{13.18} \times 100$$

$$= 4.7\%$$

Ans.(b)

Mass of dry flue gas formed for 100 kg fuel

$$= bCO_2 + dSO_2 + eO_2 + fN_2$$

$$= 7 \times 44 + 0.1 \times 64 + 0.452 \times 32 + 3.76 \times 10.002 \times 28$$

$$= 308 + 6.4 + 14.46 + 1053.01 = 1381.87 \text{ kg}$$

Mass of dfg formed per kg fuel = 13.82 kg *Ans.*(c)

Mass of water vapour formed per kg fuel

$$= gH_2O = \frac{5 \times 18}{100} = 0.90 \text{ kg}$$

Ans.(d)

Example 4.2 Propane gas is reacted w1ith air in such a ratio that an analysis of the dry products of combustion gives CO_2 11.5%, O_2 2.7%, and CO 0.7%. What is the percentage excess air used?

Solution Let x moles of propane are reacted by a moles of oxygen supplied. The combustion equation is thus,

$$xC_3H_8 + aO_2 + 3.76aN_2 \rightarrow 11.5CO_2 + 0.7CO + 2.7O_2 + 85.1N_2 + bH_2O$$

Equating moles for

Nitrogen : $3.76a = 85.1$ or $a = 22.65$

Carbon : $3x = 11.5 + 0.7 = 12.2$ or $x = 4.07$

Oxygen : $22.65 = 11.5 + \frac{0.7}{2} + 2.7 + \frac{b}{2}$ $b = 16.2$

Hydrogen : $8(4.07) = 2(16.2)$

Therefore, the combustion equation becomes

$$4.07C_3H_8 + 22.65O_2 + 85.1N_2 \longrightarrow 11.5CO_2 + 0.7CO + 2.7\,O_2 + 85.1\,N_2 + 16.2\,H_2O$$

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For 1 mole of C_3H_8,

$$C_3H_8 + 5.57O_2 + 20.91N_2 \longrightarrow 2.83CO_2 + 0.17CO + 0.66O_2 + 20.91N_2 + 3.98\ H_2O$$

Stoichiometric combustion equation is

$$C_3H_8 + 5O_2 + 5(3.76)N_2 \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$

Theoretically, 5 moles of oxygen are required. But in the actual case 5.57 moles of O_2 are supplied. Therefore, the excess air supplied is

$$\frac{5.57 - 5.00}{5.00} \times 100 \quad \text{or} \quad 11.4\%$$

Ans.

Example 4.3 An unknown hydrocarbon fuel, C_xH_y, was allowed to react with air. An Orsat analysis was made of a representative sample of the product gases with the following result: CO_2 12.1%, O_2 3.8% and CO 0.9%. Determine (a) the chemical equation for the actual reaction, (b) the composition of the fuel, (c) the air fuel ratio during the test, and (d) the excess or deficiency of air used.

Solution The combustion equation can be written as

$$C_xH_y + aO_2 + 3.76\,aN_2 \rightarrow 12.1CO_2 + 3.8O_2 + 0.9CO + 83.2N_2 + bH_2O$$

Equating moles for

Nitrogen: $3.76a = 83.2$ or $a = 22.1$

Oxygen: $2 \times 12.1 = 2 \times 12.1 + 2 \times 3.8 + 0.9 + b$ $\quad b = 11.5$

Carbon: $x = 12.1 + 0.9 = 13.0$

Hydrogen: $y = 2b = 2 \times 11.5 = 23$

The combustion equation becomes

$$C_{13}H_{23} + 22.1O_2 + 83.2N_2 \rightarrow 12.1CO_2 + 3.8O_2 + 0.9CO + 83.2N_2 + 11.5H_2O$$

Ans.(a)

The composition of the fuel is $C_{13}H_{23}$. *Ans.*(b)

Stoichiometric equation is

$$C_{13}H_{23} + 18.75O_2 + 3.76\,(18.75)N_2 \rightarrow 13CO_2 + 11.5H_2O + (18.75)3.76N_2$$

% excess air $= \dfrac{22.1 - 18.75}{18.75} \times 100 = 18\%$ *Ans.*(c)

Actual air supplied per mole of $C_{13}H_{23}$

$$= \frac{22.1 \times 32}{0.232} = 3048.276 \text{ kg}$$

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$$\therefore \qquad \text{Air–fuel ratio} = \frac{3048.276}{179} = 17.03 \qquad \textit{Ans.}(d)$$

Example 4.4 A steam generator operates under the following conditions:

Coal analysis: carbon 61.0, hydrogen 4.0, oxygen 3.0, nitrogen 2.0, sulphur 1.0 moisture 4.0 and ash 25.0

HHV of coal = 24.3 MJ/kg.

Dry flue gas analysis: CO_2 12.0%, CO 1.5%, O_2 7.0% and N_2 79.5%

Unburnt carbon = nil

Exhaust gas temperature = 170 °C

Energy loss other than dry exhaust loss and incomplete combustion = 3% of HHV

Steam generation rate = 150 t/h

Steam condition at boiler outlet = 100 bar, 500 °C

Feedwater inlet temperature = 160 °C

Determine (a) the amount of dry flue gas produced per kg fuel, (b) the dry exhaust loss and incomplete combustion loss per kg fuel, (c) the boiler efficiency, (d) the fuel burning rate, (e) the percentage of excess air used, and (f) the percentage of energy absorbed in the superheater. Given: heat of reaction of $C + \frac{1}{2}O_2 \longrightarrow CO_2$ is 33.083 MJ/kg carbon and that of $C + \frac{1}{2}O_2 \longrightarrow CO$ is 9.5 MJ/kg carbon, $\bar{c}_p$ of dfg is 1.05 kJ/kg K, and the ambient air temperature is 30 °C.

Solution Mass of dry flue gas produced per kg fuel

$$= \frac{C_{ab}(44CO_2 + 28CO + 32O_2 + 28N_2)}{12(CO_2 + CO)}$$

$$= \frac{0.61(44 \times 0.12 + 28 \times 0.015 + 32 \times 0.07 + 28 \times 0.795)}{12(0.12 + 0.015)}$$

$$= \frac{0.61 \times 30.2}{1.62} = 11.37 \text{ kg} \qquad \textit{Ans.}(a)$$

Energy loss due to dry exhaust gases

$$= w_{dfg}\,\bar{c}_p\,(t_g - t_a) = 11.37 \times 1.05\,(170 - 30)$$

$$= 1671.39 \text{ kJ/kg fuel} \qquad \textit{Ans.}(b)$$

Energy loss due to incomplete combustion

$$= w_{dfg}\,\frac{28CO}{44CO_2 + 28CO + 32O_2 + 28N_2} \times (33083 - 9500)\frac{12}{28}$$

$$= 11.37\,\frac{28 \times 0.015}{44 \times 0.12 + 28 \times 0.015 + 32 \times 0.07 + 28 \times 0.795} \times 10107$$

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$$= \frac{11.37 \times 0.42 \times 10107}{30.2} = 1598.18 \text{ kJ/kg fuel} \qquad \textit{Ans.}(c)$$

Energy loss other than dry exhaust loss and incomplete combustion loss = 0.03 × 24.3 = 0.729 MJ/kg fuel

Total energy loss = 1.671 + 1.598 + 0.729 = 3.998 MJ/kg fuel

$$\text{Boiler efficiency} = \frac{\text{HHV} - \Sigma\text{losses}}{\text{HHV}} \times 100 = \frac{24.3 - 3.998}{24.3} \times 100$$

$$= 83.54\% \qquad \textit{Ans.}(c)$$

Again, $\eta_{\text{boiler}} = \dfrac{w_s(h_1 - h_f)}{w_f \times \text{HHV}}$

At 100 bar, 500 °C, h_1 = 3373.7 kJ/kg

At 160 °C, h_f = 675.55 kJ/kg

On substitution, $0.8354 = \dfrac{150{,}000(3373.7 - 675.55)}{w_f \times 24300}$

$\therefore$ w_f = 19937 kg/h = 19.937 t/h = 5.538 kg/s *Ans.*(d)

Theoretical air required per kg fuel,

$$W_{\text{Th}} = 11.5\text{C} + 34.5\left(\text{H} - \frac{\text{O}}{8}\right) + 4.3\text{S}$$

$$= 11.5 \times 0.61 + 34.5\left(0.04 - \frac{0.03}{8}\right) + 4.3 \times 0.01$$

$$= 7.015 + 1.25 + 0.043 = 8.31 \text{ kg}$$

Actual air supplied per kg fuel

$$W_A = \frac{3.04\,\text{N}_2 C_{ab}}{\text{CO}_2 + \text{CO}} - \frac{1}{0.768}\,\text{N}$$

$$= \frac{3.04 \times 0.795 \times 0.61}{0.12 + 0.015} - \frac{1}{0.768} \times 0.02 = 10.89 \text{ kg}$$

$$\text{Percentage excess air used} = \frac{W_A - W_{\text{Th}}}{W_{\text{Th}}} \times 100 = \frac{10.89 - 8.31}{8.31} \times 100$$

$$= 31.05\% \qquad \textit{Ans.}(e)$$

At 100 bar, h_g = 2724.7 kJ/kg

Percentage of energy absorbed in the superheater

$$\frac{h_1 - h_g}{h_1 - h_f} \times 100 = \frac{3373.7 - 2724.7}{3373.7 - 675.55} \times 100 = 24.05\% \qquad \textit{Ans.}(f)$$

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Example 4.5 The ultimate analysis of a fuel oil is given to be: carbon 83.7%, hydrogen 12.7, sulphur 0.7, nitrogen 1.7 and oxygen 1.2. The combustion air has a dry bulb temperature of 27 °C and a wet bulb temperature of 21°C. With 30% excess air and assuming complete combustion, find (a) the total volume of combustion products at 200 °C and 1.013 bar, and (b) the dry flue gas analysis based on CO_2, O_2 and N_2.

Solution Theoretical air required per kg fuel,

$$W_{Th} = 11.5 \times 0.837 + 34.5\left(0.127 - \frac{0.012}{8}\right) + 4.3 \times 0.007$$

$$= 13.98 \text{ kg}$$

$$W_A = 1.3 \times 13.98 = 18.174 \text{ kg/kg fuel}$$

From the psychrometric chart, at DBT = 27 °C and WBT = 21°C, specific humidity = 0.0132 kg moisture/kg dry air.

Water vapour entering with air per kg fuel

$$= 18.174 \times 0.0132 = 0.24 \text{ kg vap/kg fuel}$$

Total water vapour formed per kg fuel = $9H + M + \gamma_A W_A$

$$= 9 \times 0.127 + 0 + 0.24$$

$$= 1.383 \text{ kg}$$

Mass of combustion gases per kg fuel:

$$CO_2 = \frac{44}{12} \times 0.837 = 3.07; \; CO = 0 \text{ (complete combustion)}$$

$$O_2 = 0.232 \times 0.30 \times 13.98 = 0.97$$

$$N_2 = 0.768\,(1 + 0.3) \times 13.98 + 0.017 = 13.99$$

$$SO_2 = \frac{64}{32} \times 0.007 = 0.02; \qquad H_2O = 1.383$$

Moles of combustion gases formed per kg fuel

$$= \frac{3.07}{44} + \frac{0.97}{32} + \frac{13.99}{28} + \frac{0.02}{64} + \frac{1.383}{18}$$

$$= 0.0698 + 0.0303 + 0.4992 + 0.0003 + 0.0769 = 0.6765$$

Volume of flue gases at 200 °C, 1.013 bar per kg fuel

$$V_G = 0.6765 \times 22.4 \times \frac{273 + 200}{273} \times \frac{1.013}{1.013} = 26.26 \text{ m}^3 \qquad \textit{Ans.}(a)$$

Composition of dry flue gas by volume:

$$CO_2 = \frac{0.0698}{0.0698 + 0.0303 + 0.4992} = 0.1164 \text{ or } 11.64\%$$

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$$O_2 = \frac{0.0303}{0.5993} = 0.0505 \quad \text{or} \quad 5.05\%$$

$$N_2 = \frac{0.4992}{0.5993} = 0.8331 \quad \text{or} \quad 83.31\% \qquad \textit{Ans.}(b)$$

Example 4.6 The volumetric analysis of a fuel gas is given as: C_2H_6 22.6%, CH_4 73.6%, CO_2 2.4% and N_2 1.4%, Assuming combustion air to be dry and in 25% excess, find (a) the molecular weight of the combustion products, (b) the total gas volume for complete combustion at 260 °C, 1.013 bar, and (c) the dry flue gas analysis based on CO_2, O_2 and N_2.

Solution Assuming 100 moles of fuel gas, the combustion equation can be written as

$$22.6C_2H_6 + 73.6CH_4 + 2.4CO_2 + 1.4N_2 + 226.3\,(O_2 + 3.76N_2) \times 1.25$$
$$= 121.2CO_2 + 215.0H_2O + 56.6O_2 + 1065.4N_2$$

Molecular weight of fuel gas

$$= 0.226 \times 30 + 0.736 \times 16 + 0.024 \times 44 + 0.014 \times 28 = 20 \text{ kg/kg mol}$$

$$\text{Moles of air supplied} = \frac{226.3(32 + 3.76 \times 28) \times 1.25}{28.96} = 1340$$

Total reaction moles = 100 moles of fuel gas + 1340 moles of air

= 1440 moles of combustion gas

Molecular weight of combustion gas

$$= \frac{121.2 \times 44 + 215 \times 18 + 56.6 \times 32 + 1065.4 \times 28}{1440}$$

$$= \frac{40845.2}{1440} = 28.36 \text{ kg/kg mol} \qquad \textit{Ans.}(a)$$

Total number of moles of combustion gas per kg fuel gas

$$= \frac{1440}{100} \frac{\text{kg mol}}{\text{kg mol fuel gas}} \frac{\text{kg mol of fuel gas}}{20 \text{ kg}} = 0.72$$

Volume of combustion products per kg fuel gas

$$V_G = 0.72 \times 22.4 \frac{273 + 260}{273} = 31.49 \text{ m}^3\text{/kg fuel} \qquad \textit{Ans.}(b)$$

Gas analysis by volume:

$$CO_2 = \frac{121.2}{121.2 + 56.6 + 1065.4} = 0.098 \quad \text{or} \quad 9.8\%$$

$$O_2 = \frac{56.6}{1243.2} = 0.046 \quad \text{or} \quad 4.6\%$$

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$$N_2 = \frac{1065.4}{1243.2} = 0.856 \quad \text{or} \quad 85.6\% \qquad \textit{Ans.}(c)$$

Example 4.7 Test data show the following gas analyses before and after the air preheater.

	Entering	*Leaving*
CO_2	9.7	9.2
O_2	4.0	4.9
N_2	86.3	85.9

The coal used shows a carbon percentage of 72% by mass. Estimate the air leakage into the1air preheater per kg of coal fired.

Solution Before air preheater:

$$W_A = \frac{3.04\ N_2\ C_{ab}}{CO_2 + CO} = \frac{3.04 \times 0.859 \times 0.72}{0.097}$$

$$= 19.42 \text{ kg air/kg fuel}$$

After air preheater:

$$W'_A = \frac{3.04 \times 0.859 \times 0.72}{0.092} = 20.42 \text{ kg air/kg fuel}$$

$\therefore$ Air leakage $= W_A - W'_A = 1$ kg air/kg fuel

Example 4.8 A textile factory has a battery of 6 Lancashire boilers, each supplying 6 t/h of steam at 16 bar, 250 °C from feedwater at 30 °C. The boilers burn fuel oil of calorific value 43.96 MJ/kg with an overall efficiency of 75%. For efficient combustion 16 kg of air per kg fuel is required for which a draught of 20 mm of water gauge is required at the base of the chimney. The flue gases leave the boilers at 320 °C. The average temperature of the gases in the stack may be taken to be 300 °C. The atmosphere is at 30 °C.

Assuming the velocity of gases at stack exit to be negligible, determine the height of the stack and the diameter at its base.

Solution The draught produced is given by

$$Dp = gH(\rho_a - \rho_g) = gH\frac{p_a}{R_a}\left(\frac{1}{T_a} - \frac{1}{T_g}\right)$$

$$20 \times 10^{-3}\text{m} \times g \times 10^3 \frac{\text{kg}}{\text{m}^3}\frac{\text{m}}{\text{s}^2} = g \times H \times \frac{1.013 \times 100}{0.287}\left(\frac{1}{303} - \frac{1}{573}\right)$$

$$\therefore \qquad H = \frac{20 \times 0.287 \times 303 \times 573}{101.3 \times 270}\ \text{m} = 36.44\ \text{m} \qquad \textit{Ans.}$$

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Now $$\eta_{boiler} = \frac{w_s(h_1 - h_{feed})}{w_f \times CV}$$

$$0.75 = \frac{6000 \times 6(2919.2 - 125.8)}{w_f \times 43,960}$$

$\therefore$ $$w_f = \frac{36000 \times 2793.4}{0.75 \times 43,960} = 3050.12 \text{ kg/h} = 0.847 \text{ kg/s}$$

$$w_a = 16 \times 0.847 \text{ kg/s}$$

$\therefore$ $$w_{fg} = 17 \times 0.847 = 14.4 \text{ kg/s}$$

Again, $$w_{fg} = \rho_{fg} A V_g = \frac{p_{fg}}{R T_g} \frac{\pi}{4} D^2 (2gH)^{1/2}$$

$$14.4 = \frac{101.3}{0.287 \times 593} \frac{\pi}{4} D^2 (2 \times 9.81 \times 36.44)^{1/2}$$

$\therefore$ $$D^2 = \frac{14.4 \times 0.287 \times 593 \times 4}{101.3 \times \pi \times 26.74} = 1.152$$

or $$D = 1.073 \text{ m}$$ *Ans.*

Example 4.9 Compute the required motor capacity needed for the FD fan under the following conditions:

Coal rate	:	10 tonnes/h
Coal analysis	:	C 78.0, H 3.0, O 3.0, S 1.0, M 7.0, A 8.0
Excess air	:	30%
Plenum chamber pressure	:	180 mm water gauge
Mechanical efficiency of the fan	:	60%
Room temperature	:	30 °C

Solution

$$W_{Th} = 11.5C + 34.5\left(H - \frac{O}{8}\right) + 4.3S$$

$$= 11.5 \times 0.78 + 34.5\left(0.03 - \frac{0.03}{8}\right) + 4.3 \times 0.01$$

$$= 9.907 \text{ kg air/kg fuel}$$

$$W_A = 9.907 \times 1.3 = 12.9 \text{ kg air/kg fuel}$$

$$v_{air} = \frac{RT}{p} = \frac{0.287 \times 303}{101.325} = 0.858 \text{ m}^3\text{/kg}$$

$$\text{FD fan motor capacity} = \frac{W_A \times w_f \times v_{air} \Delta p}{\eta_{fan}}$$

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$$= \frac{12.9 \times 10000 \times 0.858 \times 180 \times 10^{-3} \times 9.81 \times 10^3}{3600 \times 0.6}$$

$$= 9048 \text{ W} = 90.48 \text{ kW} \qquad \textit{Ans.}$$

Example 4.10 For the above boiler (Example 4.9), if the draught produced by the ID fan is 250 mm water gauge, the gas temperature is 180 °C and the fan efficiency is 52%, find the motor capacity of the ID fan.

Solution

$$w_{fg} = w_f + w_a = 10 \times 10^3 + 12.9 \times 10 \times 10^3$$
$$= 139 \times 10^3 \text{ kg/h} \quad = 38.61 \text{ kg/s}$$

$$v_{fg} = v_a \frac{t_g + 273}{t_a + 273} = 0.858 \frac{453}{303} = 1.283 \text{ m}^3/\text{kg}$$

ID fan motor capacity

$$= \frac{38.61 \times 1.283 \times 250 \times 10^{-3} \times 9.81 \times 10^3}{0.52} \text{ W}$$

$$= 233632 \text{ W} = 233.63 \text{ kW}$$

Example 4.11 During a boiler trial a partial analysis of dry flue gas showed 13.2% CO_2 and 3.2% O_2 by volume. Some CO was probably present but its percentage was not measured. The analysis of the coal burnt was 88% carbon, 4.4% hydrogen and 7.6% ash. The moisture in the fuel was nil. Assuming that all the carbon and hydrogen have been burnt, estimate (a) the complete volumetric composition of the dry flue gas, (b) the actual amount of air supplied per kg coal, (c) mass of water vapour formed per kg coal, and (d) the dew point temperature of the flue gas.

Solution For 100 kg of coal, let a moles of oxygen are supplied.

$$\frac{88}{12}C + \frac{4.4}{2}H_2 + aO_2 + 3.76aN_2 \longrightarrow bCO_2 + dCO + eO_2 + fN_2 + gH_2O$$

Carbon balance: $\frac{88}{12} = 7.333 = b + d$ (1)

Hydrogen balance: $2.2 = g$

Oxygen balance: $a = b + \frac{d}{2} + e + \frac{g}{2} = b + \frac{d}{2} + e + 1.1$ (2)

Nitrogen balance: $3.76a = f$ (3)

From dfg analysis, $0.132 = \frac{b}{b + d + e + f}$ (4)

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$$0.032 = \frac{e}{b+d+e+f} \quad (5)$$

Dividing (4) by (5), $\frac{b}{e} = 4.125$ (6)

From Eqs (1), (2) and (6),

$$a = 7.333 - d + \frac{d}{2} + \frac{7.333-d}{4.125} + 1.1 = 10.21 - 0.742\,d$$

$\therefore$ $$f = 38.39 - 2.79\,d$$

From Eq. (4),

$$0.132 = \frac{7.333-d}{7.333-d+d+\frac{7.333-d}{4.125}+38.39-2.79d} = \frac{7.333-d}{47.5-3.032d}$$

$\therefore$ $$d = 1.772$$

$\therefore$ $$a = 10.21 - 0.742 \times 1.772 = 8.895$$

$$b = 7.333 - 1.772 = 5.561$$

$$e = \frac{5.561}{4.125} = 1.348$$

$$f = 3.76 \times 8.895 = 33.45$$

Mass of air supplied for 100 kg coal = $\frac{8.895 \times 32}{0.232} = 1227$ kg

$\therefore$ Mass of air supplied per kg coal = 12.27 kg *Ans.*(b)

Total number of moles of dfg = $b + d + e + f = 42.131$

$$\%\ CO_2 \text{ by volume} = \frac{5.561}{42.131} \times 100 = 13.20\%$$

$$\%\ O_2 \text{ by volume} = \frac{1.348}{42.131} \times 100 = 3.20\%$$

$$\%\ CO \text{ by volume} = \frac{1.772}{42.131} \times 100 = 4.20\%$$

$$\%\ N_2 \text{ by volume} = \frac{33.45}{42.131} \times 100 = 79.40\%$$ *Ans*(a)

Mass of water vapour formed per kg coal = $\frac{2.2 \times 18}{100} = 0.396$ kg *Ans.*(b)

Mole fraction of water vapour in flue gas

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$$s = \frac{g}{b+d+e+f+g} = \frac{2.2}{42.131+2.2} = 0.0496$$

∴ Partial pressure of water vapour

$$= x_{H_2O}\, p_{total} = 0.0496 \times 101.325 = 5.026 \text{ kPa}$$

∴ Dew point temperature (from steam tables) = 32.9 °C *Ans.*(d)

Example 4.12 A 200 m high 4 m dia stack emits 1000 kg/s of 100 °C gases into 5 °C air. The prevailing wind velocity is 50 km/h. The atmosphere is in a condition of neutral stability. Calculate the height of the gas plume.

Solution Using the correlation of Carson and Moses,

$$\Delta H = 2.62\,\frac{(Q_c)^{1/2}}{V_w} - 0.029\frac{V_s D}{V_w}$$

where, V_s = stack gas exit velocity, m/s

D = stack dia, m

V_w = wind velocity, m/s

Q_c = heat emission from plume, watts = $\dot{m}\, c_p\,(T_s - T_a)$

$\dot{m}$ = gas flow rate, kg/s

T_s = stack exit gas temperature, K

T_a = ambient air temperature, K

$Q_c = 1000 \times 1.005\,(100 - 5)$

$= 95475 \text{ kW} = 95.475 \times 10^6 \text{ W}$

$V_w = 50000/(60 \times 60) = 13.89$ m/s

$V_s = \dot{m}/\rho A$, where $\rho = (101.325)/(0.287 \times 373)$

$= 0.9465 \text{ kg/m}^3$

$$V_s = \frac{1000}{0.9465 \times \frac{\pi}{4} \times 4^2} = 84.08 \text{ m/s}$$

$$\Delta H = 2.62\,\frac{(95.475)^{1/2} \times 10^3}{13.89} - 0.029\,\frac{84 \times 4}{13.89}$$

$= 1843 - 0.7015 = 1842.3$ m *Ans.*

Example 4.13 A gas turbine is supplied with a fuel of calorific value of 20 MJ/kg and gravimetric analysis 65% C, 25% H_2, 10% O_2. The compressor takes in air at 1 bar, 27 °C and compresses it isentropically to a pressure of 4 bar. The exhaust gases heat the air leaving the compressor, before it enters the combustion chamber.

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Assuming that the regenerator effectiveness is 78%, the expansion in the turbine is isentropic, and the analysis of the dry exhaust gas is 6% CO_2, 1.5% CO, determine the maximum temperature in the cycle, thermal efficiency of the plant, and specific fuel consumption. For air, c_p = 1.005 kJ/kg K, g = 1.4 and for combustion products, c_p = 1.15 kJ/kg K, g = 1.33.

Solution Let us consider 1 kg fuel and let x kg mol of air is supplied. Then the combustion equation can be written as

$$\frac{0.65}{12}C + \frac{0.25}{2}H_2 + \frac{0.10}{32}O_2 + 0.21\,x\,O_2 + 0.79\,x\,N_2$$
$$\longrightarrow aCO_2 + bCO + CO_2 + dN_2 + eH_2O$$

A carbon balance gives, $\frac{0.65}{12} = a + b = 0.0542$

A hydrogn balance gives, $\frac{0.25}{2} = e = 0.125$

An oxygen balance gives, $\frac{0.10}{32} + 0.21x = a + \frac{b}{2} + c + \frac{e}{2}$

A nitrogen balance gives $0.79x = d$

Also,

$$\%\,CO_2 = \frac{a}{a+b+c+d} = 0.06$$

$$\%\,CO = \frac{b}{a+b+c+d} = 0.015$$

$$\therefore \qquad \frac{b}{a} = \frac{0.015}{0.06} = \frac{1}{4} = 0.25$$

$$\therefore \qquad a + b = a + 0.25\,a = 1.25a = 0.0542$$

or $\qquad a = 0.0434 \quad \text{and} \quad b = 0.0108$

$$\frac{0.10}{32} + 0.21\,x + d = a + \frac{b}{2} + c + \frac{e}{2} + d$$

Also, $0.0031 + 0.21x + 0.79x = (a + b + c + d) - \frac{1}{2}b + \frac{e}{2}$

$$= 0.7233 - \frac{0.0108}{2} + \frac{0.125}{2}$$

$$= 0.7804$$

$$\therefore \qquad x = 0.7773 \text{ k mol.}$$

Hence, air supplied = 0.7773 × 28.96 = 22.54 kg/kg fuel

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$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 4^{0.4/1.4} = 1.486$$

$$T_2 = 446 \text{ K}$$

An energy balance on the combustion chamber gives

$$w_a\, c_{pa}\,(T_2 - T_a) + w_f(\text{CV}) = (w_a + w_f)\, c_{pg}\,(T_3 - T_0)$$

$$22.54\,(1.005)\,(446 - 288) + 1.0\,(20{,}000) = (22.54 + 1)\,(1.15)\,(T_3 - 288)$$

T_o = datum temperature = 15 °C assumed

T_3 = 1159 K = maximum temperature of the cycle *Ans.*

$$\frac{T_3}{T_4} = (4)^{0.33/1.33}$$

$$\therefore \quad T_4 = 820 \text{ K}$$

Regenerator effectiveness, e is

$$e = \frac{T_5 - T_2}{T_4 - T_2} = \frac{T_5 - 446}{820 - 446} = 0.78$$

$$\therefore \quad T_5 = 738 \text{ K}$$

$$W_c = w_a c_{pa}\,(T_2 - T_1) = 22.54 \times 1.005\,(446 - 300) = 3307.3 \text{ kW}$$

$$W_T = 23.54 \times 1.15\,(1159 - 820) = 9177.1 \text{ kW}$$

$$Q_1 = 23.54\,(1.15)\,(11.59 - 738) = 11396.9 \text{ kW}$$

$$\therefore \quad \eta_{cycle} = \frac{9177.1 - 3307.3}{11396.9} = 0.515, \quad \text{or} \quad 51.5\%$$ *Ans.*

$$\text{Specific fuel consumption} = \frac{3600}{9177.1 - 3307.3} = \frac{3600}{5869.8}$$

$$= 0.613 \text{ kg/kWh}$$ *Ans.*

Example 4.14 A boiler uses 2000 kg/h of coal. The temperature of air supplied is 300 K, and the average temperature of the flue gas leaving the chimney is 650 K. The 33-m high steel chimney produces a draught of 20 mm of water column. Determine the (a) quantity of air supplied per kg of coal, (b) the draught in terms of column of hot gases, and (c) the base diameter of the chimney, assuming that 10% of the theoretical draught is used for creating the flow velocity of gases through the chimney.

Solution

$$\text{(a)} \quad h = 353\, H \left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right]$$

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$$20 = 353 \times 33 \left[\frac{1}{300} - \frac{m+1}{m}\frac{1}{650}\right]$$

$\therefore$ $$m = 19.727 \text{ kg air/kg coal}$$ *Ans.* (a)

(b) Let H_g be the height of the hot gas column producing the draught. Then

$$\Delta p = \rho_g g H_g = 353\, gH \left[\frac{1}{T_a} - \frac{m+1}{m}\frac{1}{T_g}\right]$$

where $$\rho_g = 353\,\frac{m+1}{m}\frac{1}{T_g} = 353 \times \frac{20.727}{19.727} \times \frac{1}{650} = 0.5706\ \frac{kg}{m^3}$$

$$H_g = H\left[\frac{m}{m+1}\frac{T_g}{T_a} - 1\right] = 33\left[\frac{19.73}{20.73} \times \frac{650}{300} - 1\right] = 35 \text{ m}$$

Ans. (b)

(c) Mass flow of flue gas = mass of coal + mass of air

$$= 2000\ (1 + 19.727)/3600$$

$$= 11.515 \text{ kg/s}$$

$\therefore$ Volume flow of flue gas $= \dfrac{11.515}{0.5706} = 20.18 \text{ m}^3/\text{s}$

Flue gas velocity, $V_g = \sqrt{2gH_g} = \sqrt{2 \times 9.18 \times 0.1 \times 35}$

$$= 8.29 \text{ m/s}$$

$$\frac{\pi}{4} d^2 \times V_g = 20.18$$

$$\frac{\pi}{4} d^2 = \frac{20.18}{8.29}$$

$\therefore$ d = base diameter of chimney

$$= 1.76 \text{ m}$$ *Ans.* (c)

Example 4.15 A forced draught fan supplies air at 10 m/s against a draught of 20 mm of water across the fuel bed. Estimate the power required to run the fan if 2500 kg/h of coal is consumed and 16 kg of air is supplied per kg of coal burned. The temperature of the flue gas and the ambient air may be taken as 600 K and 300 K respectively.

If the forced draught fan is replaced by an induced draught fan, what will be the power required to drive the fan?

Solution

Density of air, $\rho_a = \dfrac{p}{RT} = \dfrac{101.3}{0.287 \times 300} = 1.176 \text{ kg/m}^3$

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$$\text{Velocity head} = \rho V^2/2 = 1.176 \times \frac{100}{2} = 58.83 \text{ N/m}^2$$

Static pressure head = 20 mm of water

$$= 0.02 \times 10^3 \times 9.81 = 196.2 \text{ N/m}^2$$

Total pressure head to be produced by the fan

$$\Delta p = 196.2 + 58.83 = 255.03 \text{ N/m}^2$$

Volume of air to be handled

$$\dot{V} = \frac{2500 \times 16 \times 1}{3600 \times 1.176} = 9.45 \text{ m}^3/\text{s}$$

$$\therefore \quad \text{Power required by the FD fan} = \frac{\dot{V}\,\Delta p}{1000}$$

$$= \frac{9.45 \times 255.03}{1000} = 2.41 \text{ kW} \qquad \textit{Ans.}$$

If an ID fan replaces the FD fan, since the ID fan handles flue gases,

$$\text{Density of flue gas, } \rho_g = \frac{m+1}{m} \times \frac{353}{T_g}$$

$$= \frac{17}{16} \times \frac{353}{600} = 0.625 \text{ kg/m}^3$$

$$\text{Velocity head} = 0.625 \times \frac{100}{2} = 31.25 \text{ N/m}^2$$

Draught to be produced

$$\Delta p = 196.2 + 31.25 = 227.45 \text{ N/m}^2$$

Volume flow rate of flue gas, $\dot{V}_g$

$$= \frac{2500 \times 17}{3600} \times \frac{1}{0.625} = 18.88 \text{ m}^3/\text{s}$$

Power required by the ID fan

$$= 18.88 \times 227.45 \times 10^{-3} = 4.29 \text{ kW} \qquad \textit{Ans.}$$

SHORT-ANSWER QUESTIONS

4.1 What are the fuels used for generation of steam?
4.2 How was coal formed?
4.3 What is rank of coal? What are the different ranks of coal?
4.4 What are the types of coal analysis?
4.5 What is proximate analysis of coal?

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4.6 What do you understand by ultimate analysis of coal?
4.7 What are the important coal properties relevant to boilers?
4.8 What is the difference between a caking coal and a free-burning coal?
4.9 What is swelling index?
4.10 Define grindability index of a coal.
4.11 What do you understand by weatherability of coal?
4.12 What is a bomb calorimeter?
4.13 How is lower heating value of coal different from the higher heating value?
4.14 What is ash softening temperature? What is its effect on boiler performance?
4.15 How is fuel oil formed?
4.16 How is oil brought out from the oil-well?
4.17 What are the main constituents of fuel oil?
4.18 What is the ultimate analysis of fuel oil?
4.19 What are the required properties of fuel oil for boiler firing?
4.20 What is pour point?
4.21 What is flash point?
4.22 How is the natural gas formed?
4.23 What is the approximate heating value of natural gas?
4.24 What are the major constituents of natural gas?
4.25 What is compressed natural gas? What is its probable application?
4.26 What is LPG? What are its constituents? Where is it used?
4.27 What is an emulsion?
4.28 How does an emulsion of water in heavy oil promote combustion?
4.29 What is a coal-oil mixture (COM)?
4.30 How is COM prepared?
4.31 What are the merits of COM as a boiler fuel?
4.32 What is the stabilizer used for COM?
4.33 Explain the merits and demerits of the coal-water mixture as a boiler fuel.
4.34 What are the industrial wastes and byproducts used as boiler fuels?
4.35 Evaluate wood waste as a boiler fuel.
4.36 What is an incinerator? Where is it used?
4.37 What are synfuels?
4.38 How is coal gasified?
4.39 What is a town gas?
4.40 What is water gas reaction?
4.41 How are gases classified according to heating value?
4.42 What is synthesis gas?
4.43 What is pipeline gas?
4.44 What is hydrogasification?
4.45 What is underground gasification of coal?
4.46 What are the attractive features of underground gasification of coal?
4.47 How is coal liquefied? What are the different products?
4.48 What is Fischer—Tropsch process?
4.49 What is Sasol plant in South Africa famous for?
4.50 What are the basic modes of liquefying coal?
4.51 What is hydrogenation process?

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4.52 What do you understand by catalytic conversion process?
4.53 What is hydropyrolysis?
4.54 What is biomass? How is it formed?
4.55 What are biofuels? How can these be produced?
4.56 What are the routes of bioconversion?
4.57 What is thermochemical conversion?
4.58 Explain what you understand by biohemical conversion?
4.59 What is anaerobic digestion?
4.60 What is fermentation?
4.61 How is gasohol produced?
4.62 What is an energy farm?
4.63 Explain the scope of an aquatic farm.
4.64 What is combustion?
4.65 What are the four basic conditions to be fulfilled to burn a fuel efficiently?
4.66 What is MATT?
4.67 Why is excess air always required to be supplied for combustion? What is dilution coefficient?
4.68 What are the different methods of measuring the amount of excess air supplied?
4.69 Why is the flue gas analysis said to be measured on the dry basis?
4.70 How can the degree of burnout of carbon in a fuel be determined?
4.71 What do you understand by the dew point temperature of flue gases? Why is it important in boiler design?
4.72 What is Dulong's formula? How is it arrived at?
4.73 How does the proper control of the right amount of excess air maintain optimum combustion efficiency?
4.74 Why is the oxygen measurement considered the preferred method for combustion control?
4.75 Give the mass balance of a steam generator.
4.76 Give the dry refuse analysis by mass.
4.77 Enlist the various energy losses taking place in a steam generator.
4.78 Define the efficiency of a steam generator.
4.79 What is draught? What are the functions of the draught system?
4.80 How is draught produced in a steam generator?
4.81 What is natural draught? How is it caused?
4.82 What is the function of stack in a boiler?
4.83 What do you understand by (a) dispersion, (b) plume height with reference to a stack?
4.84 What is mechanical (or artificial) draught?
4.85 What is the function of the FD fan? Where is it located?
4.86 What is the function of the ID fan? Where is it located?
4.87 Explain what you understand by balanced draught.
4.88 What is a pressurized furnace? What are its characteristic features?
4.89 How are pressure and volume flow of flue gases measured?
4.90 What are the different types of fans used for producing draught?

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4.91 Why are centrifugal fans with backward-curved blading normally used for FD fans?
4.92 Why are centrifugal fans with forward-curved blading normally used for ID fans?
4.93 Explain the two common methods of controlling the output of fans.
4.94 Explain the characteristic features of damper control.
4.95 What are the advantages of the variable speed control of fans? What is the effect of speed on fan performance? What are the types of drives for the fan?
4.96 What is (a) a primary air fan, (b) a gas recirculation fan?
4.97 How is the noise of fans reduced?
4.98 Define (a) enthalpy of formation, (b) internal energy of combustion, and (c) enthalpy of combustion.
4.99 What do you understand by (a) heating value at constant pressure, and (b) heating value at constant volume?
4.100 Explain what you understand by adiabatic flame temperature. What is its importance in the design of a combustor? How is it controlled?
4.101 Explain the effect of dissociation on the combustion temperature.
4.102 What is free energy of formation? What is the standard free energy change?
4.103 Define the equilibrium constant of a chemical reaction. How is it related to the free energy change?

PROBLEMS

4.1 A bituminous coal has the following composition: carbon 71.6, hydrogen 4.8, oxygen 6.3, nitorgen 1.3, sulphur 3.4, moisture 3.5 and ash 9.1. Calculate for complete combustion per kg of coal (a) theoretical weight of air per kg of coal, (b) actual weight of air required with 60% excess air, (c) actual volume of air at 1.013 bar, 25 °C, and (d) HHV of coal. Write the complete combustion equation and estimate the LHV of coal and the dew point temperature.

4.2 The analysis of fuel oil is as follows; carbon 83.9, hydrogen 13.9, sulphur 2.2, oxygen 0.0, nitrogen 0.0. Calculate for complete combustion with 20% excess air (a) the flue gas volume in m^3/kg of oil at 200 °C and 1.013 bar, and (b) dry gas volumetric analysis.

4.3 A fuel gas consists of 85% methane and 15% ethane by volume. What should be the CO_2 meter reading if the above fuel is burnt with 10% excess air?
[*Ans.* 10.9%]

4.4 The wet products of combustion from an oil-fired steam generator furnace pass through a tubular air preheater. The entering temperature is 320 °C. the exit temperature is 130 °C and the mean c_p is 1.09 kJ/kg K. The oil has a percentage composition by weight: carbon 87.6, hydrogen 9.0, oxygen 2.0, moisture 1.4. The dry volumetric analysis of the products: CO_2 14.6% and O_2 2.92%, the remainder being nitrogen. The temperature rise of preheated

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air is 180°C. Calculate (a) the air-fuel ratio, and (b) the heat exchanger effectiveness.

[*Ans.* (a) 15, (b) 0.82]

4.5 The following data refer to a certain steam generator with pulverized coal system:
Evaporation rate: 150 t/h
Design pressure: 150 bar
Superheater outlet temperature: 500 °C
Economiser inlet temperature of feedwater: 180 °C
Overall boiler efficiency: 90%
Calorific value of coal: 23.1 MJ/kg
Heat release rate in furnace: 186.5 kW/m^3
Width: Length: height of furnace = 1 : 2 : 3
Determine (a) the dimensions of the furnace, (b) the fuel burning rate, (c) the percentage of total heat that each of the surface (economiser, evaporator and superheater) contributes.

4.6 The coal supplied to a furnace had the following composition: carbon 85.5, hydrogen 5.0, oxygen 4.0 and incombustibles 5.5. During a test it was found that the ashes discharged from the furnace contained 0.015 kg of unburnt carbon per kg of coal supplied to the furnace. The analysis of the dry flue gas showed CO_2 14.5% and CO 1.3% by volume. Calculate the mass of air supplied and the complete volumetric composition of the flue gas.

[*Ans.* 13 kg/kg coal, CO_2 13.76, CO 1.24, O_2 3.0, N_2 76.65 and H_2O 5.35% by volume]

4.7 The percentage analysis by mass of the coal used in a boiler was: Carbon 83, hydrogen 6, oxygen 5 and ash 6. The dry flue gas contained CO_2 10.50, CO 1.30, O_2 7.67 and N_2 80.53, % by volume. The temperatures of the air and flue gas were 15 °C and 215 °C respectively. Find (a) the total mass of air supplied per kg coal, (b) the partial pressure of the steam in the hot flue gas, if the pressure of the flue gas is 1 bar, (c) the heat carried away by the moist flue gas per kg of coal fired including that due to the CO present.

[*Ans.* (a) 17.13 kg/kg coal, (b) 0.05 bar, (c) (1624 × 4.187) kJ/kg coal]

4.8 A coal analysis by mass is 73% C , 12% ash, 15% H_2O. When burned in a boiler the residue contains 18% C by mass. The analysis of the dry flue gas, by volume, is CO_2 11.8%, CO 1.3%, O_2 5.5%. Calculate the % C in the coal which undergoes combustion and the actual air used.

[*Ans.* 97%, 13.47]

4.9 The analysis of a coal by mass is 82% C, 6% H, 6% ash, 2% O and 4% H_2O. Calculate the stoichiometric air-fuel ratio. The actual air supplied is 18 kg/kg fuel. Given that 80% of the carbon is completely burnt and all the hydrogen, calculate the volumetric analysis of the dry products.

[*Ans.* 11.42 kg/kg; 9.1% CO_2, 2.3% CO, 7.2% O_2, 81.4% N2]

4.10 A boiler is supplied with duel-fuel burners, using natural gas and fuel oil. Analysis of natural gas, by volume, is 94% CH_4, 2% C_2H_6, and 4% CO_2. Analysis of fuel oil, by mass, is 87% C, 13% H_2. The analysis of dry

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combustion product is CO_2 10%, CO 0.64%, O_2 4.56%, N_2 84.80%. Calculate the m^3 gas used/kg oil.

[*Ans.* 2.26]

4.11 A circular tank 10 m long and 1.5 m diameter is used for oil storage. Calculate the number of days' oil supply the tank can hold for continuous operation at the following conditions:
Steam flow, 2000 kg/h; Steam condition, 15 bar, dry saturated; Feedwater temperature, 110°C; Boiler efficiency, 75%, Specific gravity of fuel oil, 0.82; Calorific value of fuel oil, 42 MJ/kg.

[*Ans.* 4.08 days]

4.12 The following readings, representing percentage volume of the dry flue gas, were noted on a boiler instrument panel: O_2 meter 1.97% and CO_2 meter 16.52%
However, the CO_2 analyzer also absorbs SO_2. Therefore, the CO_2 meter reading includes a percentage of SO_2 as well as a percentage of CO_2. If the fuel used has an analysis by mass: carbon, 84%, hydrogen 6%, sulphur 4%, oxygen 4% and ash 2%, and assuming that no CO or SO_3 was formed,determine the correct complete dry flue gas analysis by volume and state the percentage excess air that was supplied.

[*Ans.* CO_2 15.80, SO_2 0.28, O_2 1.92 and N_2 82.0%, by vol; 9.2%]

4.13 Coal with composition by weight: carbon 75%, hydrogen 5%, oxygen 5%, moisture 8% and ash 7%, is burnt with excess air. The Orsat analysis of the resulting flue gas shows CO_2 9.09%, O_2 10.55%, CO nil and the balance nitrogen. Determine the weight of air used per kg of coal and the percentage of carbon which is not burnt.

[*Ans.* 19 kg air/kg coal, 0.048 kg unburnt carbon/kg coal]

4.14 The following stack gas was obtained from a furnace burning bituminous coal: CO_2 = 12.6, CO = 0.2, N_2 = 80.7, O_2 = 6.5
The ashpit refuse contains 8.5% combustable matter, and the coal used has the following analysis:

C	73.4	N	1.5	A	7.6
H	5.0	S	3.0	M	3.5
O	6.0				

Calculate (a) the mass of dry gas produced per kg coal, and (b) the mass of flue gas produced per kg coal, assuming dry combustion air.

4.15 Test data from a boiler are as follows: Coal analysis: C 75.0, H 5.0, O 7.0, N 2.0, S 1.0, M 1.0 and A 9.0
Dry gas analysis : CO_2 12.3, O_2 6.6, N_2 80.8 and CO 0.3
Refuse analysis : ash 92.0, carbon 8.0
Air has 1.7% moisture
Calculate per kg coal (a) actual air used, (b) air theoretically needed with given moisture and (c) percentage of excess air.

4.16 Data from a boiler test are as follows:
Coal analysis: C 57.6, H 3.73, O 5.8, N 1.07, S 3.29, M 12.00, A 16.50
HHV of coal as fired: 25 MJ/kg
Refuse analysis: C 20.0, A 80.0

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Dry flue gas analysis: CO_2 13.0, O_2 7.0, CO 0.0, N_2 80.0
Flue gas temperature, 180 °C
Air data: dry bulb, 20 °C, wet bulb, 15 °C
Make an energy balance of the boiler assuming an "unaccounted and radiation loss" as 3.7%.

4.17 A steam generator operates under the following conditions: Coal analysis: carbon 60.0, hydrogen 4.0, nitrogen 2.0, sulphur 1.5, oxygen 3.0, moisture 4.5 and ash 25.0. Dry flue gas analysis: CO_2 12.0, CO 1.5, O_2 7.0 and N_2 79.5%.
Steam condition at boiler outlet: 100 bar, 500 °C
Feedwater inlet temperature: 170 °C
Steam generation rate: 160 tonnes per hour
Steam generator efficiency: 85%
HHV of coal: 21 MJ/kg
Room temperature: 30 °C, C_{pg} = 1.08 kJ/kgK
Determine (a) the excess air coefficient, (b) the fuel burning rate, (c) the amount of dry flue gas produced per kg coal, and (d) the dry exhaust loss per kg coal. If the pressure losses before and after the furnace are 500 mm and 800 mm of w.g. respectively, and the exhaust gas temperature is 160 °C , find (e) the total power consumed by the FD and the ID fans having efficiencies 70% and 60% respectively.

[*Ans.* (a) 0.3075 or 30.75% excess air, (b) 6.622 kg/s or 23.84 t/h, (c) 11.185 kg/kg coal, (d) 1614 kJ/kg coal, (e) 427.45 kW by FD fan and 1246.4 kW by ID fan]

4.18 Liquid octane (C_8H_{18}) enters the combustion chamber of a gas turbine at 25 °C and air enters from the compressor at 227 °C. It is determined that 98% of the carbon in the fuel burns to form CO_2 and the remaining 2% burns to form CO. What amount of excess air will be required if the temperature of the products is to be limited to 827 °C? Given: (1) values h_f^o of C_8H_{18} (*l*), CO_2, CO and H_2O (g) are – 249952, – 393522, – 110529 and – 241817 kJ/kg mol, respectively. (2) Values of $(h_{500} - h_{298})$ for O_2 and N_2 are 6088 and 5912 kJ/kg mol, respectively. (3) Values of $(h_{1000} - h_{298})$ for CO_2, CO, H_2O (g), O_2 and N_2 are 33405. 21686, 25978, 22707 and 21406 kJ/kg mol, respectively.

[*Ans.* Eqn. $100C_8H_{18} + aO_2 + 3.76\,aN_2$ ® $784\,CO_2 + 16CO + bO_2 + 900H_2O + 3.76aN_2$, Actual A/F = 66, Theoretical A/F = 15.05, % Excess air = 338%]

4.19 The products of combustion of an unknown hydrocarbon fuel C_xH_y have the following composition as measured by an Orsat apparatus: CO_2 8.0%, CO 0.9%, O_2 8.8% an N_2 82.3%. Determine (a) the values of *x* and *y*, (b) the air fuel ratio, (c) the composition of the fuel, and (d) the percentage excess air used.

[*Ans.* (a) $C_{8.9}H_{18.5}$, (b) 24, (c) C 85.23%, H14.77%, (d) 61.6%]

4.20 For the chemical reaction,

$$CO_2 + H_2 \rightleftharpoons CO + H_2O$$

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the equilibrium value of the degree of reaction at 1200 K is 0.55. Determine the equilibrium constant and the Gibbs function change.

[*Ans.* 1.495, – 4005 J/g mol]

4.21 A fuel oil composed only of carbon, hydrogen, and sulphur is used in a steam generator. The dry flue gas analysis gives CO_2 11.7%, CO 0.44%, O_2 4.002%, SO_2 0.176% and N_2 83.682%. Find (a) the fuel mass composition, (b) the air-fuel ratio by mass, (c) the excess air used, in per cent, and (d) the dew point temperature of the flue gases if their pressure is 2 bar.

4.22 A desulphurized coal has the ultimate analysis on dry combustible basis: C 81.1%, H 4.4%, 0 2.7%, and N 1.8%, by mass. It burns in 120% air. The reactants were at 25 °C. Find the adiabatic flame temperature.

[*Ans.* 2186 K]

4.23 A power plant is situated at sea level. 1650 kg/s of flue gases enter a 5 m dia stack at 140 °C and leave at 110 °C. The outside air temperature is 10 °C. The stack is designed for a driving pressure of 0.007 bar. Using the Carson and Moses correlation, calculate the flue gas plume height if the prevailing winds are at 80 km/h.

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5

Combustion Mechanism, Combustion Equipment and Firing Methods

The rate of a chemical reaction, as in the combustion process, is greatly influenced by the temperature, concentration, preparation, and distribution of the reactants and mechanical turbulence. All these factors tend to increase contact between molecules of the reactants.

Higher temperature, for example, increases the velocity of molecular movement, permitting more frequent contacts between molecules. The concentration and distribution of the reactants in a given volume are directly related to the contact between interacting molecules. It assumes greater importance as the reaction nears completion. Mechanical turbulence induced by a properly designed burner system permits greater opportunity for the molecular contact.

In the process of generating steam, the furnaces or burner systems provide controlled, efficient conversion of the chemical energy of fuel to heat energy which, in turn, is transferred to the heat-absorbing surfaces of the steam generator. To do this, the firing systems introduce the fuel and air for combustion, mix these reactants, ignite the combustible mixture, and distribute the flame envelope and the products of combustion.

Furnaces can be broadly divided into two types:

(1) Grate-fired furnaces
(2) Chamber-type or flame furnaces

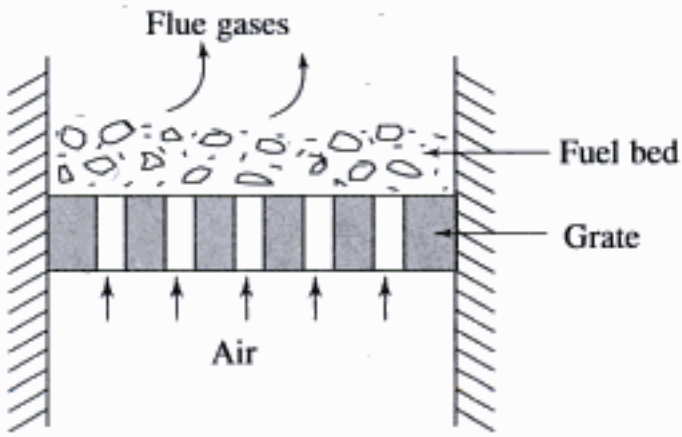

Fig. 5.1 *Grate-fired furnace*

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Grate-fired furnaces are suitable for burning solid fuels like coal, wood chips, bagasse, urban waste and so on. The schematic of a grate-fired furnace in which fuel is burned in a bed is shown in Fig. 5.1.Combustion of volatiles, fine carbon particles, and carbon monoxide occurs in the space above the fuel bed. In a chamber type or flame furnace (Fig. 5.2), which is used for firing pulverized coal, fuel oil and gas, a mixture of fuel and air is delivered into the furnace chamber. The fuel ignites and burns as it flows in the furnace space.

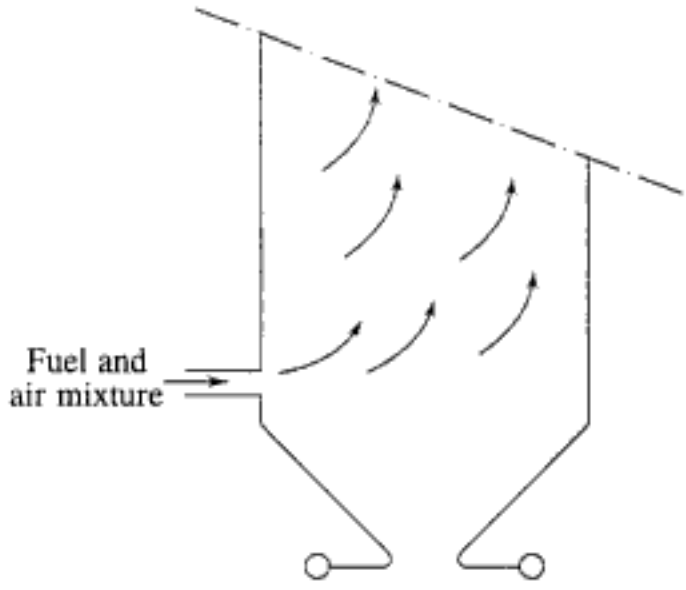

Fig. 5.2 *Chamber-type or flame furnace*

5.1 KINETICS OF COMBUSTION REACTIONS

The combustion of fuel in a furnace is determined by a number of physical and chemical factors. The physical factors include the process of mixing of fuel and air, the size of fuel particles and the surface area exposed for reaction. The chemical factors are related to the temperature and concentration of the reactants. The complex fields of velocities, temperatures and concentrations together determine the kinetics of chemical reactions.

Reactions involved in combustion of fuel proceed with evolution of heat, i.e. they are exothermic. These include the burning of carbon, hydrogen and sulphur in atmospheric air. At high temperatures, some reactions may occur with heat absorption, i.e. they are endothermic,

$$N_2 + O_2 = 2NO \qquad -180 \text{ kJ/mol}$$

$$CO_2 + C = 2CO \qquad -7.25 \text{ MJ/kg}$$

These chemical reactions can proceed in either a forward or reverse direction and are called *reversible*. During combustion of fuel in furnaces, the rate of a direct process, say $C + O_2 \longrightarrow CO_2$, is extremely higher than that of the reverse process, i.e. $CO_2 \longrightarrow C + O_2$. The equilibrium of these reactions is shifted towards the formation of the final products and therefore, these processes are irreversible.

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The intensity of combustion is characterized by the rate of reaction involved. The homogeneous single phase reaction rate is measured in terms of the variation of the concentration of one of the reacting substances, Δc_A, mol/m^3 (or g/m^3), in a given time interval $\Delta\tau$, or

$$r = \frac{\Delta c_A}{\Delta\tau} \text{ (mol/m}^3\text{s)} \tag{5.1}$$

The rate of a heterogeneous reaction, r_s, is expressed as the quantity of substance m, in kg, that has reacted on the solid particle surface, S, m^2, during a time interval $\Delta\tau$, s

$$r_s = \frac{m}{S\Delta\tau} \text{ (kg/m}^2\text{s)} \tag{5.2}$$

Reaction rates obey the *law of mass action,* according to which the rate of a reaction in a homogeneous medium at a constant temperature is proportional at any instant to the product of the concentrations of the reactants:

$$r = k\, c^m{}_A\, c^n{}_B \tag{5.3}$$

where c_A and c_B are the concentrations of the reactants A and B in the gas mixture, m and n are the number of moles of A and B respectively, and k is the reaction rate constant which depends on the temperature and the nature of the reactants.

Since the partial pressures of the components in a gas mixture at a certain temperature are proportional to their concentrations, Eq. (5.3) can be written in the form

$$r = kp^m{}_A p^n{}_B \tag{5.4}$$

In heterogeneous combustion, the concentration of the combustible substance (solid fuel) is constant and therefore, the rate of reaction depends only on the concentration of oxygen on the surface of solid fuel (c_s),

$$r = k\, c_s \tag{5.5}$$

If the concentrations of the reactants do not vary with time, the reaction rate is determined by the reaction rate constant k which depends on the temperature and nature of the reactants, as given by Arrhenius equation,

$$k = k_o\, e^{-E/RT} \tag{5.6}$$

where k_o is a constant, E is the activation energy, kJ/kg mol, $\overline{R}$ is the universal gas constant, 8.3143 kJ/(kg mol K) and T is the absolute temperature.

Figure 5.3 shows the dependence of reaction rate on (a) temperature, (b) activation energy, and (c) concentration of combustible matter. The rate of reaction increases rapidly with temperature. A chemical reaction occurs due to the collisions of the molecules of reacting substances. If all collisions resulted in a reaction, combustion would occur at an enormous rate and k would be equal to k_o. The energy that is sufficient to destroy the molecular bonds of the starting substances is called the *activation energy, E*. If the activation energy is high, it is difficult to destroy the molecular bonds of the

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original molecules, and so the reaction rate is low. Therefore, as E increases, r decreases. The activation energy of the reaction $C + O_2 = CO_2$ is $E_{CO_2} = 140$ kJ/mol and that of the reaction $C + \frac{1}{2}O_2 = CO$, $E_{CO} = 60$ kJ/mol. Thus, it follows that the rate of formation of CO in carbon oxidation is substantially higher than that of CO_2, and CO will be formed predominantly at the surface of burning carbon particles.

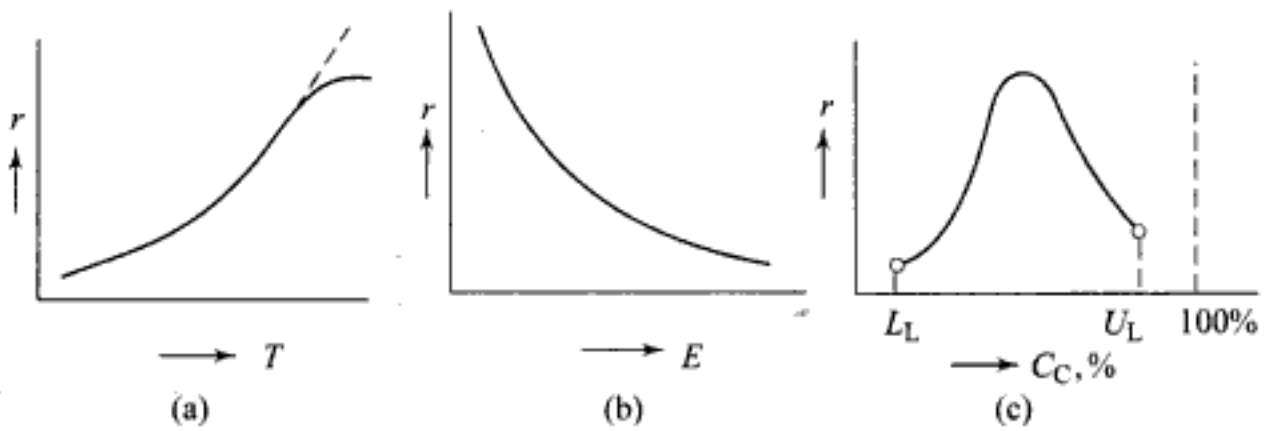

Fig. 5.3 *Effect of (a) temperature, (b) activation energy and (c) concentration of combustible matter, on the reaction rate, LL = lower limit, UL = upper limit*

Combustion cannot take place at any arbitrary concentration of fuel in the mixture, but only in a definite range of its concentrations in the air. There is a lower concentration limit below which combustion is impossible, and an upper concentration limit when any further increase of the concentration of the fuel prevents combustion. Thus, combustion is possible only in the concentration range between these two limits.

5.2 MECHANISM OF SOLID FUEL COMBUSTION

Upon heating, solid fuel particles first undergo a stage of *thermal preparation*, which consists in the evaporation of residual moisture and distillation of volatiles. Fuel particles are heated to a temperature at which volatiles are evolved rapidly (400–600 °C) in a few tenths of a second. The volatiles are then ignited, so that the temperature around a coke particle increases sharply and its heating is accelerated. The combustion of volatiles occurs in 0.2–0.5s. A high yield of volatiles produces enough heat to ignite coke particles. When the yield of volatiles is low, the coke particles must be heated additionally from an external source like a radiant furnace wall. The final stage is the combustion of coke particles at a temperature above 800–1000 °C. This is a heterogeneous reaction (gas–solid), the rate of which depends on the oxygen supply to the reacting surface. The burning of a coke particle takes up 1/2 to 2/3 of the total combustion time which is about 1-2.5 s (for pulverized coal).

In carbon–oxygen reaction, oxygen is first adsorbed from the gas volume on the surface of particles and reacts chemically with carbon to form complex

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carbon–oxygen compounds of the type C_xO_y, which then dissociate to form CO_2 and CO. The resulting reaction at about 1200 °C can be written as

$$4C + 3O_2 = 2CO + 2CO_2 \tag{5.7}$$

The ratio of the primary products, CO/CO_2, increases sharply with the increasing temperature of burning particles. At 1700 °C, the resulting reaction becomes

$$3C + 2O_2 = 2CO + CO_2 \tag{5.8}$$

where the CO/CO_2 ratio is equal to 2.

The primary reaction products are continuously removed from the surface of particles to the environment. In this process, CO diffusing out encounters the oxygen diffusing into the reacting surface and reacts with it within the boundary layer of gas to form CO_2. Consequently, the concentration of oxygen decreases sharply as it approaches the reacting surface, while the concentration of CO_2 increases (Fig. 5.4a). At a high temperature, CO can consume all the oxygen supplied, which, consequently, will not reach the reacting solid surface (Fig. 5.4b) and the endothermic reduction of CO_2 to CO will occur, with the high combustion temperature maintained due to high heat release (Fig. 5.4).

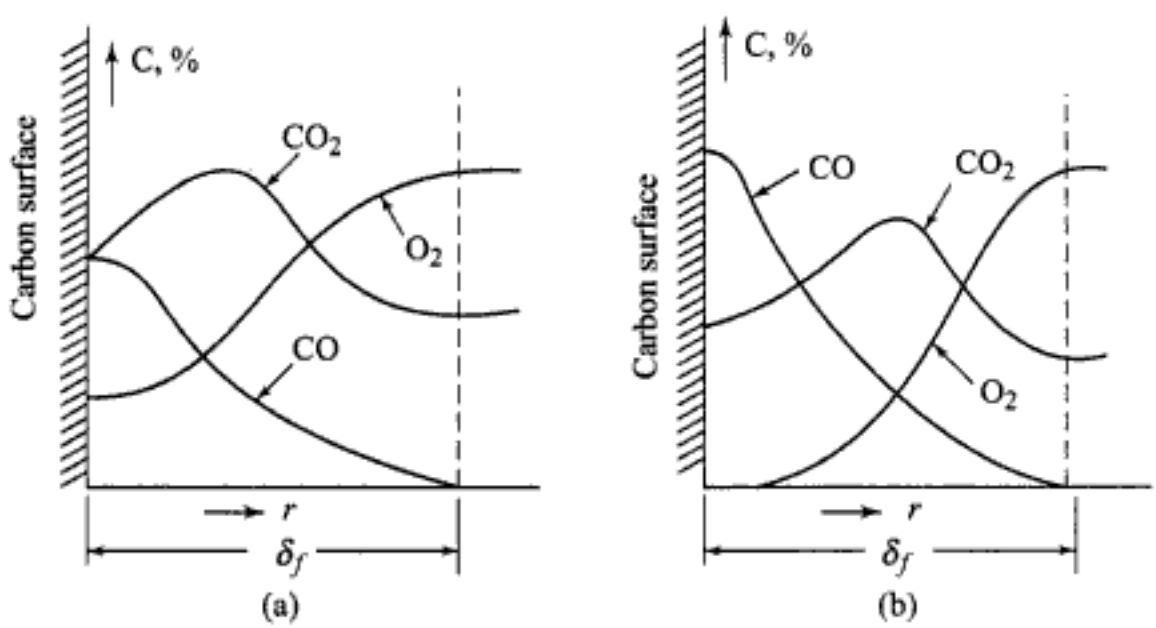

Fig. 5.4 *Variation of concentration of gases near the burning carbon surface (a) at moderate temperature, (b) at high temperature; δ_f = thickness of boundary layer of gases*

5.3 KINETIC AND DIFFUSION CONTROL

The rate of combustion of carbon particles is determined not only by the rate of chemical reaction on the carbon surface but also by the rate of oxygen supply to the reaction zone by molecular and turbulent diffusion. Ash is distributed uniformly throughout the coal substance. With the progress of combustion the reaction surface of the coke particle moves inward leaving behind a layer of porous inert ash, through which also oxygen diffuses for reaction (Fig. 5.5). The

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ash layer will be assumed to shed off the particles due to turbulence. The burning particle is surrounded by a boundary layer or gas film across which there is a gradient of oxygen gas concentration (or partial pressure) and in which CO is oxidized to CO_2 while diffusing out from the reacting surface. Outside this gas film, in the main flow, there is no gradient in concentration of any gas or temperature due to turbulent mixing. It may be assumed that the oxygen concentration in the bulk flow, c_b, is equal to the oxygen concentration at the film surface, c_f.

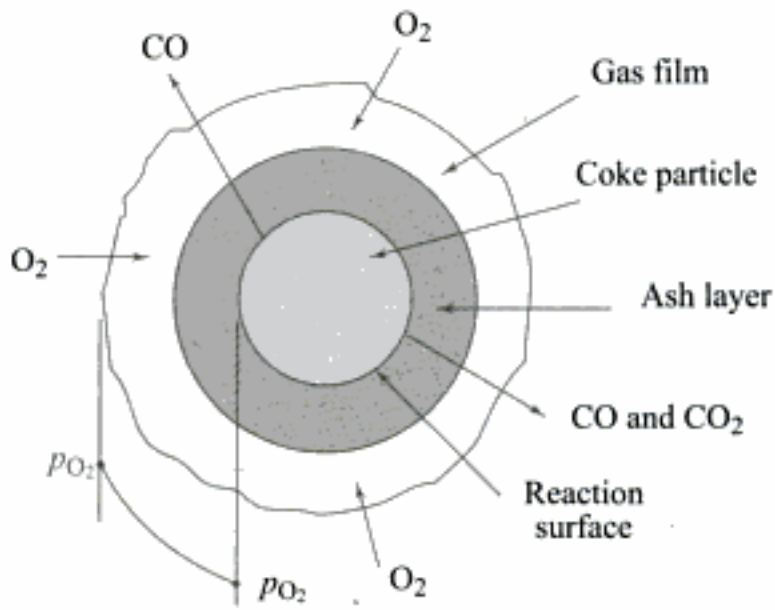

Fig. 5.5 *Burning of a coke particle*

The rate of diffusion of oxygen per unit surface area of particle is given by Fick's law:

$$r^d_s = k_g (c_b - c_s) \tag{5.9}$$

where $k_g = D/\delta$ is the mass transfer coefficient, m/s, D is diffusion coefficient, m^2/s, and δ is the boundary layer thickness, m.

The diffusion rate of oxygen will be maximum when $c_s = 0$,

$$\left(r^d_s\right)_{max} = k_g c_b \tag{5.10}$$

Let us assume that all the oxygen supplied by diffusion reacts on the particle surface. Then the rate of reaction is given by the Eq. (5.5).

$$r^r_s = kc_s \tag{5.11}$$

The maximum reaction rate will occur at $c_s = c_b$.

$$\left(r^r_s\right)_{max} = k\, c_b \tag{5.12}$$

Now, the oxygen consumption in the surface reaction must be equal to the oxygen supply through the boundary film, i.e. $r^d_s = r^r_s = r_s$. From Eqs (5.9) and (5.11).

$$r^d_s = r_s = k_g\left(c_b - \frac{r^r_s}{k}\right)$$

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or
$$r_s = \frac{k_g k}{k_g + k} c_b = \frac{1}{\frac{1}{k} + \frac{1}{k_g}} c_b \tag{5.13}$$

Temperature has only a weak effect on mass transfer by diffusion ($k_g \propto T^{0.5}$), but it considerably influences the surface reaction rate which increases exponentially with temperature ($k \propto e^{-E/RT}$). The nature of variation of the maximum rates of oxygen diffusion to the surface and oxygen consumption by that surface (Fig. 5.6) suggests two principal regions of combustion of coke particles. At temperatures less than 1000 °C, the surface reaction proceeds at a slow rate and oxygen consumption is only a small fraction of the quantity of oxygen supplied to the surface, i.e. $k << k_g$ and $c_b = c_s$. Then Eq. (5.13) reduces to

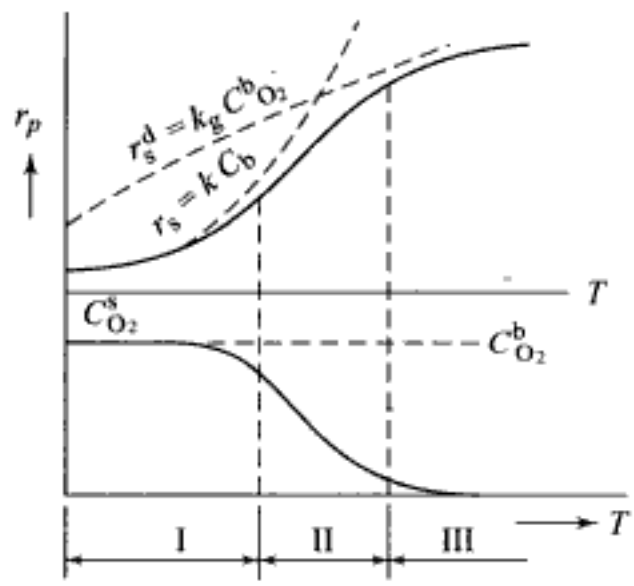

Fig. 5.6 *Variation of oxygen concentration near the burning coke particle surface. I = kinetic region, II = transition zone, III = diffusion region of burning*

$$r_s = k\, c_s \tag{5.14}$$

Thus, the total reaction rate is controlled by the kinetics of the chemical reaction on the surface. This temperature region (1) of reactions is called the *kinetic combustion* zone, and the reaction is said to be *kinetically controlled* (which depends on the temperature of the reaction surface).

At high temperatures, above 1400 °C, the rate constant of the reaction on the surface increases rapidly and exceeds the maximum rate of oxygen supplied to the surface, which varies only slightly with temperature. Then $k >> k_g$ and from Eq. (5.13),

$$r_s = k_g\, c_b \tag{5.15}$$

In this zone, the reaction rate varies slowly inspite of increasing temperature. Oxygen supplied to the surface by diffusion reacts instantaneously (at high temperature) and its concentration at the surface becomes zero. This temperature region is called the *diffusion combustion zone* (III). With oxygen deficiency at the surface, the reduction of CO_2 to CO occurs at the incandescent

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coke surface, while the diffusing oxygen is completely consumed in the gas film in oxidizing CO to CO_2. In this zone, the rate of combustion increases with the increasing rate of diffusion of gases, i.e. turbulence, and with the decreasing size of solid particles.

At intermediate temperatures (1000–1400 °C) the rate of reaction at the surface commensurates with the rate of oxygen supplied to the surface, and the total rate of the reaction is determined by both the processes. This is called the *transition zone of combustion* (II).

It can thus be inferred that when the reaction surface is incandescent and its temperature is very high, the chemical resistance at the surface is small and the reaction is almost instantaneous. Solid fuel combustion is most often *diffusion controlled*. It depends on the rate at which oxygen diffuses into the reaction surface. If the gas film resistance is reduced by inducing turbulence, the rate of diffusion of oxygen and hence the rate of reaction get enhanced. When the air-fuel mixing is very high, as in fluidized beds, the diffusional resistance becomes negligible and the combustion is then *kinetically controlled* with the combustion rate depending on the surface temperature.

Ignition of any fuel begins relatively at a low temperature with ample oxygen supply and it is essentially in the *kinetic zone*. As the temperature rises, oxygen consumption in the reaction zone rapidly increases and the process passes through the transition zone into the diffusion zone.

The end of combustion is usually well in the *diffusion zone* where the residual oxygen concentration becomes low, thus retarding the combustion of the remaining fuel.

5.4 COMBUSTION EQUIPMENT FOR BURNING COAL

Coal may be fed into the furnace for combustion in lump pieces or in powder form. Combustion of coal may occur in

(a) Fuel bed furnaces (coarse particles)
(b) Pulverized coal furnaces (fine particles)
(c) Cyclone furnaces (crushed particles)
(d) Fluidized bed furnaces (crushed small particles)

5.5 FUEL BED COMBUSTION

A grate is used at furnace bottom to hold a bed of fuel (Fig. 5.1). There are two ways of feeding coal on to the grate:

(a) overfeeding
(b) underfeeding

An overfed fuel bed section receives fresh coal on its top surface. It is characterized by the following distinct zones from top to the bottom (Fig. 5.7).

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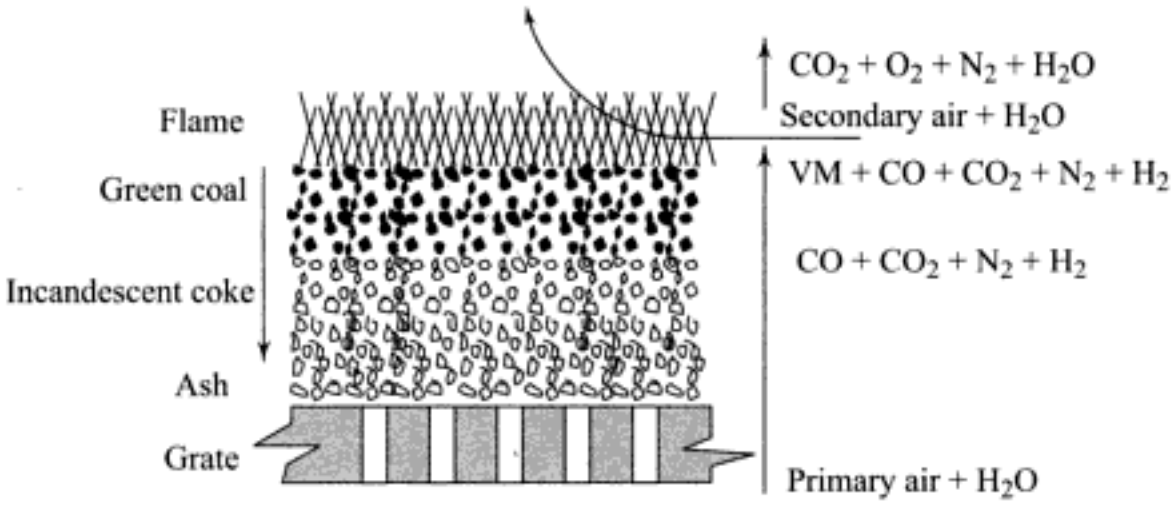

Fig. 5.7 *Overfeed fuel bed section receives fresh coal on top surface; ignition plane lies between green coal and incandescent coke*

1. A layer of fresh or green coal
2. A layer of coal losing moisture—drying zone
3. A coking layer of coal losing its volatile content-distillation zone
4. A layer of incandescent coke, where the fixed carbon is consumed—combustion zone
5. A layer of ash progressively getting cooler.

From zone 4 where heat is released by combustion, heat transfer occurs upward and downward by conduction, while the air flow from below will tend to carry away the heat upward by convection.

Primary air (N_2, O_2, H_2O) gets warmed up as it flows through the ash layer. As it passes through the incandescent coke layer (at about 1200 °C), the reaction taking place initially is $C + O_2 = CO_2$. This is an exothermic reaction and provides the heat release for continuing the combustion process. It continues till all the oxygen is consumed. If the incandescent layer is thick, CO_2 may be partly or fully reduced to CO: $CO_2 + C = 2CO$. A slight water gas reaction may also take place with the moisture from air: $H_2O + C = H_2 + CO$. These are endothermic reactions and may bring down the temperature of the bed and gas stream considerably.

The stream then passes through the distillation zone where the volatile matter is added, and then through the drying zone where the moisture is picked up, and finally emerges above the fuel bed. Its contents are nitrogen, carbon dioxide, carbon monoxide, hydrogen, volatile matter and water vapour.

If the combustion of carbon monoxide, hydrogen and volatile matter is to be completed before the gases leave the furnace, the following have to be provided:

(a) A hot ignition point—in the range 1000–1300 °C. This takes the form of a fire brick-lined arch which stores up the heat and remains at a high temperature.
(b) Sufficient fresh air—secondary or overfire air.
(c) Turbulence—secondary air is fed at right angles to the upflowing gas stream from the fuel bed so as to penetrate the gas mass and secure thorough mixing of air and gases.

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It does not help if secondary air is supplied along with primary air, since more primary air produces only more carbon monoxide.

The fixed carbon in coal leaves the fuel bed in the form of CO_2 or CO, both of which are colourless. But carbon combined with hydrogen in the volatile matter leaves the fuel bed as part of the hydrocarbon gases, the molecules of which break down or disintegrate at high temperature of the furnace releasing some free carbon in the form of finely divided particles suspended in the gas stream. If the coal feeding apparatus and the furnace are not properly designed, or if a well-designed coal burning unit is not properly operated, these fine carbon particles may not burn, in which case these will appear at the chimney top as black smoke, which is a certain indication of inefficient combustion and energy wastage.

In under-feeding, coal is fed from below the grate by a screw-conveyor or ram (Fig. 5.8). Primary air passing through holes in the tuyeres diffuses through spaces in the raw or green coal picking up moisture and then through the distillation zone where volatile matter is added. When the gas stream passes next through the incandescent coke region, the volatile matter breaks up and readily burns with secondary air fed at the top. In overfeeding, burning the volatile matter will be somewhat cooler and so need longer time to ignite and burn.

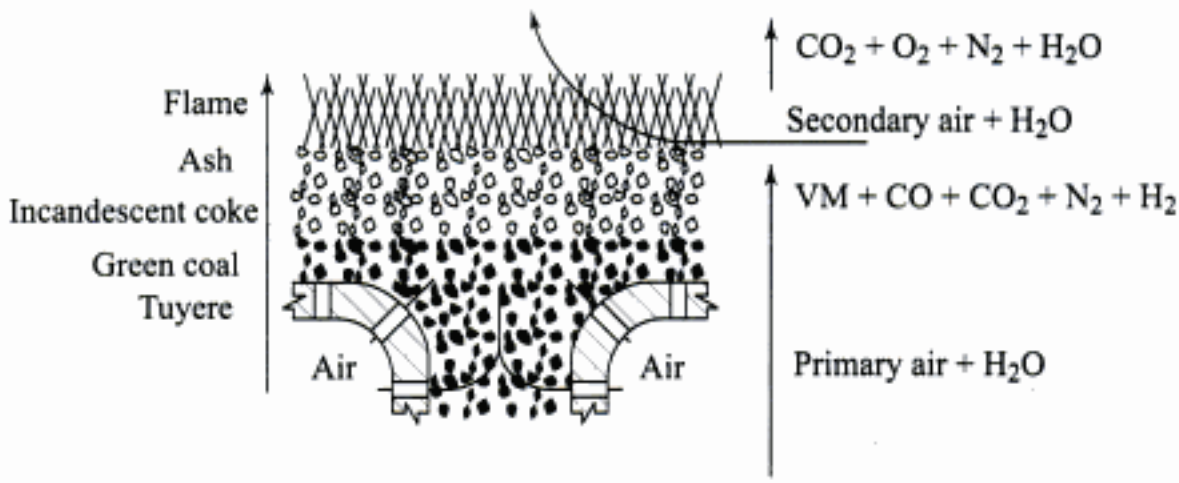

Fig. 5.8 (a) *Underfeed fuel bed section receives fresh coal from retort underneath between two tuyeres*

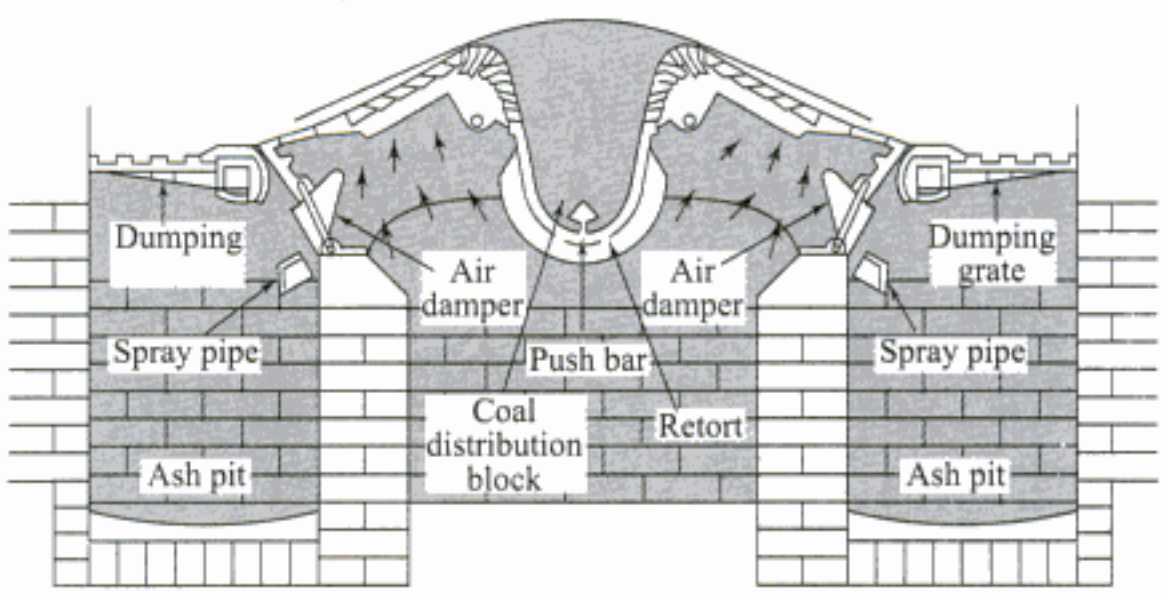

Fig. 5.8 (b) *Single-retort stoker showing contour of fuel bed and ash dumps*

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5.6 MECHANICAL STOKERS

In small boilers, the grate is stationary and coal is fed manually by shovels. But for more uniform operating condition, higher burning rate and greater efficiency, moving grates or stokers are employed. Stokers may be of the following types:

(a) Travelling grate stoker
(b) Chain grate stoker
(c) Spreader stoker
(d) Vibrating grate stoker
(e) Underfeed stoker

5.6.1 Travelling Grate Stoker

The grate surface is made up of a series of cast-iron bars joined together by links to form an endless belt running over two sets of sprocket wheels with a surface as wide as needed (Fig. 5.9). A coal gate at the rear of the coal hopper regulates the depth of the fuel bed. The gate can be raised or lowered as needed. Simultaneous adjustment of grate speed, fuel bed thickness, and air flow controls the burning rate so that nothing but ash remains on the grate by the time it reaches the furnace rear. The ash falls into the ash pit as the grate turns on the rear sproket to make the return trip.

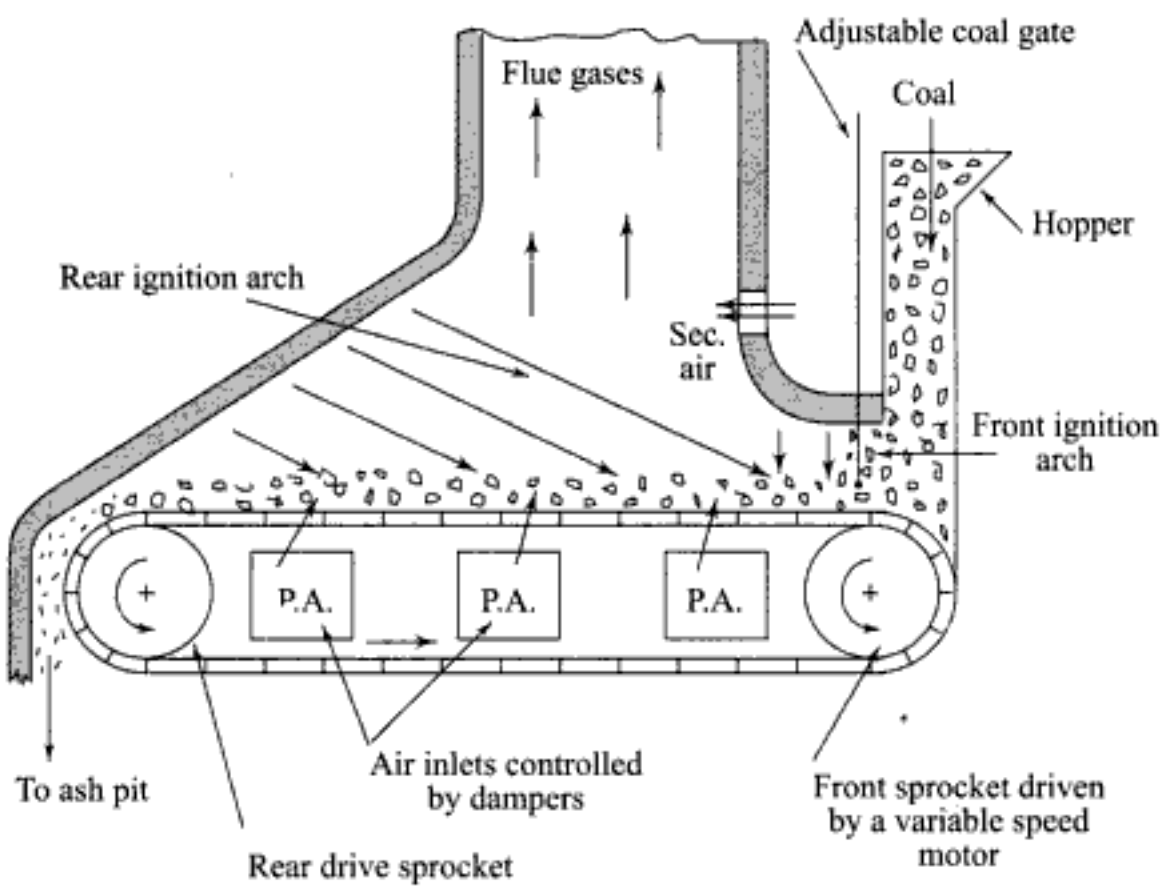

Fig. 5.9 (a) *Travelling grate stoker moving like an endless belt carrying the fuel bed from right to the left*

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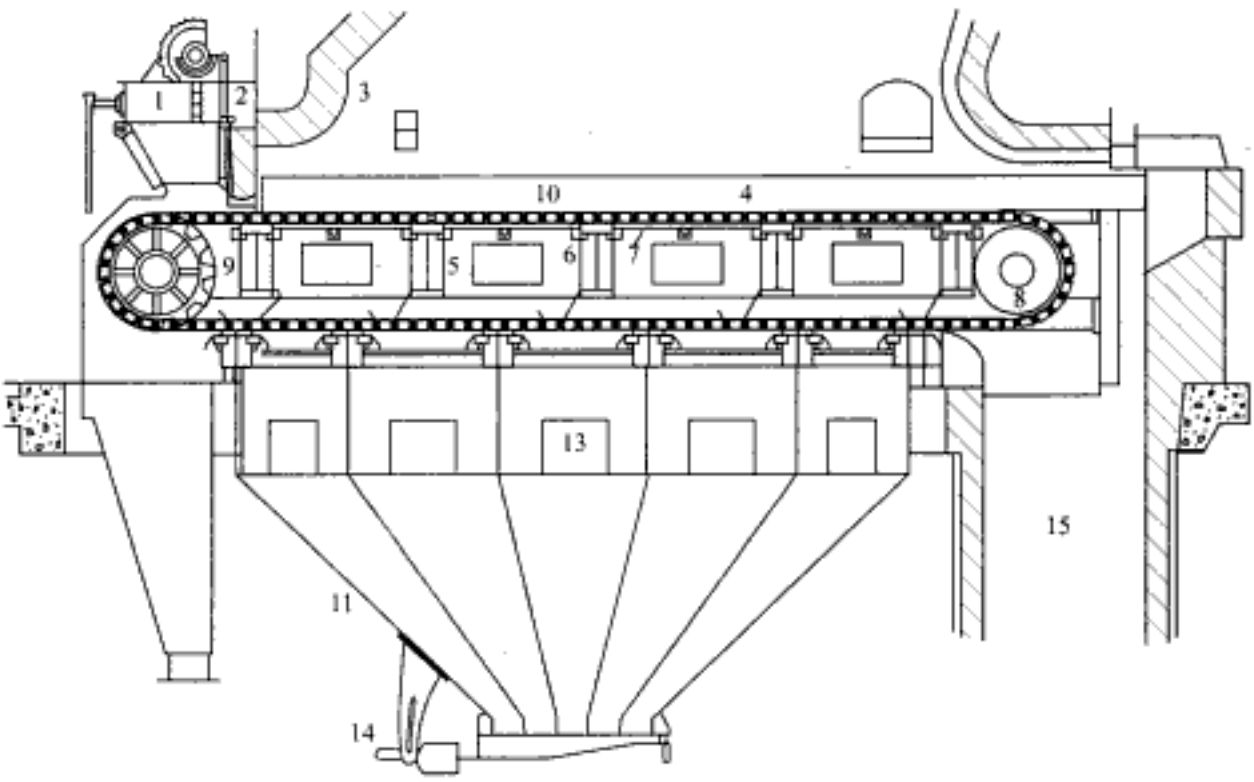
As the raw or green coal on the grate enters the furnace, the surface coal gets ignited from heat of the furnace flame and from radiant heat rays reflected by the ignition arch. The fuel bed becomes thinner toward the furnace rear as the combustible matter burns off. Undergrate air pressures are varied by dampers from front to rear of the stoker to admit gradually reduced quantity of primary air fed by the FD fan. The secondary air aids in mixing the gases and supplies oxygen to complete combustion.



1. Hopper
2. Coal gate
3. Front arch
4. Chain grate
5. Cross-girders
6. Caps
7. Skid bars
8. Rear drum
9. Sprockets
10. Water-wall boxes
11. Hoppers
12. Drag seals
13. Dampers
14. Hopper gate
15. Ash-pit

Fig. 5.9 (b) *A chain grate stoker*

5.6.2 Chain Grate Stoker

It is similar to the travelling grate stoker except that it is made up of a series of CI links connected by bars or pins to form an endless chain, the principle of its operation being the same.

Overfeed stokers are suited for industrial power plants having steady demand. The grate heat release rate should be limited to a maximum of 1340 kW/m^2 of active grate area. A relatively wide range of coals can be burned on overfeed stokers, the size varying from 19 to 38 mm with the amount of fines limited to 50% passing through a 6 mm screen. Non-caking or free-burning coal is preferred. Coal having more than 20% ash is difficult to burn efficiently, incurring considerable unburnt carbon loss.

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5.6.3 Spreader Stoker

Coal from the hopper is fed by a rotating feeder, a drum fitted with short blades on its surface, to the spreader or distributor below, which projects the coal particles in a continuous stream on to the grate holding an ignited fuel bed (Fig. 5.10). The finer particles burn in suspension and the coarser particles are consumed on the grate. The speed of the feeder varies directly with the steam output of the boiler.

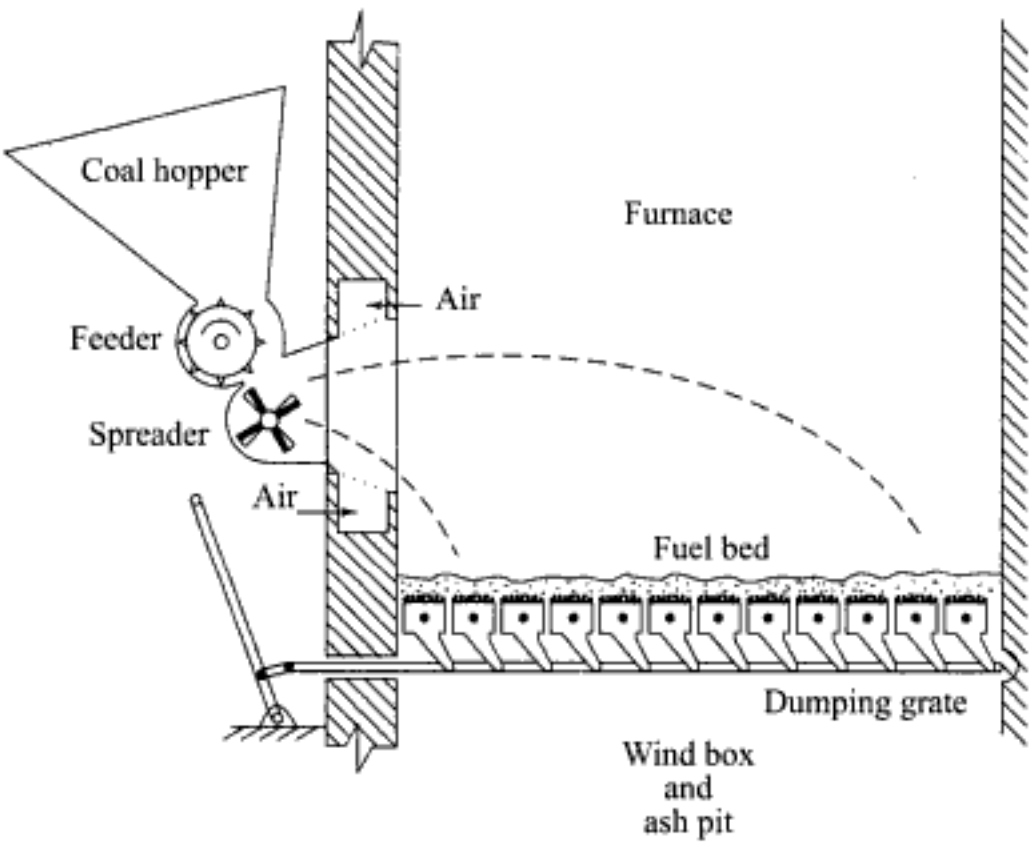

Fig. 5.10 *Spreader stoer hurling coal particles into furnace*

Secondary or overfire air promotes turbulence and completes combustion. High capacity boiler may have a travelling grate in addition to the spreader.

The grate consists of CI bars. Links underneath the grate connect all the bars to a lever. Moving the lever back and forth makes the bars rock about their pivots, which makes the ash fall through to the ash pit below.

The solid fuels burned on spreader stokers include bituminous coal, subbituminous coal, lignite, wood waste from the forest products industry, bagasse from the sugar cane industry, peat from peat bogs, peanut shells, hydrocarbon from agricultural wastes, and refuse-derived fuels (RDF) of municipal solid waste.

It is necessary to size the fuel properly for spreader stokers. Coal should have 95% less than 32 mm. Waste fuels can have top sizes up to 102 mm. Spreader stokers can be applied to a wide range of boiler sizes which can go up to 155 MW for coal and 264 MW for certain waste fuels with steaming rates of 50 kg/s and 75 kg/s respectively.

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5.6.4 Vibrating Grate Stoker

The stoker shakes the fuel bed intermittently, the frequency and amplitude of vibration depending on boiler load. The fuel bed is inclined so that the fuel moves towards the rear of the boiler by gravity with the progress of combustion and then falls into the ash pit. The grate is water-cooled to prevent slagging.

5.6.5 Underfeed Stoker

Underfeed stokers, either single or multiple retort, consist essentially of a trough or troughs into which coal is pushed by rams or screws. Part of the combustion air is introduced into the fuel bed through tuyers. Movement of the fuel particles discourages the formation of large coke masses. Volatile matter is distilled off the coal and burns above the incandenscent fuel bed. Coal forms a thick sloping pile (15 to 25°) over the whole surface of the grate and carbonization proceeds from above downward. The coal particles are pushed forward by the ram till the refuse is discharged to the ashpit (Fig. 5.11). Retorts may be set side by side according to width of furnace, up to 18 retorts being used in a furnace 9.6 m wide. Fuel bed is quite thick, which may be up to 0.6 m at the front of the grate.

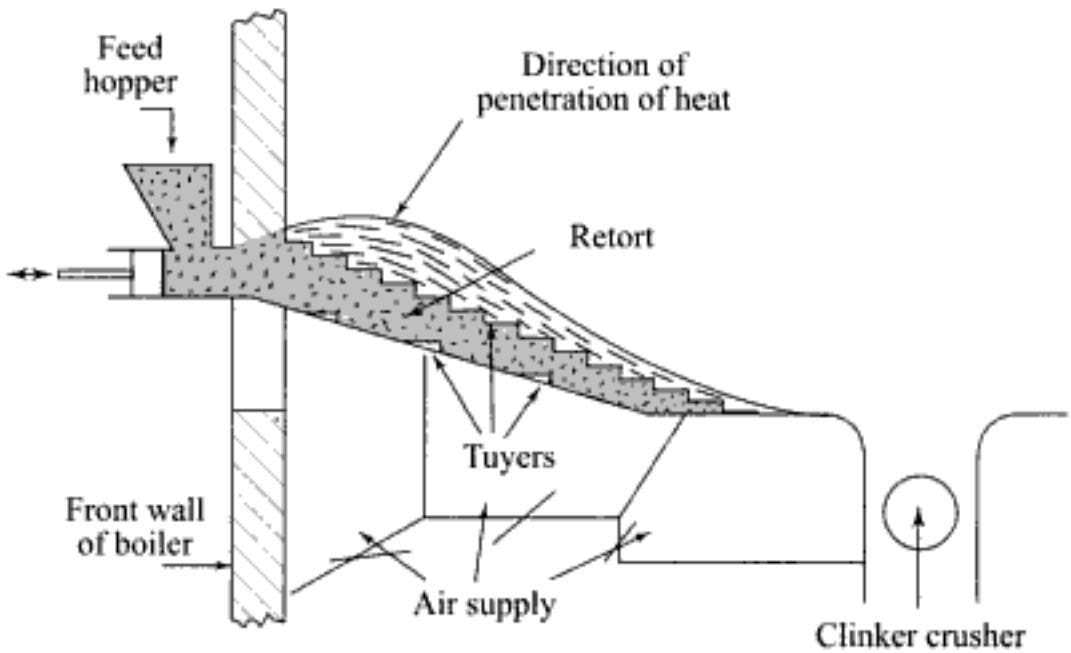

Fig. 5.11 *Principle of retort-type stoker*

In all mechanical stokers forced draught fans are used both for undergrate primary air and overfire secondary air. Zoned primary air admission is used to control excess air. Stoker firing is limited to relatively low capacities (up to 50 kg/s of steam) due to low efficiency and the physical size of the stoker. However, in their size range, stokers remain important in steam generator systems.

Beyond a certain size of boiler, stokers become impracticable and uneconomical due to increased floor area and boiler house cost. Modern high pressure boilers tend to grow vertically upward rather than horizontally, adopting pulverized coal firing through burners.

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5.7 PULVERIZED COAL FIRING SYSTEM

The first commercial application of pulverized coal firing for steam generation was made in the early 1920s. Since then it has become almost universal in central utility stations using coal as fuel. Coal is first ground to dustlike size and powdered coal is then carried in a stream of air to be fed through burners into the furnace. As the entering coal particles get heated in high temperature flames in the furnace, the volatile matter is distilled off and this reduces the coal particles to minute sponge-like masses of fixed carbon and ash. The volatile gases mix with the oxygen of the air, get ignited and burn quickly. Oxygen of the hot air reacts with the carbon surface to release energy. The combustion products form a blanket on carbon particles, which is stripped off by turbulent mixing of these particles and air. Proper burning of fuel needs the supply of correct proportion of air, mixing of fuel and air, high temperature, and adequate time to complete combustion reactions. The ash resulting from combustion (i) partly falls to the furnace bottom and (ii) the rest is carried in gas stream as *fly ash* to flue-gas outlet, or (iii) is deposited on the boiler heating surfaces. Modern central station boiler furnaces have water-cooled walls that form part of the heat-absorbing surfaces in steam generation.

To burn pulverized coal successfully, the following two conditions must be satisfied:

1. Large quantities of very fine particles of coal, usually those that would pass a 200-mesh sieve must exist to ensure ready ignition because of their large surface-to-volume ratio.
2. Minimum quantity of coarser particles should be present since these coarser particles cause slagging and reduce combustion efficiency.

A typical screen analysis of a high volatile bituminous coal sample, pulverized to 80% 200 mesh (0.074 mm opening).

99.5%–50 mesh
96.5%–100 mesh
80.0%–200 mesh

This represents a surface area of roughly 150,000 mm^2/g with 97% of the surface in the – 200 mesh portion. By overgrinding and poor classification, it would be possible to have a sample of the following analysis:

95%–50 mesh
90%–100 mesh
80%–200 mesh

This is not a satisfactory grind because of the high percentage retained on the 50 mesh, even though the surface area remains the same. Thus classification plays a major role in matching the particle size to the reactivity of the fuel.

Greater surface area per unit mass of coal allows faster combustion reactions because more carbon becomes exposed to heat and oxygen. This reduces the excess air needed to complete combustion. This also reduces the dry exhaust loss through chimney and raises the steam generator efficiency. However, the extra cost of the pulverizing equipment and grinding energy partly offset these advantages.

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Advantages of pulverized coal firing

1. Low excess air requirement
2. Less fan power
3. Ability to use highly preheated air reducing exhaust losses
4. Higher boiler efficiency
5. Ability to burn a wide variety of coals
6. Fast response to load changes
7. Ease of burning alternately with, or in combination with gas and oil
8. Ability to release large amounts of heat enabling it to generate about 2000 t/h of steam or more in one boiler
9. Ability to use fly ash for making bricks etc.
10. Less pressure losses and draught need.

Disadvantages

1. Added investment in coal preparation unit
2. Added power needed for pulverizing coal
3. Investment needed to remove fly ash before ID fan
4. Large volume of furnaces needed to permit desired heat release and to withstand high gas temperature.

However, the advantages far outweigh the disadvantages in large utility central stations, and the net gain has led to the wide use of pulverized coal firing in such systems.

In modern plants the hot air for drying coal in the pulverizer is supplied from the forced draught fan and the air preheater, as shown schematically in Fig. 5.12. Most of the air (about 70%) leaving the air preheater goes directly to the burner windbox on the boiler and is known as secondary air. The remaining air, called the primary air, is used to dry the coal in the pulverizer and convey the powdered coal from the pulverizer to the burners (or storage bins). Burners make the mixing of coal, primary air, and secondary air.

In some instances, the temperature of the air leaving the air preheater is not high enough to dry the coal properly. Then, a primary air heater using steam coils may be used to raise the air temperature, or very hot flue gases from the steam generator may be mixed with the air to raise its temperature. The flue gases must be taken from a point of the steam generator where the concentration of CO_2 is low and must be well diluted with air in the mixing chamber. Otherwise, the presence of the CO_2 will be detrimental to the combustion process.

An advantage of coal pulverizers over stokers is the ability of pulverizers to use hot air at temperatures ranging from 260 °C to 420 °C, depending on the kind of coal. These high air temperatures promote good combustion and permit lower flue gas exit temperatures.

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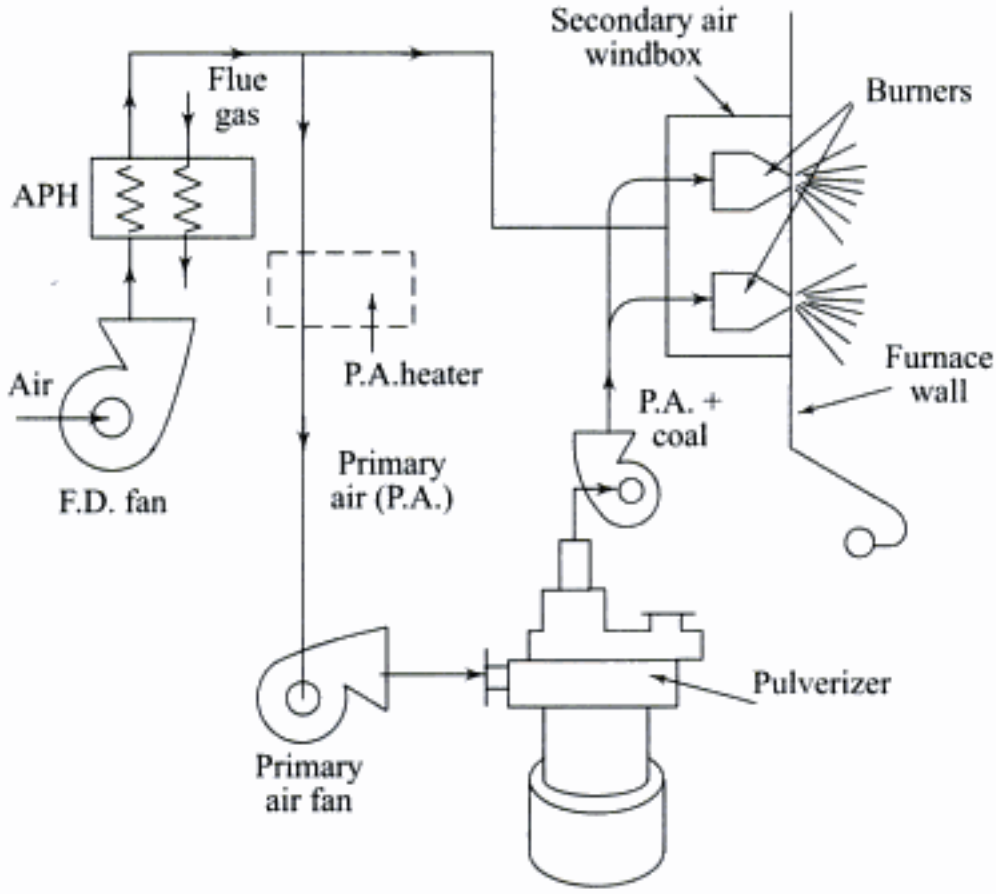

Fig. 5.12 *Primary and secondary air in a pulverizer coal firing system*

In order to maintain the desired air temperature leaving the pulverizer with variation in coal moisture and varying air temperature with boiler load (Table 5.1), tempering air from the boiler room or forced-draught fan exit is mixed with primary air at the pulverizer inlet to maintain the necessary outlet temperature. Again, if the temperature of the hot primary air is very high then in addition to moisture, some volatile matter may also be distilled off from the coal releasing heat, which may lead to a fire hazard in a cumulative effect. Tempering air may be used to offset this tendency.

Table 5.1 *Allowable pulverizer outlet temperatures*

System	Storage (°C)	Direct (°C)	Semidirect (°C)
1. High-rank high-volatile bituminous	54	77	77
2. Low-rank high volatile bituminous	54	71	71
3. High-rank low volatile bituminous	57	82	82
4. Lignite	43	43–60	49–60
5. Anthracite	93	95	95

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5.7.1 Crushers

Coal carried in conveyor belt passes through a magnetic separator to remove tramp iron pieces. Then coal is taken to crusher for being broken into the required feed size which in pulverizing mills is 30 mm or below. Two types of crushers are common: the ring crushers and the hammer mill (Figs 5.13 and 5.14). The coal is fed at the top and is crushed by the action of rings that pivot off centre on a rotor or by swinging hammers attached to it. Adjustable screen bars determine the maximum size of the discharged coal.

There is another type of crusher, called the Bradford breaker (Fig. 5.15) which is used for large capacity work. It consists of a large cylinder made up of perforated steel plates to which lifting shelves are attached on the inside. The cylinder rotating slowly at about 20 rpm receives coal at one end. The shelves lift the coal up and then the coal drops down by gravity.

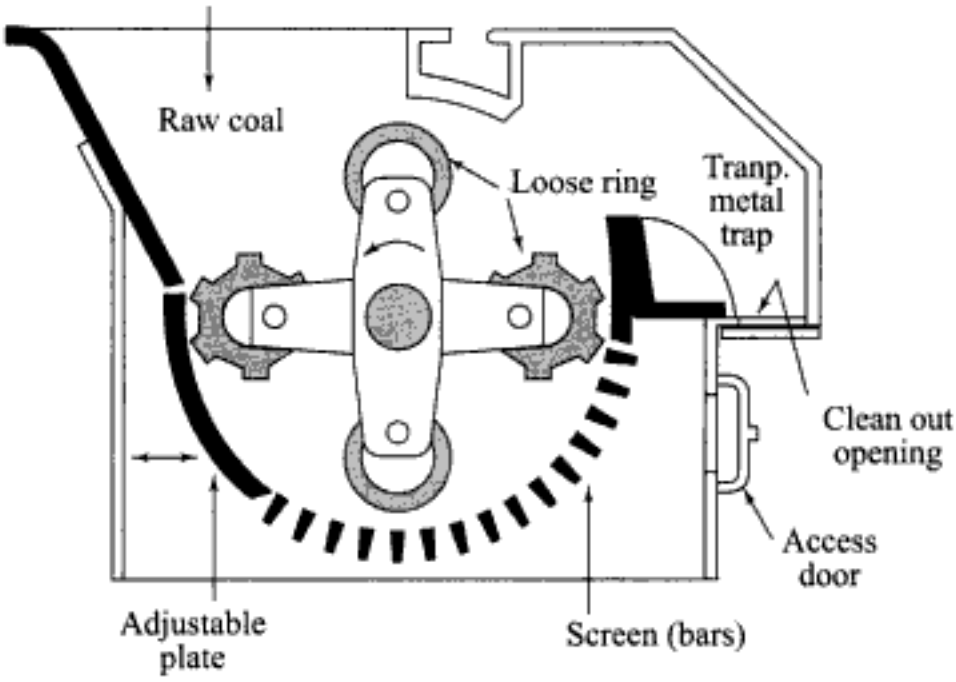

Fig. 5.13 *A ring-type coal crusher*

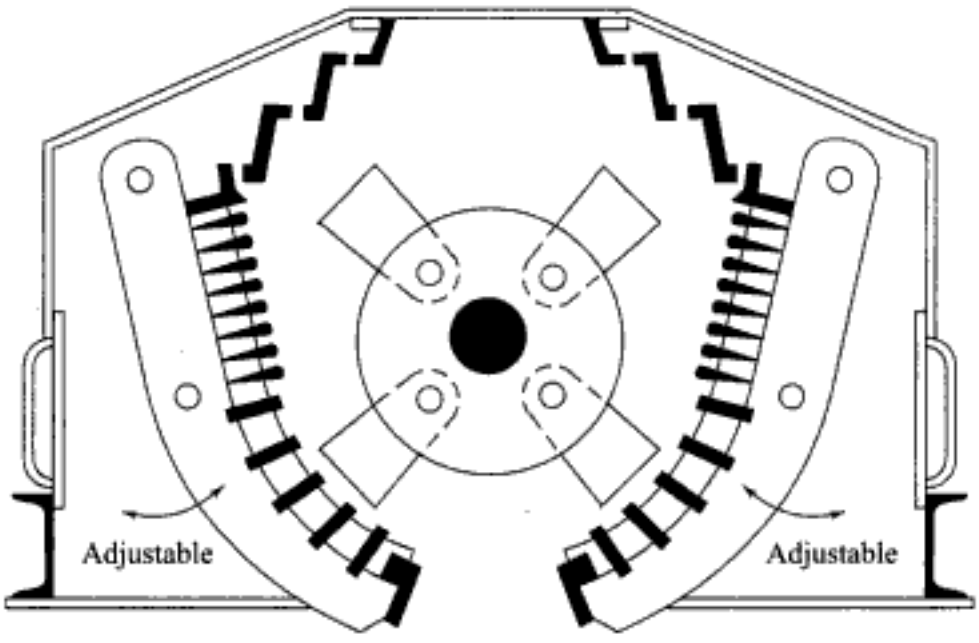

Fig. 5.14 *A hammer-mill coal crusher*

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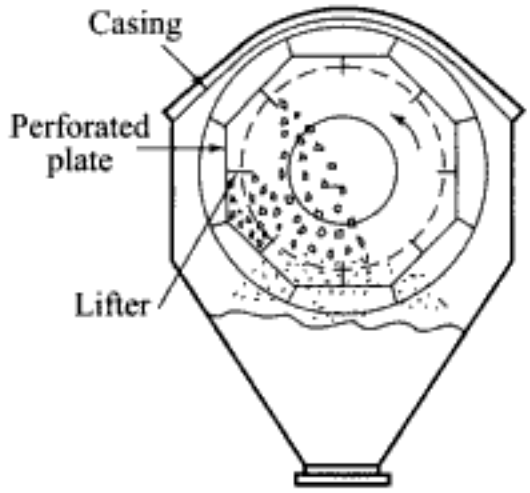

Fig. 5.15 *A Bradford breaker*

Repeated lifting and dropping crushes the coal until its size becomes small enough to pass through the performation (roughly 32 to 38 mm).

5.7.2 Pulverizers

There are three stages in the pulverizing process of coal: (a) feeding, (b) drying, and (c) grinding. The feeding system controls the fuel feed rate according to the boiler demand and the required air rate (primary air) for drying, and then transporting the pulverized fuel and primary air stream to the burner. Since coals have varying quantities of moisture and in order that lower-rank coals can be used, dryers are an integral part of the pulverizing equipment. Part of the hot air from the air preheater (primary air) is forced into the pulverizer at about 350 °C by the primary air fan. There it is mixed with the coal as it is being circulated and ground. The heart of the equipment is, however, the pulverizer or the tube mill. Grinding is performed by impact, attrition, crushing, or combinations of these. Pulverizers, commonly used, are classified by speed: (a) low speed (below 75 rpm), the ball tube mill; (b) medium speed (75 to 225 rpm), the ball-and-race mill and bowl mill; and (c) high speed (above 225 rpm), the impact or hammer mill.

The ball tube mill is a hollow horizontal cylinder with conical ends and wear-resistant liners revolving slowly at about 20 rpm with 20–35% of its volume being filled with forged steel balls of mixed size 30–60 mm (Fig. 5.16).

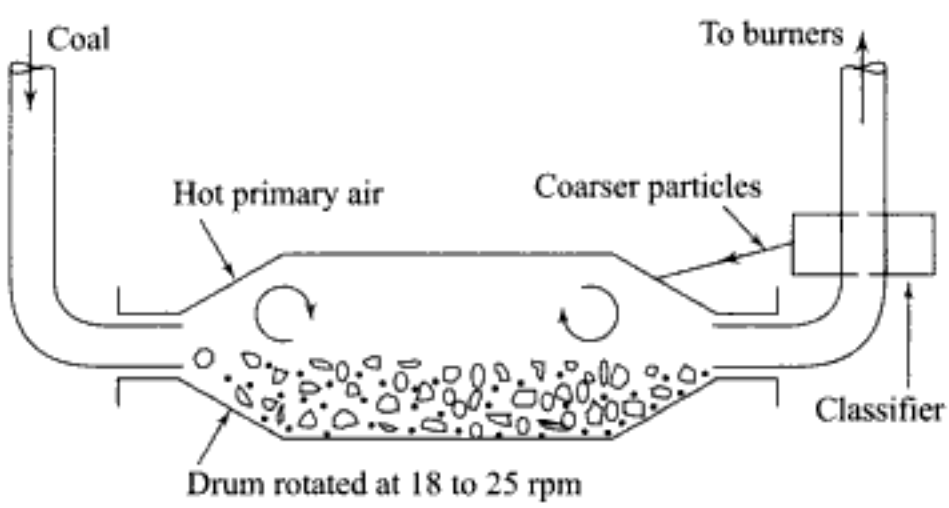

Fig. 5.16 *Ball mill*

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Coal is pulverized by attrition and impact as the balls and coal rise up and fall down with cylinder rotation. Primary air is blown over the charge to carry the pulverized coal to classifiers, which feed back the coarser particles for regrinding. The ball mill is reliable and requires low maintenance, but it is bulky and heavy in construction, consumes more power (about 22 kWh per tonne) and because of poor air circulation works less efficiently with wet coals.

The ball-and-race mill (Fig. 5.17) crushes coal between two moving surfaces, balls and races, by attrition. An upper stationary race and a lower rotating race, driven by a worm and gear, hold the balls between them. Coal enters the mill through the rotating table feeder to fall on the inner races. The moving balls and races catch coal between them to crush it to a powder. Springs hold the upper stationary race and adjust the force needed for crushing. Hot air from F.D. fan enters the mill and carries the coal dust to the classifier above. Mill, feeder and fan need about 14 kWh per tonne of coal pulverized.

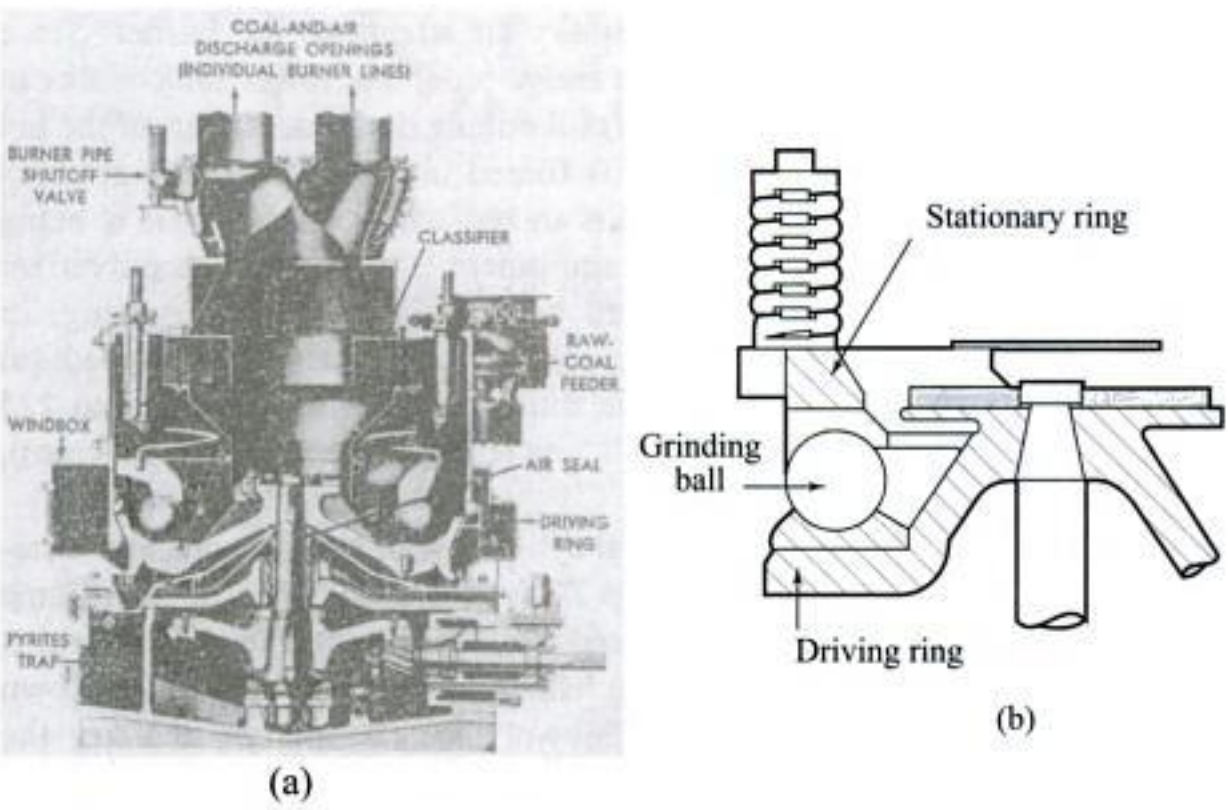

Fig. 5.17 *Ball-race mill (a) B and W type, (b) Principle of working*

The bowl mill is widely used for grinding coal. Coal is fed to a rotating bowl and thrown outward by centrifugal force to the rotating ring where it is pulverized by rolling under spring-loaded rollers (Fig. 5.18). Air sweeps by the bowl and removes the product to the classifier, from which large particles are returned for further grinding. This is an intermediate speed pulverizer (vertical

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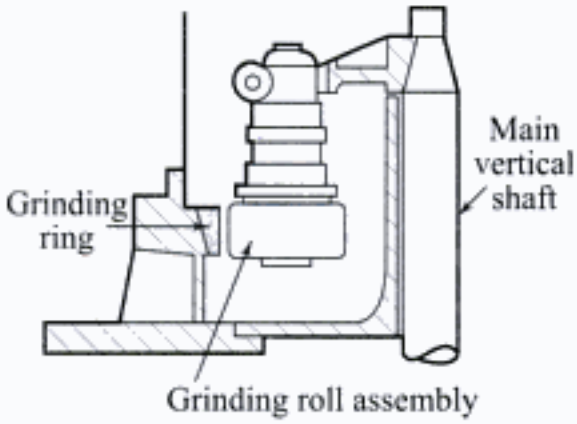
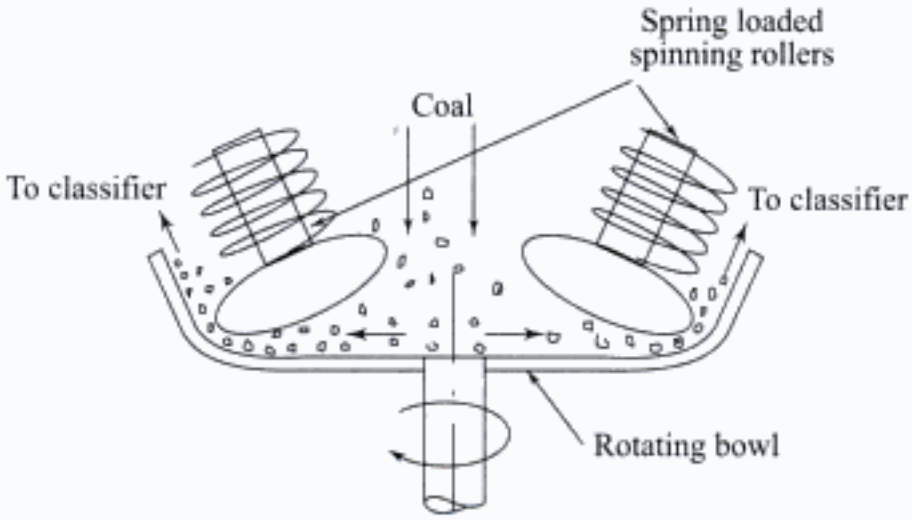


(a)



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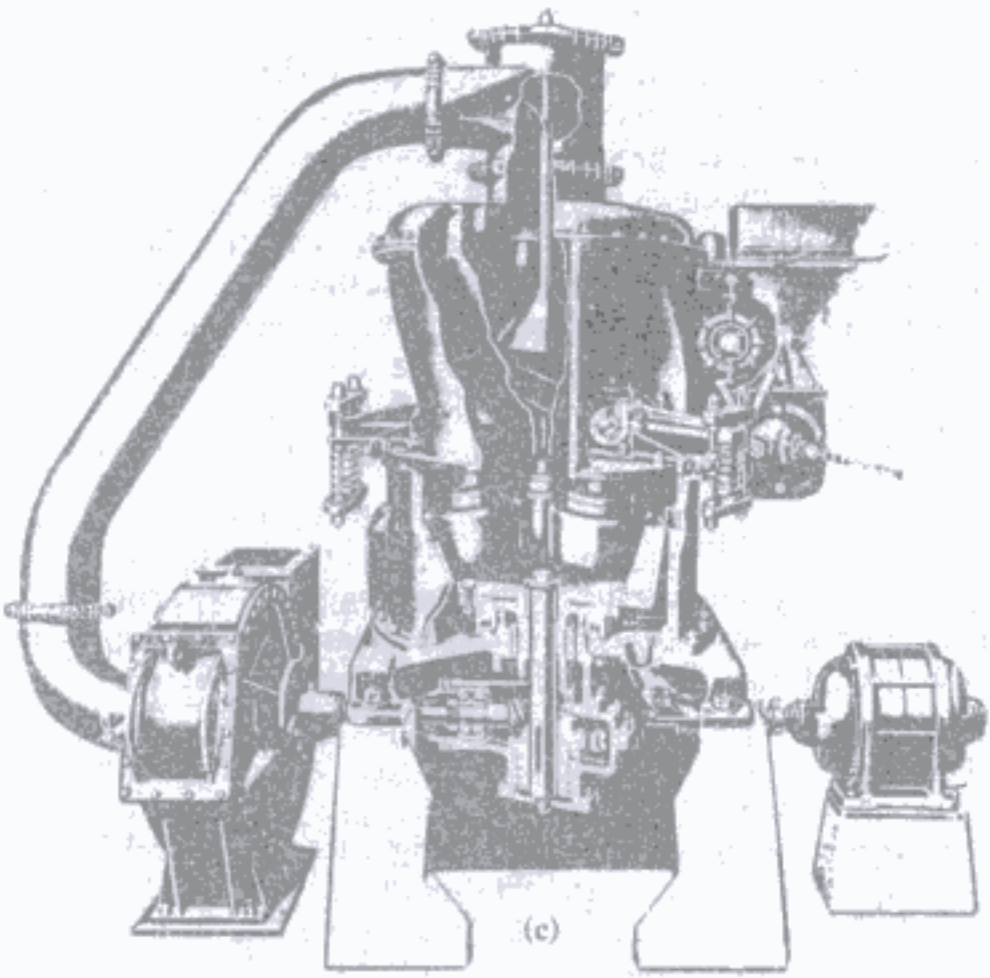

(c)

Fig. 5.18 (a) *Roller-type ring-rong mill, (b) Bowl mill, (c) Reymond bowl mill*

shaft speed about 65–100 rpm) which, with its quiet, vibrationless operation, combines satisfactorily the desirable characteristics of low power consumption and low maintenance cost.

High speed pulverizers use hammer beaters that revolve in a chamber with wear-resistant liners. They are mostly used with low-rank coals with high moisture content and often use flue gas for drying.

5.7.3 Performance of Pulverizer

The power consumption of the pulverizer should be low to minimize operating costs. The main factor of performance depends on (a) the grindability of the coal, (b) the surface moisture on the coal, and (c) the fineness of the grind needed.

Hardgrove index is most often used to measure the grindability of a coal, Properly prepared and measured coal samples are ground in a small laboratory mill for 60 revolutions and then the amount of fines that passes through a 200-mesh sieve indicates the Hardgrove index, which normally varies from less than 26 to over 112. A low index number designates coals hard and difficult to grind, while a high index applies to coals soft and easy to grind.

The fineness of the powdered coal needed for successful burning depends largely on the ratio of volatile matter (VM) to fixed carbon. Since the carbon burns slower than the volatile matter, low-VM coals must be ground finer to expose more surface to permit burning more coal in a given residence time in the furnace. Thus, for anthracite (low VM) about 85% of the powdered coal should pass through a 200-mesh sieve, with less than 2% being retained on a 50-mesh screen, whereas only about 60% need pass through a 200-mesh sieve for a high-VM coal like lignite. The finer the grind, the greater the power input needed. Overgrinding is expensive in energy and maintenance incurred by extra wear. Moisture makes fine coal particles adhere to each other, defeating the purpose of grinding to expose more surface. Heated primary air evaporates this moisture in the mill and carries the vapour to the furnace with the coal.

5.7.4 Feeding Systems

A pulverized coal system comprises pulverizing, delivery, and burning equipment. It must be capable of continuous operation as well as rapid change as required by load demands. There are three main feeding systems:

1. Bin or storage system
2. Direct firing system
3. Semidirect firing system

Indirect firing or storage system In the early pulverized coal fired installations, pulverizers did not discharge directly to the furnace, and were often housed in a separate hall outside the boiler house. The coal was usually dried externally, then fed to the pulverizers, and the pulverized product conveyed to bunkers above the pulverized fuel feeders through which it was fed to the burners.

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This direct or storage system has the advantage that the coal–feed and coal-air ratios to individual burners are under the direct control of the operator, and less mill capacity is needed for a given load since the mills always operate at full load. The mills may not be operated during peak load periods if auxiliary power saving is important, depending on the load factor and the pulverizer size selection.

Among its disadvantages are duplication of bunkers, conveying and feeding equipment and so on, which necessitate large capital expenditure, higher auxiliary power requirement, space for cyclone, disposal of dusty air from mill vents and safeguarding against fire hazard.

Direct firing system Indirect firing or storage system has mostly been replaced by direct firing system in modern power plants. It continuously processes the coal from the storage receiver bunker through a feeder, pulverizer,

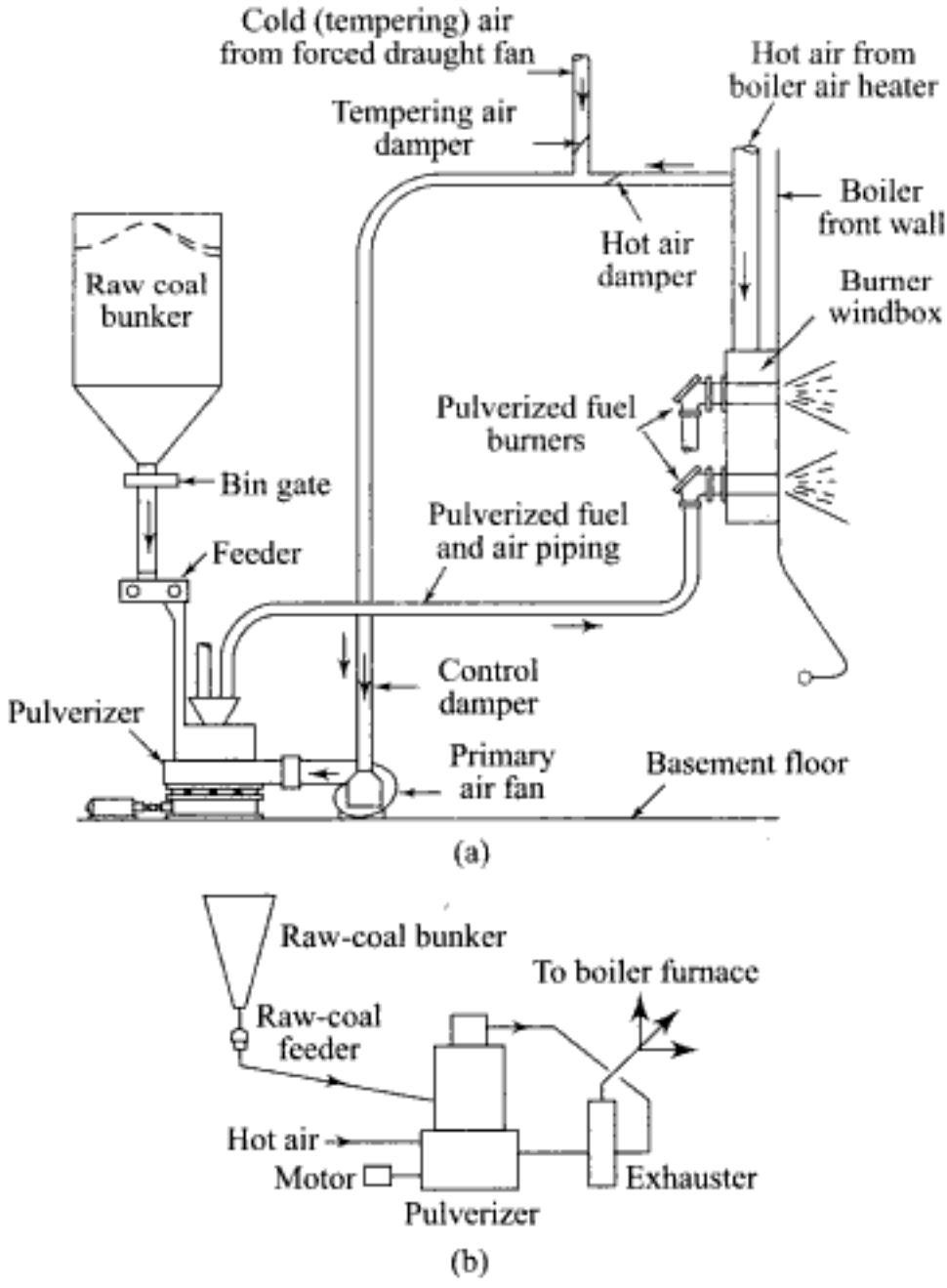

Fig. 5.19 *Pulverized coal direct firing system*

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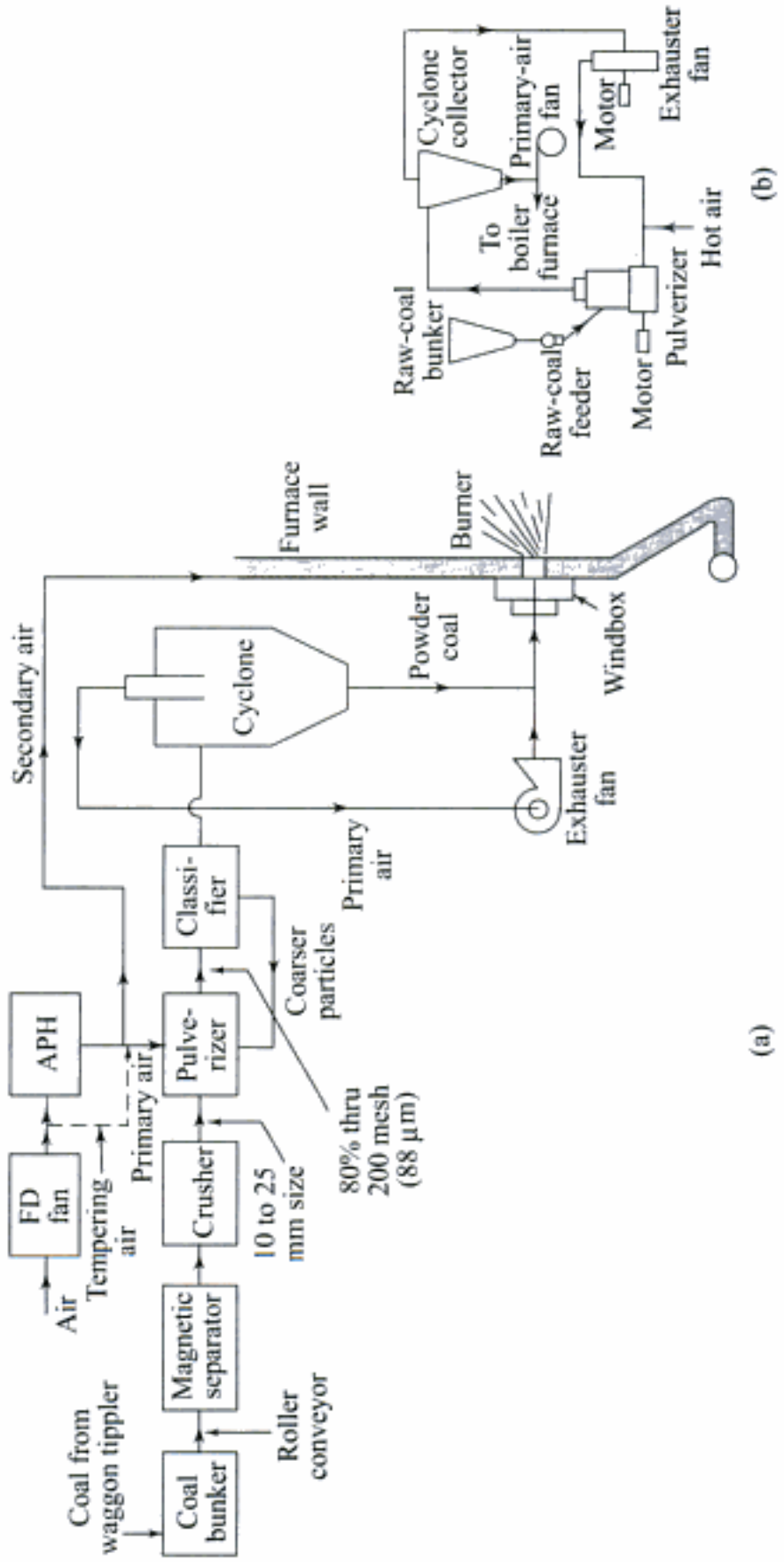

Fig. 5.20 *Semidirect firing sysstem of pulverized coal*

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and primary air fan to the furnace burners (Fig. 5.19). Fuel flow is suited to load demand by a combination of controls on the feeder and on the primary air fan in order to give air-fuel ratios suitable for various steam-generator loads. Large steam generators are provided with more than one pulverizer system, each feeding a number of burners, so that a wide control range is possible by varying the number of pulverizers and the load on each. This system has greater simplicity and hence greater safety, lower space requirements, lower capital and operating costs, and greater plant cleanliness.

Semidirect firing system A cyclone separator between the pulverizer and furnace separates the conveying medium from the coal. The hot primary air separated in the cyclone is used by the exhauster or primary air fan to push the coal particles, falling by gravity from the cyclone, through the burners into the furnace (Fig.5.20).

5.7.5 Pulverized Coal Fired Furnaces

Pulverized coal burns in suspension in the furnace space. Heat released by combustion is transferred to the water wall tubes around the furnace by radiation. The combustion gases leave the furnace at a safe temperature which will not cause clinkering to the subsequent heating surfaces. Water walls consist of vertical tubes arranged mostly in tangent connected with the boiler drum at the top and headers at the bottom. These tubes receive water from the drum via downcomer tubes and discharge the water–steam mixture to the drum.

A furnace can be characterized geometrically by its linear dimensions: front width a, depth b, and height h_f (Fig. 5.21) which are estimated according to the rated fuel consumption and the thermal, physical and chemical properties of the fuel to be used. The hot furnace gases pass through the cross-sectional area of the furnace ($a_f = ab$) at a rather high speed (7–12 m/s). The heating power of the furnace (kW) is given by

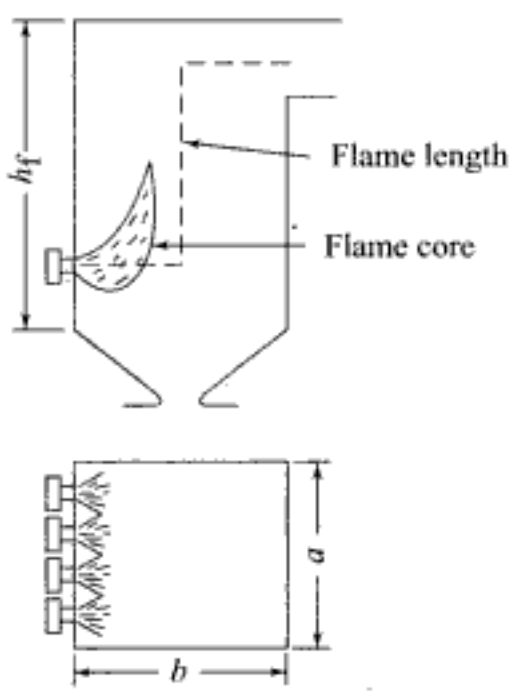

Fig. 5.21 *Main dimensions of a furnace*

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$$Q_f = w_f \times \text{HHV}$$

where w_f is the fuel consumption rate, kg/s and HHV is the heating value, kJ/kg. The heat release rate per unit bed cross-section,

$$q_f = \frac{w_f \times \text{HHV}}{a_f} \text{ kW/m}^2 \tag{5.16}$$

is an important characteristic of a furnace. The highest permissible value of q_f depends on the kind of fuel and the method of burning (arrangement and type of burners), and it may vary from 3500 kW/m^2 for brown coals to 6400 kW/m^2. An increase in q_f results in an increase in w_f and temperatures of gases as well as heat absorption by the furnace walls. The rated value of heat release q_f^r is taken to be somewhat lower than this highest permissible limit. If this is known, then for a certain heating power of a furnace (Q_f) its cross-sectional area can be obtained from

$$a_f = \frac{Q_f}{q_f^r} \tag{5.17}$$

The depth b of a furnace is chosen in the range of 6 m to 10.5 m so that the flame does not impinge the waterwall on the opposite side, which depends on the steaming or fuel rate of the boiler. The depth can be increased to 8–10.5 m when using burners of larger ports in several tiers on the furnace walls.

The front width of the furnace "a" varies from 9.5 m to 31 m depending on the steaming capacity of the boiler, which can be approximately estimated from

$$a = 0.67\sqrt{D} \tag{5.18}$$

where D is the steaming capacity in tonnes per hour.

The height h_f of the furnace is taken to be in the range 15–65 m to ensure complete combustion of fuel along the flame length and allow space for arranging the water tubes on the furnace walls to cool the combustion gases to the specified temperature and to ensure the desired natural circulation of water–steam mixture. If $\overline{V}_g$ is the average velocity of gases in the furnace cross-section, m/s and τ is the time a unit volume of gas is present in the furnace, s, then the furnace height required to ensure complete combustion is given by

$$h_f = \overline{V}_g \cdot \tau \tag{5.19}$$

The furnace is also characterized by the allowable volumetric heat release rate, i.e. energy release per unit volume, kW/m^3, given by

$$q_V = \frac{Q_f}{V_f} = \frac{w_f \text{HHV}}{V_f} \tag{5.20}$$

where V_f is the volume of the furnace ($a \times b \times h_f$). The value of q_V varies from 120 to 210 kW/m^3, the magnitude of which determines the average time the gases are present in the furnace space, τ. As q_V increases, τ decreases (Fig. 5.22).

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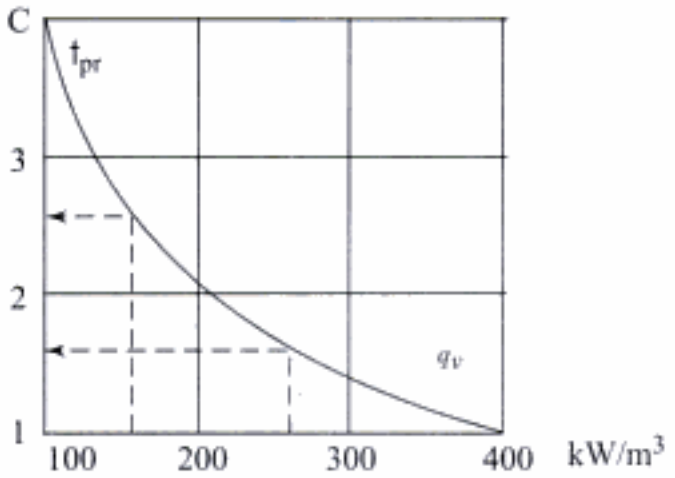

Fig. 5.22 *Relation between heat release rate per unit volume (q_v) and the time of residence in furnace(t)*

The surface areas provided inside the furnace, i.e. the water walls and secondary superheater, absorb heat by radiation, whereas the surface areas of the primary superheater, reheater, economiser and air preheater absorb heat by convection. To obtain the best ratio between the radiant and convective heat transfer in boiler surfaces, it is advisable to maintain the temperature of gases at the furnace exit in the range 1100–1200 °C.

Depending upon the condition of ash coming out from the furnace bottom, pulverized coal furnaces may be of two types:

(a) Dry-bottom furnaces
(b) Wet-bottom or slag-tap furnaces

(a) Dry-bottom furnace In a dry-bottom furnace ash or slag is removed in the solid or dry state. The exit temperature of gases leaving the furnaces must be below the ash fusion temperature. Due to suction of the ID fan, about 85% of the ash in coal, known as "fly ash" flows with the flue gases and is collected in the dust collectors and electrostatic precipitator (ESP) before the ID fan. The remaining ash (about 15%) falls through furnace bottom to the clinker-grinder and then to the hydraulic sluice to which also falls the ash collected in the dust collectors and ESP. The furnace has a hopper at the bottom which has its walls inclined at an angle of 50–60° (angle of repose) so that ash does not stay on the walls and falls by gravity. The minimum gap *b* is about 1.0 to 1.2 m (Fig. 5.23). The dry-bottom hopper intensively cools the furnace gases in the furnace bottom, so that molten slag particles which enter this zone are cooled quickly, solidify and fall along the hopper sides into a slag pit containing water. Granulated slag particles are continuously removed from the bath by a screw mechanism. The water bath also serves as the hydraulic seal preventing the suction of cold air from beneath into the furnace.

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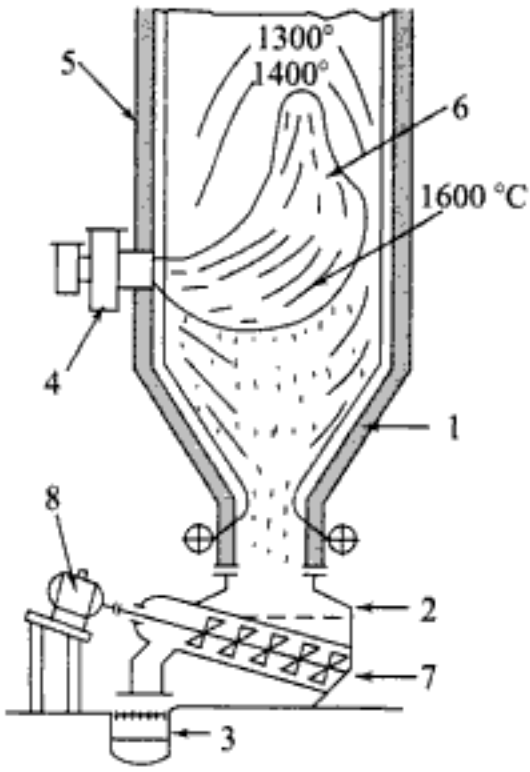

Fig. 5.23 *Dry bottom furnace: 1-dry bottom hopper; 2-slag bath with water; 3-hydraulic ash disposal channel; 4-burner; 5-water walls; 6-flame core; 7-slag removing screw mechanism; 8-electric motor*

(b) Wet-bottom furnace In a wet-bottom furnace, 40–60% of the ash in coal moves with the flue gases due to ID fan suction, and the rest falls through furnace bottom in a continuous molten stream (Fig. 5.24). The temperature of gases at the walls in the lower portion of the furnace should be higher than the melting temperature of ash to increase its fluidity by making the flame core closer to the furnace bottom. The molten ash is discharged into a slag tank containing water, quenched and shattered into a granular form. Water wall tubes at the furnace bottom are often studded and refractory coated to prevent molten ash from eating into the tubes. Coals having high ash content and low ash fusion temperature are only consumed in wet-bottom furnaces.

5.7.6 Pulverized Coal Burners

The efficient utilization of pulverized coal largely depends upon the ability of the burners to produce uniform mixing of coal and air. The air which carries the pulverized coal through the burner is primary air and the remaining secondary air required for complete combustion is admitted separately around the burner or elsewhere in the furnace. The pulverized coal burners should satisfy the following requirements:

1. It should prepare two individual flows, a coal dust air mixture and secondary air, for ignition and active burning in the furnace space, creating a turbulent environment for thorough mixing and sucking hot furnace gases for preheating.

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2. It should not only maintain stable ignition of the fuel-air mix but also control the flame shape and travel in the furnace, since ignition depends on the rate of flame propagation.

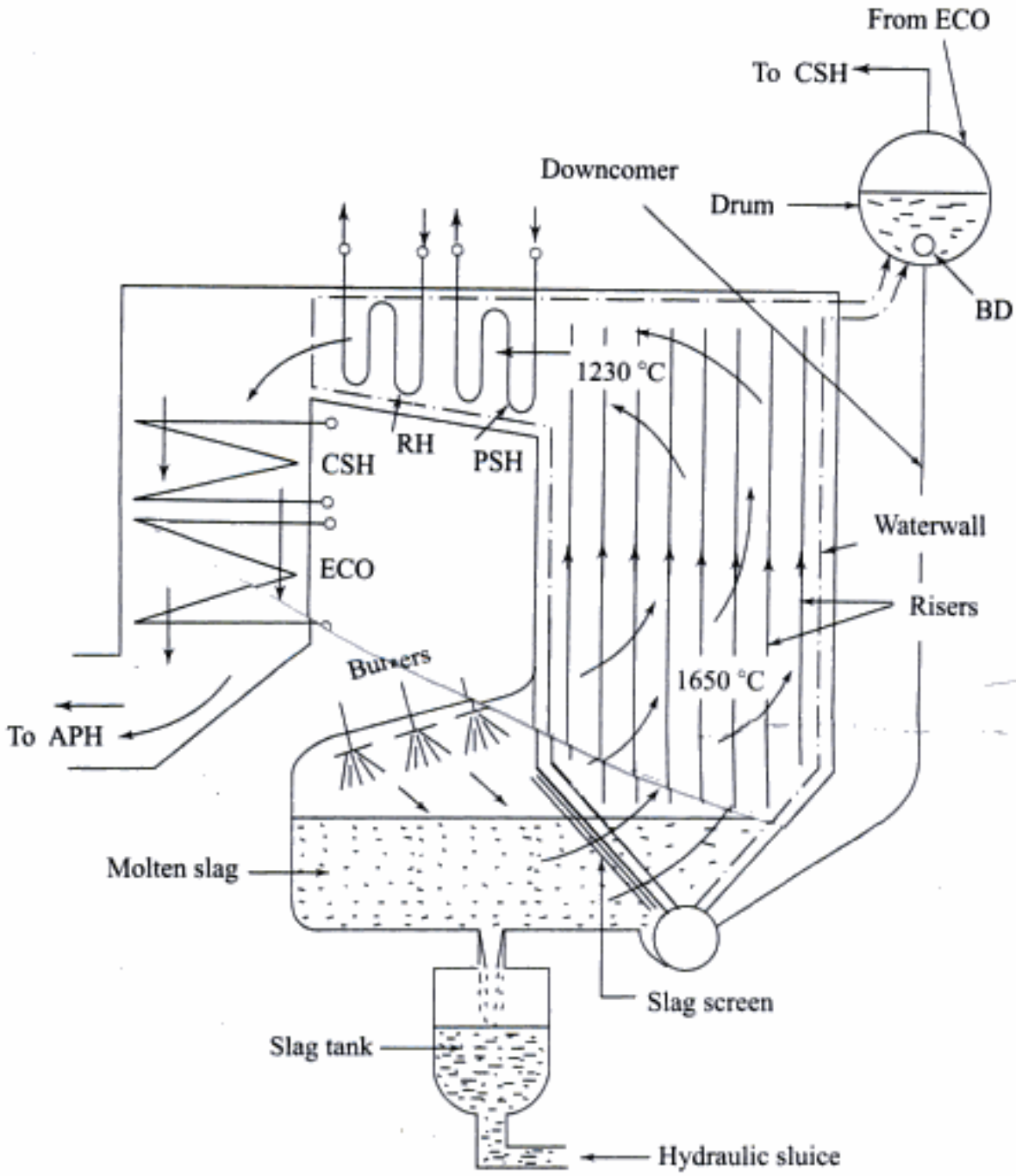

Fig. 5.24 *Wet-bottom furnace*

3. To prevent flash back into the burner, the coal-air mixture must move away from the burner at a rate equal to flame-front travel.
4. Secondary air quantity is to be controlled since too much air supply can cool the mixture and prevent its heating to ignition temperature.

Pulverized coal burners can be divided into two main groups: circular and slot type burners, as shown in Fig. 5.25. The simplest circular burner consists of a cylindrical tube through which the pulverized coal and primary air mixture is blown into the furnace, the secondary air being delivered through a separate tube. The simplest slot-type burner differs from the circular burner only in the shape of its cross-section.

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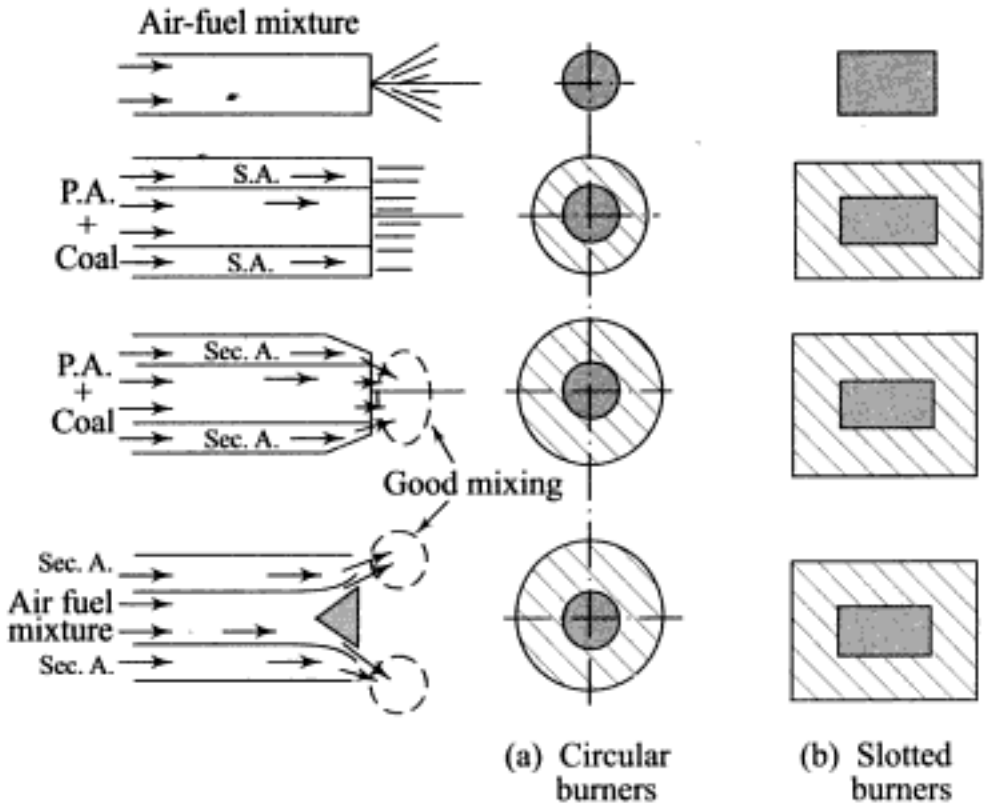

Fig. 5.25 *Pulverized coal burners: (a) circular cross-section, (b) rectangular cross-section*

Depending on the kind of flow of the two streams, air-fuel mixture and secondary air, within the burner and its method of mixing, burners can be of two types:

1. Straight-flow burner (external mixing of air-fuel mixture and secondary air in the furnace space).
2. Turbulent or vortex burner (internal mixing of air-fuel mixture and secondary air before entering the furnace).

In straight-flow burners, air–fuel mixture and secondary air are blown in as parallel jets. They may be either circular or rectangular in cross-section. Their intermixing takes place in the furnace space, not in the burners, by properly arranging the burners on the furnace walls. Efficient combustion is achieved by making the jets from various burners interact with one another in the furnace space. These burners can be either fixed or tilted, which assists combustion control. A typical straight flow burner with tilting nozzle is shown in Fig. 5.26. These burners are usually of a relatively low throughput capacity. In high-capacity boilers, a number of such burners are combined in a burner assembly (Fig. 5.27). They are suitable for high-reactive fuels like high-volatile coals. The velocity of pulverized coal-primary air mixture at the burner outlet is taken as 20–28 m/s to ensure keeping the powdered coal in suspension, and the optimal velocity of secondary air is 35–45 m/s.

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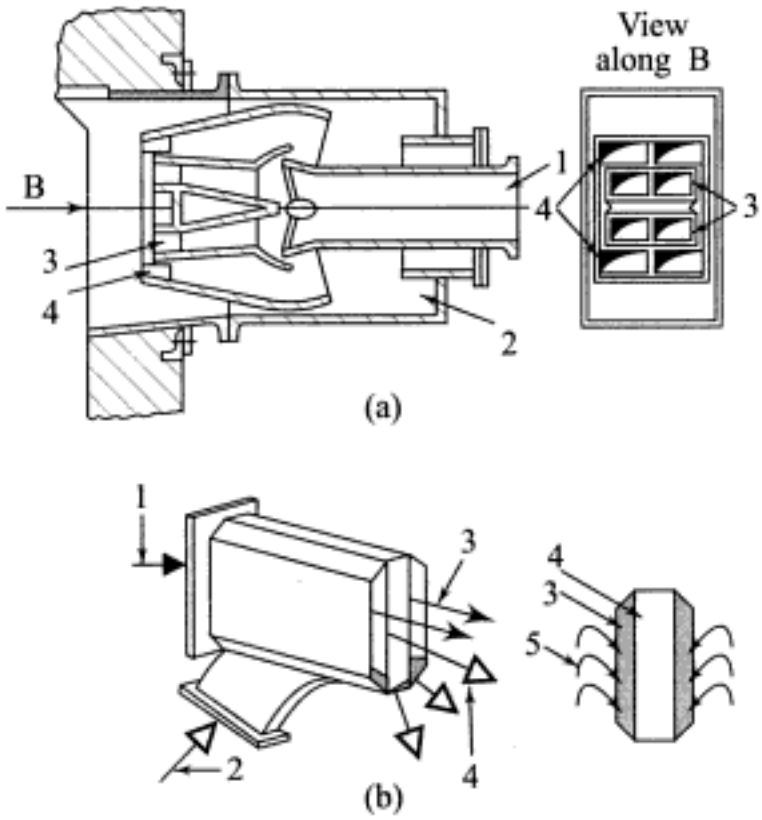

Fig. 5.26 *Straight-flow burners for pulverized coal: (a) with tilting nozzle at the exit of fuel-air mixture; (b) with central channel for hot air; 1- supply of fuel-air mixture; 2-supply of hot air; 3-fuel-air mixture outlet; 4-hot air outlet; 5-suction of furnace gases*

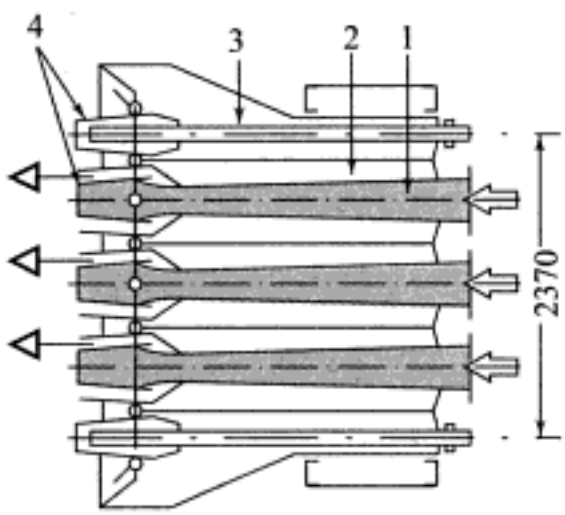

Fig. 5.27 *An assembly of three straight-flow pulverized coal burners; 1-supply of fuel-air mixture to burner; 2-secondary air supply; 3-pipe for mounting the fuel-oil burner with electric gas igniter; 4-tilting air pipes*

In turbulent or vortex burners, a whirling rotary motion is imparted to the coal-air mixture by entering it tangentially to a central nozzle and also to the secondary air that flows through the annular space surrounding the nozzle (Fig. 5.28). This mixes the two streams as they enter the furnace to distribute the coal dust uniformly throughout the total air available for combustion. This is

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what is called a *two-scroll burner*, where two scrolls are provided for whirling the dust air mixture and secondary air. In a straight scroll burner the dust air mixture is supplied through a central straight flow nozzle and spread by a deflector, and the secondary air is whirled in a scroll (Fig. 5.29). There are also vane-type burners in which whirling of the dust-air mixture and the secondary air is effected by axial and tangential vanes. A burner for horizontal firing of coal is also shown.

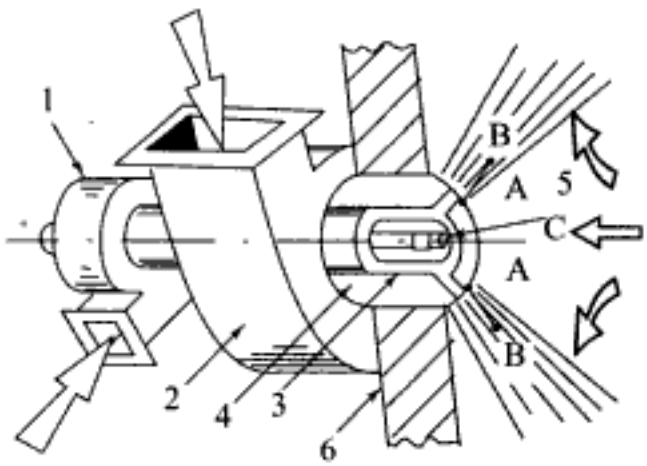

Fig. 5.28 *Two scroll turbulent burner. 1-Scroll for fuel-air mixture; 2-secondary air scroll; 3-annular channel for discharging fuel-air mixture into furnace; 4-ditto, for secon-dary air; 5-main fuel oil burner; 6-furnace lining; AB-boundary of fuel-air mixture ignition; C-suction of gases to flame root*

Turbulent burners have a throughput capacity of the range 1–3.8 kg/s. The completeness of fuel burning depends on the ratio of axial velocities of the primary and secondary air flows in a burner. The velocity of primary flow (dust-air mixture) is usually 16–25 m/s, while that of secondary air is 1.3 to 1.4 times the primary flow velocity.

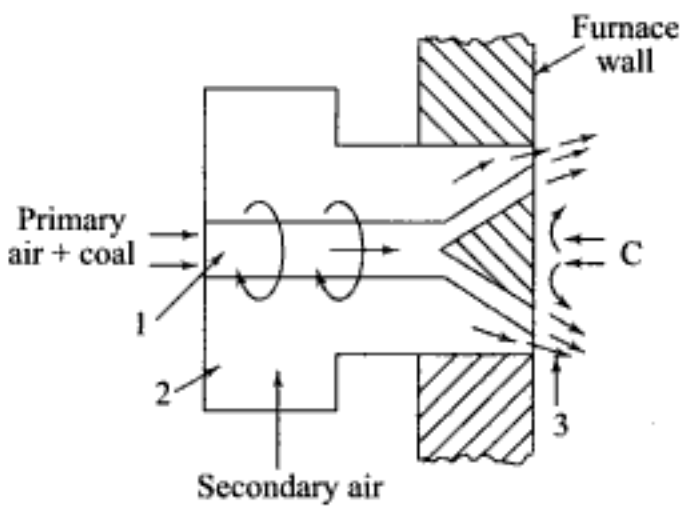

Fig. 5.29 (a) *Straight-scroll burner; 1-channel for primary air and coal; 2-scroll for secondary air; 3-mixing zone and ignition; C-suction of gases to the flames*

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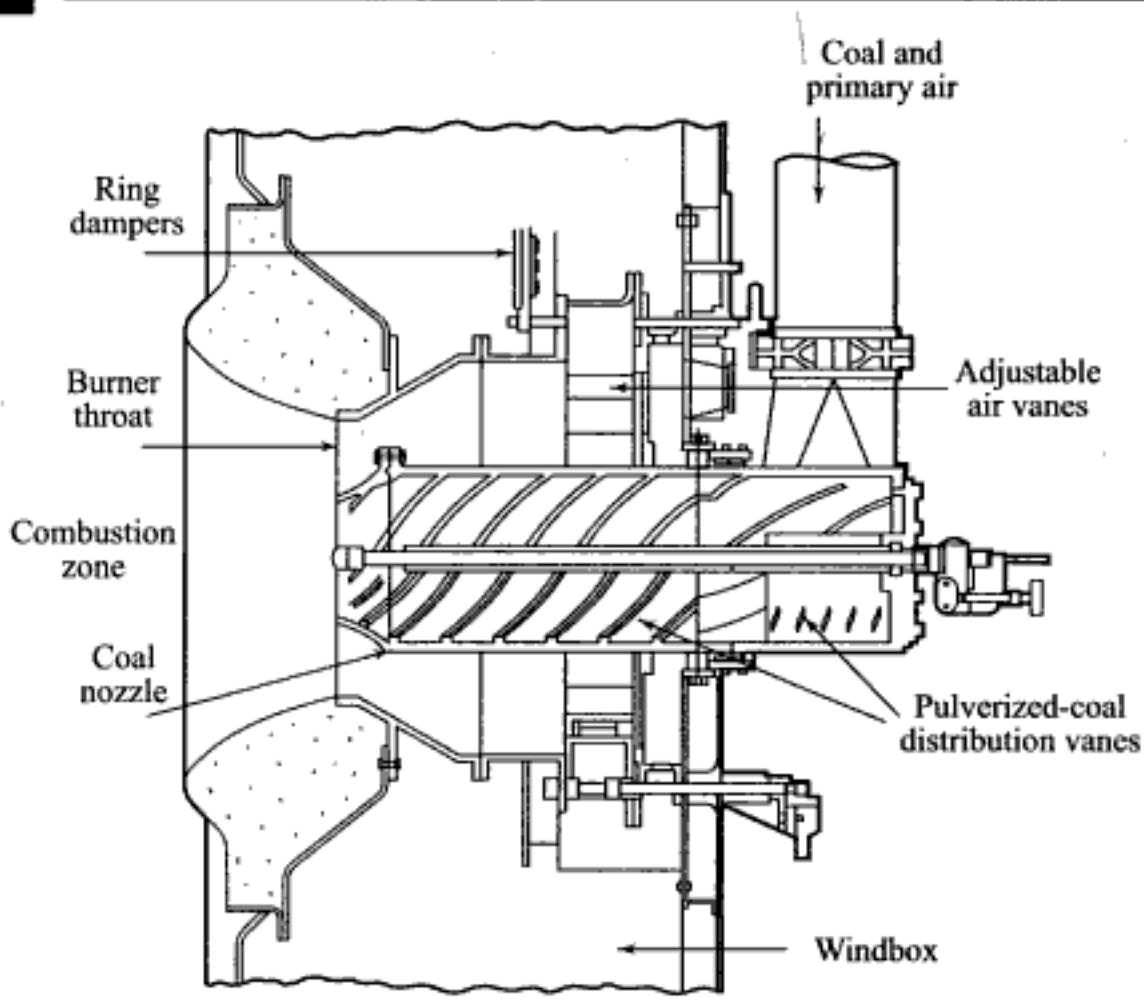

Fig. 5.29(b) *Burner for horizontal firing of coal*

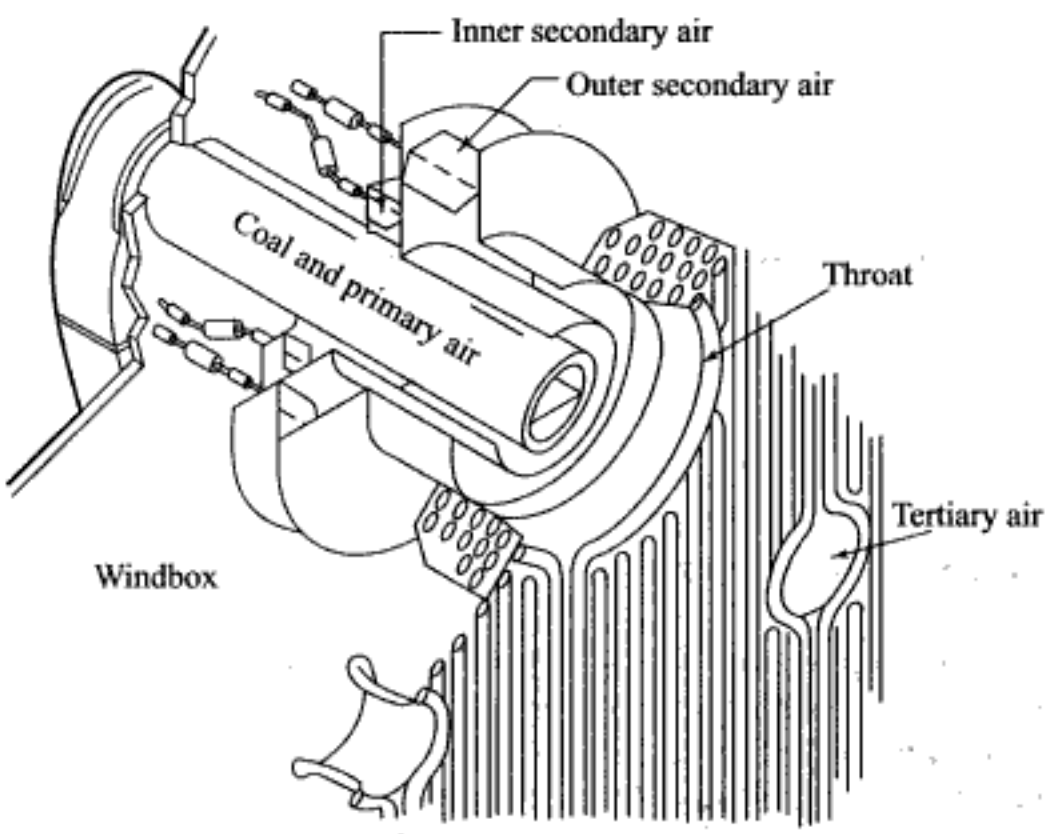

Fig. 5.30(a) *Distributed mixing burner (DMB)*

Distributed mixing burner (DMB) In this type of burner, the total air supplied is distributed in such a way that complete combustion of fuel is achieved

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with minimum excess air (Fig 5.30a). This is a very efficient burner with the sequential stages of fuel-air mixing as shown in Fig 5.30b. Conceptually, there are three zones into which the total input is divided. In the first zone, the stoichiometry is maintained at a very rich condition when the devolatilization starts. Supplementary secondary air is then added to the rich products so as to decay the volatilized fuel-nitrogen compounds. Tertiary air provides the oxygen necessary for complete combustion. This oxidizing zone around the rich primary zone prevents corrosion due to a reducing atmosphere in the furnace and limits the interactions between adjacent burners.

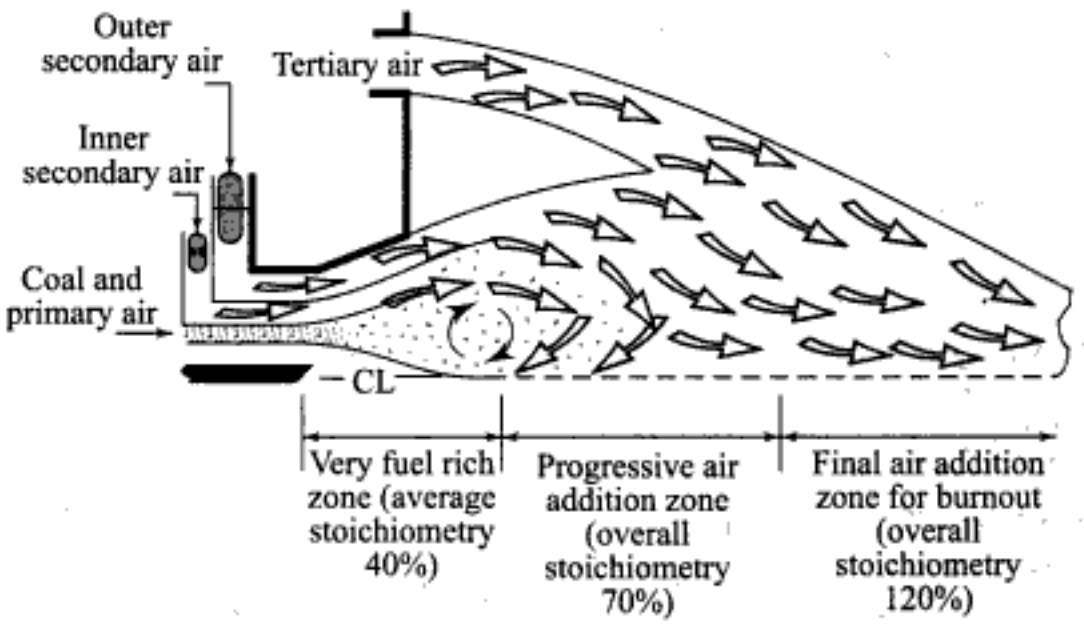

Fig. 5.30 (b) *Distributed burner mixing concept*

Low NO_X burners NO_X is produced within the furnace combustion system by the oxidation of both atmospheric nitrogen (thermal NO_x) and the nitrogen chemically combined with the fuel (fuel NO_x). In order to limit the NO_x output, it is necessary to reduce the oxygen available to the nitrogen in the primary combustion process (by fuel-rich operation) and subsequently to control the overall gas temperature to limit the further oxidation of nitrogen while completing combustion (fuel-lean operation). Thus, the main NO_x reducing parameters are met by the following measures in a corner-fired unit: (1) Reduction of secondary air quantity injected through the fuel nozzles. (2) Diversion of the secondary air injected through the auxiliary secondary air nozzles towards the furnace walls. (3) Injection of the remaining secondary air through ports above the top fuel nozzles on the original firing circle (overfire air). Figure 5.31(a) shows the effect on NO_x production of varying quantities of overfire air.

A CEGB 500 MW corner-fired boiler has been converted to this system and NO_x reduction of up to 40% obtained without increase in carbon-in-ash losses and an improvement in the slagging characteristics of the boiler (Modern Power Station Practice, Vol. B, 1991).

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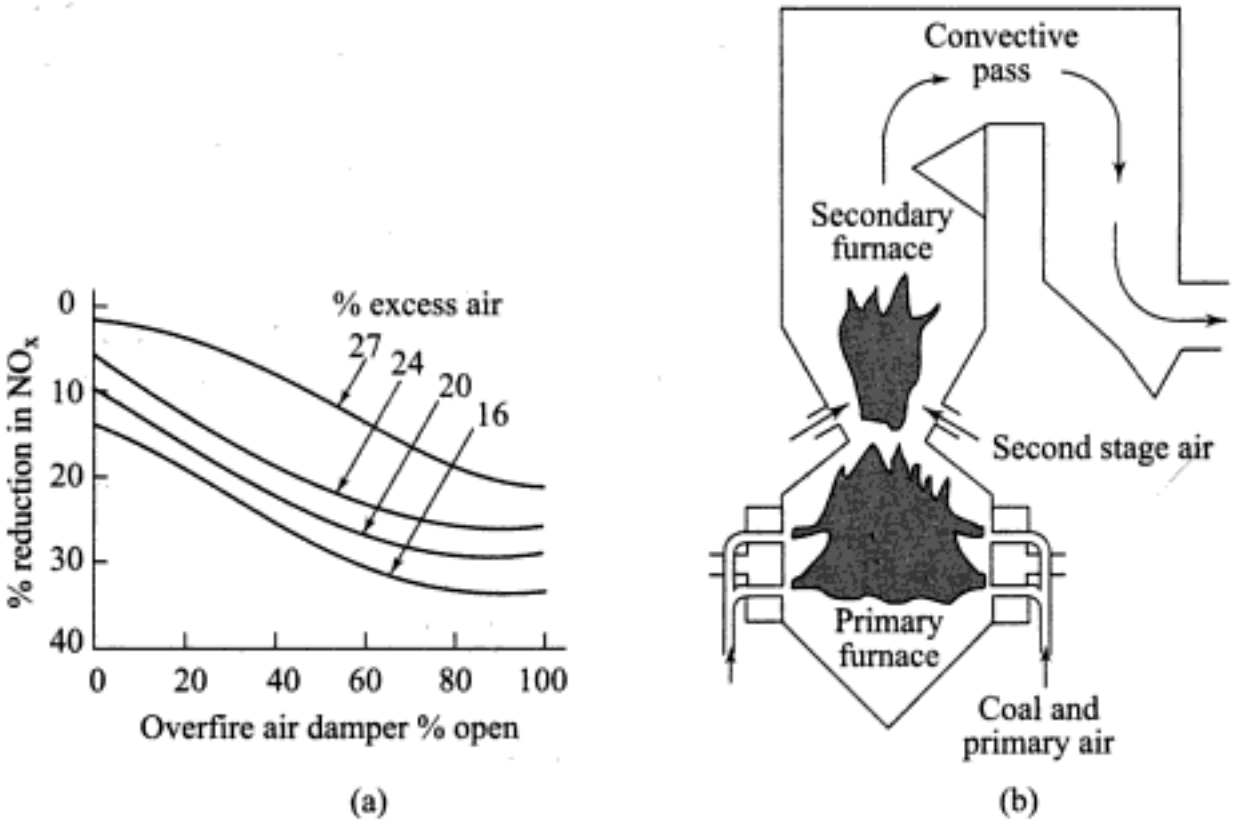

Fig. 5.31 *(a) Effect of overfire air on NO_x production*
(b) Babcock and Wilcox two-stage venturi furnace (for low NO_x production)

Figure 5.31(b) shows a two-stage venturi furnace for reducing NO_x emissions. Coal is burned in two separate chambers. In the initial chamber coal is burned at substoichiometric conditions, resulting in incomplete combustion. This partial burning also allows the desired N_2 producing reactions to occur. Secondary air is added at the entrance to the second chamber in order to achieve oxidizing conditions during transport through the second combustion chamber.

NO_x emissions can also be reduced by flue gas recirculation where bulk flame temperature and flame oxygen concentration are reduced.

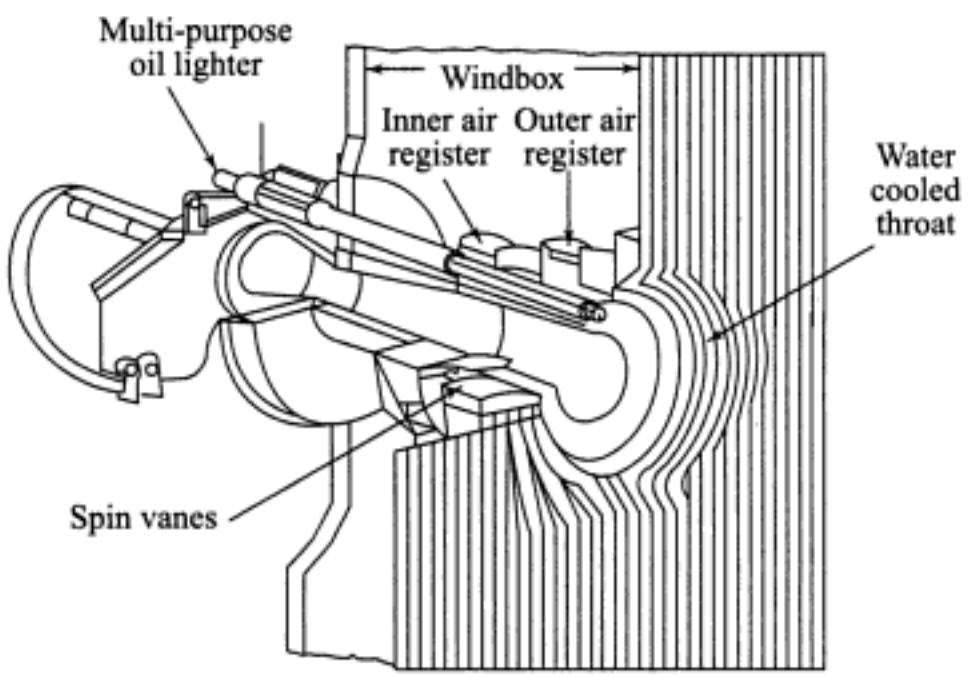

Fig. 5.32 (a) *Multifuel burner for pulverized coal, oil and gas*

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Combined burners Power stations may have to operate simultaneously or alternatively on a variety of fuels. Combined burners which can burn efficiently each kind of fuel are then employed. Figure 5.32 shows a combined burner in a high capacity boiler which can operate on three kinds of fuel: pulverized coal (basic), fuel oil and natural gas. The burner has an enlarged diameter in the central channel in which the main fuel oil burner and a whirling device for turbulizing the axial air flow are arranged. Natural gas is fed through feeder tubes in thin jets between the axial and secondary air flows ensuring thorough intermixing and stable burning of fuel.

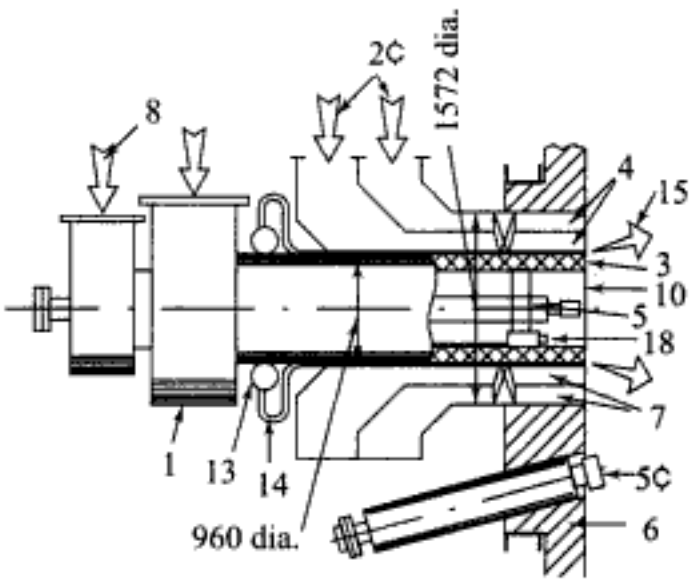

Fig. 5.32 (b) *Multifuel burner: Numbering as in Fig. 5.28 and 8-tertiary air inlet; 2¢-secondary air inlet; 7-swirl vanes for sec. air; 10-Swirler for axial air; 5¢-firing fuel oil burner; 13-natural gas inlet; 14-pipes for gas inlet; 15-gas to furnace; 16-electric gas igniter.*

Burner arrangement Burners are arranged on the furnace walls so as to ensure almost complete combustion of fuel in the flame core itself. Care must be taken for the removal of slag from the furnace (in either dry or molten state) and prevention of clinker formation on the furnace walls. Burners are optimally arranged depending on their type and characteristics.

1. *Turbulent burners*: These burners give a shorter and wider flame than do the straight-flow type. The primary and secondary air flows are intensively intermixed due to the energy of turbulent motion, which secures a thorough burning off of fuel in the flame cope.

Turbulent burners can be positioned for

(a) front firing
(b) double-front or opposed firing
(c) side firing
(d) double-side firing

These arrangements are shown in Figure 5.33.

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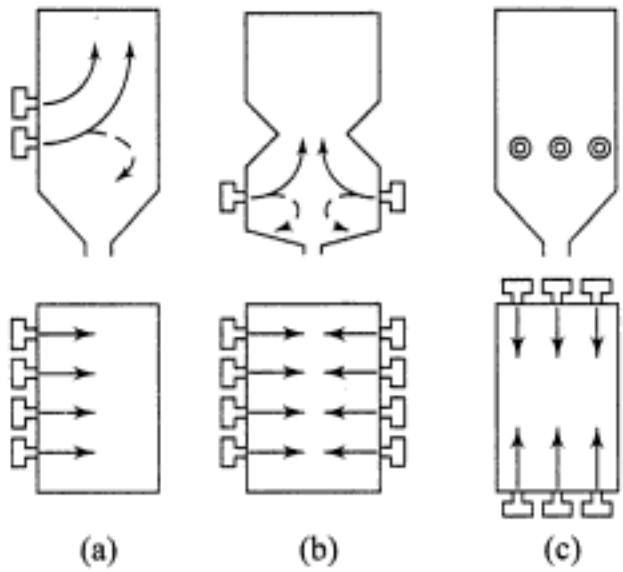

Fig. 5.33 *Arrangement of turbulent burners on furnace walls: (a) front firing; (b) opposed firing; (c) opposed firing on side walls*

The important parameter of tubulent burners is the diameter of their ports, D_p. Burners are arranged (2.2 to 3) D_p from one another and at a distance of (1.6 to 2) D_p from the side walls of the furnace, which helps to prevent the flames from interacting prematurely and touching the walls. With a single wall firing (either front or side), the opposite water wall of the furnace has a high heat absorption (10–20% above the average value), so that the furnace depth in a dry-bottom furnace must not be less than b = (6 to 7) D_p so as to prevent clinkering on this wall. The opposed firing, either front or side, is employed in high-capacity steam boilers where the required number of burners cannot be arranged on a single wall. With the opposite arrangement of burners, heat absorption by water walls in the furnace is much more uniform.

2. *Straight-flow burners*: Burners of this type secure complete combustion of fuel by the huge mixing of the flames from individual burners upon their collision in the furnace space. Typical arrangements of these burners are

(a) opposite displaced (offset) firing
(b) corner firing with encountering flame jets
(c) corner firing with tangential jets
(d) vertical firing

These are shown in Fig. 5.34.

With opposite displaced jets (a) peat and brown coal can be efficiently burnt as a result of the intensive turbulization of the flame in the main combustion zones, as a result of large velocity gradients between adjacent jets moving in opposite directions.

With corner burners and encountering jets (b), the flame core is highly turbulized. However, if the flame moves away from the furnace centre to both sides of the wall, clinkering of the walls may occur due to intense thermal loading near those walls.

The tangentially fired system (c) with corner burners and flame jets directed tangentially to an imaginary circle 1–2.5 m in diameter in the furnace centre has found wide application in pulverized coal fired boilers. Both fuel and air are

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projected horizontally from the corners of the furnace along lines tangent to a vertical cylinder at the centre of the furnace. Intensive mixing occurs where the streams meet. A rotating motion, similar to that of a cyclone, is imparted to the flame body, which spreads out and fills the furnace area. Dampers control the air to each compartment according to the need. The vertical arrangement of fuel nozzles permits great fuel flexibility for multifuel burning. Fuel and air nozzles most commonly tilt in unison to raise and lower the flame in the furnace, to control furnace heat absorption and thus heat absorption in the superheater and reheater sections.

A tangential burner arrangement can, however, be employed in furnaces with a nearly square cross-section, i.e. with a/b ratio equal to 1–1.2. This ensures good aerodynamics in the furnace space.

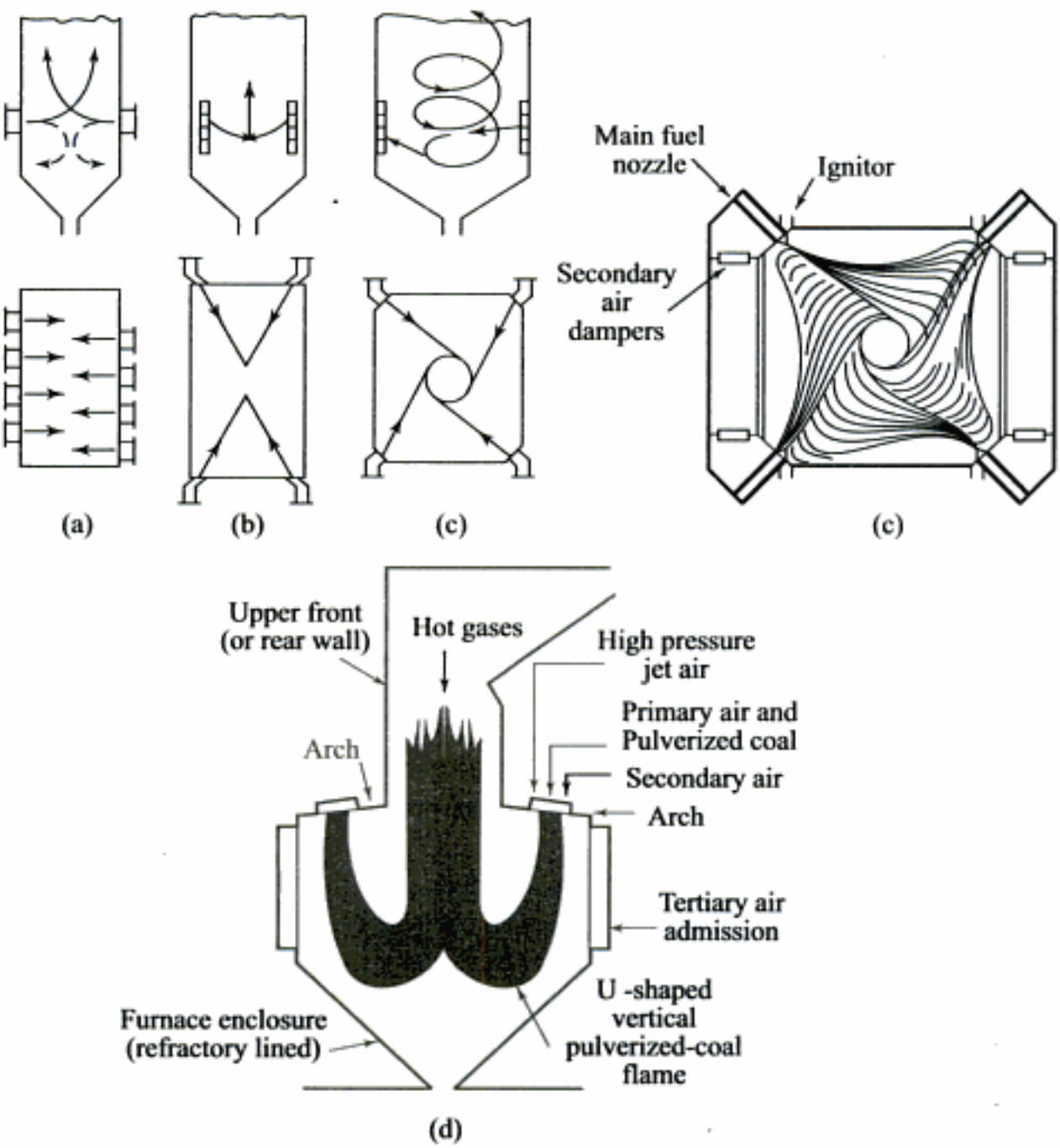

Fig. 5.34 (i) *Arrangement of straight flow burners on furnace walls: (a) opposed displaced firing, (b) corner firing with encountering jets; (c) corner firing with tangential jets; (d) vertical firing*

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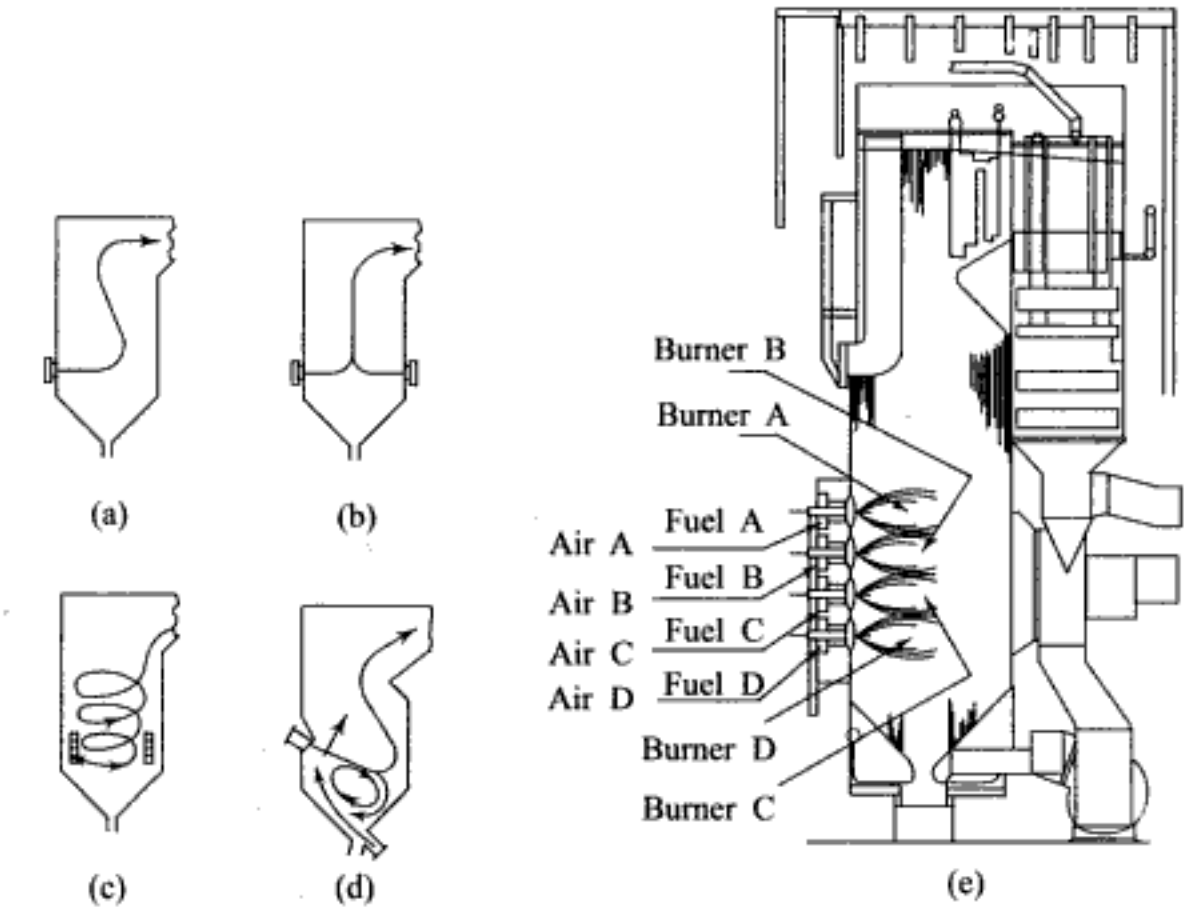

Fig. 5.34 (ii) *Patterns of flame motion in dry-bottom furnaces: (a) straight-flow S-shaped flame; (b) opposite straight-flow flame; (c) vertically turbulized flame; (d) combination of straight-flow and horizontally turbulized flame; (e) flow pattern of horizontal (wall) firing*

Vertically fired systems (d) are used only to fire solid fuels which are difficult to ignite, like anthracite and low volatile coals. Pulverized coal and primary air mixture is discharged through vertical nozzles. Heated combustion air is introduced around the fuel nozzles and through auxiliary ports. High pressure jets help avoid short-circulating the fuel air streams to the furnace discharge. Tertiary air from the front and rear rows provides needed turbulence. Long travel (and so long residence time) in the combustion chamber helps in complete combustion.

Figure 5.34(ii) shows the flow pattern of flame in dry-bottom furnaces which can be straight, vertically turbulized or horizontally turbulized. The flow pattern of horizontal firing is shown in Fig. 5.34(e).

5.8 CYCLONE FURNACE

Cyclone furnace firing, developed in the 1940s, represents one of the most significant steps in coal firing since the introduction of pulverized coal firing in the 1920s. It is now widely used to burn poorer grades of coal having high ash and moisture content. Biofuels like rice husks can also be successfully burned in cyclone furnaces for steam generation.

The cyclone is essentially a water-cooled horizontal cylinder (Fig. 5.35) located outside the main boiler furnace, in which crushed coal (60 mm size or less) is fed and fired with very high rates of heat release. The cyclone is made

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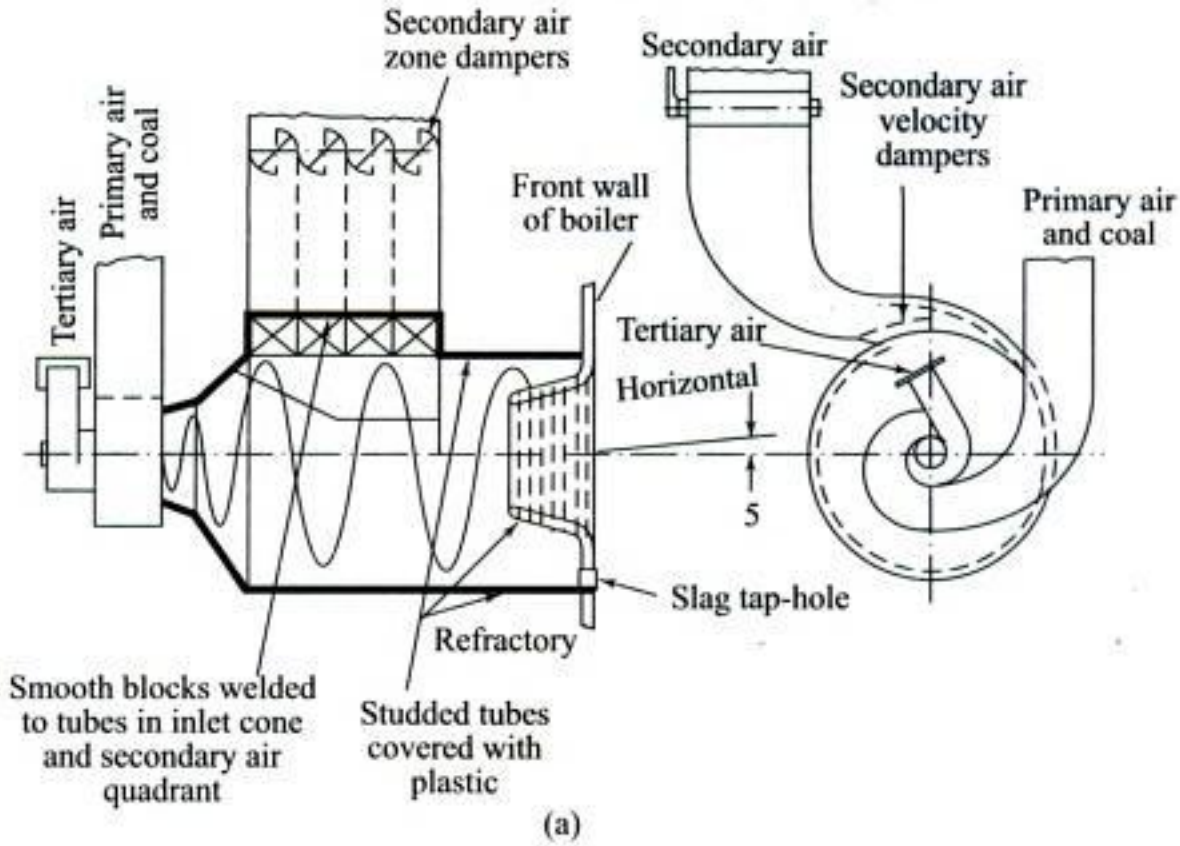

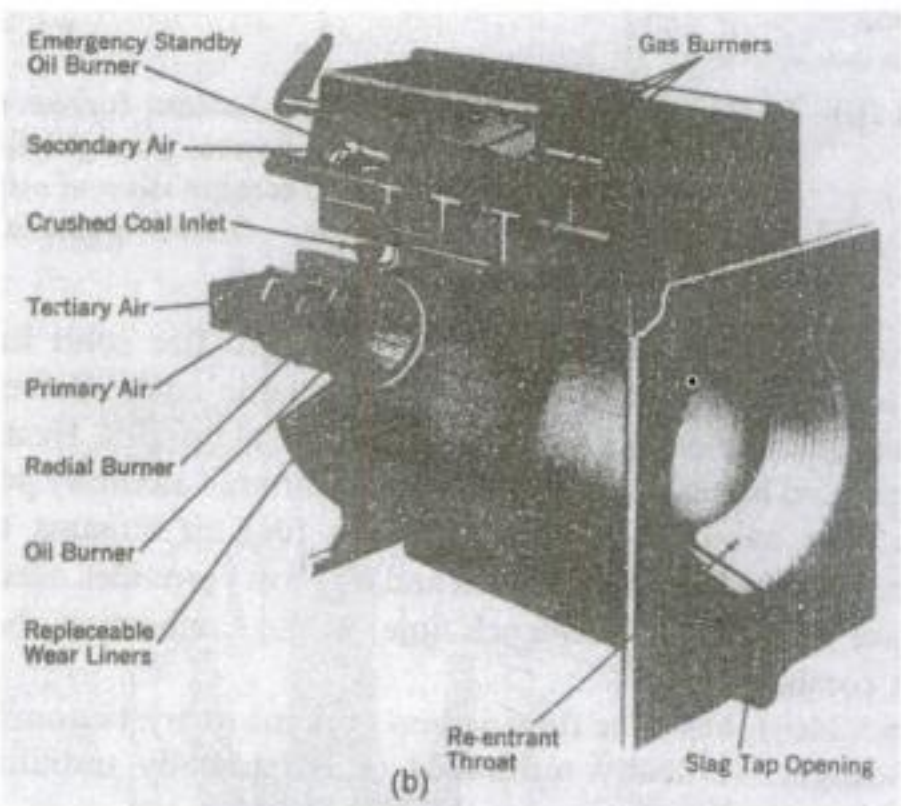

Fig. 5.35 *Cyclone furnace*

to a diameter of 1.8–4 m and its length is 1.2–1.3 times its diameter. The crushed coal is fed into the cyclone from the left along with primary air, which is about 20 per cent of combustion or secondary air. The coal air mixture is entered tangentially, thus imparting a centrifugal motion to the coal particles. The secondary air is also admitted tangentially at the top of the cyclone at high speed (80–120 m/s) imparting further centrifugal motion. A small quantity of air, called

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tertiary air, is admitted at the centre. Combustion of the coal is completed before the resulting hot gases enter the boiler furnace. The whirling motion of coal and air results in large volumetric heat release rates ranging from 4.7 to 8.3 MW/m^3 and high combustion temperatures, more than 1650 °C. These high temperatures melt the ash into a liquid slag that covers the surface of the cyclone and eventually drains through the slag-tap opening to a slag tank at the furnace bottom, where it is solidified and fragmented for removal.

The main advantage of cyclone firing is the removal of ash, about 60 per cent, as molten slag through the slag tank. Thus only 40 per cent ash leaves with flue gases, compared with 85 per cent ash for dry-bottom pulverized coal furnaces, this reduces erosion and fouling of boiler surfaces as well as the size of dust-removal precipitators or bag houses at boiler exit. Another advantage is that only crushed coal is used and no pulverization equipment is needed and that the boiler size is reduced. The disadvantages are higher forced draught fan pressures and therefore, higher power requirement, and formation of relatively more oxides of nitrogen, NO_X, which are air pollutants.

5.9 FLUIDIZED BED COMBUSTION

When air is passed through a fixed or packed bed of particles, air simply percolates through the interstitial gaps between the particles. As the air flow rate through the bed is steadily increased, a point is eventually reached at which the pressure drop across the bed becomes equal to the weight of the particles per unit cross-sectional area of the bed. This critical velocity is called the *minimum fluidization velocity*, U_{mf}, at which the bed is said to be *incipiently fluidized*. As the air velocity is increased further, the particles are buoyed up and imparted a violently turbulent fluidlike motion, with the drag forces exerted by the fluid on the particles exceeding their weight. There is a high degree of particle mixing and equilibrium between gas and particles is rapidly established. This is called a fluidized bed.

In Fig 5.36 (a), air supplied by a centrifugal blower is passed through a perforated or porous plate, called the *distributor*, and then a bed of particles of wide size distribution. A few distributors are shown in Fig. 5.36(b). The air flow rate is regulated by a bypass valve along with a control valve, and it is measured by a rotameter. Dividing the mass flow rate, so measured, by the product of the bed cross-sectional area and density of air, the *superficial velocity* of air, U, is estimated. For each mass flow rate or superficial velocity, which is gradually increased, the pressure drop across the bed is measured. Figure 5.37(a) demonstrates the variation of bed pressure drop with superficial velocity. The pressure drop Δp varies with the superficial velocity linearly along AB till it approaches W/A_t, where W is the weight of particles in the bed and A_t is the bed cross-sectional area. This is the fixed bed regime. With further increase in air flow, the particles are buoyed up by the drag force only to fall back into the bed yielding high turbulence, with $\Delta p = W/A_t$ remaining constant. This continues till there is considerable pressure fluctuation and finally the superficial velocity

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reaches the state C when the particles start getting entrained in the air stream. The superficial velocity of air at which particle entrainment starts is called the *terminal velocity*, U_t, of the particles. The state of bed in the range of superficial velocity from B to C, i.e. between U_{mf} and U_t is the *fluidized bed regime*. With further increase in velocity, the entrained particles are carried away, the weight of particles in the bed and so Δp decrease till it reaches the state D when all the particles are carried away by the air stream, the bed becomes empty and Δp is zero.

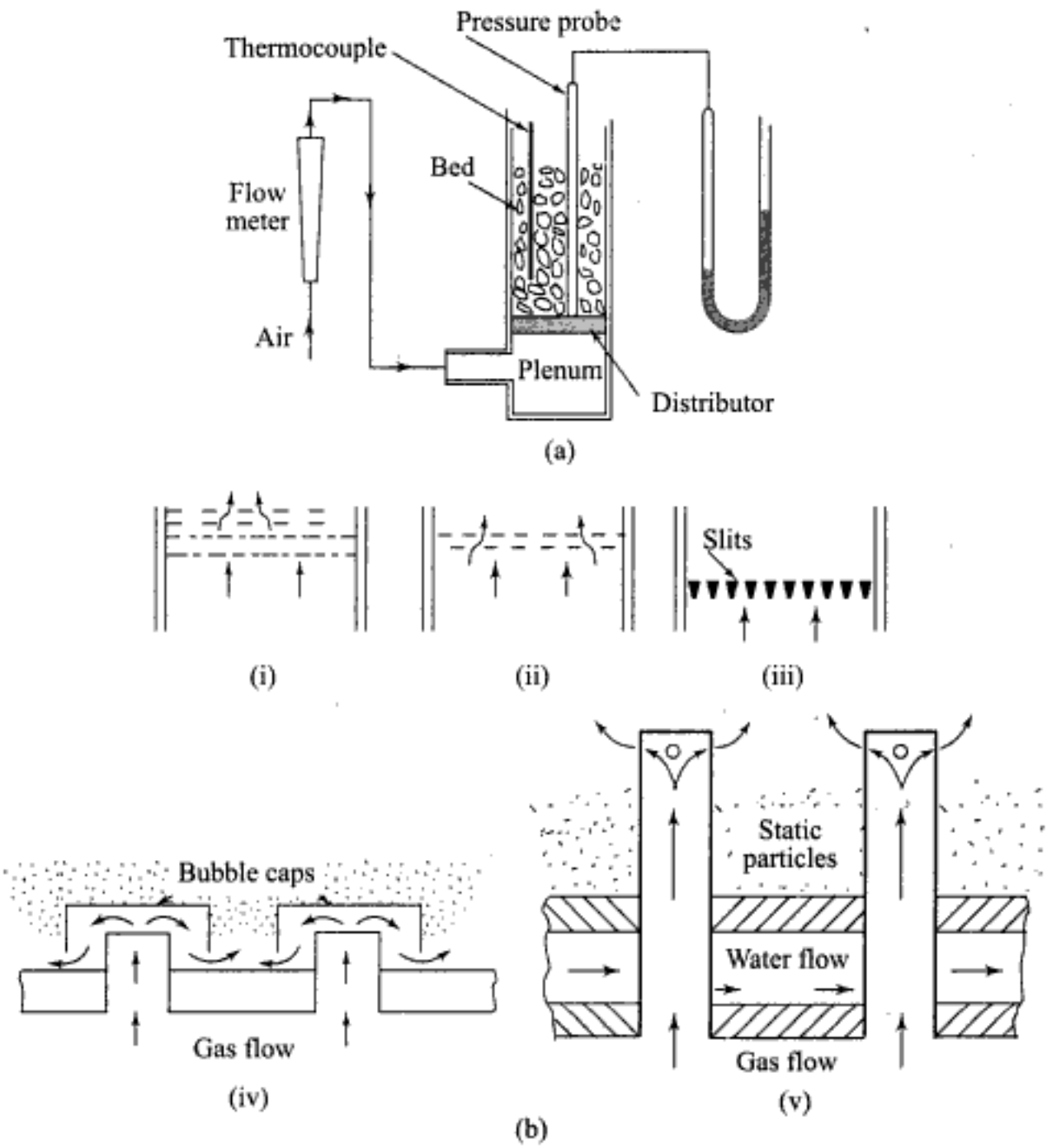

Fig. 5.36 *(a) Measurement of U_{mf} and U_t of a fluidized bed*
(b) Few types of distributors; (i) sandwiching perforated plates, (ii) staggered perforated plates; (iii) grate bars, (iv) bubble cap type, (v) water-cooled type

At incipient fluidization, a small "hump" may be recorded. This arises because extra pressure is required to unlock the particles from their pattern of

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packing. Once the particles have been released from this pattern, the voidage increases, and the pressure drop falls back to a value sufficient to support the weight of particles of the bed.

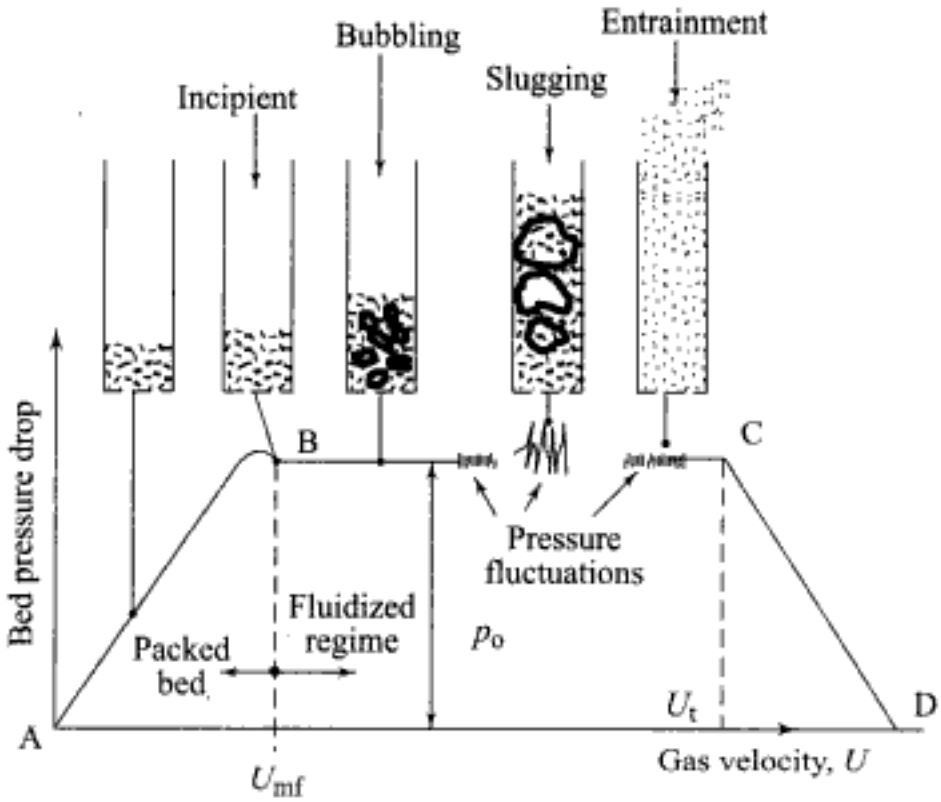

Fig. 5.37 (a) *Variation of bed pressure drop with superficial velocity*

5.9.1 Regimes of Fluidization

Fluidization, as explained above, is an operation by which fine solids are transformed into a fluid like state through contact with a gas (or a liquid). A fluidized bed displays characteristics similar to those of a liquid, e.g. (a) the static pressure at any height is approximately equal to the weight of bed solids per unit cross-section above that height, (b) the bed surface maintains a horizontal level, irrespective of how the bed is tilted, (c) the solids from the bed may be drained like a liquid through an orifice at the bottom or on the side, and (d) an object denser than the bulk of the bed (steel ball) will sink, while one lighter than the bed (wood piece) will float.

With an increase in gas velocity through a bed of granular solids, the bed moves from one regime to another. Table 5.2 gives a comparison of some characteristic features of different gas solid processes in various types of boilers.

Different commercial combustion systems operate under different gas–solid flow regimes, as shown in Fig. 5.37(b). A stoker-fired boiler burns coal in a fixed bed (E) having the highest density and lowest gas velocity. It is followed by a bubbling fluidized bed (A) boiler, an Ignifluid boiler, where the bed is in turbulent

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fluidization (*B*); a CFB boiler, where parts of the combustor are in fast fluidization (*C*); and pulverized coal firing where coal burns under entrained or pneumatic transport condition (*D*) (Basu and Fraser, 1991).

Table 5.2 *Comparison of characteristic features of gas–solid processes in various boilers**

Property	*Packed bed*	*Fluidized bed*	*Fast bed*	*Pneumatic transport*
Application in boilers	Stoker fired	Bubbling fluidized	Circulating fluidized	Pulverized coal fired
Mean particle diameter (mm)	<300	0.03–3	0.05–0.5	0.02–0.08
Gas velocity through combustor zone (m/s)	1–3	0.5–3	3–12	15–30
Typical U/Ut	0.01	0.3	2	40
Gas motion	Up	Up	Up	Up
Gas mixing	Near plug flow	Complex two phases	Dispersed plug flow	Near plug flow
Solids motion	Static	Up & down	Mostly up, some down	Up
Solid-solids mixing	Negligible	Usually near perfect mixing	Near perfect mixing	Near plug flow
Overall voidage	0.4–0.5	0.5–0.85	0.85–0.99	0.98–0.998
Temperature gradient	Large	Very small	Small	May be significant
Typical bed-to-surface. heat transfer coefficient, (W/m^2K)	50–150	200–550	100–250	50–100
Attrition	Little	Some	Some	Considerable
Agglomeration	Considerable	Some	No problem	No problem

* P. Basu and S. Fraser, *Circulating Fluidized Bed Boiler: Design and Operations*, Butterworth - Heineman, 1991.

1. Packed bed A packed or fixed bed consists of a bed of stationary particles on a perforated grid through which a gas is flowing (Fig. 5.38a). The pressure drop per unit height of a packed bed of uniformly sized particles, $\Delta p/L$, is given by Ergun's equation

$$\frac{\Delta p}{L} = 150\frac{(1-\varepsilon)^2}{\varepsilon^3}\frac{\mu U}{(\phi d_p)^2} + 1.75\frac{1-\varepsilon}{\varepsilon^3}\frac{\rho_g U^2}{\phi d_p} \tag{5.21}$$

where U is the superficial velocity, ε is the void fraction in the bed (volume of gas or voids divided by total volume of bed), d_p and ϕ are the diameter and sphericity of particles respectively. μ is viscosity and ρ_g is the density of the gas. The voidage

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of a bed of particles, ε, is the fraction of the bed volume which is occupied by the space between the solid particles. So,

$$\text{Voidage, } \varepsilon = \frac{\text{volume of bed} - \text{volume of solids}}{\text{volume of bed}}$$

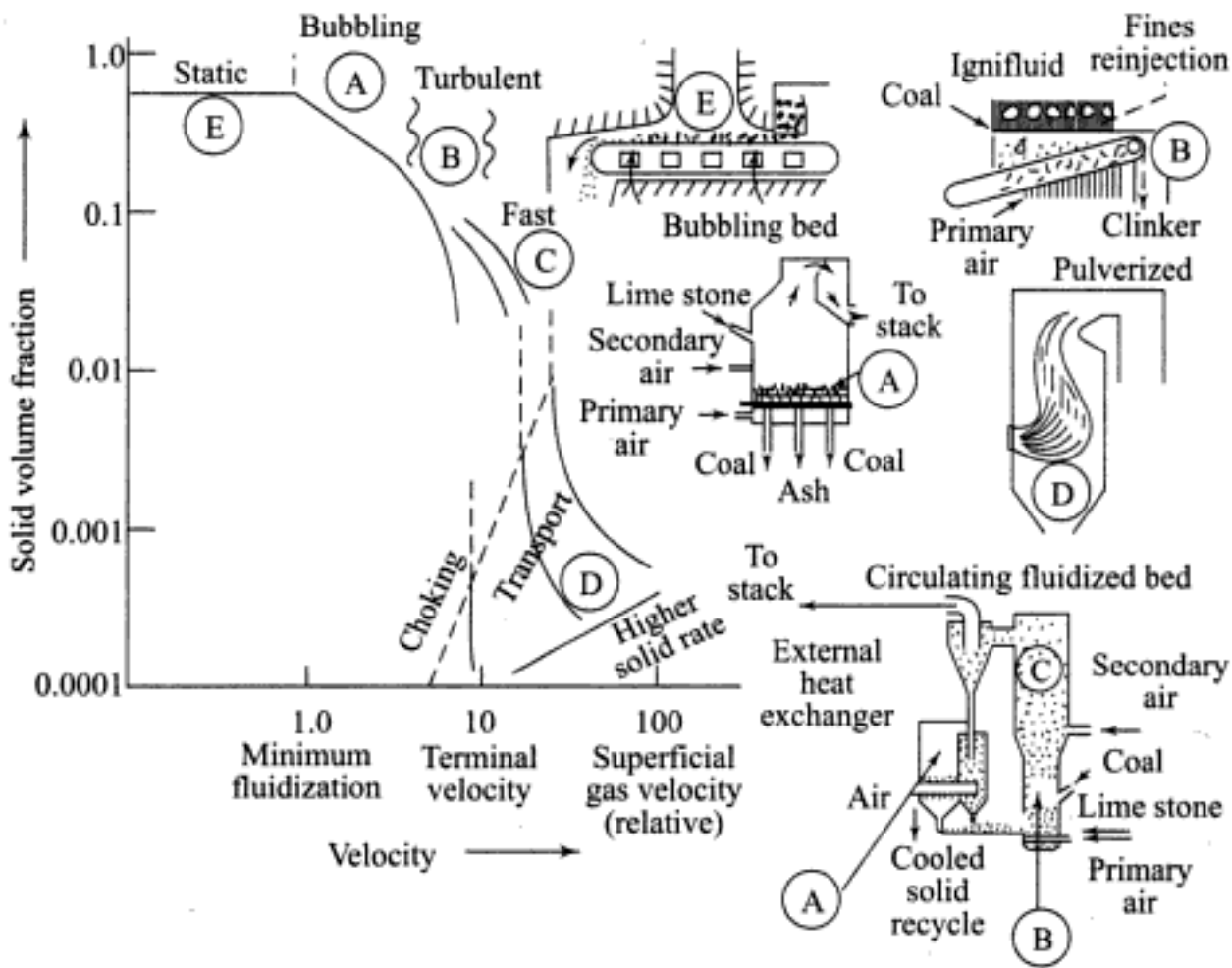

Fig. 5.37 (b) *Different commercial combustion systems operating under different gas-solid flow regimes*

$$= 1 - \frac{\text{volume of solids}}{\text{volume of bed}} = 1 - \frac{m_s}{\rho_s} \times \frac{\rho_b}{m_b}$$

where m_s and m_b are the masses of solids and bed respectively and ρ_s and ρ_b are their corresponding densities. Since the voids may be regarded as empty spaces, $m_s = m_b$. Therefore,

$$\varepsilon = 1 - \frac{\rho_b}{\rho_s} \tag{5.22}$$

where ρ_b is the bulk density of the bed which is equal to the mass of solids in the bed divided by the bed volume.

The sphericity, ϕ, of a particle is defined as

$$\phi = \frac{\text{surface area of a sphere of the same volume as the particle}}{\text{surface area of the particle}}$$

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In a moving packed bed, the solids move with respect to the wall, but not with respect to one another.

2. Bubbling fluidized bed When the superficial velocity of gas flowing through a fixed bed reaches the minimum fluidization velocity, U_{mf}, the fixed bed

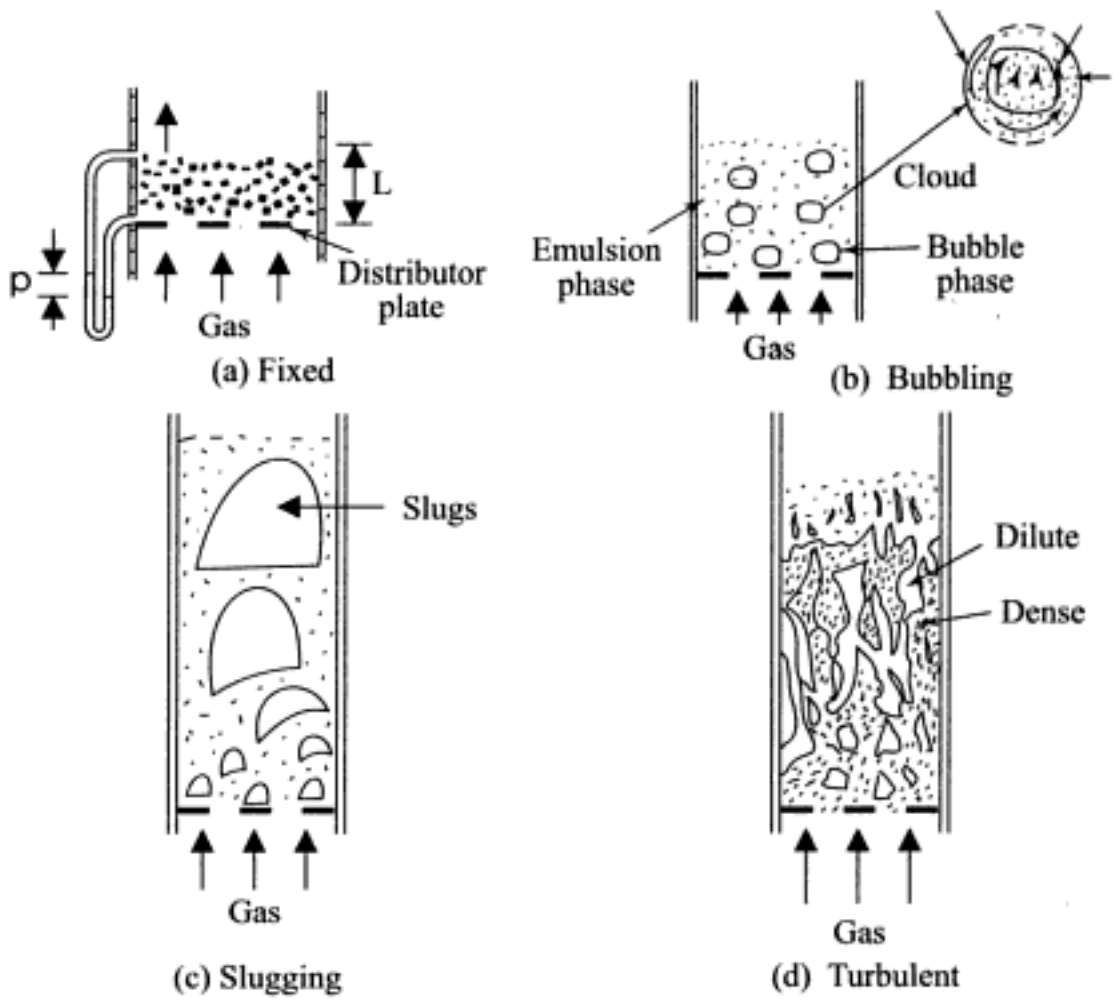

Fig. 5.38 *Gas-solid contacting mode changing from one regime to another with the increase of gas velocity*

transforms into an incipiently fluidized bed and the bed starts behaving as a liquid (Fig. 5.38b). Since the pressure drop across the bed is equal to the weight of the bed, the fluid drag F_D is given as

$$F_D = \Delta pA = AL\,(1-\varepsilon)\,(\rho_s - \rho_g)\,g$$

where A and L are the cross-section and height of the bed, respectively, and ρ_s is the density of solids.

$$\Delta p/L = (1-\varepsilon_{mf})\,(\rho_s - \rho_g)\,g \tag{5.23}$$

The minimum superficial velocity, U_{mf}, may be obtained by solving Eqs. (5.21) and (5.23) simultaneously, as given below; in terms of Reynolds number Re_{mf}:

$$Re_{mf} = \frac{\rho_g d_p U_{mf}}{\mu} = \left[C_1^2 + C_2 Ar\right]^{0.5} - C_1 \tag{5.24}$$

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$$\text{Ar} = \text{Archimedes number} = \frac{\rho_g(\rho_s - \rho_g)gd_p^3}{\mu_g^2} \tag{5.25}$$

$C_1 = 27.2$ and $C_2 = 0.0408$ (Grace, 1982)

As increase in gas flow beyond minimum fluidization can cause the extra gas to flow in the form of bubbles. The section of the bed outside the bubbles is called the *emulsion phase*. Bubbles are gas voids with very little or no solids. Due to the buoyancy force a bubble rises through the emulsion phase and its size increases with particle diameter d_p, excess gas velocity U-U_{mf}, and bed height. A bubble carrying some solid particles in its wake erupts at the bed surface throwing particles into the space above the bed, called the *freeboard*. The particles entrained travel upwards due to their momentum and local drag of gas. Some of the particles may disengage from the gas and fall to the dense bed due to gravity. Beyond a certain height, called the *transport disengaging height* (TDH), particle disengagement is negligible, and the flux rate of particles carried away is known as the *elutriation rate*.

The velocity U_b at which a single isolated bubble rises in a large diameter vessel is given by

$$U_b = 0.71\,(g\,D_b)^{1/2} \tag{5.26}$$

where D_b is the bubble diameter, taken as the diameter of a sphere having the same volume as the bubble.

In most practical situations, the fluidizing velocity is sufficiently greater than U_{mf} for bubbles to form continuously. However, the bubbles are not isolated, but they are in close proximity. They influence particle mixing, gas to particle contacting, bed expansion, gas bypass, elutriation and so on. In a bubbling bed, the bubble rise velocity may be obtained from

$$U_b = k\,(U - U_{mf}) + 0.71\,(g\,D_b)^{1/2} \tag{5.27}$$

where the value of constant k is approximately 1.

3. Slugging If the bed is small in cross-section and deep, the bubble may increase to a size comparable to the diameter or width of the bed. The bubble then passes through the bed as a slug (Fig. 5.38c). This is known as slugging and there is a large fluctuation of pressure drop across the bed which is unstable.

4. Turbulent bed As the velocity of gas through a bubbling fluidized bed is increased, the bed expands, and a point is eventually reached when the emulsion walls become very thin, the bubbles constantly collapse and re-form, resulting in a violently active bed. The bed surface is highly diffused and particles are thrown into the freeboard above. Such a bed is called a *turbulent bed* (Fig. 5.38d). The pressure drop across the bed fluctuates rapidly. The amplitude of pressure fluctuation reaches a peak at the velocity U_c and reduces to a steady value at the velocity U_k (Fig. 5.39). Grace (1982) gave the following correlations

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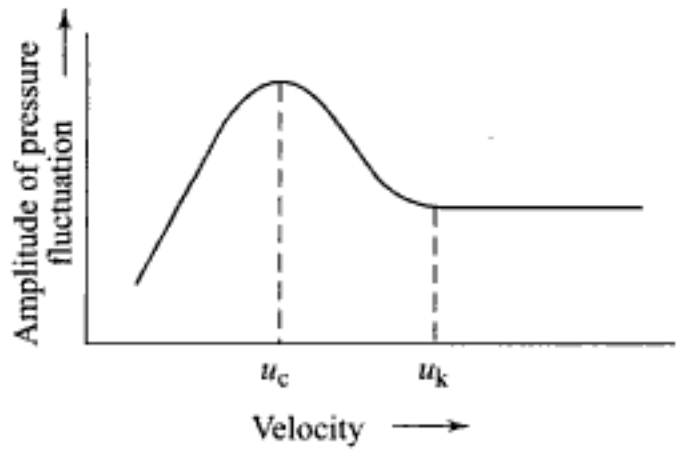

Fig. 5.39 *Transition from bubbling to turbulent bed*

$$U_c = 3(\rho_p\, d_p)^{1/2} - 0.17 \text{ m/s} \tag{5.28}$$

$$U_k = 7(\rho_p\, d_p)^{1/2} - 0.77 \text{ m/s} \tag{5.29}$$

where $(\rho_p\, d_p)$ is in the range 0.05–0.7 kg/m^2. Finer particles become turbulently fluidized at $U > U_t$, where coarser particles undergo this transition at $U < U_t$. The gas-solid contact in this regime is very good.

In the regimes discussed so far, solids are generally retained within the bed and there is no large scale migration of particles from the bed, except for some entrainment. These regimes are said to be in the *captive stage*.

5. Fast fluidized bed It is a regime lying between the turbulent fluidized bed and pneumatic transport. Following Basu and Fraser (1991), a fast fluidized bed may be defined as a high-velocity gas-solid suspension where particles, elutriated by the fluidizing gas above the terminal velocity of single particles, are recovered and returned to the base of the furnace at a rate sufficiently high as to cause a degree of solid refluxing that will ensure a minimum level of temperature uniformity in the furnace. The term "fast bed" has become synonymous with the criculating fluidized bed (Fig. 5.40). High slip velocity $(U_g - U_s)$ between gas and solid, formation and disintegration of particle agglomerates, and a very good gas-solid mixing are the characteristic features of this regime. The cross-sectional average suspension density ρ_{sus} is given from

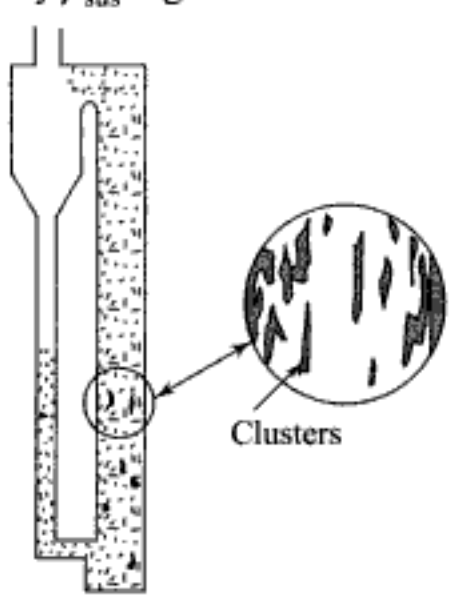

Fig. 5.40 *A fast fluidized bed in which solid agglomerates move up and down in a very dilute dispersion of solids*

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$$m = m_s + m_g$$
$$\rho_{sus} V = \rho_s (V - V_g) + \rho_g V_g$$
$$\rho_{sus} = (1 - \varepsilon)\, \rho_s + \varepsilon \rho_g \tag{5.30}$$

where V_g represents the volume of the voids and V the volume of the bed.

The main difference between bubbling beds and fast or circulating beds lies in the gas velocity used. While bubbling beds normally operate at gas velocities of around 1–3 m/s (depending on particle size), the circulating fluidized bed (CFB) is typically run at 5–10 m/s.

The circulating fluidized bed may be divided into a dense bottom region and a dilute upper region. The bottom region operates either in bubbling or turbulent mode of fluidization depending on the superficial velocity used. Above the bottom bed is the splash zone and then the upper dilute zone where the solids volumetric concentration is very low, typically about 0.3%.

The cross-section average suspension density above the entry level of the secondary air in a commercial CFB boiler generally decays exponentially along the height of the furnace, as it does in the freeboard region of a bubbling fluidized bed. The local gas-voidage at a particular height decreases towards the wall in the same way the local gas velocity through a tube decreases towards it.

The structure of the fast bed is better explained by the *core-annulus model* which divides it into two vertical zones, a relatively dilute up-flowing core in which solid particles are entrained upward by a high velocity gas stream, and a much denser annular layer near the column wall in which solid particles congregate and fall as dense structures similar to waves of strands or streamers (Fig. 5.41). In the core area the gas velocity is more and the particles are entrained in the up-flowing gas stream. Near the wall the gas velocity is less and the particles cohering to form clusters or streamers fall by gravity. After falling a certain distance along the wall the clusters disintegrate only to reform again. The solids concentration in the wall region is found to be 2–3 times the cross-sectional average solids concentration. The thickness of the annulus decrease towards the top of the bed, which may vary from a few mm in laboratory scale units to several decimeters in commercial units. The clusters are slender in shape and may have a voidage of the range 0.6–0.8. They move upwards in the core of the furnace, and when swept towards the low velocity region near the wall, start falling along the wall at the velocity of 1–2 m/s.

The axial voidage distribution in a CFB furnace (Fig. 5.41b) is found to be S-shaped. There is a *point of inflection* which separates the lean and dense regions of a fast bed and is a function of the solid circulation rate and the solids inventory in the system.

Figure 5.42 gives a pressure balance around a CFB loop. The lower section of the fast bed is denser and therefore it results in higher pressure drops per unit height of the bed than the upper leaner section. For stable operation the pressure balance around the loop may be written as

$$\Delta p_{F\text{-}A} + \Delta p_{A\text{-}B} + \Delta p_{B\text{-}C} + \Delta p_{C\text{-}D} = \Delta p_{D\text{-}E} + \Delta p_{E\text{-}F}$$

This pressure balance depends on different operating parameters.

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5.9.2 Terminal Velocity of Particles and Elutriation

If a single particle falls freely under gravity in the atmosphere, it will accelerate until its velocity is such that the drag exerted by the surrounding air is equal to the gravitational force. This velocity is called the "terminal velocity", U_t, which for a sphere of diameter d_p is given by

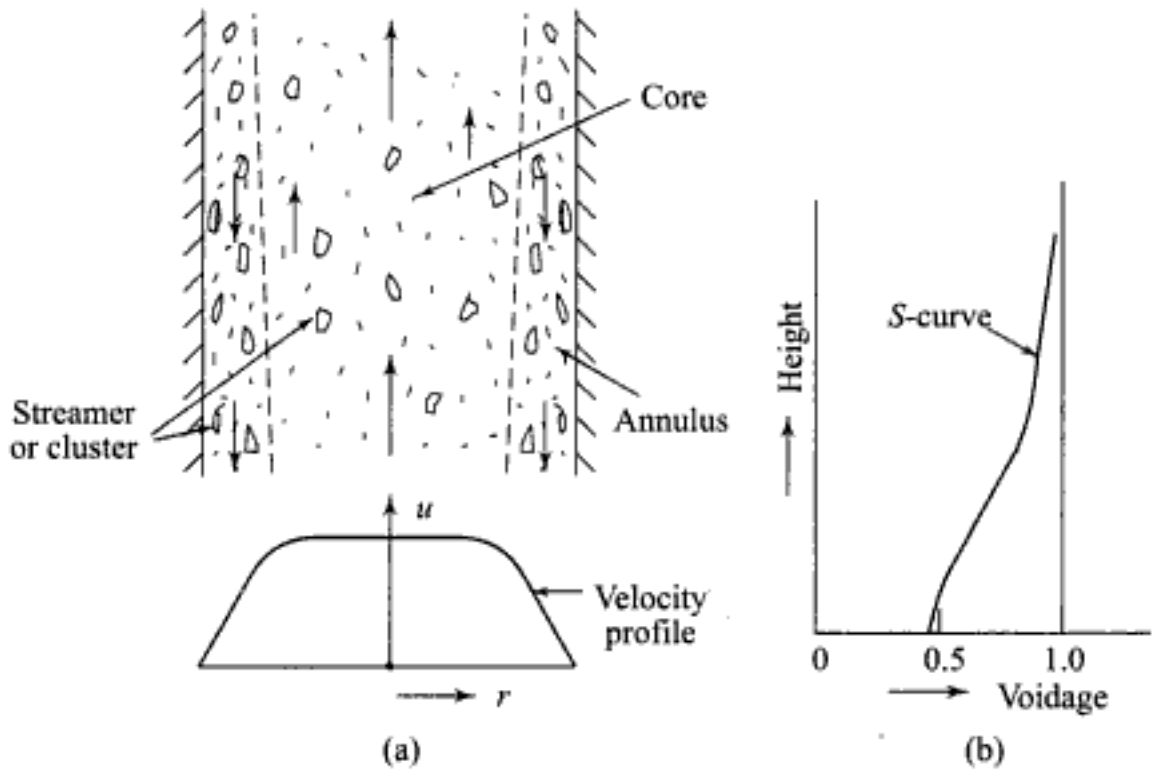

Fig. 5.41 *(a) Core-annulus structure of a CFB: (a) upflowing core (dilute suspension of gas solid mixture) and down-flowing annulus (dense phase of gas and solids), (b) voidage variation along height*

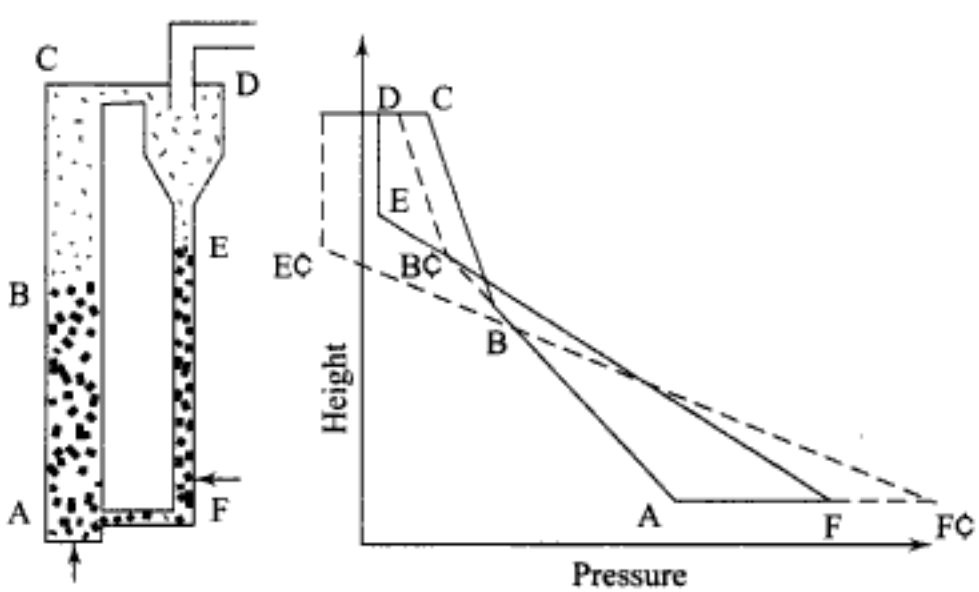

Fig. 5.42 *Pressure balance around a CFB loop. The dotted lines indicate the pressure profile at a higher circulation rate*

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$$(\rho_s - \rho_g)g\,\frac{\pi d_p^3}{6} = \frac{1}{2}\rho_g U_t^2 S C_D \tag{5.31}$$

where S is the frontal area of a sphere, $\frac{\pi}{4}d_p^2$, C_D is the experimentally determined drag coefficient, ρ_s and ρ_g are the densities of solids and gas, respectively.

$$\therefore \qquad U_t = \left[\frac{4(\rho_s - \rho_g)gd_p}{3\rho_g C_D}\right]^{1/2} \tag{5.32}$$

The drag coefficient C_D depends upon the Reynolds number Re_p defined as

$$Re_p = \frac{\rho_g U_t d_p}{\mu_g}$$

No simple relationship expresses the variation of drag coefficient with Reynolds number. Two relationships often quoted are

$$C_D = \frac{24}{Re_p} \qquad \text{for} \qquad Re_p < 0.4$$

and

$$C_D = \frac{10}{(Re_p)^{1/2}} \qquad \text{for} \qquad 0.4 < Re_p < 500$$

Substituting these expressions into Eq. (5.30) yields

$$U_t = \frac{(\rho_s - \rho_g)gd_p^2}{18\mu_g} \qquad \text{for} \qquad Re_p < 0.4 \tag{5.33}$$

and

$$U_t = \left[\frac{4(\rho_s - \rho_g)^2 g^2}{225\mu_g\rho_g}\right]^{1/3} \cdot d_p \quad \text{for} \quad 0.4 < Re_p < 500 \tag{5.34}$$

The terminal velocity of the smallest particle in the bed is important conceptually in that it constitutes the maximum velocity of the gas at which a bubbling fluidized bed should be operated, if entrainment of particles is to be avoided. Fluidized beds, however, commonly operate at significantly higher gas velocities. Particulate solids are composed of a range of particle sizes and, perhaps densities, while many solid materials undergo attrition when fluidized, or fragmentation due to thermal stresses when put into a hot fluidized bed. Attrition and fragmentation lead to the production of fines. These fines become entrained in the gas leaving the bed.

5.9.3 Fluidized Bed Combustion of Solid Fuels

In its simplest arrangement, the fluidized bed consists of inert particles such as sand, through which fluidizing air can be passed via a distributor plate at the bottom (Fig. 5.43). The bed normally contains only a small percentage of burning

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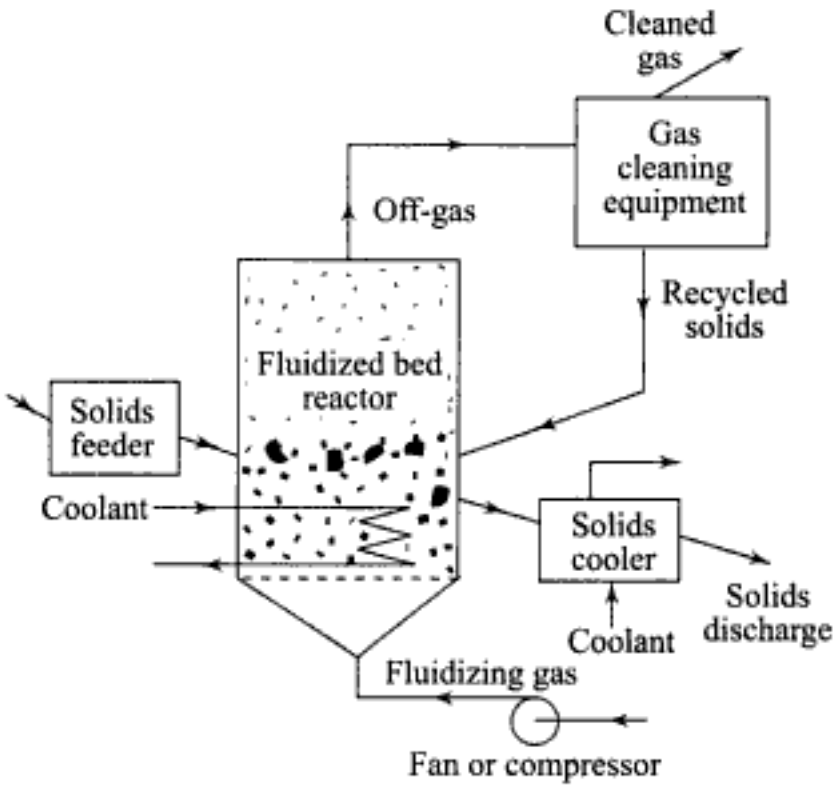
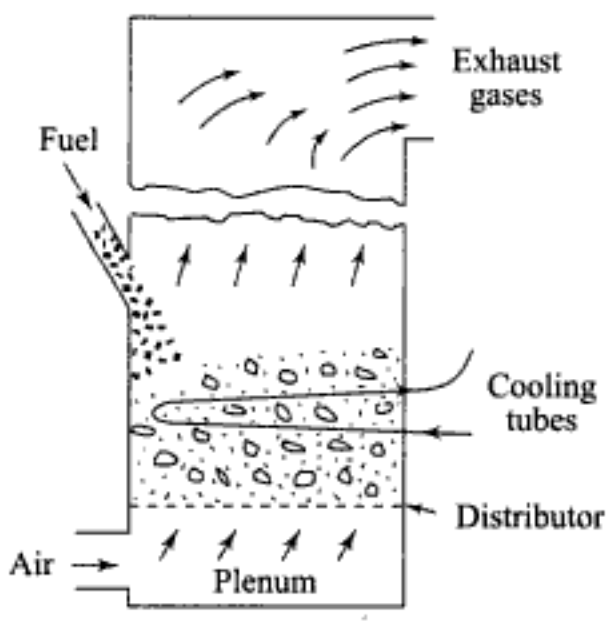
fuel, about 1 or 2%, so that the inert particles carry the heat away from the burning particles, thus keeping their temperature below that at which ash melts. The temperature of the bed when burning solid fuels is commonly controlled to about 850 °C. So, heat is required to be removed from the bed by installing cooling tubes. The bed temperature during start-up should be raised to the solid fuel ignition temperature, which may vary from 450 °C to 500 °C depending on the type of fuel by burning fuel gas or oil in the bed (Fig. 5.44). The initiated input to the bed by burning gas or oil is gradually reduced till the main fuel starts to deliver heat and self-sustaining combustion is established.



(a)



(b)

Fig. 5.43 *Fluidized bed combustor with immersed cooling surfaces*

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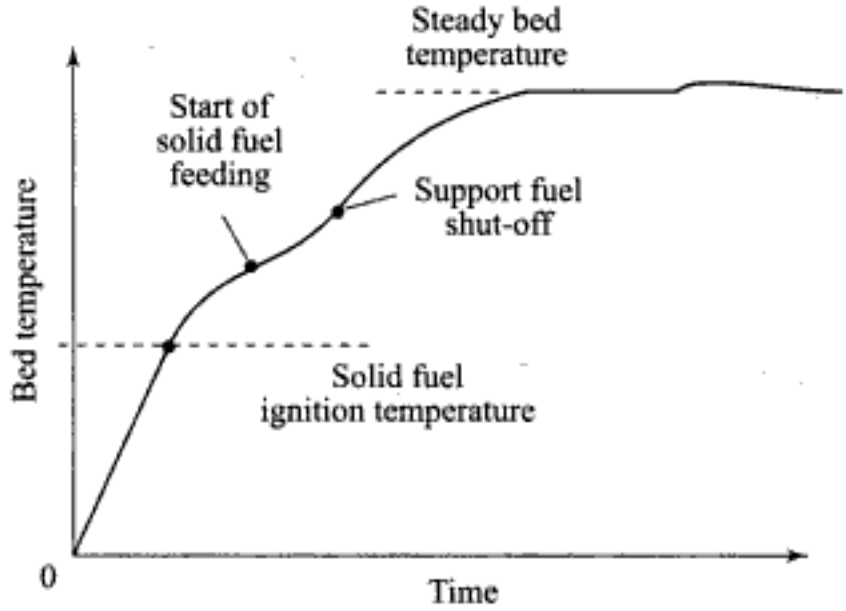

Fig. 5.44 *Start-up transient of a fluidized bed combustor*

Flue gas desulphurization is achieved by adding a sorbent such as limestone or dolomite to the fluidized bed, where sulphur is absorbed in a solid form. This sorbent is used as the bed material into which fuel is fed. A combustion temperature of 850 °C is about the optimum for absorption of sulphur oxides by the lime, and the residence time of lime in the bed is long. The amount of sorbent required to absorb about 90% of the sulphur in the fuel needs a calcium-sulphur molar ratio of about 3 (Howard, 1989).

Fluidized bed systems are generally smaller for a given thermal output than conventional equipment. The firing rate (MW/m^2) of a bubbling fluidized bed boiler having cooling tubes in the bed can be up to 50% larger. Fluidized beds can burn a wide variety of fuels quite efficiently. Indeed, with some low-grade fuels and wastes, e.g. colliery tailings, fluidized bed combustion is the only way they can be burned satisfactorily. The relatively good mixing of particles and gas promoted by bubbling action provides good conditions for both combustion and heat transfer.

5.9.4 Combustion of Fuel Particles in a Fluidized Bed

The sequence of events and temperature history of a single burning particle of solid fuel in a fluidized bed at 850 °C is shown in Fig. 5.48. Initially, the moisture is given off and the temperature of the solid fuel particle rises as it is heated by inter particle convection, gas convection and radiation. The temperature of the fuel particle soon reaches a value at which volatile matter is released. The residue left after devolatilization is char, which is essentially porous carbon with some ash bound to it, with the degree of porosity depending on the type of fuel. Once the ignition temperature is reached, the char begins to burn. As seen in Fig. 5.45, the time required to burn the particle out is much longer than the time required to devolatilize. The mass and size of the char particle diminishes with the progress of combustion, as well as by attrition and fragmentation till the particle becomes small enough to be elutriated from the bed.

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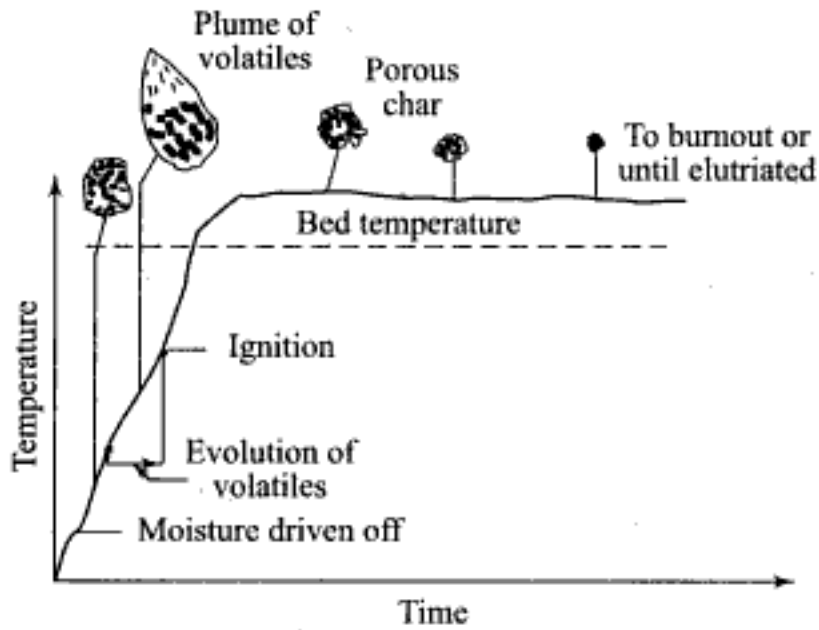

Fig. 5.45 *Temperature history of a single fuel particle introduced into a fluidized bed combustor*

5.9.5 Mechanism of Combustion of Char Particles

The general principle of coal combustion has been discussed earlier in Section 5.2. The rate of loss of mass of the burning particle per unit surface area is termed "specific burning rate", r_s (kg/m^{2}s). The faster the burning rate, the shorter the time required for the char particle to burn out.

A char particle whose surface is at sufficiently high temperature will burn rapidly if there is a large mass flow rate of oxygen towards it. There are two resistances which impede the rate at which the char reacts, *viz.*, the diffusional resistance to mass transfer of oxygen, R_g, and the chemical resistance, R_c, depending on the surface temperature or the chemical kinetics, as we have discussed in Section 5.3. To speed-up the burning rate of char, these resistances must be reduced.

Mass transfer rate of oxygen The mass transfer rate of oxygen towards the fuel surface may be characterized by Sherwood number, Sh, given by

$$\text{Sh} = \frac{k_g d_c}{D_g}$$

where k_g is the mass transfer coefficient, d_c is the diameter of the char particle and D_g is the diffusivity of oxygen. Chakraborty and Howard (1981) suggested the following expansion

$$\text{Sh} = 2e + 0.69\ \text{Re}^{1/2}\ \text{Sc}^{1/3} \tag{5.35}$$

where Re is the carbon particle Reynolds number

$$\text{Re} = \frac{\rho_g U d_c}{\mu_g}$$

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and Sc is the Schmidt number

$$\text{Sc} = \frac{\mu_g}{\rho_g d_c}$$

The specific burning rate of carbon may be written as

$$r_s = \lambda\, k_g\, (c_b - c_s) \tag{5.36}$$

where λ depends upon which particular reaction occurs at the surface, c_b is the oxygen concentration in the bulk particulate phase of the bed and c_s is the oxygen concentration at the particle surface. For the reaction,

$$C + O_2 \rightleftharpoons CO_2$$

$\lambda = 3/8$, and for the reaction,

$$C + \frac{1}{2}O_2 \rightleftharpoons CO$$

$\lambda = 3/4$. The mass transfer coefficient, k_g, is the inverse of the resistance, R_g, to mass flow of oxygen:

$$R_g = \frac{1}{k_g}$$

The diffusional resistance R_g can be reduced by increasing the Reynolds number or the velocity of the gas.

Chemical kinetic rate For a first order reaction, the specific burning rate r_s may be expressed as

$$r_s = kc_s \tag{5.37}$$

where k is the reaction rate coefficient and c_s is the oxygen concentration at the carbon surface.

The reaction rate coefficient k is given by Arrhenius equation,

$$k = k_o\, e^{-E/RT}$$

The inverse relationship between the reaction rate coefficient k and the surface kinetic resistance R_c is

$$R_c = 1/k_c$$

If the chemical kinetics controls the rate of reaction, the burning rate may be increased by increasing the surface temperature of carbon.

Combustion control by combined influence of mass transfer and chemical kinetics The oxygen concentration at the char surface, c_s, is not known directly, so if it is eliminated between Eqs (5.36) and (5.37), the resulting expression for specific burning rate is

$$r_s = Kc_s \tag{5.38}$$

$$\text{where } \frac{1}{K} = \frac{1}{\lambda k_g} + \frac{1}{k} \tag{5.39}$$

which is similar to Eq. (5.13).

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During the residence time of burning carbon particle within a fluidized bed, both chemical kinetic and diffusional resistances influence the burning rate. However, it has been observed that for particles of size 1–3 mm, combustion is mainly controlled by diffusion of oxygen to the carbon surface, whereas for particles smaller than 1 mm, combustion is controlled primarily by chemical kinetics (Ross and Davidson, 1981). Larger char particles glow more brightly than the bulk of the bed with higher surface temperature and diffusion controlling the reaction. Finer particles are seen to glow with almost the same brightness as the bed with their temperature only slightly above that of the bed. Thus, the only way to speed up the burning rate of the fines is to raise the bed temperature.

5.9.6 Circulating Fluidized Bed Combustion System

The furnace is divided into three distinct zones: (1) Lower zone (below the secondary air entry). 2. Upper zone (above the secondary air entry), and (3) Hot gas/solid separator (Fig. 5.46).

The lower zone in the furnace is fluidized by primary combustion air, which constitutes about 40–80% of the stoichiometric air required for the coal feed: coal and sorbent (limestone or dolomite) as well as the unburnt char from the hot cyclone are received in this section. Devolatilization and partial combustion of char occur in this zone, which is oxygen deficient. So, to protect the boiler tubes from corrosion, this zone is refractory lined.

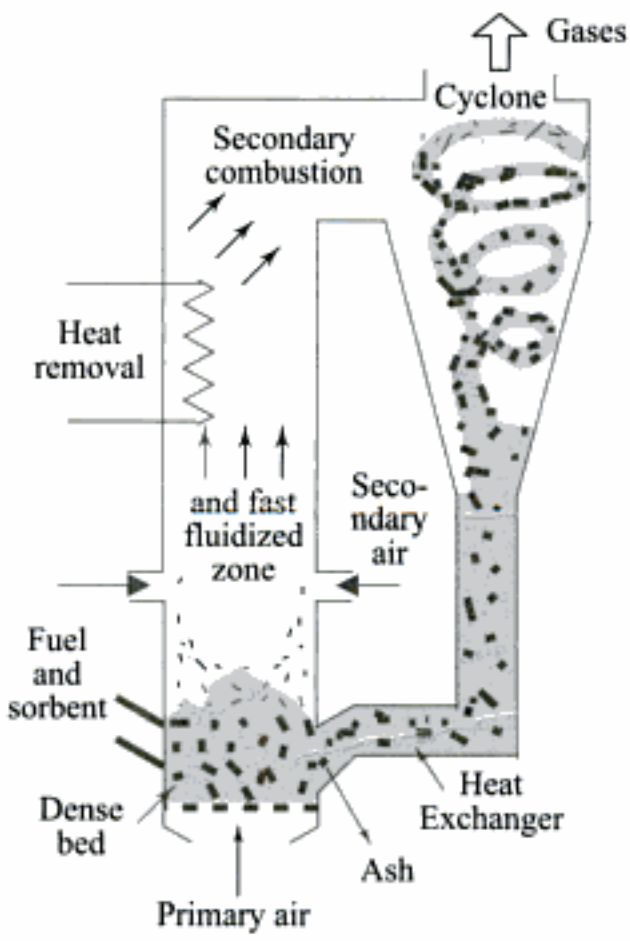

Fig. 5.46 *Fast or circulating fluidized bed combustion systyem*

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As mentioned earlier, the lower zone of a CFB furnace is much denser than the upper zone. Thus, it serves also as an insulated storage of hot solids, providing the CFB boiler with a thermal "flywheel". When the boiler load increases, the proportion of primary to secondary air is increased. As a result, a greater amount of hot solids is transported to the upper zone of the furnace and the solid circulation rate increases. When the boiler load decreases, the reverse happens and the solid circulation rate decreases.

Char particles transported to the upper zone are exposed to an oxygen rich environment, where most of the combustion occurs. The upper zone is much taller than the lower zone. Unburned char particles that are entrained out of the furnace go around the refractory-lined cyclone. The residence time of char particles within the cyclone is short and the oxygen concentration is the lowest there. So the extent of combustion inside the cyclone is small compared to the rest of the combustion loop. However, carbon monoxide and the volatiles often burn in the cyclone.

Basu and Halder (1989) presented a model for estimation of the burning rate of a single char particle in a fast bed. The ratio of fixed carbon to volatile matter in fuel, the attrition rate, the intrinsic reactivity of char, the particle size and temperature, swelling and fragmentation, and finally, the temperatures of coal ignition and ash agglomeration affect the combustion characteristics in a CFB.

5.9.7 Pressurized Fluidized Bed Combustion (PFBC)

If a coal-fired fluidized bed combustor is operated at elevated pressure, the products of combustion can be expanded through a gas turbine to produce electricity. The products of combustion have to be sufficiently clean for a gas turbine to accept without excessive erosion, corrosion or fouling of the turbine. Pressurized fluidized bed combustion (PFBC) has been under extensive research and development, not only because of its potential as a coal fired gas turbine but also when used in conjunction with a steam plant (Fig. 5.47), a significantly higher efficiency of electricity generation is possible than that from either a gas turbine or steam turbine plant alone. The thermodynamic aspects of combined cycle power generation have been discussed in great details in Chapter 3.

The first PFBC plants with an electrical power of about 80 MW were almost simultaneously installed by ABB Carbon at Värtan (2 units) in Sweden, Escarton in Spain and TIDD in the USA. Their details are as follows.

	Värtan	Escarton	TIDD
Put into operation	1991	1991	1991
Electrical power	2 × 220 MW (th)	79	71
Gas turbine inlet	830 °C, 12 bar	840 °C, 12 bar	830 °C, 12 bar
Steam condition	137 bar, 530 °C	94 bar, 513 °C	90 bar, 496 °C
Coal	Bituminous	Lignite	Bituminous

Operational success of these demonstration units gave an increased confidence in this technology. The Värtan PFBC plant in Stockholm is a

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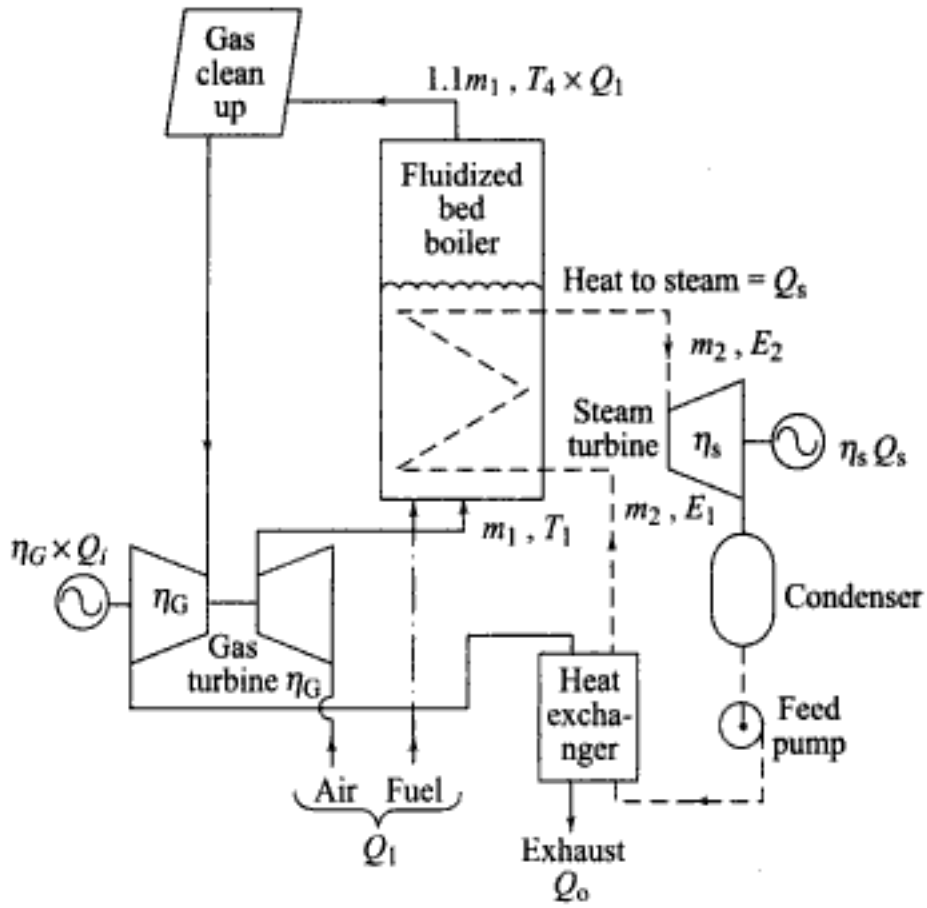

Fig. 5.47 *A combined cycle plant with a pressurized fluidized bed combustor (PFBC)*

cogeneration unit, which consists of two modules, each containing one boiler with gas turbine and associated fuel and ash systems (Fig. 5.48). The fuel is a mixture of coal, sorbent and water supplied as a paste by means of six piston pumps. Coal is crushed to 0–6 mm size and the sorbent (dolomite) to 0-3 mm size, before they are mixed in water to form the paste.

Figure 5.50 shows a simplified flow diagram of the TIDD plant at Ohio, USA, where the PFBC operates at 12 bar increasing the gas density and reducing the bed area for a given heat release. To further improve the overall cycle efficiency, there is a growing interest in combining PFBC with partial or mild gasification of coal to take advantage of a higher gas turbine inlet temperature. This is known as the second generation PFBC, or the Topping Cycle. Mild gasification is a devolatilization process designed to produce a series of alternate fuels by decomposing coal into simpler components at relatively mild temperatures (540°C–650°C) and pressure (1–2 bar). The coal heated by limited air gets pyrolyzed to yield a solid carbon char (20–40% of feed), a complex of hydrocarbon liquids and hydrocarbon gas. The char is combusted in a conventional fluidized bed boiler to raise steam and hence electricity. The gaseous and liquid hydrocarbon products are cleaned and then combusted in a gas turbine, also producing electricity. The plant (Fig. 5.50) is similar to the British Coal Topping Cycle (Fig. 3.24) and the CFBC topping cycle (Fig. 3.25). The significant advantage of partial gasification over total gasification is that devolatilization and gasification of coal are achieved relatively easily, whereas

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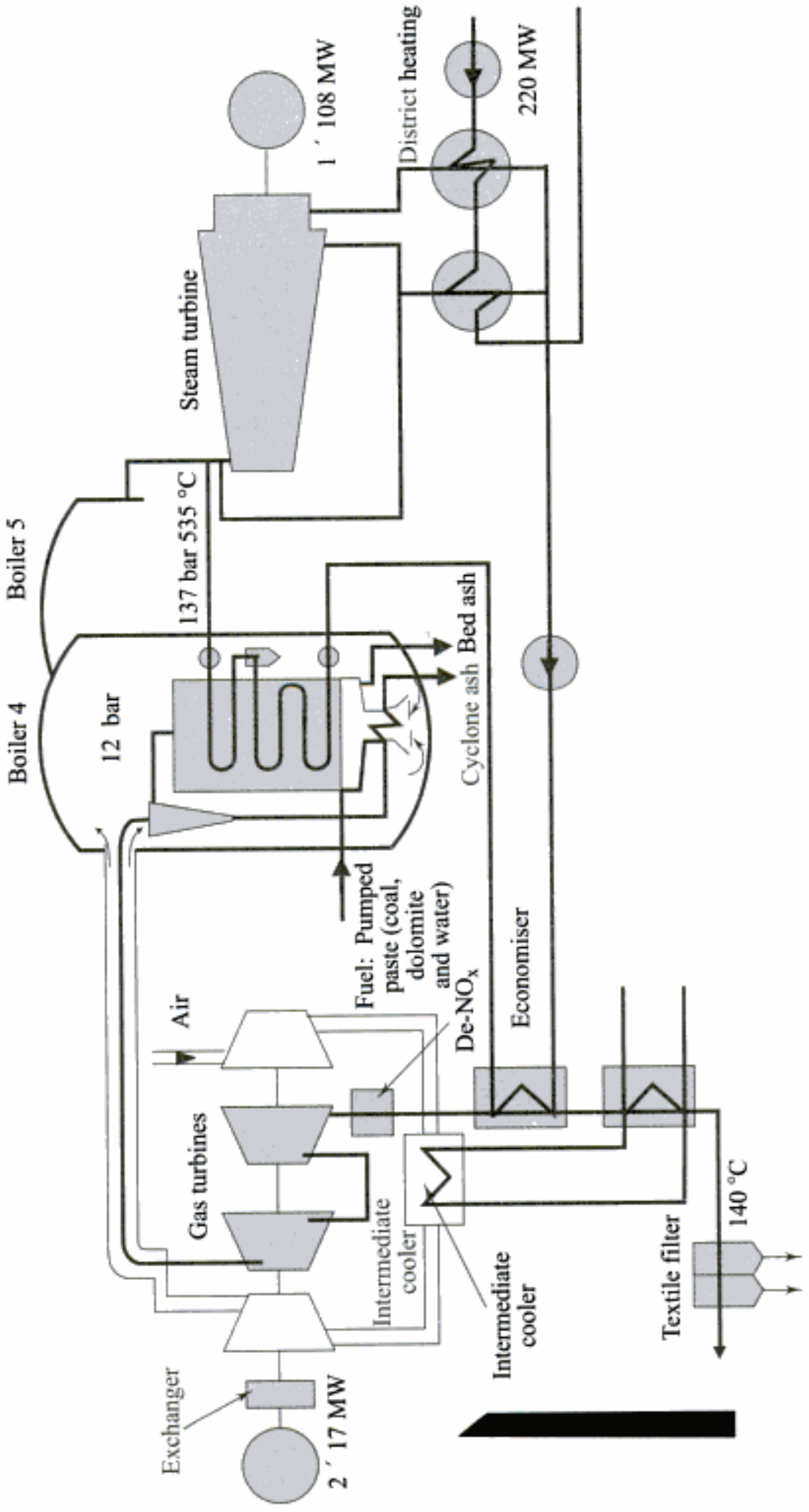

Fig. 5.48 *Simplified Flow Diagram for the PFBC Plant at Värtan (Sweden)*

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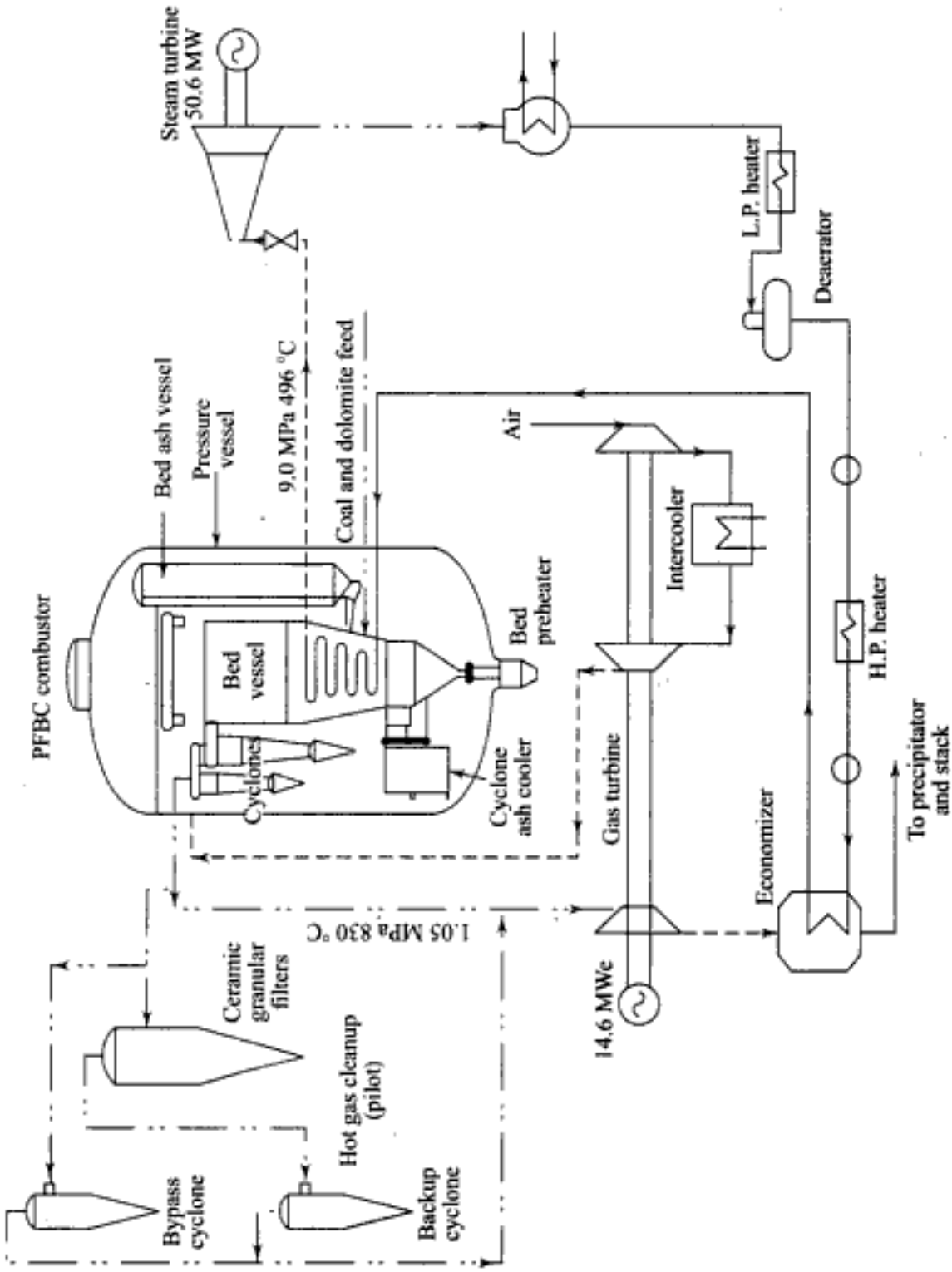

Fig. 5.49 *TIDD PFBC simplified plant cycle*

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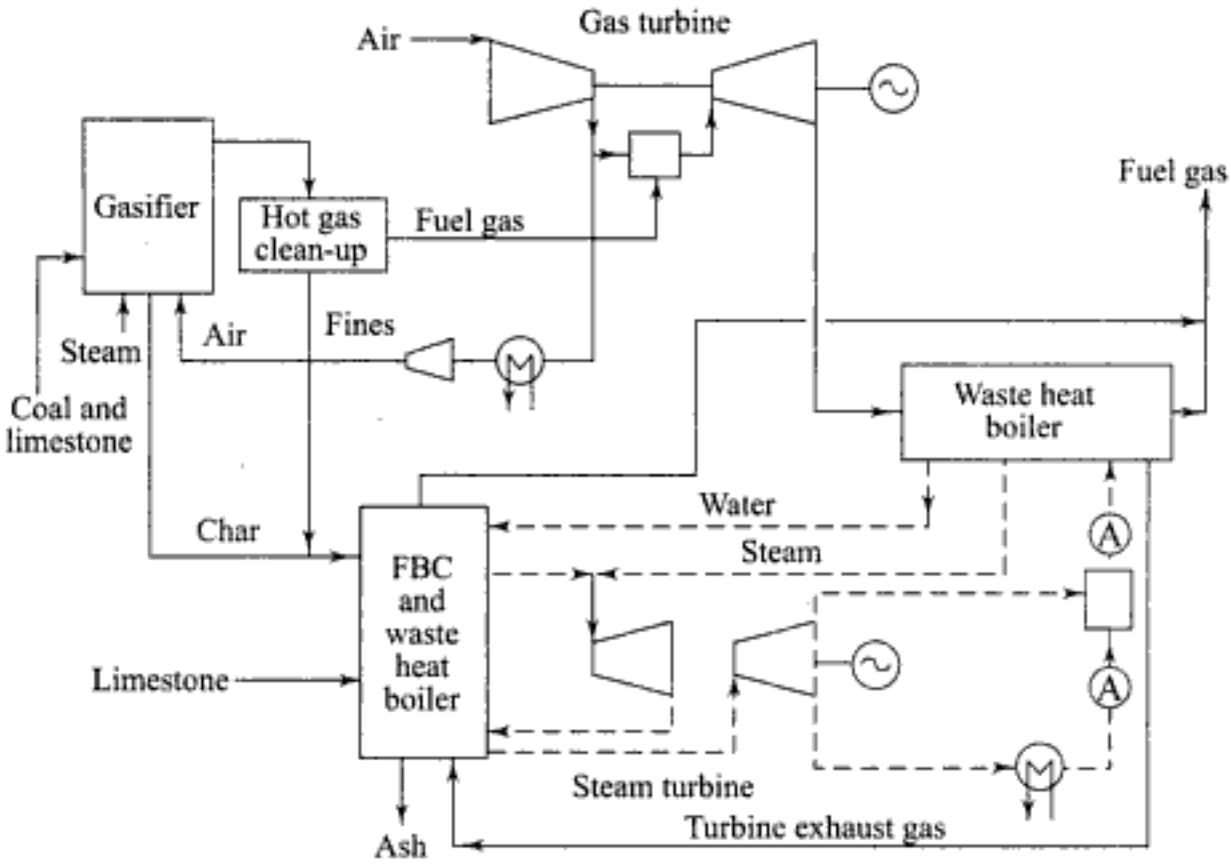

Fig. 5.50 *Outline of the Topping Cycle*

the total gasification of the char requires more energy and time. Consequently, by partially gasifying and then combusting the char in a fluid bed combustor to raise steam, an optimum can be achieved between the gasifier and the combustor sizing as well as the steam and gas turbines. Also, advantage can be made of higher GT inlet temperatures.

Table 5.3 shows the PFBC projects in Japan and Fig. 5.51 gives the flow diagram of the Wakamatsu plant, where the PFBC boiler is housed inside a thick pressure vessel and the combustion gases are de-dusted by cyclone and ceramic filter. The outline of the 85 MWe Tomatoh-Azuma (Atsuma-Cho) PFBC plant is given in Fig. 5.52. Coal fed together with limestone is burnt at an elevated pressure. Desulphurization of the flue gas takes place along with combustion. After being de-dusted in cyclones and ceramic filters, the flue gas, still at high pressure, expands in the gas turbine, generating electricity. The exhaust gas at about atmospheric pressure flows through SCR (Selective Catalytic Reduction), where NO_x is removed and SGC (Stack gas cooler), where heat is removed, before leaving through the stack. The boiler is of bubbling fluidized bed drum-type with membrane water wall. The steam turbine is a single casing configuration which accommodates HP/IP and LP turbines within one cylinder. Gas and steam turbines generate electrical power in a ratio of 15 per cent and 85 per cent respectively.

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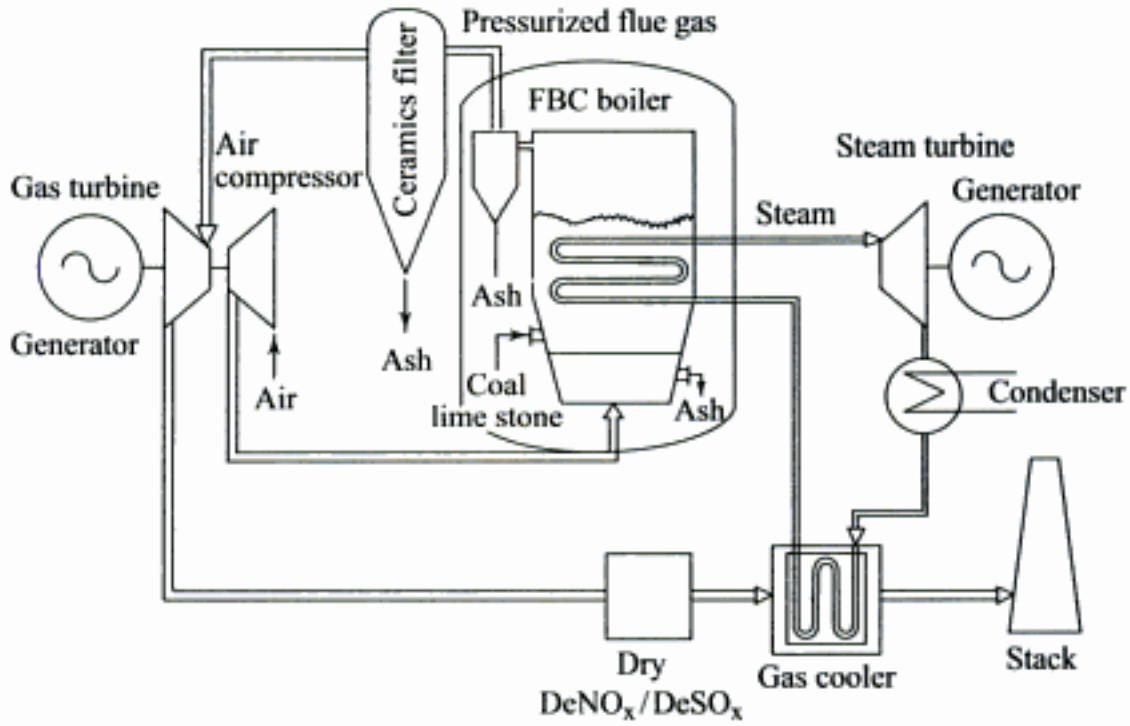

Fig. 5.51 *Simplified Cycle of Wakamatsu Plant, Japan*

Table 5.3 *PFBC Projects in Japan*

	MW	*Commissioning*	*Manufacturer*
Wakamatsu	71	1994 Nov	IHI/ABB
Tomatoh-Azuma	85	1996 Oct	MHI
Karita	350	1997 July	IHI/ABB
Ohsaki	250 × 2	1999 Mar	Hitachi

The schematic diagram of the 250 MWe PFBC system at Ohsaki, Japan is shown in Fig. 5.52, while Table 5.4 gives its principal features.

Table 5.4 *Principal Specifications of PFBC Plant at Ohsaki,Japan*

Gross electrical output :	*250 MW mty × 2 (ST : 210 MWe, GT : 40 MWe) × 2*		
Boiler	:	Pressurized bubbling fluidized bed combustion regenerative once-through type	
		Steam output at MCR	: 520 t/h × 2
		Bed temperature	: 865°C
		Combustion pressure	: 10 bar
		Coal and limestone feed system	: Wet paste feed
Gas turbine	:	Single shaft simple open cycle	
Steam turbine	:	Tandem compound regenerative condensing type	
		Steam condition	: 166 bar, 566/593°C
Emission control	:	SO_x	: In-site limestone desulfurization
		NO_x	: NH_3-injection SNCR and SCR
		Particulate	: Cyclones and Ceramic filters
Stack		Height 200 m.	

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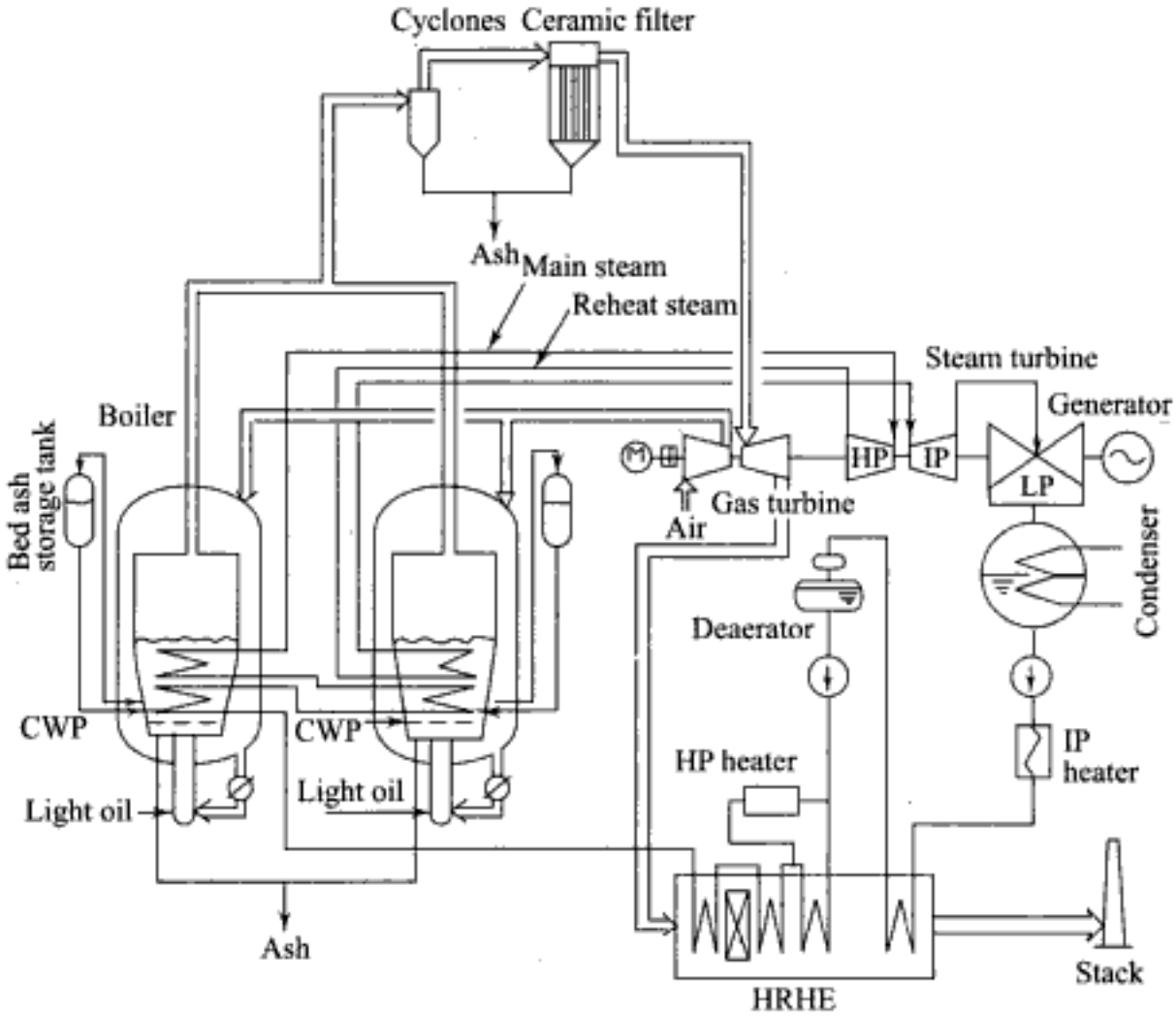

Fig. 5.52 *Schematic diagram of the 250 MWe PFBC system at Ohsaki, Japan*

Fuel is fed into the bed as paste. Raw coal is crushed and screened to less than 6 mm and fed into a mixer with limestone of 3 mm top size. With water added, the mixer makes a coal-water paste (CWP), which is then pumped into the pressurized fluidized bed through fuel nozzles using piston pumps. Hot flue gas leaving the bed is cleaned by cyclones and ceramic filters and is then delivered to the gas turbine. The heat recovery heat exchanger (HRHE) recovers heat from the GT exhaust of about 430 °C and heats up the boiler feedwater.

The 350 MW Karita Unit 1 (Fig. 5.53) at Fukuoka, Japan is the largest PFBC power plant in the world. The schematic PFBC hot loop is shown in Fig. 5.54 and an advanced PFBC process diagram is shown in Fig. 5.55.

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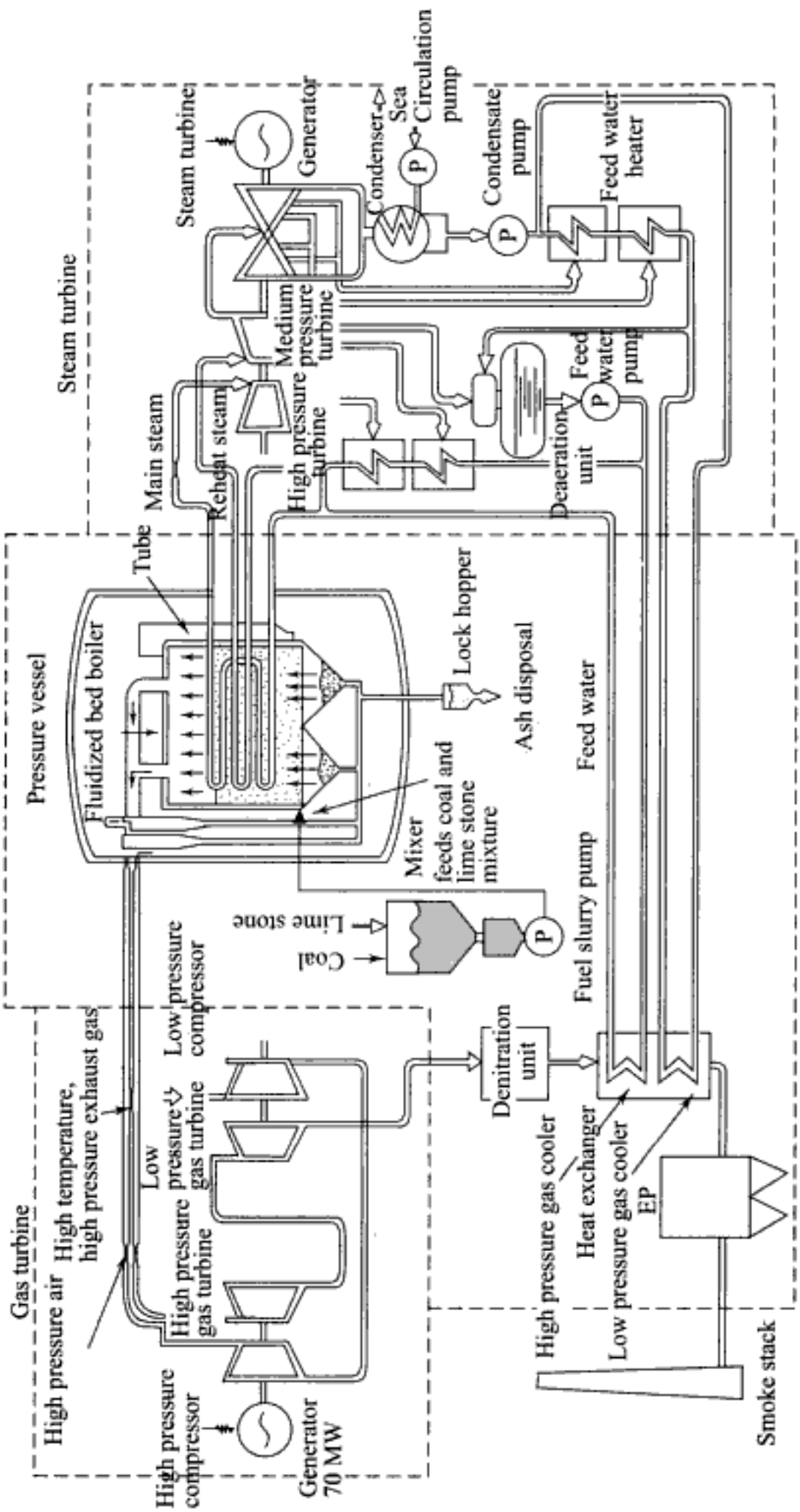

Fig. 5.53 *350 MW Karita Plant, Japan*

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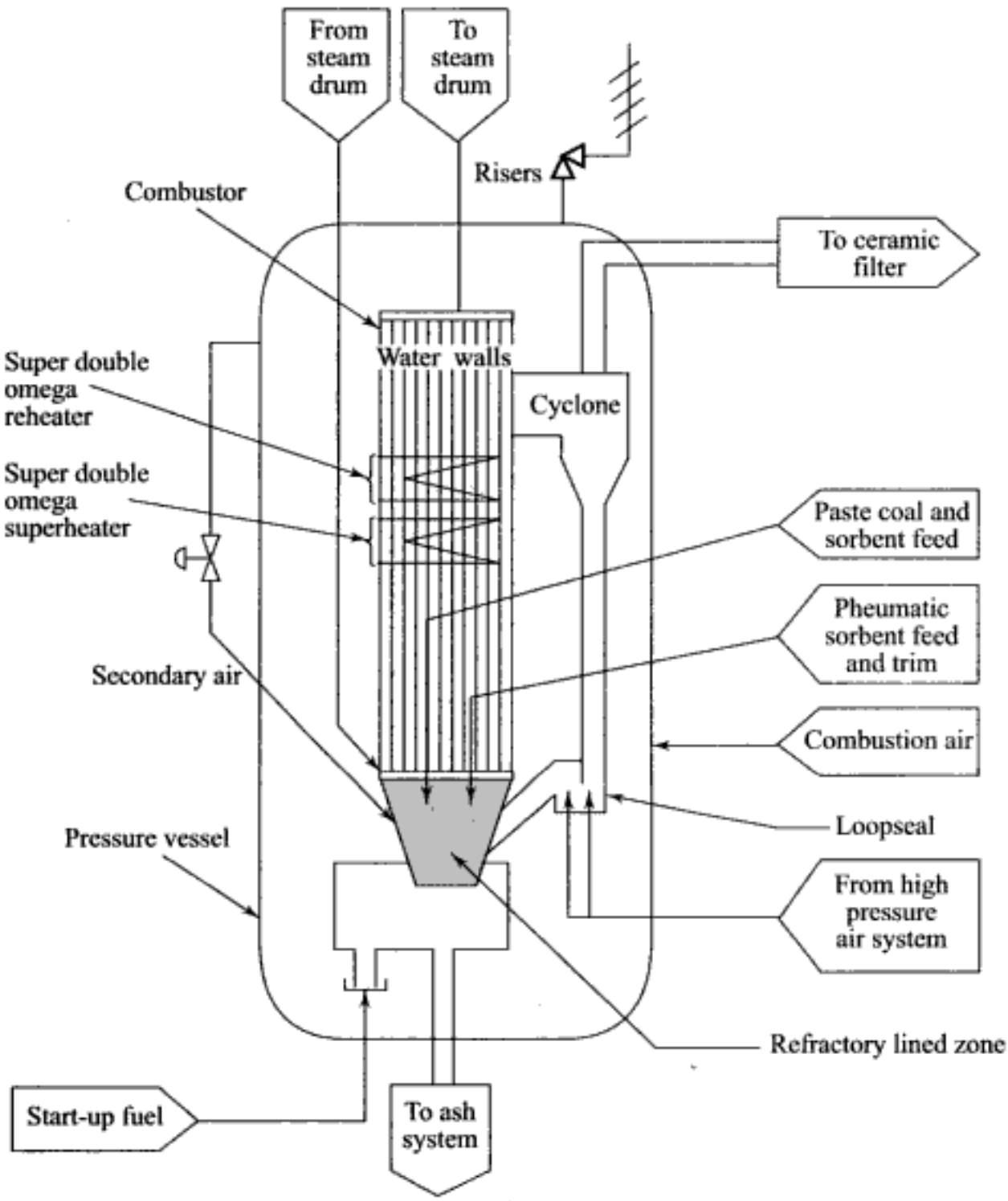

Fig. 5.54 *Schematic PFBC hot loop*

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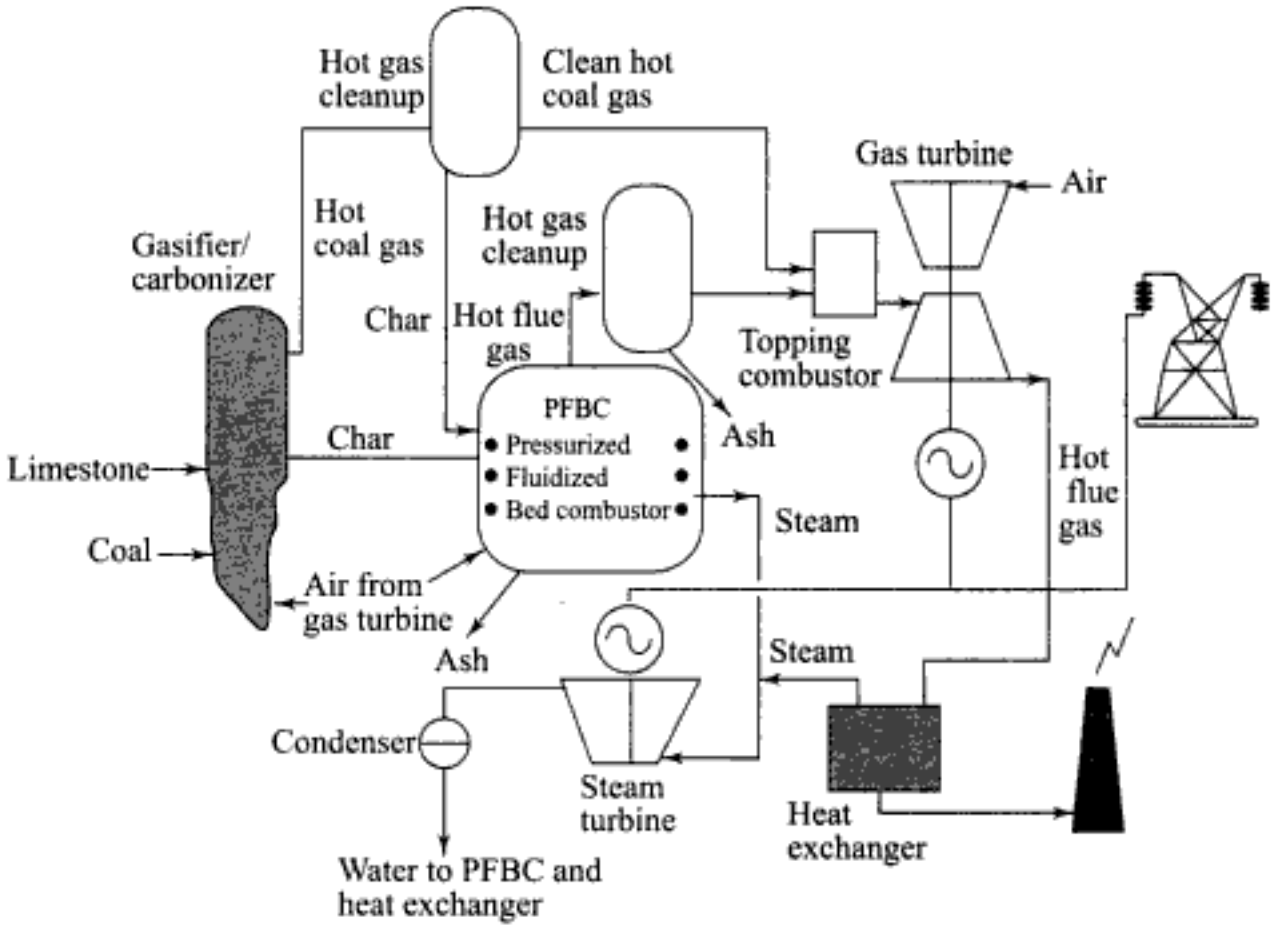

Fig. 5.55 *Advanced PFBC system*

5.10 COAL GASIFIERS

The gasification of coal is a very old technology. Before natural gas was introduced in the market, coal gasification was used to produce fuel gas (town gas) for distribution in urban areas. The basic principles of coal gasification have been discussed in section 4.8.1.

In auto thermal coal gasification, partial combustion of coal takes place to generate the heat required for the gasification process, the overall reaction of which is endothermic. The oxygen required for combustion is supplied to the gasifier by injecting air or oxygen into it.

Gasification process can, therefore, be classified as:

(1) Air-blown gasifiers, which produce a gas with a low calorific value, typically 5000–6000 kJ/kg.

(2) Oxygen-blown gasifiers, which produce a gas with a medium calorific value, typically 15,000 kJ/kg, approximately one-third of the calorific value of natural gas.

The typical gas composition, relative heating value and adiabatic flame temperature of natural and coal-derived gases are given in Table 5.5. Depending on the flow pattern in the gasifier itself, there are three main types of gasifiers:

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Table 5.5 *Typical gas composition, in % volume*

% volume	Natural gas	Coal gasification	
		Oxygen-blown	Air-blown
CO	—	60	25
H_2O	—	30	18
CO_2	—	03	07
N_2	10	02	49
CH_4	90	05	01
Relative heating value	1 (reference)	1/3	1/9
Adiabatic flame temperature	2100 °C	2300 °C	1900 °C

1. Fixed bed gasifiers with counter-current flow
2. Fluidized bed gasifiers with bubbling or circulating bed
3. Entrained bed gasifiers with co-current flow

The gasifiers which are closer to large-scale commercial application for combined cycle power generation are either the fixed bed type (British Gas/Lurgi slagging gasifier) or the entrained type (Texaco, Shell. Dow, Prenflow), both oxygen-blown. Table 5.6 and Fig. 5.56 to 5.58 show the above types of gasifiers.

Table 5.6 *Different types of gasifiers*

British gas/Lurgi	Texaco	Shell	Prenflow	Dow	KRW (Kellogg Rust Westing house)	High Temperture Winkler
Oxidant	Oxygen	Oxygen	Oxygen	Oxygen	Oxygen	Oxygen/ air	Oxygen/ air
Coal gas value	Medium	Medium	Medium	Medium	Medium	Medium/ low	Medium/ low
Type of gasifier	Fixed bed	Entrained bed	Entrained bed	Entrained bed	Entrained bed	Fluidized bed	Fluidized bed
Flow pattern	Counter curre-ent flow	Cocurrent flow	Cocurrent flow	Cocurrent flow	Cocurrent flow	Bubbling or circulating bed	Bubbling or circulating bed
Raw gas temperature (°C)	500–600	1400–1700	1400–1700	1400–1700	1400–1700	850–900	850–900
Gasifier pressure	Compatible with gas turbine requirement						
Raw gas cooling	No	Yes	Yes	Yes	Yes	Yes	Yes

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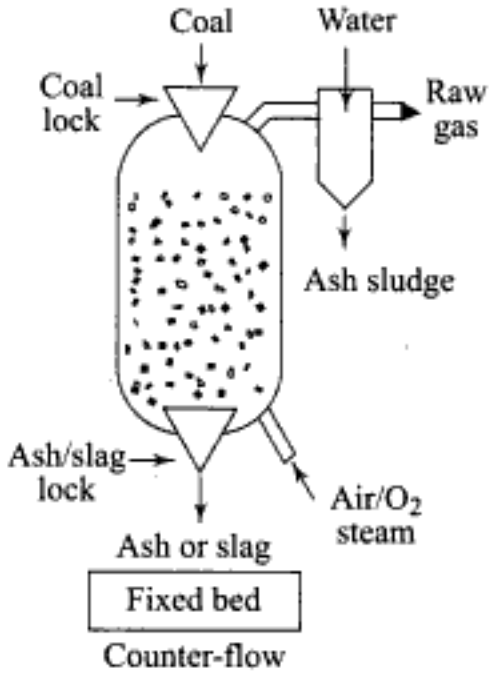

Fig. 5.56 *Fixed bed counterflow gasifier*

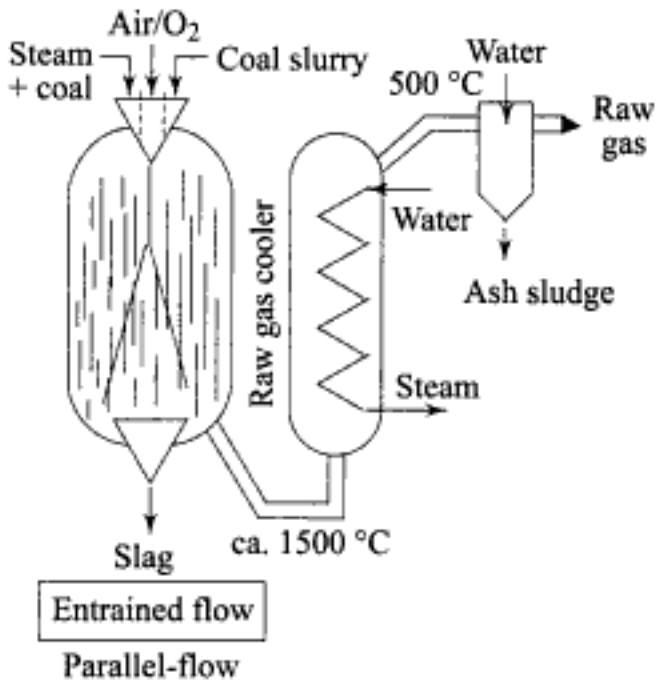

Fig. 5.57 *Entrained parallel flow gasifier*

Gasifiers can also be classified according to whether they operate at

1. atmospheric pressure, or
2. pressure above atmospheric.

Pressurized gasifiers are of interest for combined cycle plants since the gas pressure at inlet to the gas turbine must be at least 20 bar. So the gasifiers for combined cycle plants operate at a pressure between 20 to 30 bar.

By contacting coal, oxygen or air, and steam in a reactor at elevated temperatures and pressures, a product gas comprising CO, H_2, CH_4, CO_2, H_2O (g), N_2, H_2S and other trace gases (HCN, HCl) is liberated. The reactions within the gasifier are complex (see Section 4.8.1) and the basic ones are:

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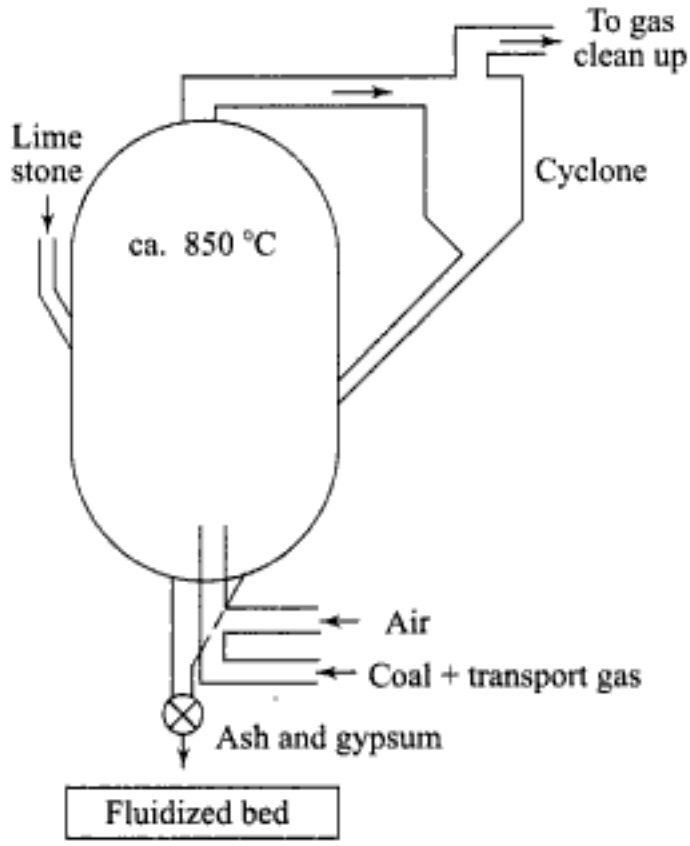

Fig. 5.58 *Fluidized bed gasifier*

1. $C + \frac{1}{2}O_2 \longrightarrow CO$ (partial combustion) – 124 kJ/mole (exothermic)
2. $C + O_2 \longrightarrow CO_2$ (complete combustion) – 406 kJ/mole (exothermic)
3. $C + H_2O \longrightarrow CO + H_2$ (gasification) + 119 kJ/mole (endothermic)
4. $C + CO_2 \longrightarrow 2CO$ + 162 kJ/mole (endothermic)
5. $CO + H_2O \longrightarrow CO_2 + H_2$ (water–gas shift) – 42 kJ/mole (exothermic)
6. $C + 2H_2 \longrightarrow CH_4$ (methanation) – 87 kJ/mole (exothermic)

Carbon in the coal reacts with oxygen, steam and carbon dioxide as shown in the above equations. The products of these reactions then react with one another.

Fixed bed (sometimes also called moving bed) gasifiers operate at 500–600 °C, while the entrained bed gasifiers operate at 1500 °C. Therefore, the latter are equipped with gas coolers to generate saturated or superheated steam, which is used in a combined cycle plant to generate more power.

The gas produced in a fixed bed gasifier contains tars and phenol which must be separated out and recycled to the gasifier. This is not true with an entrained bed gasifier because of the much higher process temperature which cracks the heavy hydrocarbon (tars and phenols).

Coal gasification is particularly interesting from the point of view of emission control. It is far easier to obtain a very clean fuel by cleaning fuel gas than by cleaning the combustion gas from a coal-fired boiler because:

1. The volume flow rate of coal gas is less than 1% of that of the exhaust gas from the boiler.
2. It is much easier to remove H_2S than SO_2.
3. Most other pollutants, i.e. heavy metals, chlorides and so on are also removed in the gasification process.

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4. The major byproduct of the desulphurization process is elemental sulphur, which is easy to transport and sell.
5. Most oxygen-blown gasifiers produce unleachable slag. Waste disposal is thus easier.

Coal gasification is the most promising method to produce a truly clean coal burning plant. Table 5.6 gives the characteristics of different types of gasifiers.

5.10.1 IGCC

The integrated gasification combined cycle (IGCC) has been discussed in Chapter 3. The synthetic gas (syngas) exiting a typical gasifier is cooled by generating high pressure steam which is subsequently expanded in a steam turbine generator to produce electricity. The syngas is then cleaned of particulates, by cyclones, filters, and/or water scrubbing, followed by heat exchange to cool the gas to near ambient temperature (Fig. 5.59). Sulphur compounds in the gas, primarily, H_2S and some carbonyl sulphide (COS), are removed, which is ultimately converted to elemental sulphur for sale. The cleaned fuel is finally sent to the combined cycle where it is combusted and expanded in the gas turbine to produce electricity. Hot exhaust gases from the gas turbine are also used to generate steam for power generation.

Typical gasification plant (Fig. 5.59) includes the following blocks:

1. Coal preparation to prepare coal to meet gasifier specification.

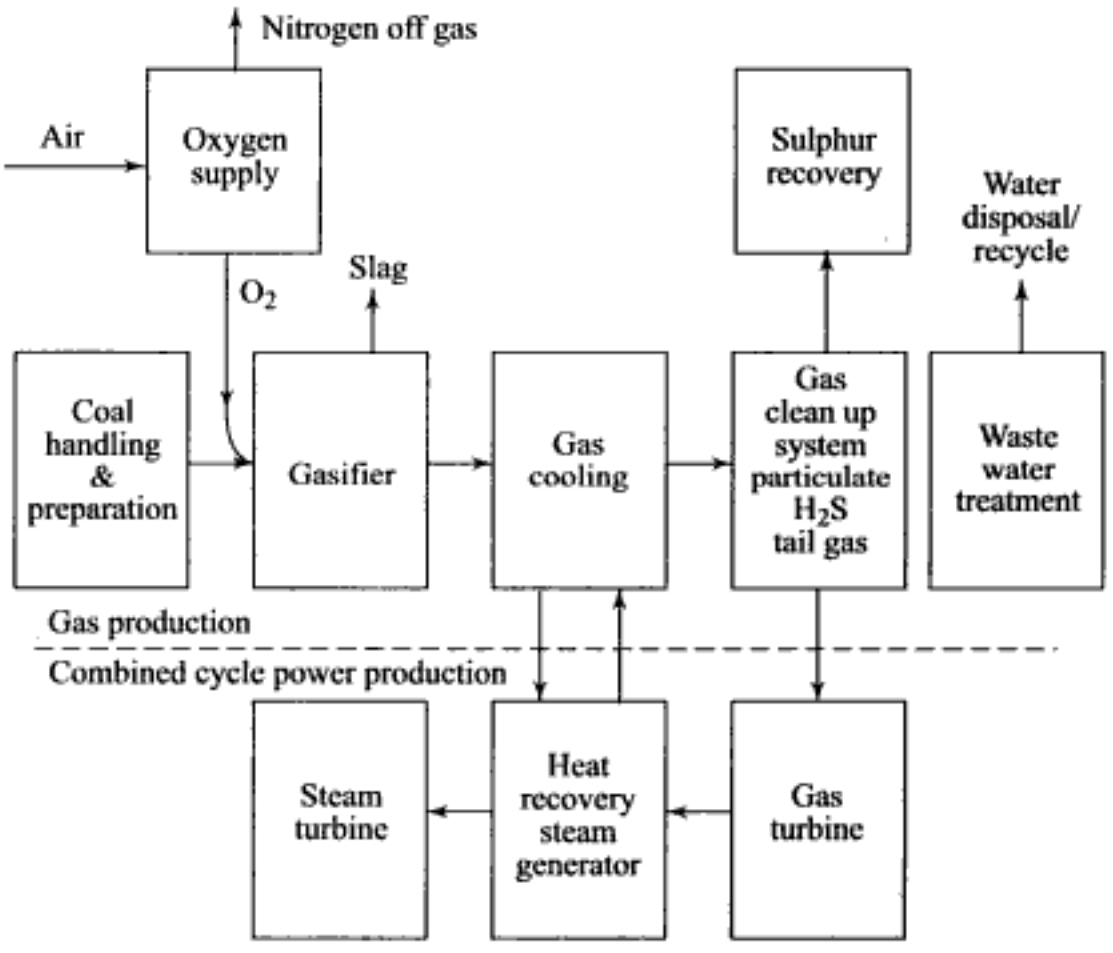

Fig. 5.59 *Basic IGCC plant arrangement*

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2. Air separation—to generate 95% pure oxygen for feeding the gasifier, and to generate gaseous nitrogen for injection into the gas turbine and for plant inert gas uses.
3. Coal gasification—to convert the coal to gas by partial oxidation.
4. Ash/slag removal—includes the removal of ash/slag from the gasifier, dewatering and disposal.
5. Syngas cooling—to remove sensible heat from the gasifier effluent gas and to facilitate gas cleaning.
6. Particulate removal—to remove fine particulates in the raw gas stream.
7. Acid gas removal—to remove sulphur bearing gases in the syngas.
8. Sulphur recovery—to convert the sulphur bearing gases to elemental sulphur.

The system incorporates one advanced gas turbine–generator similar to a General Electric MS 7001F gas turbine (inlet temperature 1260 °C), which, apart from the engine, includes auxiliaries such as fuel system, lube oil system and control system. The unit also includes one heat recovery steam generator (HRSG) having superheaters, reheater and economiser, one steam turbine–generator, a surface condenser, an integral deaerator, boiler feedwater pumps, and a stack.

The steam turbine normally produces 30–40% of the total energy output of an IGCC plant. Typical steam conditions and approach temperature differences are given below (Blamire, 1994).

HP steam pressure	101 bar
HP steam superheat temperature	540 °C
IP steam pressure	22 bar
Reheat temperature	540 °C
LP steam pressure	4 bar
Boiler pinch temperature difference	14 °C
Economiser approach	25 °C
Deaerator pressure	1.25 bar
Condenser pressure	0.047 bar
Stack temperature	120 °C

The gas turbine exhaust flows successively through the superheater, reheater, boiler and economiser sections.

A unique characteristic of an IGCC plant is its ability to take advantage of phased construction. Peaking turbines and/or a combined cycle plant of required capacities initially fuelled with fuel oil or natural gas may first be installed. It may then be followed by the addition of a coal gasification plant to refuel the combined cycle with coal-derived fuel gas. It has a higher overall efficiency and is more friendly with environment.

5.11 COMBUSTION OF FUEL OIL

Highly volatile petroleum fuels are too expensive for boiler firing. The heavy fuel oils, the residue left after refining of crude petroleum, are used. An oil burning installation is shown in Fig. 5.60.

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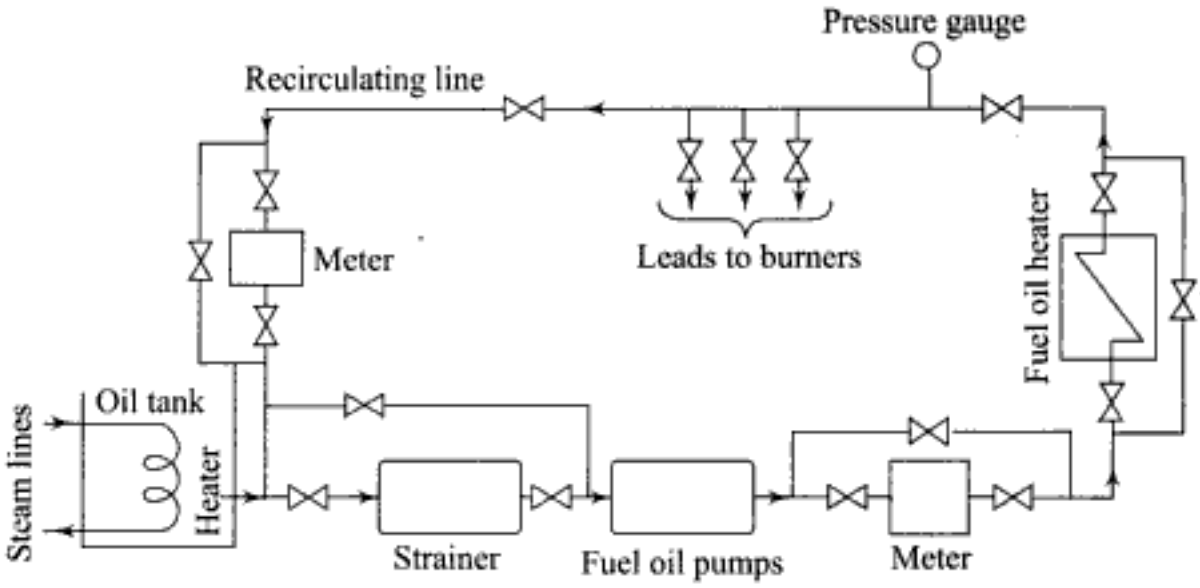

Fig. 5.60 *Schematic of an oil burning installation*

Because of the high viscosity of fuel oil, steam heating coils are installed in the oil tank to facilitate pumping. Oil pumps, mostly of the reciprocating type, receive oil from the strainers and discharge it to the burners through heaters and a meter. Oil is required to be heated, either by steam or electricity, to further reduce viscosity and improve atomisation. Oil temperature should vary between 65 °C to 115 °C. It should not exceed 115 °C because then the oil may vaporize and may even decompose or crack. Recirculation of oil is necessary initially to purge the line so that heavy oil does not collect and plug anywhere in the pipeline. Usually a light oil is used for half-an-hour or so when starting and stopping the boiler to wash the pipes clean of the heavy oil.

5.11.1 Mechanism of Fuel Oil Combustion

The sequential steps of spray combustion of fuel oil are:

1. Atomization
2. Evaporation
3. Mixing of evaporated spray with air
4. Combustion of spray

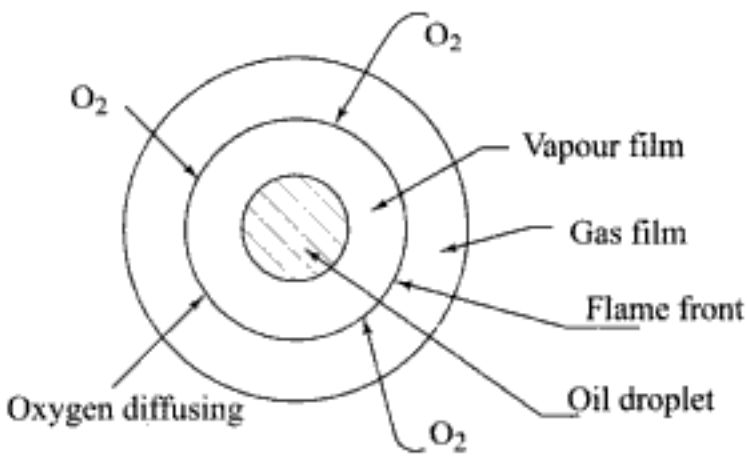

Fig. 5.61 *Combustion of an oil droplet*

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Immediately on being atomized, the oil vaporizes and a film of vapour surrounds the oil droplet (Fig. 5.61). Around the vapour film there is a gas layer through which oxygen diffuses. Reaction occurs at the flame front where the concentration or partial pressure of oxygen is zero. The combustion products diffuse out from the flame front to the bulk fluid. The flame front is an area of very rapid chemical reaction and is a boundary between burned and unburned gas. The more rapid the combustion, the thinner the flame front.

5.11.2 Oil Burners

The major parts of an oil burner are the atomizer, air register, and necessary oil lines with strainers and valves. Atomization of the liquid exposes more surface area per unit mass, and thus promotes ignition and completes combustion. It is accomplished mechanically or with steam or air. In twin-fluid burners, high pressure steam or air is used to break up oil drops into fine droplets, enabling better mixing with the air for combustion (Fig. 5.62).

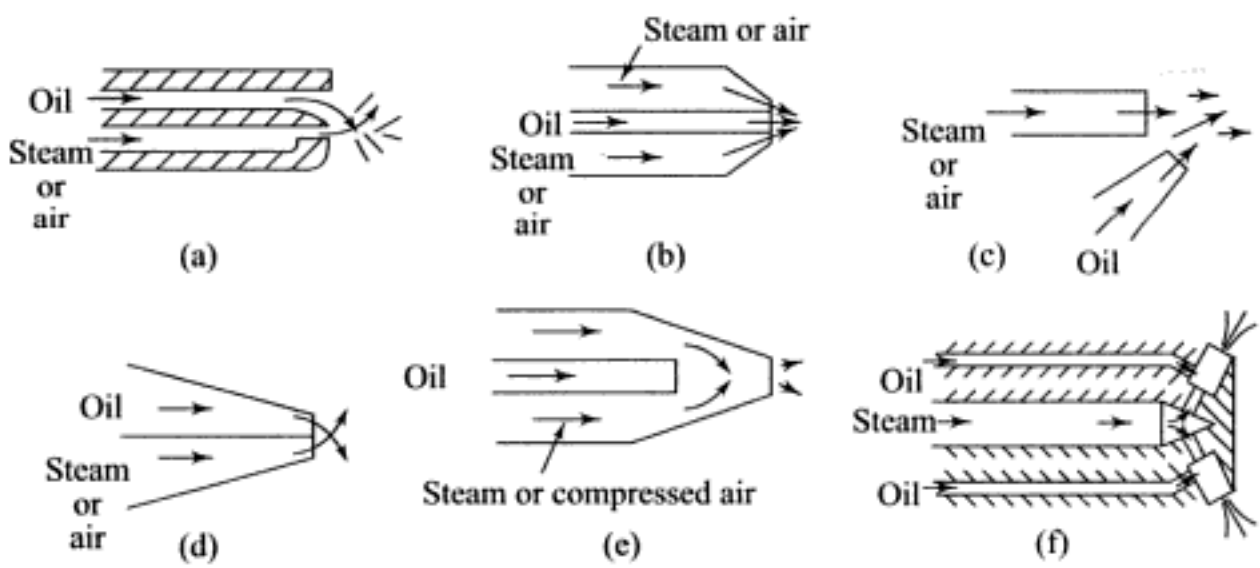

Fig. 5.62 *Steam- or air-atomised oil burners*

For large boilers, oil burners with pressure atomisation are suitable, in which oil is supplied under pressure (6–20 bar) and atomised by ejecting the oil through a small hole after passing through a swirl chamber (Fig. 5.63). A fine oil mist in the shape of a funnel forms in front of the nozzle outlet. The control of fuel oil consumption in pressure atomisers with load variation is done either by oil recirculation, part of the oil being returned to the oil tank from the swirl chamber through the return pipeline, or by means of a control piston which closes the tangential slots in the burner (Fig. 5.64).

Rotary cup atomisers are also important for the furnaces of large boilers (Morse, 1953). The oil flows through a hollow shaft into a rapidly rotating cup at the edge of which it is atomised under centrifugal force. The primary air is introduced by a fan around the outside of the cup and assists oil atomisation since it possesses swirl in the opposite direction of the rotation of the cup.

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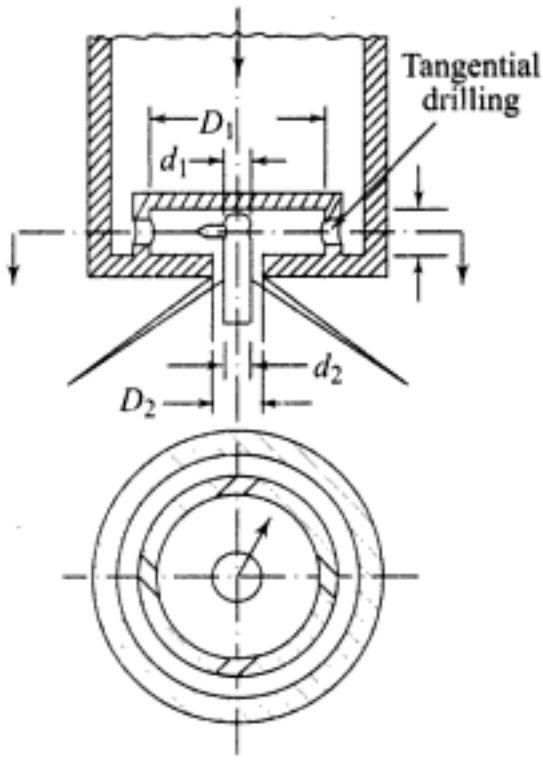

Fig. 5.63 *Pressure atomiser with tangential oil feed*

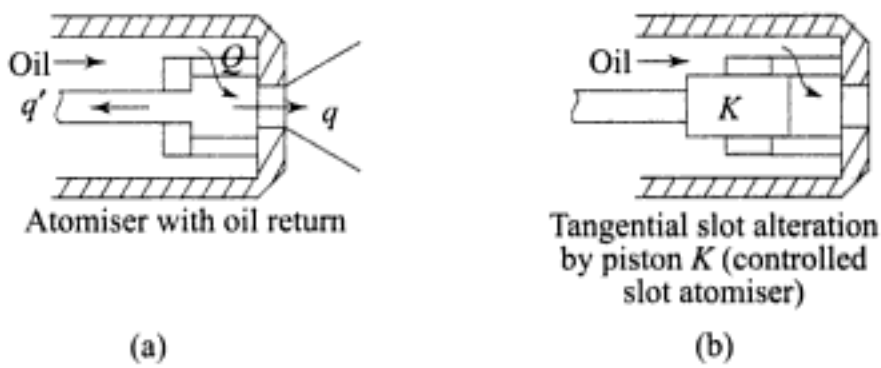

Fig. 5.64 *Methods of output control in pressure atomisers*

5.12 COMBUSTION OF GAS

Burning of gas is easy and clean. No atomization is required. Combustion of 1 m^3 of natural gas requires roughly 20 m^3 of hot air. Proper mixing of gas and air can be ensured by introducing the gas into the air flow in the form of thin jets of high penetrability (Fig. 5.65). Because of good mixing excess air required for combustion is less.

5.13 COMBINED GAS FUEL OIL BURNERS

One advantage of combined gas-fuel oil burners is that the change from one fuel to the other can be done quite easily. In addition, both fuels can be burned under almost optimal conditions. Natural gas flows from the central annular header through two rows of holes (Fig. 5.66). Air is supplied through a tangential vane register and controlled by disc valve. Fuel oil is atomized in a mechanical burner mounted in the central channel. The mixture is ignited by electric means.

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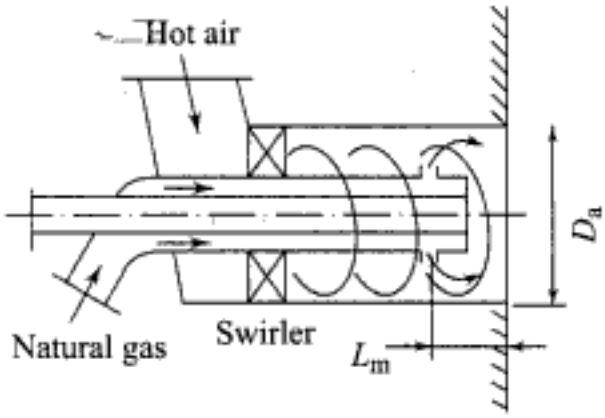

Fig. 5.65 *Gas burner with central gas supply*

Example 5.1 A bed of solid particles has a mass of 2500 kg. The density of solid is 2650 kg/m^3. The mean particle size is 800 mm and the sphericity of particles is 0.84. Estimate the total surface area of the particles in the bed.

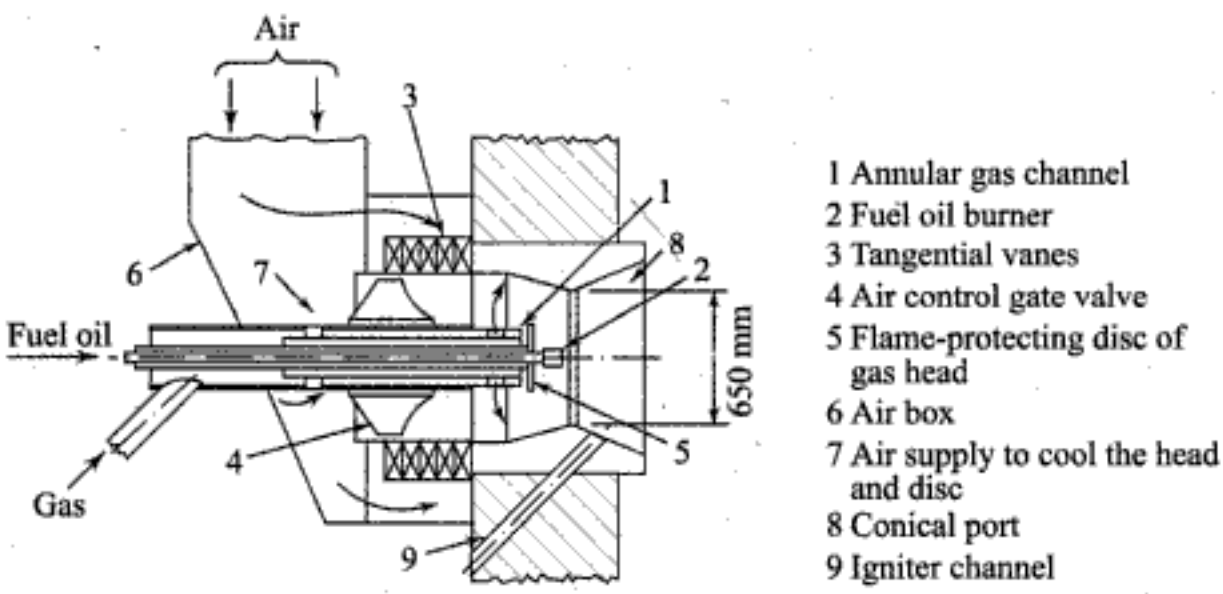

Fig. 5.60 *Coaxial gas-fuel oil burner with central gas supply*

Solution Surface-to-volume ratio of particles,

$$S_v = \frac{A_s}{V_s} = \frac{N_p \pi d_p^2}{N_p \frac{\pi}{6} d_p^3} = \frac{6}{d_p}$$

where N_p is the number and d_p is the diameter of particles. For non-spherical particles, the surface-area of particles is

$$A_s = \frac{6V_s}{\phi d_p}$$

where V_s is the volume and ϕ is the sphericity of the particles.

$$A_s = \frac{6 \times 2500}{2650 \times 0.84 \times 800 \times 10^{-6}} = 8423 \text{ m}^2 \qquad \textit{Ans.}$$

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Example 5.2 A bed of particles of mean size 427 μm is fluidized by air under the ambient conditions, where the air density is 1.21 kg/m^3 and the viscosity is 1.82×10^{-5} kg/m-s. The density of the loosely packed bed is 1620 kg/m^3. If the density of solids is 2780 kg/m^3, find (a) the voidage of the bed, and (b) the minimum fluidization velocity.

Solution (a) Bulk density of bed, $\rho_b = 1620$ kg/m^3

Density of solids, $\rho_s = 2780$ kg/m^3

$$\therefore \qquad \varepsilon_{mf} = 1 - \frac{\rho_b}{\rho_s} = 1 - \frac{1620}{2780} = 0.417 \qquad \textit{Ans.}$$

(b) Archimedes number, $Ar = \dfrac{\rho_g(\rho_s - \rho_g)\, g\, d_p^3}{\mu_g^2}$

$$\therefore \qquad Ar = \frac{1.21(2780 - 1.21) \times 9.81 \times (427 \times 10^{-3})^3}{(1.82 \times 10^{-5})^2} = 7753$$

From Eq. (5.24),

$$\mathrm{Re}_{mf} = [C_1^{\,2} + C_2\, Ar]^{0.5} - C_1$$
$$= [27.2^2 + 0.0408 \times 7753]^{0.5} - 27.2 = 5.299$$

Now,
$$\mathrm{Re}_{mf} = \frac{U_{mf} \times \rho_g \times d_p}{\mu_g}$$

$$\therefore \qquad U_{mf} = \frac{5.299 \times 1.82 \times 10^{-5}}{1.21 \times 427 \times 10^{-6}} = 0.185 \text{ m/s} \qquad \textit{Ans.}$$

Example 5.3 For the above bed (Ex. 5.2), taking $U_{mf} = 0.14$ m/s, estimate the sphericity of particles.

Solution Using Ergun's equation,

$$\frac{\Delta p}{L} = 150 \frac{(1 - \varepsilon_{mf})^2}{(\varepsilon_{mf})^3} \frac{\mu_g\, U_{mf}}{(\phi\, d_p)^2} + 1.75 \frac{(1 - \varepsilon_{mf})}{\varepsilon_{mf}^3} \frac{\rho_g\, U_{mf}^2}{(\phi\, d_p)}$$
$$= [1 - \varepsilon_{mf})\,(\rho_s - \rho_g) g$$

Multiplying each side by $\rho_g\, d^3{}_p / m_g^{\,2}\,(1 - \varepsilon_{mf})$ gives

$$Ar = 150 \frac{1 - \varepsilon_{mf}}{\phi^2\, \varepsilon_{mf}} \mathrm{Re}_{mf} + \frac{1.75}{\phi\, \varepsilon_{mf}^3} \mathrm{Re}_{mf}^2$$

Now,
$$\mathrm{Re}_{mf} = \frac{U_{mf}\, \rho_g\, d_p}{\mu_g} = \frac{0.14 \times 1.21 \times 427 \times 10^{-6}}{1.82 \times 10^{-5}} = 3.974$$

$$Ar = 7753$$

$$\therefore \qquad 7753 = \frac{150(1 - 0.417)}{\phi^2\, 0.417^3} \times 3.974 + \frac{1.75}{\phi 0.417^3} \times (3.974)^2$$

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$\therefore \quad 7753 f^2 - 381.1 f - 4793 = 0$

$\therefore \quad f = 0.811 \quad \text{or} \quad -0.762$

$\therefore \quad$ Sphericity = 0.811 *Ans.*

Example 5.4 A fluidized bed combustion system having an output of 35 MW at 80% efficiency when using a coal of heating value 26 MJ/kg with a sulphur content of 3.6% requires a particular limestone to be fed to it at a calcium-sulphur ratio of 3.0, so as to limit emissions of SO_2 adequately. The limestone used contains 85% $CaCO_3$. Determine the required flow rate of limestone.

Solution Coal burning rate $= \dfrac{35}{0.80 \times 26} = 1.683$ kg/s = 6057.7 kg/h

$\therefore$ Flow rate of sulphur $= \dfrac{6057.7 \times 0.036}{32} = 6.81$ kmol/h

For a Ca : S molar ratio of 3.0, the flow rate of calcium is $= 6.81 \times 3.0 = 20.44$ kmol/h

$\therefore$ Mass flow rate of $CaCO_3$ required $= 20.44 \times 100$

= 2044 kg/h (since molecular weight of $CaCO_3$ is 100)

$\therefore$ Mass flow of limestone = 2044/0.85 = 2404.7 kg/h which is about 40% of the coal burning rate.

Example 5.5 A fluidized bed combustor burns a solid fuel of high volatile matter with a calorific value of 24 MJ/kg. The combustion conditions are such that 65% of the calorific value is released in the bed and the remainder in the above-bed zone from which the products leave at 850 °C. The air inlet temperature is 30 °C, the bed temperature is 850 °C and the air fuel ratio by mass is 13.5 : 1. The specific heat of the products leaving the bed surface is 1.035 kJ/kg K. If the burning rate of coal is 7000 kg/h, estimate the rate of heat removal (a) from the bed, and (b) from the above-bed zone.

Solution (a) Heat removal from the bed per kg fuel

= heat released – heat carried away by the products

$= 0.65 \times 24000 - 14.5 \times 1.035 \times (850 - 30)$

= 3294 kJ/kg fuel

$\therefore$ Rate of heat removal from the bed $= 3294 \times \dfrac{7000}{3600}$

= 6405 kW *Ans.*

(b) Rate of heat removal from the above bed zone

$= \dfrac{7000}{3600} \times 0.35 \times 24000 = 16333$ kW *Ans.*

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Example 5.6 Fluidized bed is required to operate at atmospheric pressure and a bed temperature of 850 °C. The fuel has a calorific value of 25 MJ/kg. The stoichiometric air fuel ratio is 9.5 by mass and 20% excess air is used. The total fueling rate is 4.8 MW. The density of air at 850 °C is 0.3145 kg/m^3. Find the planform area required if (a) the firing rate is 2 MW/m^2, and (b) the fluidizing velocity is 2.7 m/s.

Solution (a) Planform area = 4.8/2 = 2.4 m^2 *Ans.*

(b) Fuel burning rate, $\dot{m}_f = \dfrac{4.8 \times 1000}{25000} = 0.192$ kg/s

$\therefore$ Air flow rate, $\dot{m}_a = 9.5 \times 1.2 \times 0.192 = 2.1888$ kg/s

$\therefore$ Planform area $= \dot{m}_a/(r_a U) = \dfrac{2.1888}{0.3145 \times 2.8} = 2.48$ m^2 *Ans.*

Example 5.7 Crushed coal particles, with diameters varying from 6 mm to 20 mm, and average density of 1250 kg/m^3, are used in a fluidized bed combustor. The coefficient of drag is 0.6 under fluidized conditions. Calculate (a) the minimum gas velocity that fluidizes all the coal particles if the gas is at 850°C and 9 bar pressure, and (b) the pressure drop in the bed assuming that the coal in the collapsed state ($\varepsilon_0 = 0.35$) has a height H0 of 30 cm and voidage fraction ($\varepsilon = 0.50$) and the gas density is approximated by pure air.

Solution Minimum fluid velocity required for fluidization is calculated by equating the drag force on a particle to the weight of the particle, or

$$C_D \rho_f A V^{-2}/2 = \rho_5 \times q_5 \times g$$

where A = projected area, $\bar{V}$ = velocity of flow, ρ_f = fluid density, ρ_5 = solid density. If γ is the radius of the spherical particle, $A = \pi\gamma^2$ and $V_s = \dfrac{4}{3}\pi r^3$

$$\therefore \quad \bar{V} = \left[\frac{8}{3C_D} \times \frac{\rho_s}{\rho_f} \times gr\right]^{1/2}$$

For minimum $r = 3$ mm $= 3 \times 10^{-3}$ m,

$$\rho_f = \rho_a = \frac{p}{RT} = \frac{9 \times 10^5}{287 \times (850 + 273)} = 2.792 \text{ kg/m}^3$$

$$\therefore \quad \bar{V} = \left[\frac{8}{3 \times 0.6} \times \frac{1250}{2.792} \times 9.81 \times 3 \times 10^{-3}\right]^{1/2}$$

$= 7.65$ m/s *Ans. (a)*

Height of fluidized bed, H

$$= H_0 \frac{1-\varepsilon_0}{1-\varepsilon} = 0.3 \times \frac{1-0.35}{1-0.5}$$

$= 0.39$ m

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∴ Pressure drop in the bed

$$\Delta p = (1 - \varepsilon) H \rho_s g$$
$$= (1 - 0.5) \times 0.39 \times 1250 \times 9.81$$
$$= 2.59 \text{ kPa}$$ *Ans.*

SHORT-ANSWER QUESTIONS

5.1 On what factors does a chemical reaction depend?
5.2 What are the two broad types of furnaces?
5.3 Explain the physical and chemical factors determining the combustion rate of a fuel.
5.4 What are reversible and irreversible chemical reactions?
5.5 What is the law of mass action?
5.6 What is Arrhenius equation?
5.7 Explain what you understand by activation energy.
5.8 Explain why carbon monoxide is formed predominantly on the surface of burning carbon particles.
5.9 What is the average burning time of a pulverized coal particle?
5.10 Why does the ratio of CO/CO_2 increase with temperature?
5.11 Explain briefly the mechanism of burning of a carbon particle.
5.12 Explain what you understand by kinetically controlled reaction.
5.13 What is diffusion controlled reaction?
5.14 What do you mean by transition zone of combustion?
5.15 Explain diffusion zone and kinetic zone of combustion.
5.16 What are the different types of furnaces in which coal may be burnt?
5.17 What are the characteristics of an overfed fuel bed furnace?
5.18 Explain the characteristics of an underfed fuel bed furnace.
5.19 What is a stoker? What are the different types of stokers?
5.20 Explain the operation of a travelling grate stoker with the help of neat sketch.
5.21 Explain the operation of a spreader stoker with the help of a neat sketch.
5.22 What are the two conditions to be satisfied to burn pulverized coal successfully?
5.23 What are the advantages and disadvantages of pulverized coal firing?
5.24 What is the function of hot primary air?
5.25 What is the function of secondary air?
5.26 What is tempering air? When is it required?
5.27 What is the function of the coal crusher?
5.28 What are the three stages of pulverization process?
5.29 How are pulverizers classified?
5.30 Explain the characteristic features of a ball mill.
5.31 What is a bowl mill? What are its advantages?
5.32 On what factors does the performance of a pulverizer depend?

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5.33 What is Hardgrove index?
5.34 What is the effect of ratio of VM/FC on the fineness of powdered coal?
5.35 What are the feeding systems of pulverized coal into the furnace?
5.36 Explain with a sketch the direct firing system? What are its merits and demerits?
5.37 What is the heat release rate of a p.f. - fired furnace?
5.38 Explain how the linear dimensions of a p.f. furnace are ascertained.
5.39 Explain the characteristics of a dry bottom furnace.
5.40 What is a wet bottom furnace? Where is it used? How is slag removed continuously?
5.41 What are the conditions to be satisfied by pulverized coal burners?
5.42 What are the different types of pulverized coal burners?
5.43 Distinguish straight flow and vortex burners.
5.44 Explain the distributed mixing burner.
5.45 Discuss the different arrangements of burners in a pulverized coal fired furnace.
5.46 What is tangential or corner firing? What are its merits and demerits?
5.47 What is a cyclone furnace? Where is it used? Mention its advantages and disadvantages.
5.48 Why is tertiary air required in a cyclone furnace? Where is it admitted?
5.49 What is a fluidized bed? What is minimum fluidization velocity?
5.50 What is a distributor? Why is it needed in a fluidized bed?
5.51 How is superficial velocity estimated?
5.52 What are the various regimes of fluidization?
5.53 Define (a) Voidage, (b) Sphericity of a particle.
5.54 How would you estimate U_{mf}?
5.55 What are particulate and emulsion phases?
5.56 What is transport disengaging height (TDH)?
5.57 What do you mean by elutriation rate?
5.58 What is slugging? How is it to be avoided?
5.59 Explain a turbulent fluidized bed.
5.60 What is the captive stage of a fluidized bed?
5.61 What is fast fluidization? On what factors does it depend?
5.62 Explain with a sketch a circulating fluidized bed (CFB).
5.63 Explain the variation of voidage, both radially and axially, in the riser column of a CFB.
5.64 Explain the core–annulus model of the structure of a CFB riser.
5.65 What is a cluster? What is its average voidage? With what velocity does it fall along the wall?
5.66 Explain the pressure balance along a CFB loop.
5.67 What do you understand by terminal velocity of particles? How does it depend on particle Reynolds number?
5.68 What is drag coefficient? On what does it depend?
5.69 What do you understand by attrition and fragmentation?
5.70 How is the fluidized bed combustion in a furnace started?

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5.71 What is the advantage of using limestone or dolomite as the bed material in a fluidized bed? Why are they called "sorbents"?
5.72 What are the optimum combustion temperature and the approximate calcium-sulphur molar ratio for maximum absorption of sulphur oxides?
5.73 What is a PFBC? Discuss its scope, advantages and disadvantages.
5.74 Briefly explain the sequence of events of burning of a fuel particle in a fluidized bed.
5.75 What are the two resistances which impede the reaction of a char particle?
5.76 What is Sherwood number? What is Schmidt number? How are they related in diffusional mass transfer of oxygen?
5.77 On what factors does the specific burning rate depend?
5.78 Explain the influence of mass transfer by diffusion and chemical kinetics in controlling combustion rate in a fluidized bed.
5.79 What are the three distinct zones of combustion in a CFB furnace?
5.80 How does the lower zone of a CFB furnace act as a thermal flywheel with respect to boiler load variation?
5.81 What are the different factors which influence the combustion characteristics of coal in a CFB?
5.82 What is the distinction between air-blown and oxygen-blown gasifiers?
5.83 What are the different types of gasifiers? Briefly explain their characteristics.
5.84 What are the basic reactions taking place in coal gasifiers?
5.85 Why does an entrained bed gasifier require gas coolers?
5.86 How is coal gasification important from emission viewpoint?
5.87 Explain the IGCC plant with the help of certain relevant blocks.
5.88 What is the unique characteristic of an IGCC plant?
5.89 Explain with a schematic diagram the oil-burning installation of a steam generator.
5.90 Why is fuel oil required to be heated? Is there any restriction to its heating?
5.91 Explain the mechanism of combustion of a drop of fuel oil.
5.92 Discuss the different types of oil burners.
5.93 How is the fuel consumption in a pressure atomiser-cum-burner controlled?
5.94 What are the advantages of using combined gas-oil burners?

PROBLEMS

5.1 The pressure drop (N/m^2) across the distributor when fluidized at ambient pressure and temperature is related to the superficial velocity U (m/s) by the equation

$$\Delta p = 0.13\ U + 0.03\ U^2$$

Silica sand particles are loosely packed in a bed to a depth of 0.3 m. The bulk density of the bed is 1600 kg/m^3 and the bed diameter is 1.7 m. Determine the pressure drop across the bed and distributor, when the bed is incipiently fluidized. Take U_{mf} = 0.19 m/s. Also find the pumping power required by the system when fluidized with ambient air at 0.5 m/s.

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5.2 A fluidized bed is operating at a velocity 2.8 times the minimum fluidizing velocity. Visual observation suggests that the largest bubbles formed are of 10 mm diameter. Estimate the residence time of a single, isolated bubble if the bed depth when fluidized is 650 mm. Take $U_{mf} = 0.15$ m/s.

5.3 A bed of particles of mean size 483 mm is fluidized by air under the ambient conditions, where the air density is 1.22 kg/m^3 and the viscosity is 1.87×10^{-5} kg/ms. The bulk density of the loosely packed bed is 1680 kg/m^3. If the density of solids is 2650 kg/m^3, estimate (a) the voidage of the bed, (b) the minimum fluidization velocity, and (c) the sphericity of particles.

5.4 A fluidized bed combustion system having an output of 40 MW operating at 78% efficiency with a coal of heating value of 25 MJ/kg with a sulphur content of 3.8% requires a particular limestone to be fed to it at a calcium-sulphur ratio of 3.1, so as to limit emissions of SO_2 adequately. The limestone used contains 85% $CaCO_3$. Determine the consumption rate of limestone.

5.5 A fluidized bed furnace operating at ambient conditions produces a stream of hot combustion products at a bed temperature of 850 °C. The mass flow rate of combustion products is to be 7.5 kg/s and the heat losses from the system are such that the air fuel ratio by mass is 25 : 1. Estimate the planform area required for the bed if the fluidizing velocity is 1.2 m/s. At 850 °C, take the density of air as 0.3145 kg/m^3. What is the firing rate in μW/m^2?

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6 Steam Generators

A steam generator generates steam at the desired rate at the desired pressure and temperature by burning fuel in its furnace. Steam generators are used in both fossil-fuel and nuclear-fuel electric generating power stations. This chapter will, however, cover only fossil-fuel steam generators. Nuclear fuel steam generators are of very different design and will be covered in Chapter 9.

A steam generator is a complex integration of furnace, superheater, reheater, boiler or evaporator, economiser, and air preheater along with various auxiliaries such as pulverizers, burners, fans, stokers, dust collectors and precipitators, ash-handling equipment, and chimney or stack. The boiler (or evaporator) is that part of the steam generator where phase change (or boiling) occurs from liquid (water) to vapour (steam), essentially at constant pressure and temperature. However, the term "boiler" is traditionally used to mean the whole steam generator.

6.1 BASIC TYPES OF STEAM GENERATORS

Classification of steam generators or boilers can be made in different ways. From the point of view of applications, they can be

(a) utility steam generators
(b) industrial steam generators
(c) marine steam generators

Utility steam generators are those used by utilities for electric-power generating plants. Depending on whether the pressure of steam is below or above the critical pressure (221.2 bar), they can be either subcritical or supercritical units. The subcritical steam generators are water tube–drum type and they usually operate at between 130 and 180 bar steam pressure. The supercritical steam generators are drumless once-through type and operate at 240 bar pressure or higher. Majority of the utility steam generators are of the 170–180 bar water tube–drum variety, which produce superheated steam at about 540–560 °C with one or two stages of reheating. The steam capacities of modern

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utility steam generators range from 120 to 1300 kg/s with power plant unit output ranging from 125 to 1300 MW. Pulverized coal is mainly the fuel used. Some circulating fluidized bed steam generators have also been installed recently in some utilities.

Industrial steam generators are those used in process industries like sugar, paper, jute and so on, and institutions like hospitals, commercial and residential building complexes. They are smaller in size. They can be pulverized coal fired, fluidized bed or stoker fired units, with coal mostly as the fuel. They can also be heat recovery types which use waste heat from various industrial processes, and are termed waste heat steam generators. They operate at pressures ranging from 5 to 105 bar with steam capacities upto 125 kg/s. Normally, they do not produce superheated steam. They supply wet or saturated steam, and sometimes even only hot water.

Marine steam generators are used in many marine ships and ocean liners driven by steam turbines. They are usually oil-fired. They produce superheated steam at about 60–65 bar and 540 °C.

Steam generators or boilers can also be classified according to the relative flows of products of combustion or flue gases and water. In one type of boiler, called the *fire-tube boiler*, the hot flue gases flow through tubes surrounded by water in a shell. In another type of boiler, called the *water-tube boiler*, water from a drum flows through the tubes and hot flue gases flow over them.

6.2 FIRE-TUBE BOILERS

Since the late eighteenth century, fire-tube boilers have been used in various forms to produce steam for industrial purposes and also for hauling railway locomotives and river launches. They are no longer used in utility power plants and steam locomotives have also mostly disappeared. However, they are still often used in industrial plants to produce saturated steam at the upper limits of about 18 bar pressure and 6.2 kg/s steaming capacity.

For small steam requirements, fire-tube boilers are suitable. They have certain inherent advantages like (1) low first cost, (2) reliability in operation, (3) need of only unskilled labour, (4) less draught required, and (5) quick response to load changes.

A fire-tube boiler is so named because the products of combustion pass through its tubes or flues, which are surrounded by water. They may be either:

(a) Externally fired (e.g. locomotive type boilers, Lancashire boilers, horizontal return tubular (HRT) boiler etc.), or
(b) Internally fired (e.g. Scotch-marine boilers, package boilers etc.).

Figure 6.1 shows a typical externally fired fire-tube boiler in which the furnace is outside the boiler shell. Coal is entered manually by shovels on to the grate by opening the fire-door. The products of combustion flow through the tubes which are immersed in the shell containing water. A fusible plug made up of a low melting point alloy (lead-based) is installed on the roof of the crown in the furnace. If the water-level in the shell falls below a certain level, the fusible

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plug melts due to overheating and water pours down through the hole formed and puts out the fire. There is a spring-loaded safety-valve provided to keep the boiler pressure within the safety limit. The spring is set in such a way that the upward thrust of steam against the lid is balanced by the downward thrust of the spring. If the operating steam pressure exceeds this value, the upward thrust of steam will then be greater than the downward spring thrust and the difference will force open the lid upward, as a result of which steam will be released with a hissing sound, the steam pressure inside the shell will go down till the lid is forced down to be back on its seat.

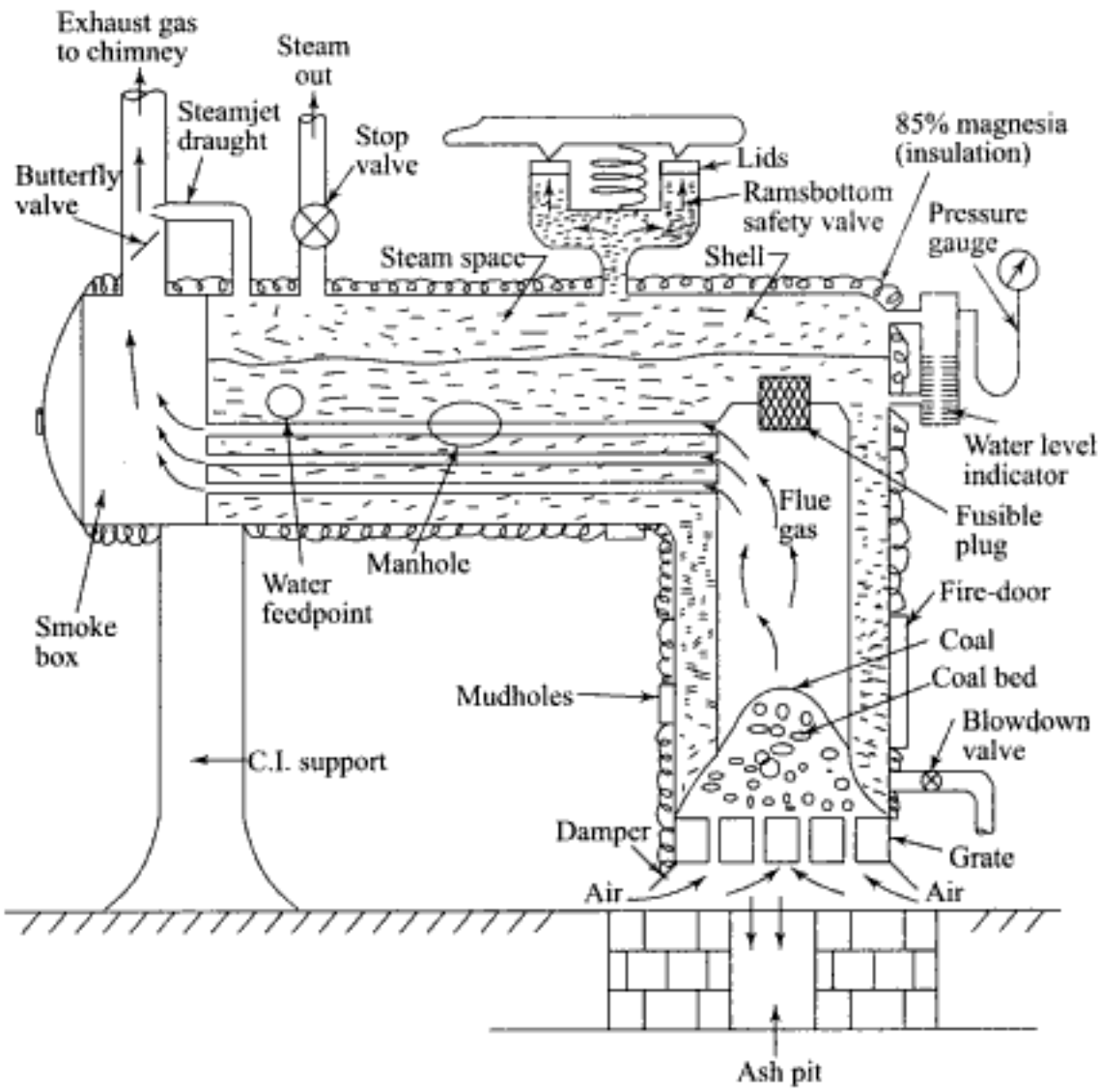

Fig. 6.1 *A typical fire-tube boiler*

As the hot flue gases flow through the tubes, heat is transferred from gas to water all along the length. The gas is cooled and the water is heated till there is nucleate boiling around the tubes and steam is formed. Steam is taken out at the required rate by opening the main stop valve. Auxiliary steam may also be taken to operate a steam jet water injector to feed water into the shell. The shell is insulated all around by asbestos and 85% magnesia to reduce heat loss to the surroundings. Air flow from below the grate is regulated by operating dampers according to the requirement of combustion. An elliptic manhole is provided for a man to go in to do cleaning or repair as the need arises.

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An oil-fired pressurized package type boiler is shown in Fig. 6.2. It is an internally fired fire-tube boiler, since the furnace is within the cylindrical shell. Oil is filtered, adequately heated either by electricity or steam, and is then fed through the burner. Air is supplied by an FD fan. Excess oil is returned to the main feedline. Combustion occurs in a horizontal corrugated flue, and the combustion gases flow through the tubes in two passes till they are let out through a chimney to atmosphere. The corrugations of the flue increase strength and help in allowing thermal expansion and contraction, and also in cracking and removing the soots deposited. The gases get cooled as they flow through the tubes and water is heated enough to form steam. The wet steam taken from the top flows through a bank of coils for superheating and the superheated steam is taken out as desired. Since the pressure inside the furnace is above atmospheric, caution is required to make the casing sealed against leakage. If it is not tightly sealed, hot gases will exfiltrate causing energy losses and polluting the atmosphere.

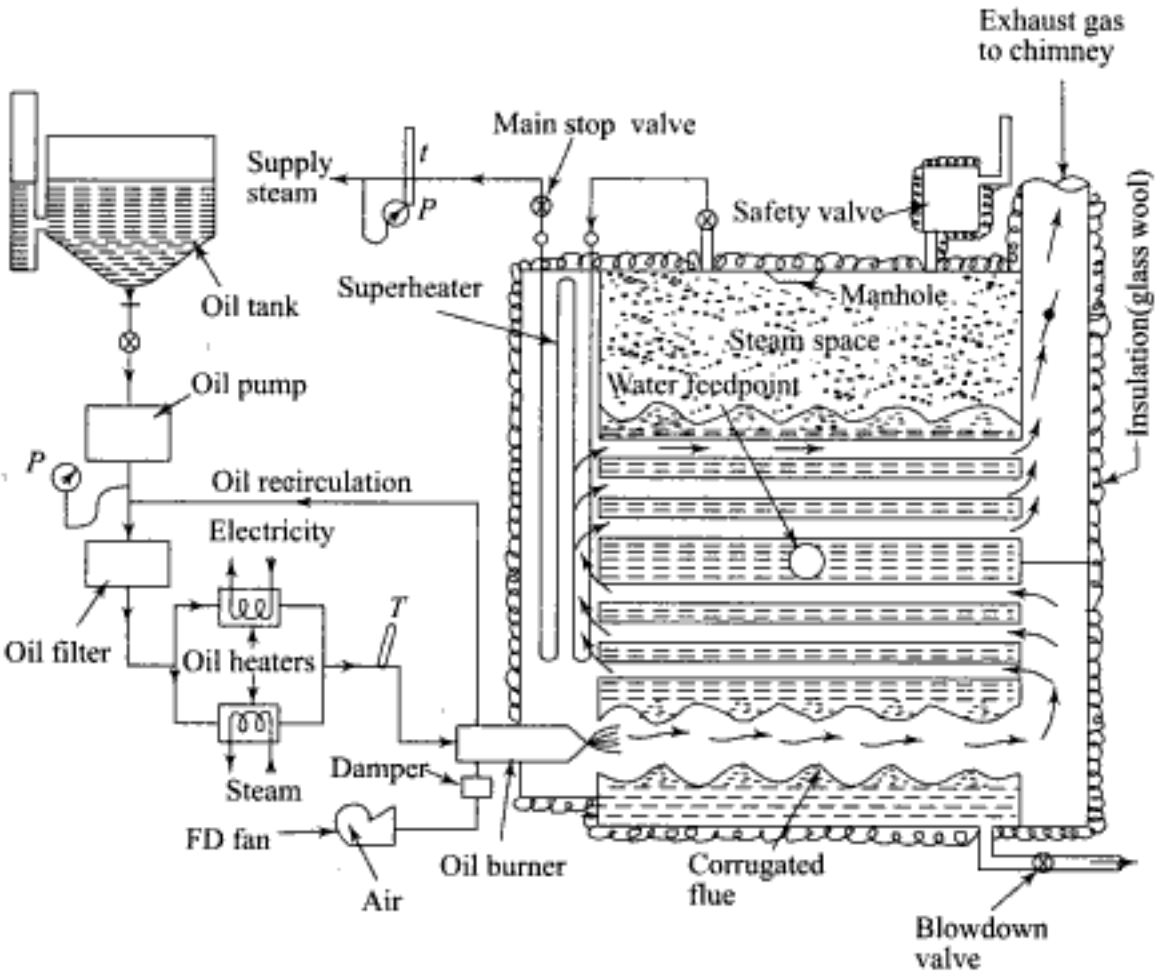

Fig. 6.2 *An oil-fired package boiler*

The package-type boiler has certain inherent advantages: (1) It is highly compact. A large surface area of heat transfer is provided in a small volume. The furnace volume is less since it is pressurized. (2) No ID fan is required. A short chimney is provided just for disposal of flue gases. (3) The entire unit is mostly fabricated in the factory itself. It is transported to the site and installed there with relative ease. (4) It occupies less space. (5) It is easy to operate.

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Because of these advantages, package boilers were favoured in process industries. But the paucity of oil supply has reduced its demand giving way to the use of fluidized bed boilers mostly based on coal.

As stated earlier, fire-tube boilers are relatively inexpensive. They have large water storage capacity and can thus meet relatively large and sudden load demands with only small pressure changes. Reduction in pressure leaves the stored water superheated and causes part of it to flash into steam. However, the large water storage increases the explosion hazard of the unit, and also because of it, a longer period of time is required to bring the unit to steaming from a cold condition.

The major shortcoming of a fire-tube boiler is that definite size and pressure limitations are inherent in its basic design, i.e. the maximum size of the unit and the maximum operating pressure are limited. The tensile stress on the drum wall is a function of the drum diameter and the internal pressure given by

$$\sigma = \frac{pd}{2t} \tag{6.1}$$

where σ = tensile stress, N/m^2

p = gauge pressure, N/m^2

d = internal diameter of shell, m

t = thickness of wall, m

The growing needs for increased quantities of steam at higher and higher pressures could not be met by fire-tube boilers, for as Eq. (6.1) shows, both higher pressures and larger sizes mean larger wall thicknesses. Thus, high pressures and large diameters lead to prohibitively thick shells, and the thicker the shell, the higher the cost.

6.3 WATER-TUBE BOILERS

Water-tube boilers were developed to permit increases in boiler pressure and capacity with reasonable metal stresses. As mentioned above, with higher steam pressures and capacities, fire-tube boilers would need large-diameter shells, and with such large diameters, the shells would have to operate under such extreme pressure and temperature stresses that the thicknesses would become very large. They are also subjected to large scale deposit and susceptible to boiler explosions, and become very costly.

The water-tube boiler, where water flows through the tubes and flue gases flow outside them, puts the pressure in the tubes and the relatively small-diameter drums, which are capable of withstanding extreme pressures of the modern steam generator.

6.3.1 Straight-Tube Boiler

The water-tube boiler went through several stages of development. The earliest design employed straight tubes rolled into headers at each end, since straight

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tubes could be made, installed and replaced easily. The tubes were of 75 to 100 mm O.D., inclined upward at about 15° to the horizontal and staggered. Nearly saturated water leaving the drum flowed through one header, called the downcomer, into the tubes. While flowing upward in the tubes some of this water on being heated by flue gases flowing outside get transformed to vapour and the two-phase water-steam mixture went back to the drum through the other header, called the riser. The density of nearly saturated water in the downcomer was larger than the density of the two-phase mixture in the riser, and this density difference caused natural circulation. The drum was arranged either parallel to the tubes (the longitudinal drum, Fig. 6.3a) or pependicular to them (the cross drum, Fig. 6.3b). The feedwater from the feedwater heater flowed through an economiser to the drum, which supplied saturated steam to the superheater. The lower end of the downcomer was connected to a mud drum, which collected sediments from the circulating water and discharged through blowdown.

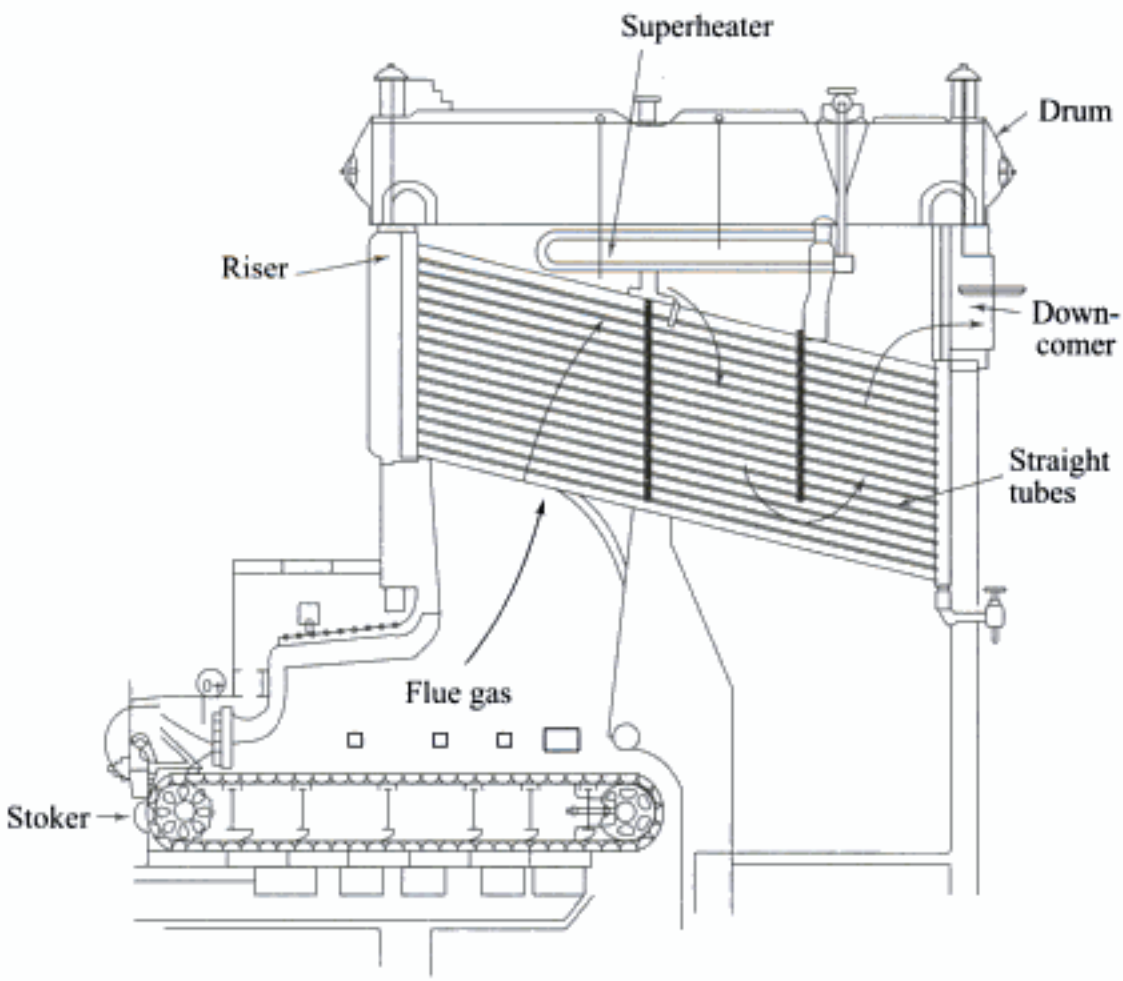

Fig. 6.3 *Early water-tube boiler: (a) longitudinal drum*

A single longitudinal drum, about 1.2 m in diameter, could accommodate only a limited number of tubes and hence a limiting heating surface. Depending on capacity, more than one drum could be installed in parallel. Longitudinal drum boilers were limited to low pressures (12–23 bar) and steam capacities (0.63–10 kg/s), having heating surfaces of 93–930 m^2.

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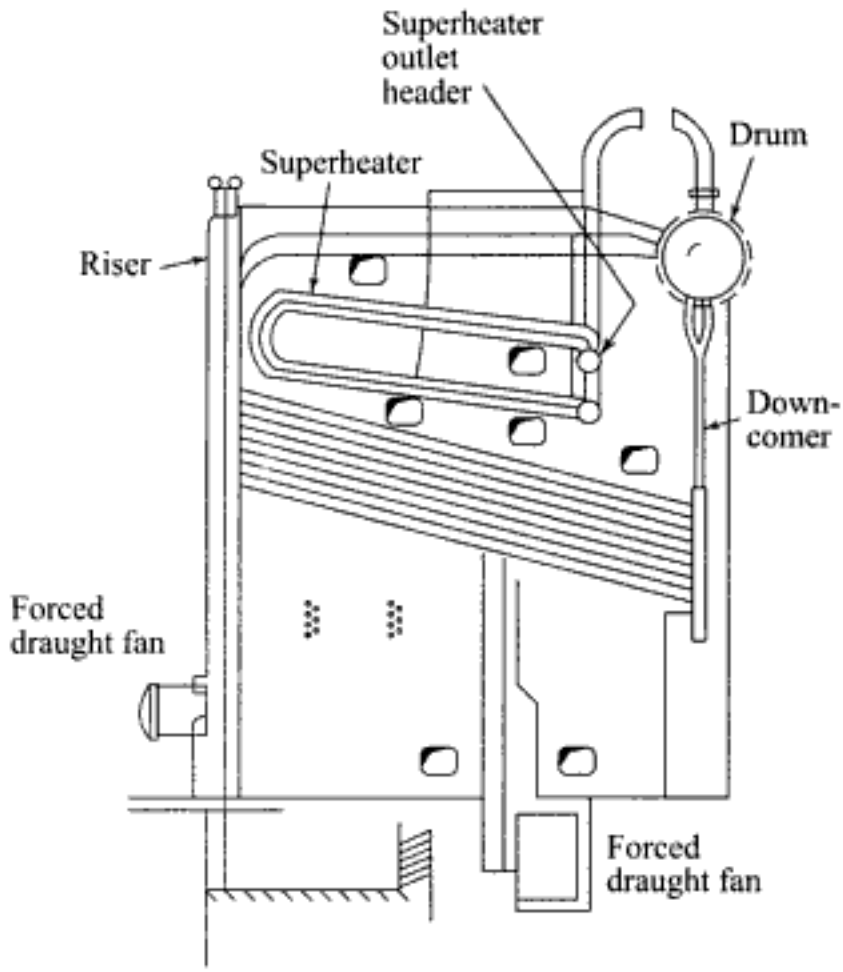

Fig. 6.3 *Early water-tube boiler: (b) Cross drum*

Cross-drum boilers could accommodate many more tubes than longitudinal drum boilers because of their geometry and were built with heating surfaces of 93–2200 m^2, pressures 12–100 bar and steam capacities of 0.63–63 kg/s.

To ensure maximum exposure of tubes to the hot flue gases, baffles were installed across the tubes in both kinds of boilers to permit up to three gas passes.

6.3.2 Bent-tube Boilers

Straight-tube boilers had many disadvantages, like:

1. They had less accessibility and poorer inspection capability. Considerable time, labour and expense were required to open up or close the bolts in the headers, and to remove or replace the gaskets and handhole caps for tube inspection and replacement.
2. Inadequate design and imperfect fabrication of handhole caps resulted in much leakage, which was a continuous source of annoyance to the operators.
3. Circulation was sluggish due to low head, and limited steam disengaging surface made inadequate separation of steam and water reducing steaming rate.

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Bent-tube boilers introduced in the 1880's offered many advantages over the straight-tube ones, the notable among them being greater accessibility for inspection, cleaning, and maintenance, and ability to operate at higher steaming rates and to deliver drier steam.

In a bent-tube boiler, the tubes were so bent that they entered and left the drums radially. As an example, a four-drum Stirling boiler is shown in Fig. 6.4. There was easy access to every tube. The combustion gases flowed upward from the furnace through the first bank of tubes connecting the front steam drum and the mud drum, through the superheater, and by proper baffling through the second and third tube banks connecting to the second and rear steam drums. The gases then flowed past an economiser in counterflow arrangement. Feedwater from the economiser entered the rear steam drum. Water circulated from the rear to the lower drum through the rear bank of tubes (downcomers) and then up through both the centre and front banks of tubes (risers) to the centre and front drums. All three drums had their steam and water regions interconnected at top and bottom. The tubes were typically 75–77 mm in OD and spaced 125–175 mm on centres, for easy replacement of defective tubes without removing neighbouring tubes.

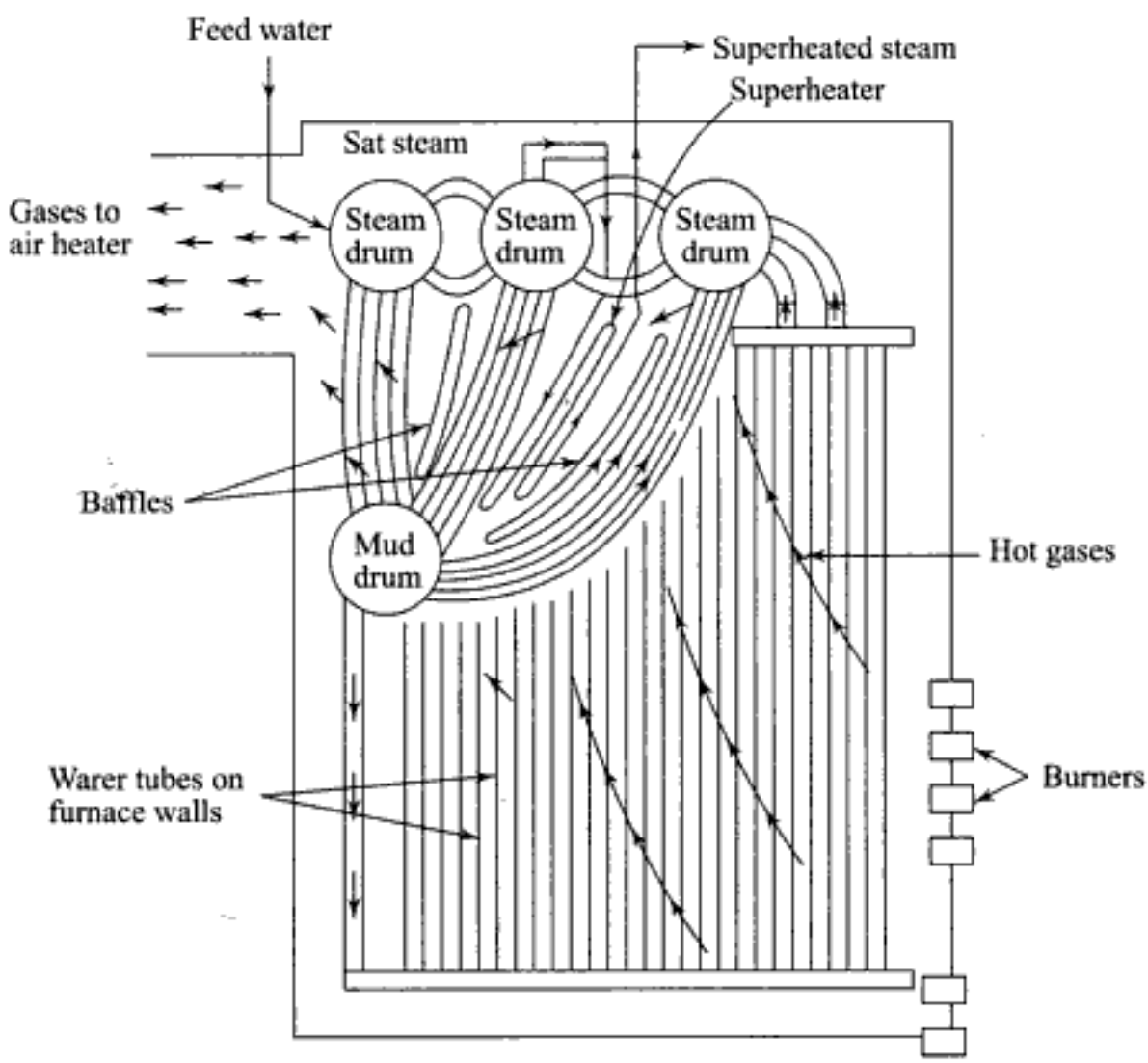

Fig. 6.4 *A four-drum bent tube boiler*

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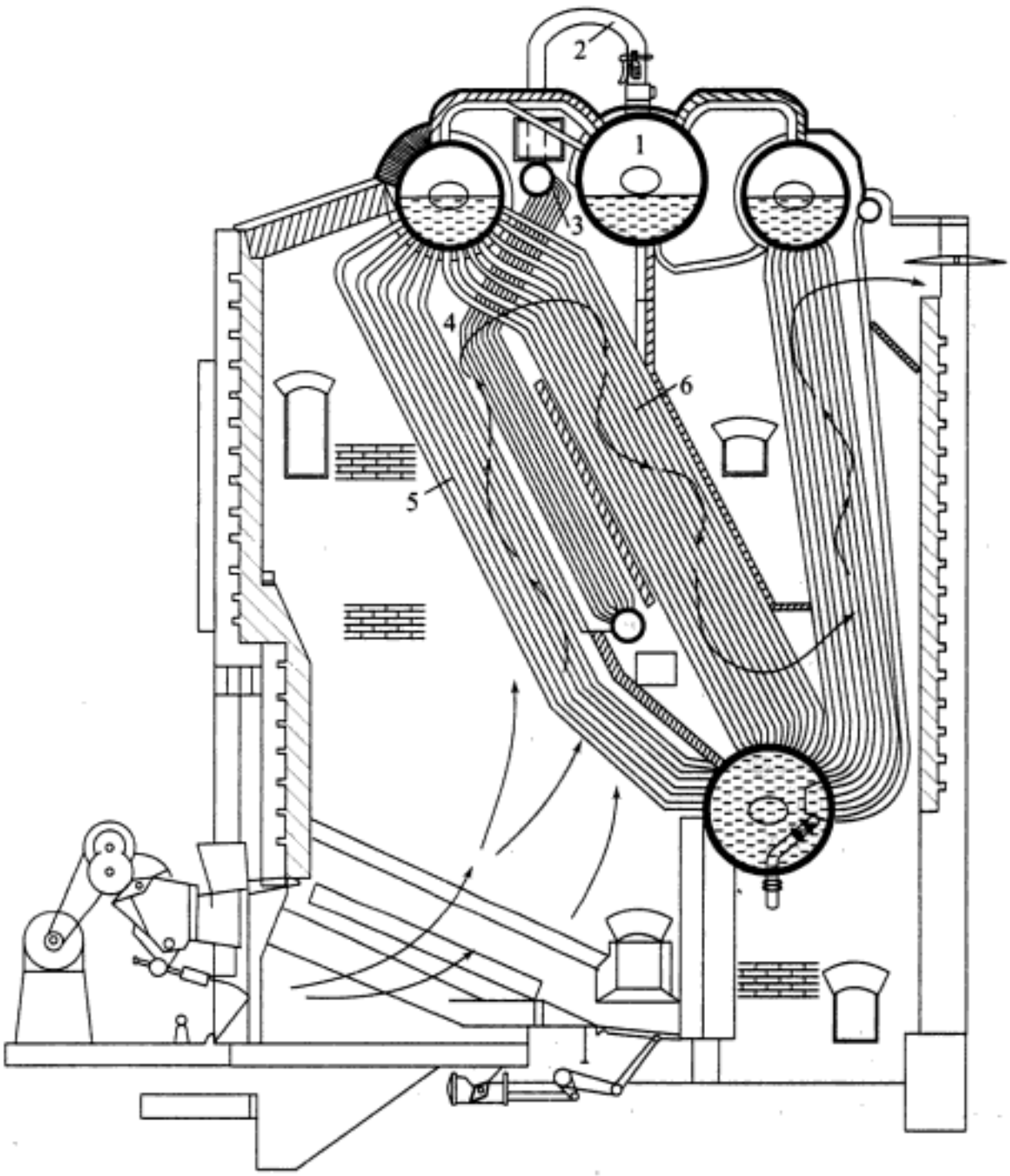
More recent designs of Stirling boilers used water cooled furnace walls by lining the interior of the walls with tubes carrying the same boiler water, where partial boiling of water occurs. Fig. 6.5 shows a schematic of a natural circulation four-drum Stirling boiler. Water flows downwards from the mud drum to headers feeding the tubes lining the walls of the radiant furnace. The low-density steam-water mixture rises up to the steam drum at the upper right. The steam is separated and flows to the central drum, where it is removed. Feedwater enters the drum at the left and mixes with the sturated liquid in the left and central drum. The cooled liquid flows down to the mud drum.



1. Steam delivery drum
2. Saturated steam transfer tubes
3. Superheater inlet header
4. Superheater tubes
5. First-pass tubes
6. Second-pass tubes

Fig. 6.5 *Natural circulation four-drum Stirling boiler*

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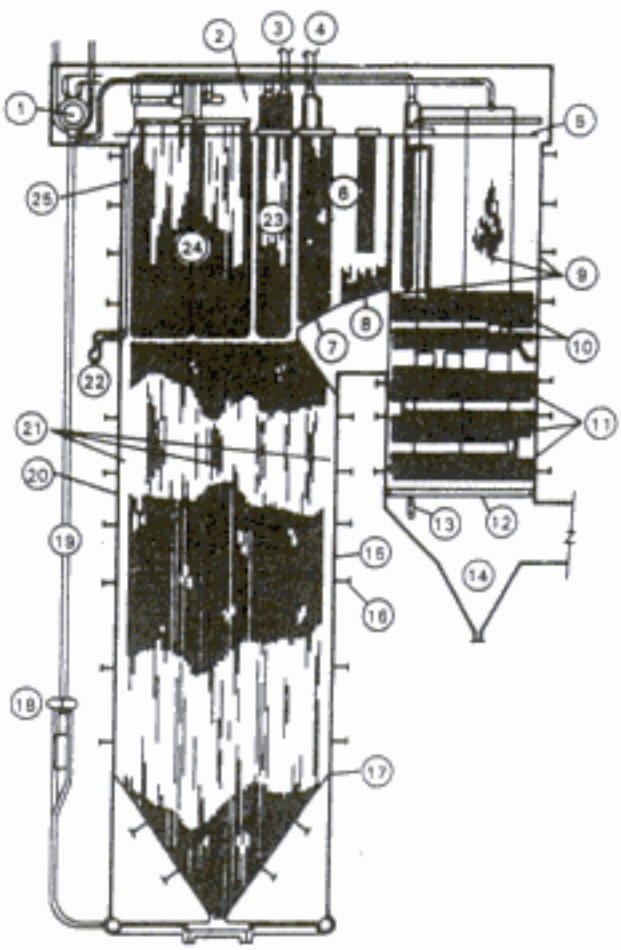
The four-drum Stirling boiler was superseded by a simpler two-drum design in which the steam drum was directly above the water drum with one bank of bent tubes to the front and another to the rear. Later designs of the two-drum boiler had a single gas-path. The Stirling boiler was generally capable of meeting conditions of rapidly varying loads and was adaptable to various fuels. It was employed in both stationary and marine applications.

Modern pulverized coal-fired boilers stand up vertically with a furnace, a steam drum, downcomers, risers, superheaters, reheaters, economiser, air preheater, tube mills, FD and ID fans, coal feeders and burners, windbox, various control systems, ductwork and all accessories and mountings. The penthouse encloses the drum, high-pressure headers and connecting links on top of the steam generator. The floor of the penthouse is formed by the roof tubes of the boiler. The principal structural members consist of vertical structural steel columns, a top support grid and beams (buckstays) which provide stiffness to the furnace, combustion air and flue gas flow enclosures. Igniters and warmup burners ignite and stabilize the coal flame. A ductwork system directs the flow of combustion air and flue gas. Soot blowers are devices that direct a high pressure fluid (steam or water) onto heat transfer surfaces to remove ash or slag deposits. Selected components of modern steam generators are shown in Figs 6.6 (a) and (b).



1. Steam drum
2. Penthouse
3. Superheater outlet
4. Reheater outlet
5. Steam-cooled roof
6. Pendant convection superheater or reheater
7. Furnace nose
8. Furnace arch
9. Steam-cooled walls
10. Horizontal convection superheater or reheater
11. Economizer
12. Superheater junction header
13. Economizer inlet header
14. Economizer ash hopper
15. Rear wall
16. Buckstays
17. Furnace knuckle
18. Boiler circulating pumps
19. Downcomer
20. Front wall
21. Furnace water-cooled walls
22. Reheataer inlet header
23. Platen type superheater or reheater
24. Panel type superheater
25. Radiant wall reheater

Fig. 6.6 (a) *Boiler Nomenclature*

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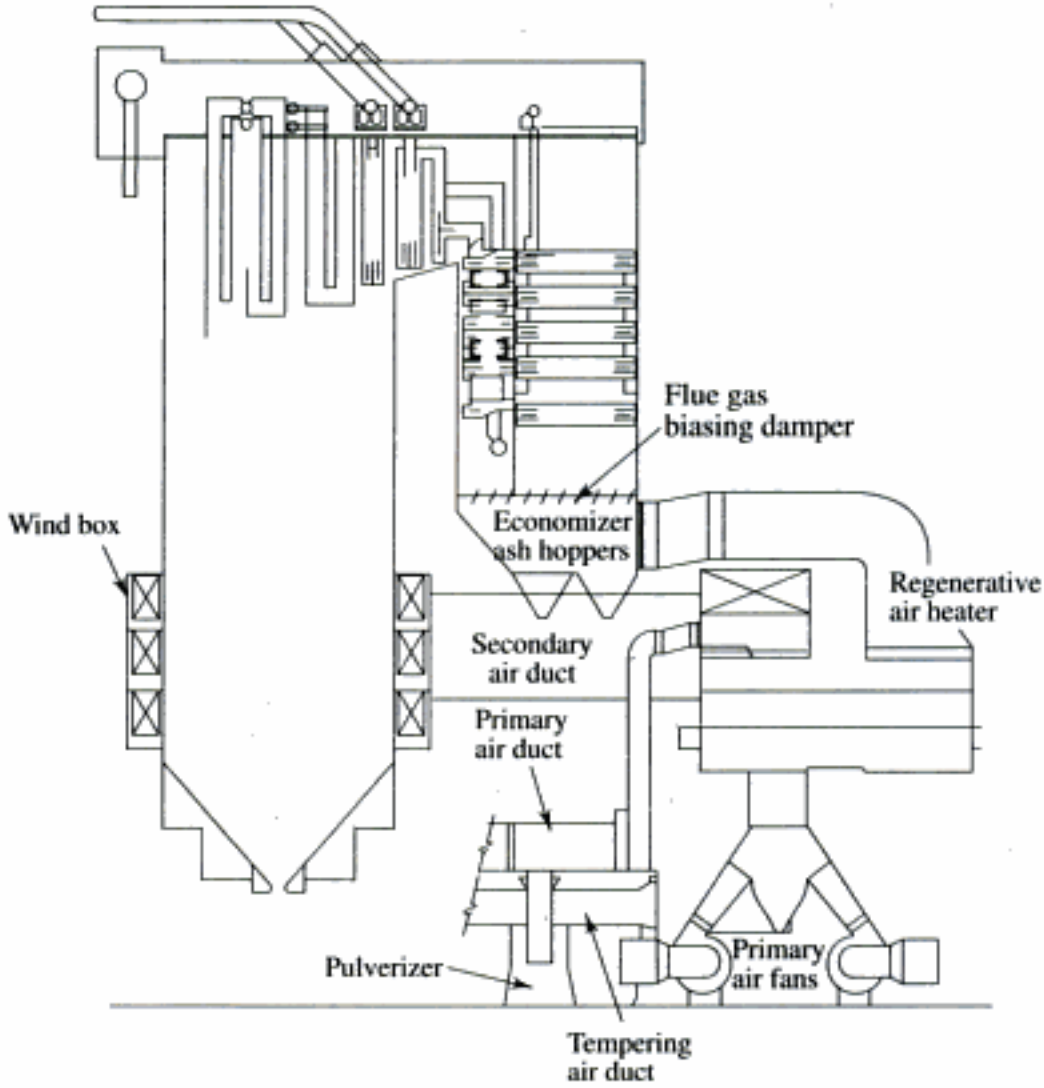

Fig. 6.6 (b) *Typical wall-fired boiler cross-section with divided convection pass*

6.3.3 Heat Absorption in Water-Tube Boilers

In a water-tube boiler, feedwater is heated in three kinds of heat exchangers, *viz.*, economizer, evaporator (downcomer-riser circuit) and superheaters (Fig. 6.6). Feedwater from the h.p. heater enters the economiser where it is heated by the outgoing flue gases till it is saturated liquid at that pressure and then it is fed to the drum. Saturated water falls through the downcomer into the bottom header and moves up through the riser where water is partially boiled back into the drum (Fig. 6.7(a)). Saturated steam from the drum goes to the superheaters for being heated to the desired temperature. If the pressure drop during the entire heating process and any external heat loss are neglected, the line 1 – 4 represents the heating process in *T–s* diagram (Fig. 6.7(b)). For each kg of steam formed, the heats absorbed in the economiser (in the liquid phase), in the evaporator (liquid to vapour transition or the latent heat of vaporization), and in the superheater (in the vapour or gas phase) are given by

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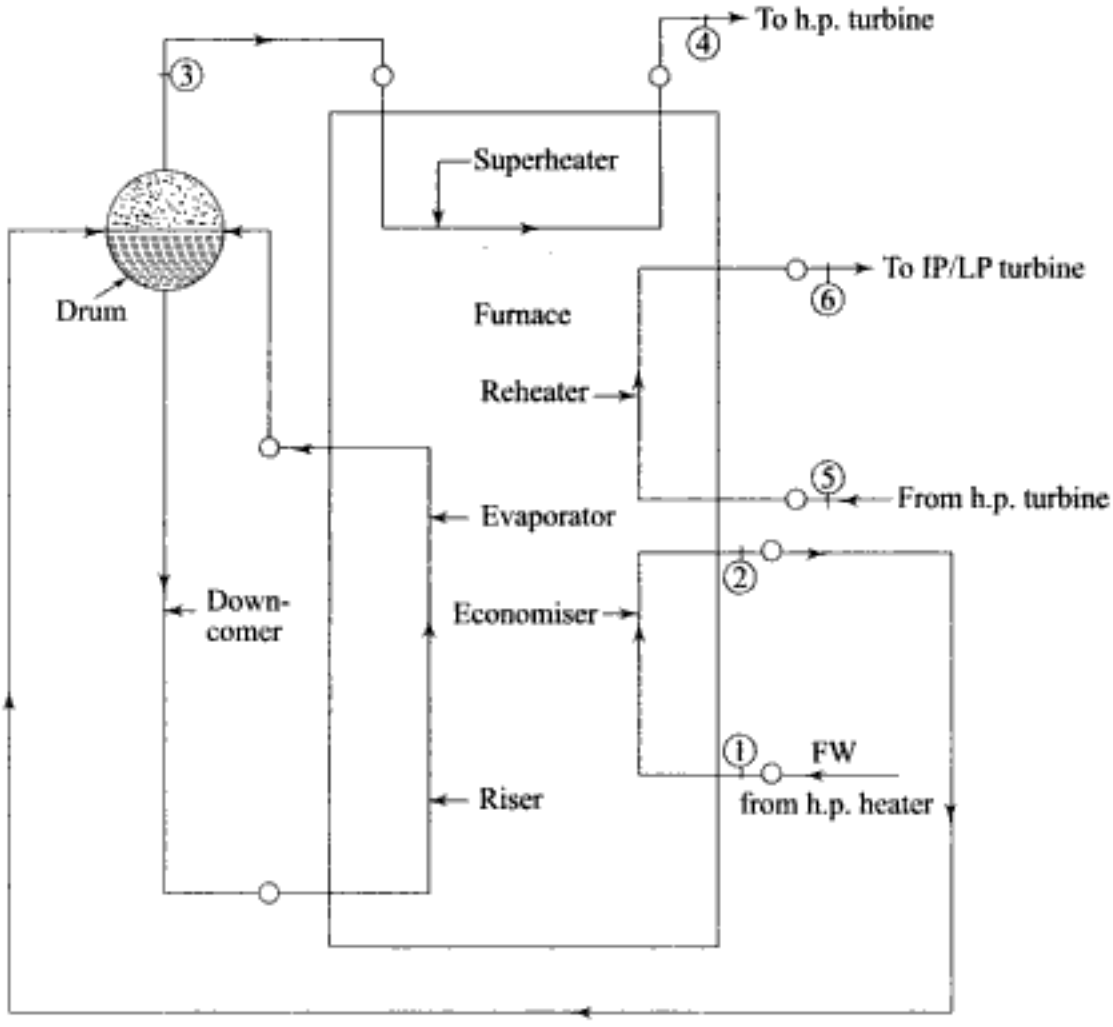

Fig. 6.7(a) *Heat absorption in a water-tube boiler*

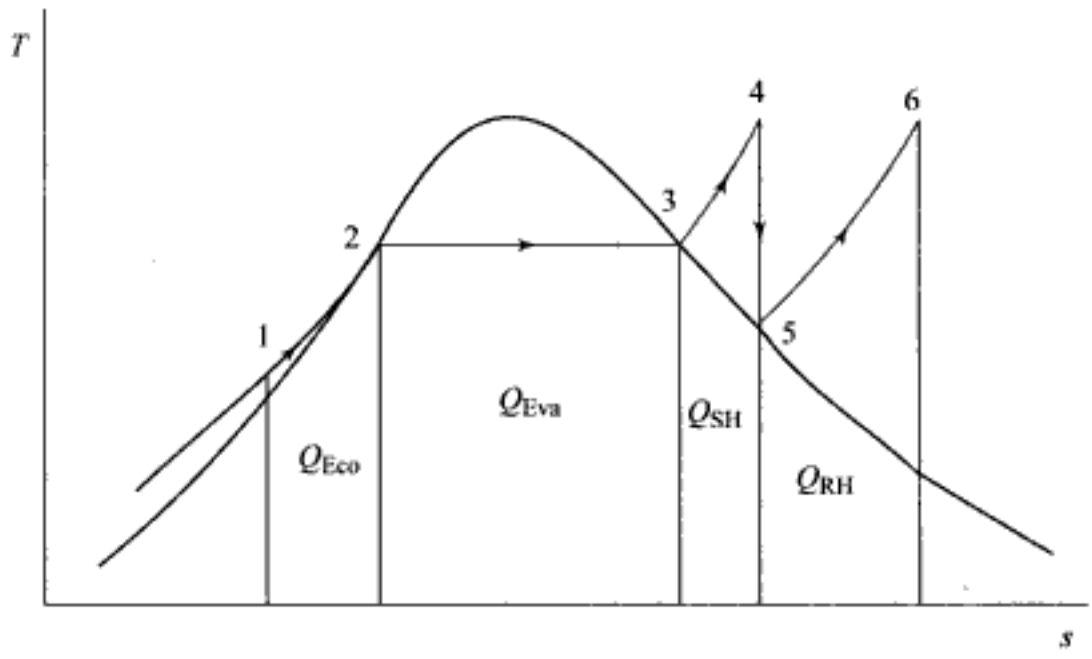

Fig. 6.7(b) *Heat absorption in different heat exchangers of a water-tube boiler*

$$\left.\begin{aligned} Q_{eco} &= h_2 - h_1 \\ Q_{eva} &= h_3 - h_2 \\ Q_{SH} &= h_4 - h_3 \end{aligned}\right\} \tag{6.2}$$

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The percentage of total heat absorbed in the economiser, evaporator and superheater are:

$$\% \text{ economiser} = \frac{Q_{eco}}{Q_{eco} + Q_{eva} + Q_{SH}} \times 100 = \frac{h_2 - h_1}{h_4 - h_1} \times 100$$

$$\% \text{ evaporator} = \frac{Q_{eva}}{Q_{eco} + Q_{eva} + Q_{SH}} \times 100 = \frac{h_3 - h_2}{h_4 - h_1} \times 100$$

$$\% \text{ superheater} = \frac{Q_{SH}}{Q_{eco} + Q_{eva} + Q_{SH}} \times 100 = \frac{h_4 - h_3}{h_4 - h_1} \times 100 \qquad (6.3)$$

If a reheater is used, the fraction of the total heat absorbed in the reheater can similarly be estimated. The use of a large number of feedwater heaters (up to seven or eight) means a smaller economiser, and a high pressure means a smaller boiler surface (risers) because the latent heat of vaporization decreases rapidly with pressure. Thus, a modern high pressure steam generator requires more superheating and reheating surfaces and less boiler surface than older units. The fraction of total heat absorbed in superheaters alone may be as high as 60% in a modern boiler.

6.3.4 Circulation

The flow of water and steam within the boiler circuit is called circulation. Adequate circulation must be provided to carry away the heat from the furnace. If circulation is caused by density difference, the boiler is said to have natural circulation. If it is caused by a pump, it has forced or controlled circulation.

A simple downcomer–riser circuit connecting a drum and a header is shown in Fig. 6.8. The downcomer, which is insulated, is outside the furnace, and the riser is inside it.

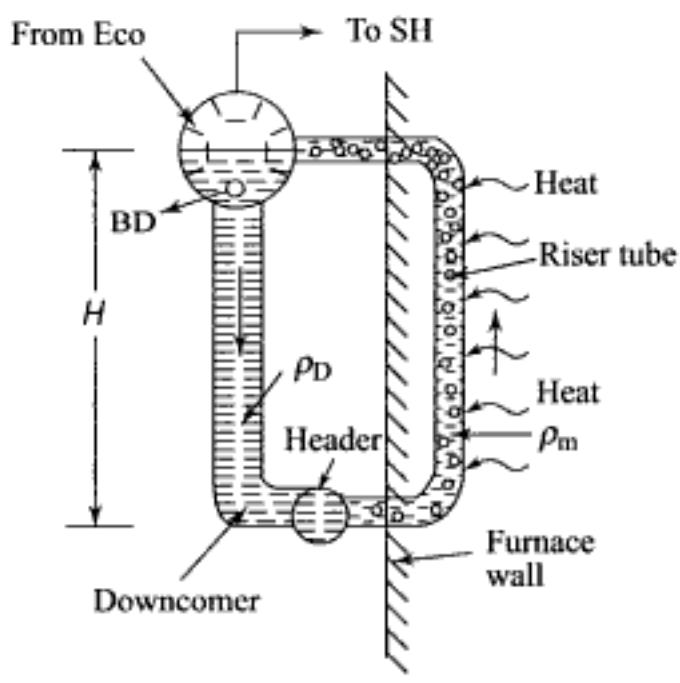

Fig. 6.8 *Natural circulation in a downcomer–riser circuit*

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Nearly saturated water falls by gravity from the drum through the downcomer into the bottom header. From the header water flows up along the riser where it partially boils with the formation of bubbles and then back into the steam drum. The density of steam–water mixture in the riser is less than that of saturated water in the downcomer, and as a result of this density difference a circlation current is set up within the downcomer–riser circuit. The feedwater from the economiser enters the drum and saturated steam is taken out of the drum to the superheater.

The pressure head available for natural circulation is given by

$$\Delta p = g H (\rho_D - \rho_m) \tag{6.4}$$

where H is the height of the riser (~ furnace), ρ_D is the density of saturated water in the downcomer, and ρ_m is the mean density of steam–water mixture in the riser. For simplicity, the mean density, ρ_m, may be taken as the arithmetic mean of the densities at the bottom and top of the riser, or

$$\rho_m = \frac{\rho_{bottom} + \rho_{top}}{2} \tag{6.5}$$

Now, $\rho_{bottom} = \rho_D$ and $\rho_{top} = 1/v_{top}$, $v_{top} = v_f + x_{top} v_{fg}$, where x_{top} is the quality of the mixture at the top of the riser and is often referred to as the top dryness fraction (TDF).

The maximum height of the furnace (H) is fixed by the rate of burning of fuel and the heat release rate. For a certain H,

$$\Delta p \, \alpha (\rho_D - \rho_m) \tag{6.6}$$

Higher is the density difference, more will be the pressure head available for natural circulation. However, the density differential decreases as pressure increases (Fig. 6.9). It is observed in the figure that the difference in specific volume ($v_g - v_f$) or the difference in density ($\rho_f - \rho_g$) of saturated liquid and saturated vapour decreases with increase in pressure. At the critical pressure, $v_g = v_f$ and $\rho_g = \rho_f$ and there can be no natural circulation.

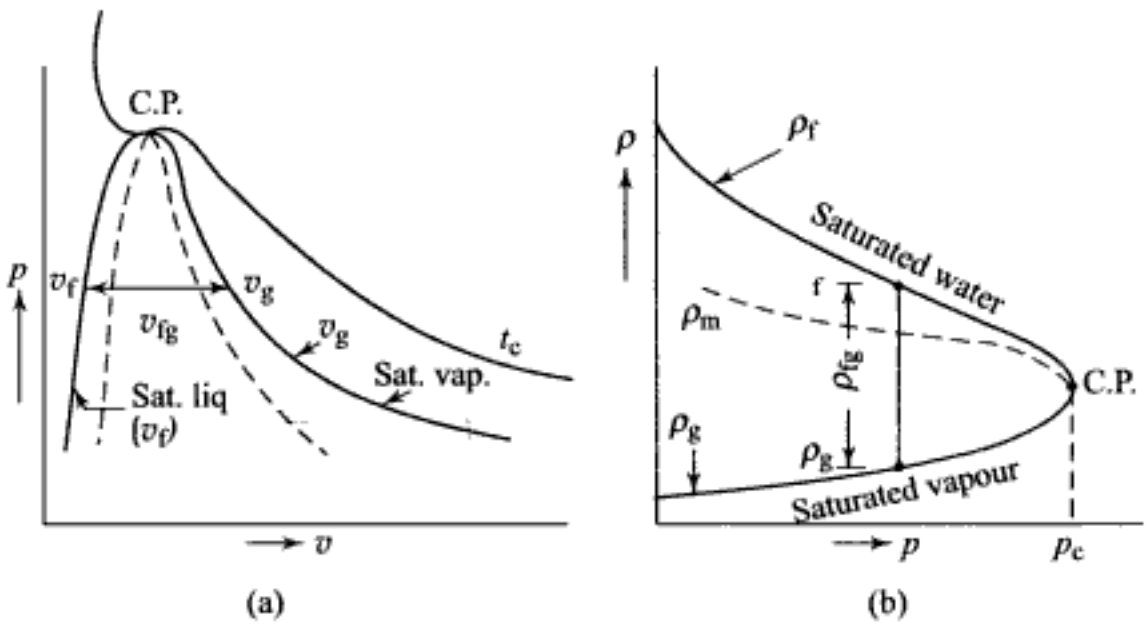

Fig. 6.9 *Differentials in specific volume and density varying with pressure*

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Risers installed all around the four walls of the furnace act as cooling tubes or a water wall and carry away the heat from the furnace at the same rate at which heat is released in it by the burning of fuel. For this reason, adequate circulation must be provided in the circuit. If the circulation is not adequate, the rate at which heat is carried away will be less than the rate of heat release, and the difference will be stored in the metal of the riser tubes leading to their overheating and ultimately rupturing when the tube temperature exceeds the melting point of the metal.

If the boiler pressure is greater than 30 bar, the downcomers are placed outside the furnace so as to secure more density difference for natural circulation. If both downcomers and risers are placed inside the furnace and heated, the density difference will be less. However, for pressures less than 30 bar, both are placed inside, with the risers installed in the hotter zone. The tubes which are heated more will act as risers and the tubes which are heated less act as downcomers.

If the pressure of steam exceeds 180 bar, the density difference $(\rho_f - \rho_g)$ becomes so small (Fig. 6.9) that natural circulation cannot be relied upon and then forced circulation is used where a pump circulates saturated water through all the risers around the furnace walls (Fig. 6.10).

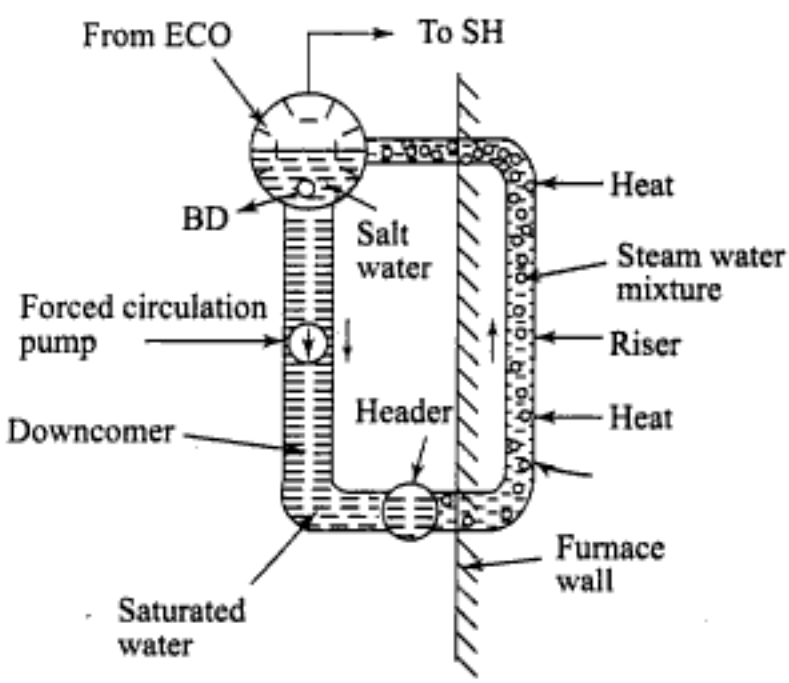

Fig. 6.10 *Forced circulation system*

There is a term called "circulation ratio" (CR) used in this connection. It is defined as

$$\text{Circlation ratio} = \frac{\text{Flow rate of saturated water in downcomers}}{\text{Flow rate of steam released from the drum}}$$

It refers to the amount of saturated water to be circulated through the downcomer–riser circuit per kg of steam released from the drum. Therefore,

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$$\text{CR} = \frac{m}{m_g} = \frac{m_g + m_l}{m_g} = \frac{1}{m_g/(m_g + m_l)} = \frac{1}{\text{TDF}} \tag{6.7}$$

where m = mass of saturated water flowing through the downcomer–riser circuit during a certain time,

m_g = mass of steam released from the drum during the same time,

$m_l = m - m_g$ = mass of liquid water (saturated) at the riser exit

TDF = top dryness fraction, i.e. the quality of liquid vapour mixture discharged from the riser into the drum.

The circulation ratio in a natural circulation boiler varies from one riser tube to another. The riser tubes located opposite to the burners have more thermal loading and generate more steam. So, these will have less circulation ratio. While the risers located in the corners of the furnace are relatively cooler, and so these will have more circulation ratio. However, the circulation ratio in any tube should not be less than 6, i.e. at least six kg of saturated water must circulate in the downcomer–riser circuit for each kg of steam released. Otherwise, the tube will get overheated and fail prematurely. The circulation ratio should again not be higher than 25 for effective utilization of the tube in steam generation.

Too much steaming (or steam formation) is not desired in a riser tube and circulation ratio needs to be maintained above 6. From heat transfer consideration a wetted surface is always desired and nucleate boiling should always persist. Bubbles originate on the heated surface. If there is a high rate of heat transfer to the riser, there will be too much bubble formation with the result that the bubbles may coalesce and first form an unstable vapour film which continually collapses and reforms. With a still higher heat transfer, the vapour film may be stable. Since a vapour film has a much lower thermal conductivity than a liquid film, it will offer a large thermal resistance, almost blanketing the surface where it forms, as a result of which heat absorbed and carried away (Q_2) will be much less than the heat transferred to the wall (Q_1). The difference (Q_1–Q_2) will be stored in the metal of the tube with the increase in its internal energy (Fig. 6.11). Consequently, the temperature of the metal may exceed the melting point and the tube may rupture allowing tube leakage.

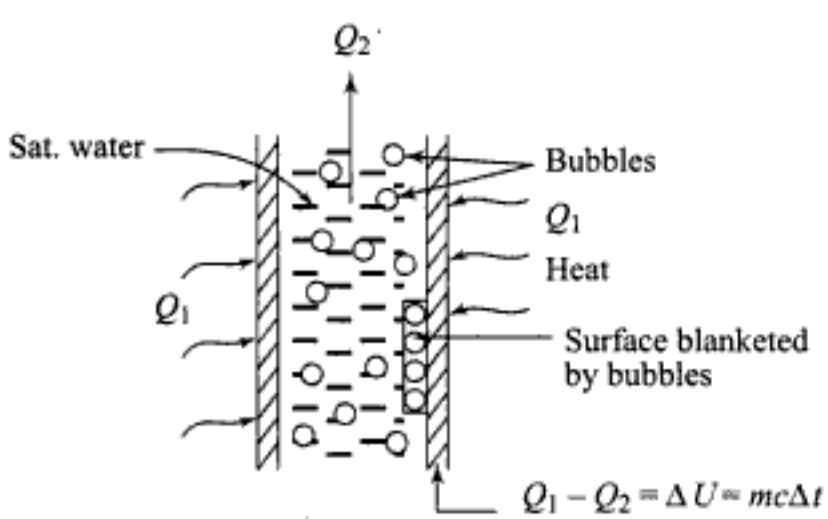

Fig. 6.11 *Nucleate and film boiling in a riser tube*

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Downcomers are fewer in number and bigger in diameter, whereas risers are more in number and smaller in diameter. Downcomers are meant to make the water fall by gravity. Bigger the diameter, less the pressure drop due to friction, since pressure drop is inversely proportional to the tube diameter.

$$\Delta p = \frac{fL}{D} \rho \frac{V^2}{2} \tag{6.8}$$

So, the downcomers are made bigger in diameter, which may vary from 150 to 200 mm or even higher. Now,

$$m_f = \left(n \frac{\pi}{4} D^2\right) \rho_f V \tag{6.9}$$

where m_f = mass flow rate of saturated water in the downcomers, kg/s

n = number of tubes

D = diameter of downcomer, m

ρ_f = density of saturated water at the boiler drum pressure, kg/m^3

V = average velocity of water in downcomers, m/s

The circulation velocity V depends on the circulation ratio in the tube, and may vary from 0.4 to 1.4 m/s depending on capacity. Thus, the number of downcomers required can be estimated from Eq. (6.9). It can be 6, 8 or as high as 20, again depending on the capacity of the steam generator.

Risers absorb heat from the furnace. For the same total cross-sectional area, the smaller the diameter, the larger the surface area exposed to hot gas for heat transfer. Therefore, the risers are of smaller diameter, 62.5 to 76.5 mm, and larger in number compared to downcomers.

It is required that nucleate boiling should occur in each riser tube and film boiling is avoided. Too much steaming and less circulation ratio in a riser may cause a departure from nucleate boiling (DNB) and an onset of film boiling. Recently, the water tubes in high heat absorbing areas of the furnace are provided with internal twisters and springs which break the vapour film and retard the onset of DNB. Tubes ribbed helically on their inside surface are often used now. The ribbing creates a centrifugal action that directs the water droplets to the vapour film clinging to the surface and to wash it away.

The void fraction (α) of a two-phase mixture, often called volumetric quality, is defined as

$$\alpha = \frac{\text{volume of vapour}}{\text{volume of liquid + volume of vapour}}$$

Thus, α is the volume fraction of vapour in a liquid vapour mixture. The quality (x) is the mass fraction of vapour in the two-phase mixture. While the specific volume of a mixture of quality x is given by

$$v = (1 - x)v_f + x\, v_g = v_f + x\, v_{fg}$$

the density of the mixture of voidage α is given by

$$\rho = (1 - \alpha)\rho_f + \alpha\, \rho_g = \rho_f + \alpha\, \rho_{fg}$$

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Let us consider the two-phase flow in a riser (Fig. 6.12). Let A_f = cross-section occupied by saturated liquid

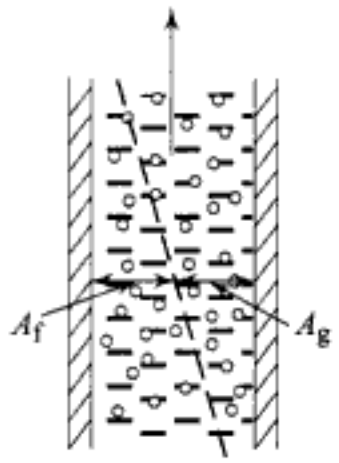

Fig. 6.12 *Two phase flow in a riser*

A_g = cross-section occupied by vapour bubbles

Then, $$\alpha = \frac{A_g}{A_f + A_g} \quad \text{and} \quad 1 - \alpha = \frac{A_f}{A_f + A_g}$$

$$\frac{A_f}{A_g} = \frac{1-\alpha}{\alpha} \tag{6.10}$$

When a two-phase mixture is moving, the vapour, because of its buoyancy, has a tendency to slip past the liquid, i.e. move at a higher velocity than a liquid. The slip ratio S is defined as

$$S = \frac{\bar{V}_g}{\bar{V}_f} = \frac{\text{velocity of vapour}}{\text{velocity of liquid}} \tag{6.11}$$

at any cross-section of the riser, and it is greater than unity. Now, the quality x is given by

$$x = m_g/m = \frac{\rho_g A_g \bar{V}_g}{m} = \frac{A_g \bar{V}_g}{m v_g}$$

$$\therefore \quad \bar{V}_g = \frac{m \times v_g \times x}{A_g} \tag{6.12}$$

Similarly,

$$1 - x = \frac{A_f \bar{V}_f}{m v_f}$$

$$\bar{V}_f = \frac{(1-x)\, m\, v_f}{A_f} \tag{6.13}$$

From Eqs (6.12) and (6.13)

$$\frac{\bar{V}_g}{\bar{V}_f} = \frac{x\, v_g\, A_f}{A_g (1-x) v_f} \tag{6.14}$$

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Therefore, $$S = \frac{x}{1-x} \times \frac{A_f}{A_g} \times \frac{v_g}{v_f}$$

Substituting A_f/A_g from Eq. (6.10),

$$S = \frac{x}{1-x} \times \frac{1-\alpha}{\alpha} \times \frac{v_g}{v_f} \tag{6.15}$$

The slip ratio has been measured experimentally and found to vary between 1 and 10, approaching 1 at high pressures (where the liquid and vapour densities approach each other). As pressure increases, the slip ratio decreases.

From Eq. (6.15), by re-arranging

$$\alpha = \frac{x\, v_g}{x v_g + (1-x)\, v_f\, S} = \frac{1}{1 + \frac{1-x}{x} \times \frac{v_f}{v_g} \times S}$$

or, $$\alpha = \frac{1}{1 + \frac{1-x}{x}\psi} \tag{6.16}$$

where $$\psi = \frac{v_f}{v_g} S \tag{6.17}$$

Similarly, $$x = \frac{1}{1 + \frac{1-\alpha}{\alpha} \times \frac{1}{\psi}} \tag{6.18}$$

The axial heat flux distribution to the riser can be used to determine the quality distribution. Using a reasonable value of S (between 1 and 2), a void fraction (α) distribution along the height can be obtained. The mixture density distribution ρ is now found from

$$\rho = (1-\alpha)\rho_f + \alpha \rho_g \tag{6.19}$$

The average mixture density in the riser is now obtained from

$$\rho_m = \frac{1}{H} \int_0^H \rho(z)\, dz \tag{6.20}$$

where z is the axial distance from the bottom of the riser.

In the case of uniform axial heating, the above equation has been solved (ElWakil, 1985) to yield

$$\rho_m = \rho_f - \frac{\rho_f - \rho_g}{1-\psi} \left\{ 1 - \left[\frac{1}{\alpha_e (1-\psi)} - 1 \right] \ln \frac{1}{1-\alpha_e (1-\psi)} \right\} \tag{6.21}$$

where α_e is the void fraction at the riser exit.

The total pressure head produced due to density difference, Eq. (6.4), is balanced by the pressure losses in the loop.

$$\Delta p = gH(\rho_D - \rho_m) = \Delta p_D + \Delta p_R + \Delta p_{bends} + \Delta p_{header} \tag{6.22}$$

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where Δp_D = pressure loss in the downcomer, Pa

Δp_R = pressure loss in the riser, Pa

Δp_{bends} = pressure loss in the bends etc., Pa

Δp_{header} = pressure loss in the header, Pa

provided there is no conversion of pressure to kinetic energy, as the fluid is discharged from the riser into the drum.

Figure 6.13 shows three downcomers, a lower (collecting) drum or header from which water enters the riser tubes, nine heated risers, an upper collecting drum or header, and four pipes for delivery of the steam. Each circulation circuit, operating at the designed boiler pressure, has a definite circulation rate associated with it, and this depends on the thermal loading of that circuit. Variation of this loading causes automatic variation in the circulation and it is impossible to alter it by any external means. The resistance to fluid motion along a pipe is proportional to the square of the velocity, to the configuration (vertical or horizontal) and dimensions of the pipe, and on the density of the fluid. With a given system, the configuration and dimensions are given and so is the density of water and steam, and for a given thermal load, the steam formation rate is known. The circulation rate and the velocity of steam–water mixture are unknown for each circuit. However, there exist definite circulation rates for which the expression, Eq. 6.22, applies.

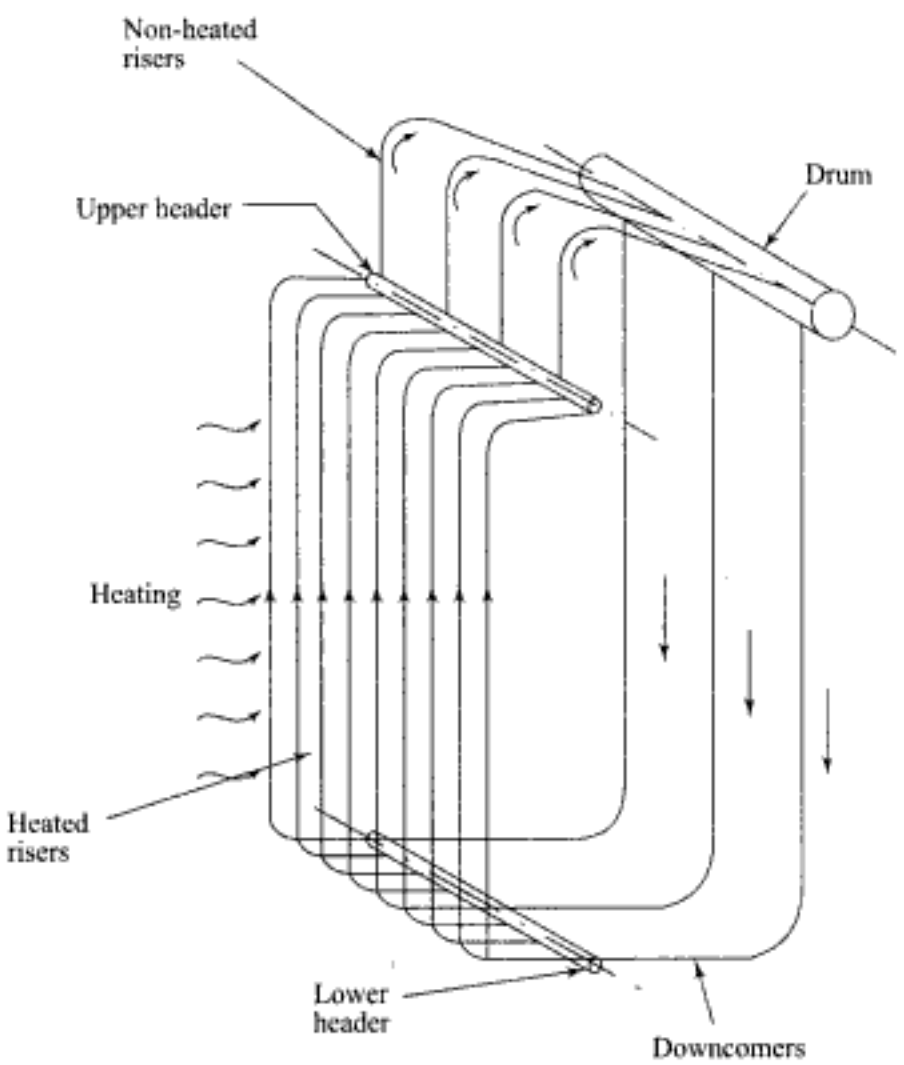

Fig. 6.13 *Circulation circuit in a natural circulation boiler*

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6.3.5 Steam Drum

Risers discharge into the drum a mixture of water, steam, foam and sludge. Steam must be separated from the mixture before it leaves the drum. Any moisture carried with steam to the superheater tubes contains dissolved salts. In the superheater, water evaporates and the salts remain deposited on the inside surface of the tubes to form a scale, which is difficult to remove. This scale reduces the rate of heat absorption, ultimately leading to the failure of the superheater tubes by overheating and rupture. The superheater tubes are exposed to the highest steam pressure and temperature on the inside and the maximum gas temperature on the outside. They are made of the costliest material. So, utmost care is taken so that no damage is done to it by the excessive carryover of moisture with steam. Some of the impurities in the steam may be vaporized silica, which may cause turbine blade deposits.

No vapour bubble should flow along with saturated water from the drum to the downcomers. This will reduce the density difference and the pressure head for natural circulation. The bubbles tending to flow upward may also impede the flow in the downcomer and thus affect circulation. The drum has to secure moisture-free steam going to the superheater and bubble-free water going to the downcomer (Fig. 6.14).

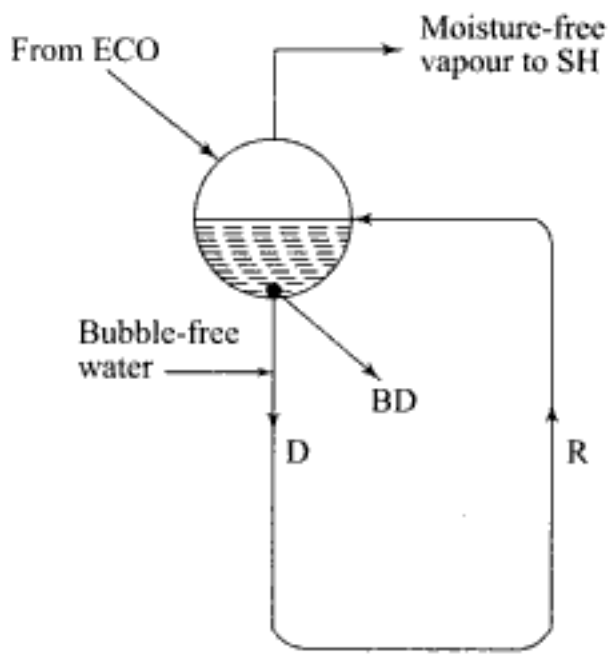

Fig. 6.14 *Separation of two-phase mixture in the drum*

Water is adequately treated prior to feeding it to the boiler as make-up. Still, it has some impurities in the form of total dissolved solids (TDS). These solids are expressed in ppm (parts per million). One ppm has one part impurity in million parts of water, by mass. Saturated water from the economiser is continuously entering the drum. Steam is separated in the drum and is taken to the superheater. So the solid content of water (TDS) in the drum goes on increasing. To maintain a certain ppm in the drum, blowdown is necessary. Blowdown may be intermittent or continuous. In utility boilers, continuous

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blowdown is preferred for proper control of solid concentration and for energy recovery from it, if possible. Solids precipitate at the drum bottom and are removed along with some hot pressurized water by opening the blowdown valve at the drum bottom. Trisodium phosphate or tannin is injected into the drum periodically in suitable doses to help precipitate salts at the drum bottom. If the steam generator is viewed as a steady flow device, continuity of mass requires that the total solids entering be equal to the total solids leaving. Therefore, at steady state for continuous blowdown,

$$\text{Impurities in} = \text{Impurities out}$$

$$\omega_{fw} \times \text{ppm}_{fw} = \text{BD}(\text{ppm})_{drum} + (1-x)\omega_s(\text{ppm})_{carryover} \tag{6.23}$$

where

ω_{fw} = rate of feedwater entering the drum, kg/s
BD = rate of blowdown necessary, kg/s
ω_s = rate of steam flow from the drum, kg/s
x = quality of steam leaving the drum

The ppm of drum water depends on the pressure and varies from 1000 to 2000. The ppm of moisture carryover is maintained at 1–3 ppm.

The functions of the drum in a water-tube boiler can then be summarized as follows:

1. To store water and steam sufficiently to meet varying load requirement.
2. To aid in circulation.
3. To separate vapour or steam from water–steam mixture, discharged by the risers.
4. To provide enough surface area for liquid–vapour disengagement.
5. To maintain a certain desired ppm in the drum water by phosphate injection and blowdown.

Separation of steam from steam–water mixture discharged by the risers is one of the important functions of the drum. At low pressures (up to 20 bar) gravity separation is simply used, if sufficient disengaging surface is provided (Fig. 6.15a). Water being heavier separates out by gravity and steam moves up ($\rho_f >> \rho_g$). But at high pressures, separating force due to gravity is low. As pressure increases, the density difference ($\rho_f - \rho_g$) decreases and so the separating force also decreases. At 100 bar, e.g. the separating force is only 35% of that at 20 bar, and it is difficult to achieve adequate separation and there is considerable moisture carried with steam. An improvement of the situation by increasing the drum size to provide greater disengaging area definitely is not economical in view of the large cost of high pressure drums. Therefore, a positive mechanical means of steam–water separation is required.

Mechanical separators like baffles, screens and cyclones which are housed inside the drum for separation of steam–water mixture are known as *drum internals*. Baffle plates (Fig. 6.15b) act as primary separators. They change or reverse the steam flow direction, thus assisting gravity separation, and act as

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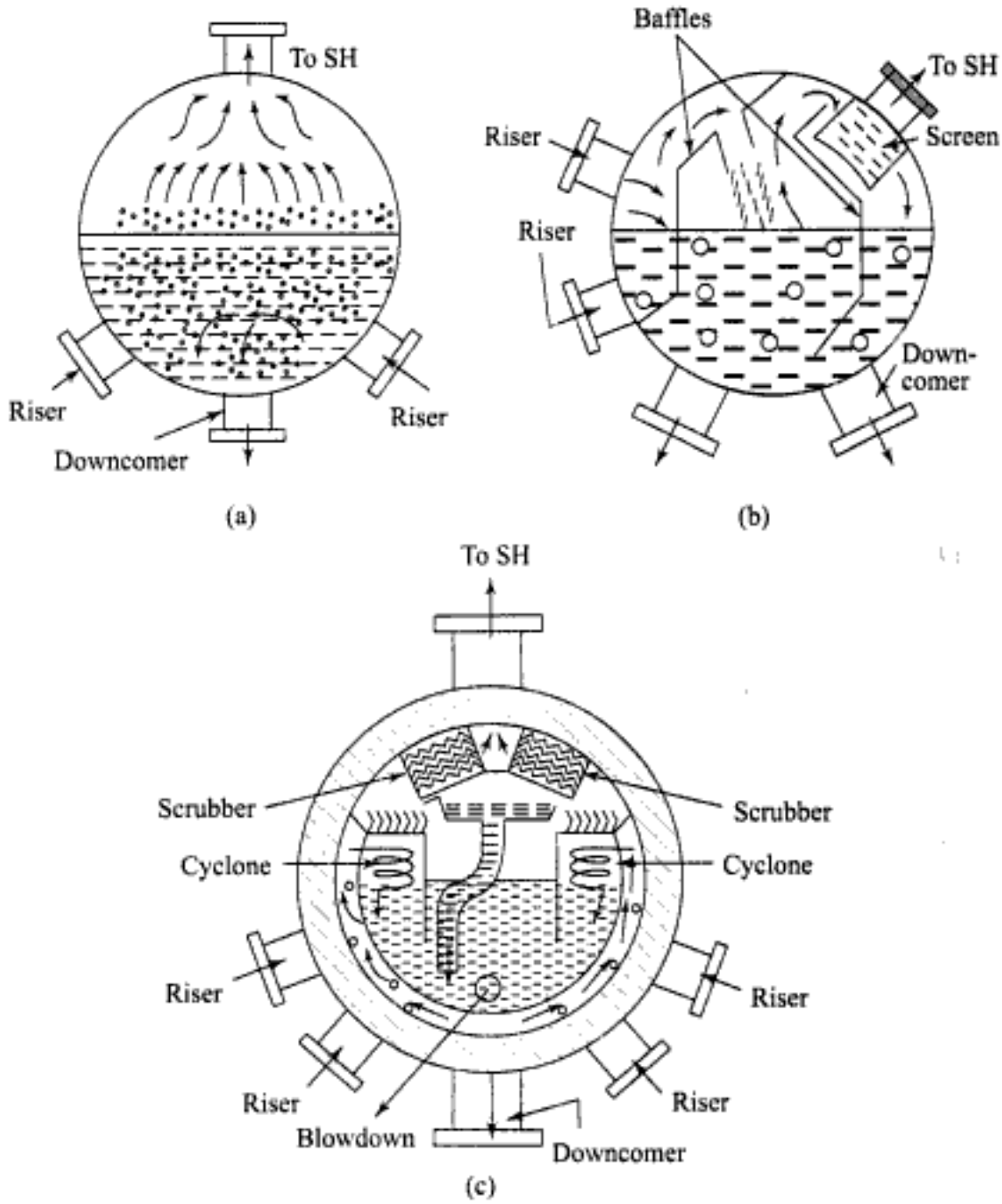

Fig. 6.15 *Steam drum separation (a) gravity, (b) baffle and screen, (c) cyclone and scrubber*

impact plates that cause water to drain off. Screens made of wire mesh act as secondary separators where the individual wires attract and intercept the fine droplets, just as fabric filters attract dust from gases. The accumulating water drops then fall by gravity back to the main body of water.

At high pressures, where the density differential between water and steam diminishes, centrifugal forces, much greater than the gravity forces, are used. Cyclone separators (Fig. 6.15c) utilize the centrifugal forces for separation of two-phase mixture, which is entered tangentially to direct the water downward and to make the steam flow upward. The steam then goes through the zig-zag path in corrugated plates, called the scrubber or dryer, on the way out to help

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remove the last traces of moisture. Finally, perforated plates or screens under the drum exit provide the final drying action. Drum internals of a controlled circulation boiler are shown in Fig. 6.15(d).

Figure 6.15(e) shows the drum of a CEGB boiler (UK) where water from the economiser is supplied to the drum via a feedpipe along its length. A small pipe with a number of small holes along its length is used to distribute chemicals into the drum at intervals. The bottom of the cyclone discharges below the water level to prevent bubbles from going along with saturated water into the downcomers. Steam, while flowing through the scrubbers gets completely dried by a number of sharp changes in direction along its flow path.

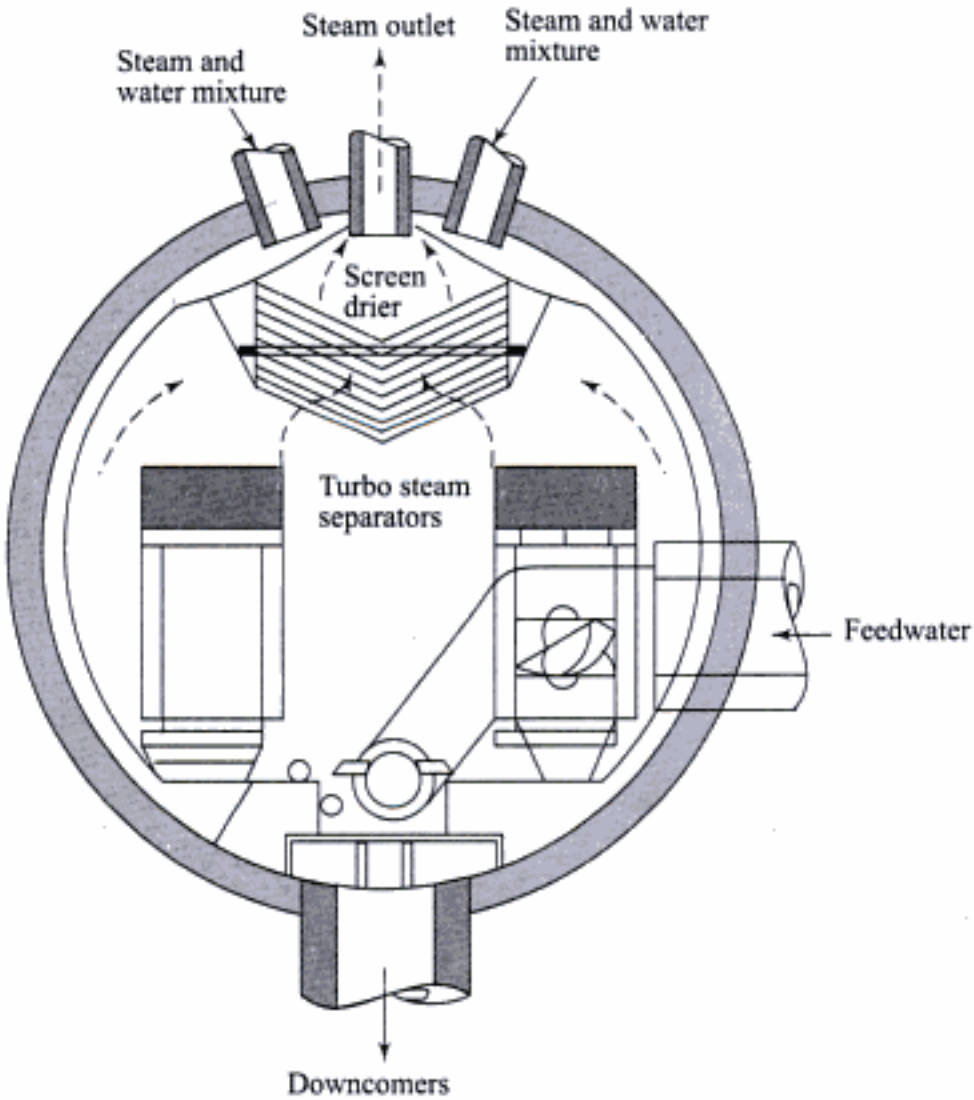

Fig. 6.15 (d) *Drum internals of a controlled circulation boiler*

Sometimes, the separation process involves a "washing" of the steam by passing it through a rain of the relatively pure feedwater before passing through the scrubber. Water of very low ppm is injected to reduce the ppm of the moisture carry-over. The steady state impurity balance gives

$$\omega_i\,(\text{ppm})_i + \omega_{CO}\,(\text{ppm})_{drum} = (\omega_i + \omega_{CO})\,(\text{ppm})_{CO} \tag{6.24}$$

where ω_i = rate of low-ppm water injection, kg/s

ω_{CO} = rate of moisture carry-over, kg/s

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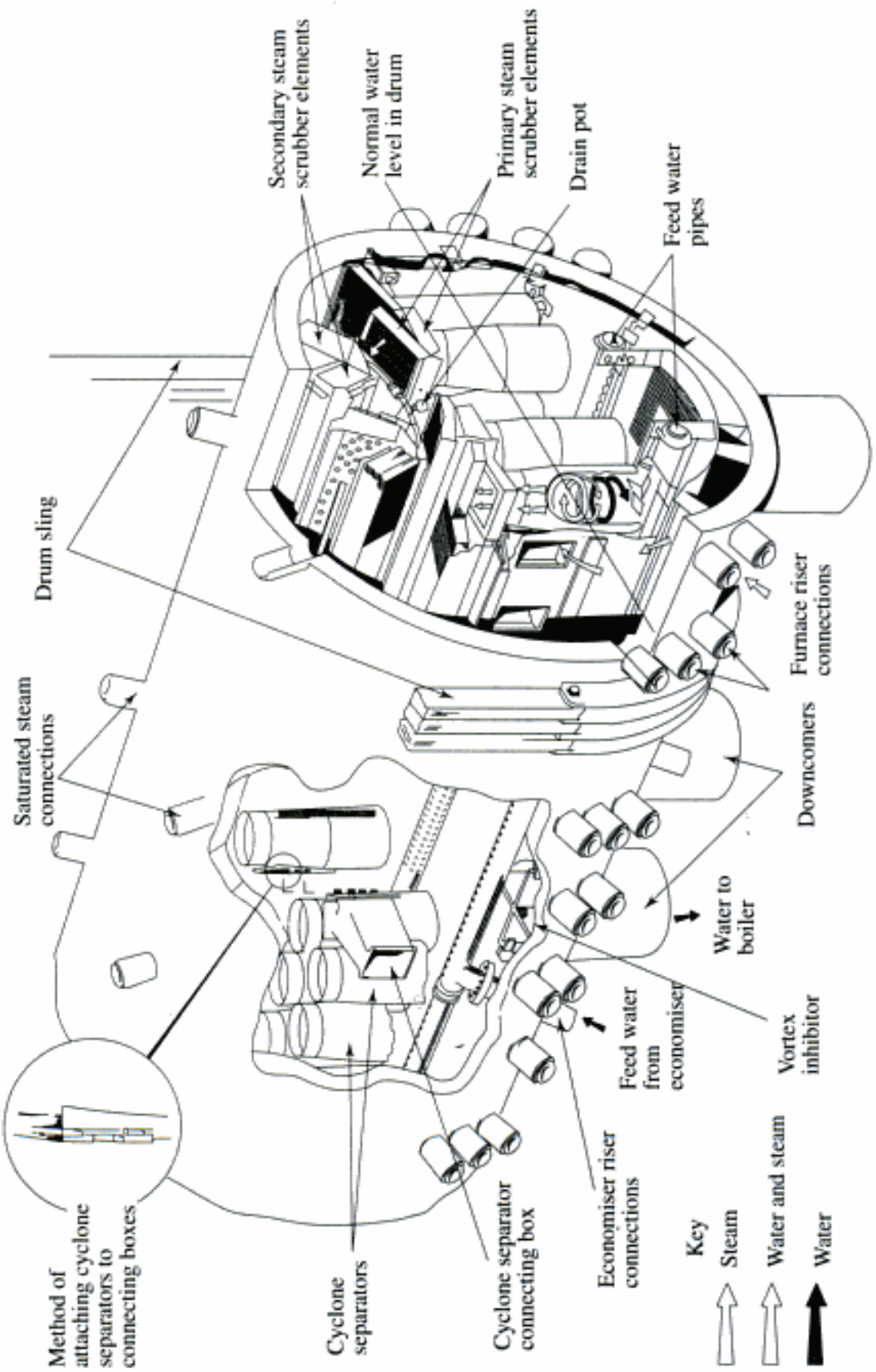

Fig. 6.15 (e) *Exploded view of boiler drum*

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with the suffices "i" and "CO" denoting injection and carry-over respectively. The moisture carry-over will have the same ppm as the drum water. Steam washing also helps in reducing silica concentration in steam. The silica vapour is absorbed in the feedwater and so silica content in steam is reduced to acceptable levels.

A boiler drum is said to be "priming" if there is too much moisture carry-over because of a high water level or a high steaming rate. Erratic feedwater control and rapid changes in steaming rate can induce priming. A certain water level in the drum should be maintained to give adequate steam space and reduce priming.

Another phenomenon encountered in the delivery of impure steam is foaming. Foaming is a condition resulting from the formation of bubbles on the water surface. It is caused by the presence of saponification agents in the boiler water, like oil, certain dissolved salts and high alkalinity. Frequently, a surface blowdown connection is located on the drum at about the normal water level to allow some of the foam to be drawn off the water surface and wasted.

Steam drums in utility boilers have diameters as high as 5 m, lengths more than 30 m and weigh a few hundred tonnes. They are made in cylindrical sections, called courses, which are welded together, and two nearly hemispherical heads which are welded to the ends. Drum size largely depends on the space requirements of all the internals to perform the steps of purification.

6.3.6 Modern Water Tube Boilers

It is now usual in public utilities to have only one boiler per turbine. This has made it possible to build even the largest power plant in unit design thus simplifying the piping systems and facilitating boiler and turbine control, especially in plants using steam reheating.

The appearance of water-cooled furnace walls, called water walls, eventually led to the integration of furnace, economiser, boiler, superheater, reheater, and air preheater into the modern steam generator. Water cooling is also used for superheater and economiser compartment walls and various other components, such as screens, dividing walls, etc.

Three design concepts of water tube boilers are illustrated in Fig. 6.16. Type A is a boiler with natural circulation as is type (a). Heat transfer to the water tubes around the walls is mostly by radiation from the fuel flame and less by convection from flue gases. Natural circulation is used up to steam pressures of approximately 180 bar, with separation of the steam from the water taking place in the boiler drum. Boilers with forced circulation by a special pump, originally known as La Mont boilers, are shown schematically as type B and also (b) in Fig. 6.16. They offer a certain amount of freedom in the arrangement of evaporator tubes and the boiler drum. Such boilers can be adapted to limitations in height and space. They are suitable for steam pressures up to 200 bar. Boilers operating at subcritical pressures (<221.2 bar) which rely on a drum and recirculation, either natural or forced, are commonly known as drum *boilers*. Circulation in the boiler is often controlled by orifices, usually located at the

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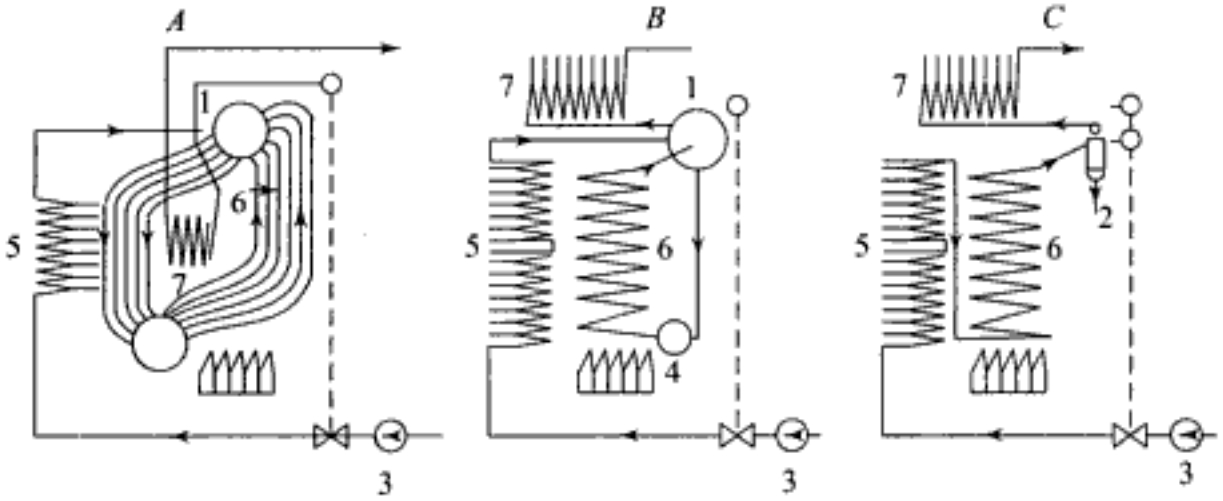

1. Boiler drum
2. Steam separator
3. Boiler feed pump
4. Circulation pump
5. Economizer
6. Evaporator
7. Superheater

A. Natural circulation B. Forced circulation C. Once-through boiler

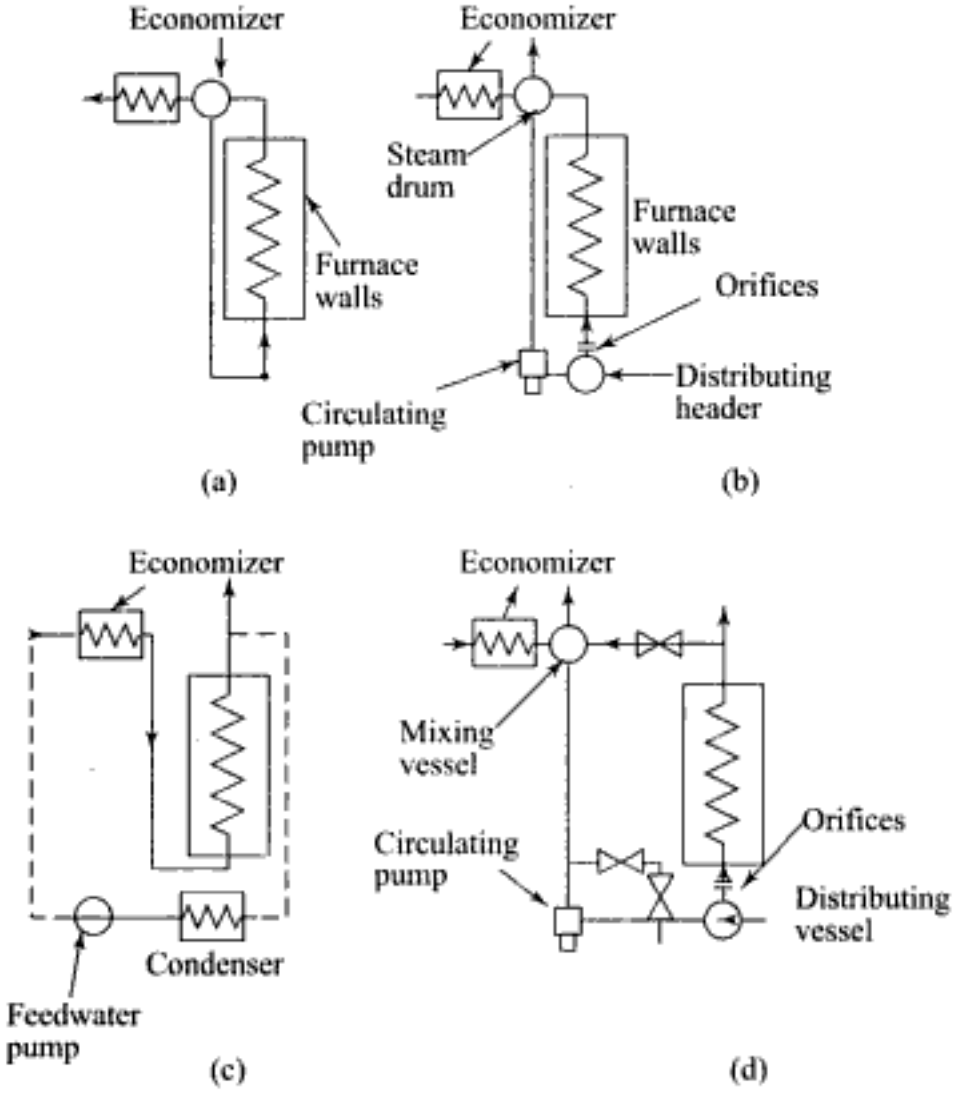

Fig. 6.16 *Types of boiler circulation: A, (a) Natural circulation; B, (b) Forced circulation; C, (c) Once-through; D, (d) Once-through with recirculation (forced)*

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bottom of the tubes. They ensure even distribution of flow through the tubes in the waterwalls.

Boilers can be designed to operate at pressures above the critical pressure (221.2 bar). No drum is necessary in such boilers since no separation of the water and steam occurs, the two being at the same density. Thus, there is no recirculation. The water enters the bottom of the tubes and is completely transformed to steam by the time it reaches the top, passing through the tubes only once. For this reason it is known as a *once-through boiler*, often referred to as drumless boiler. It is illustrated in type C as well as (c) and (d) of Fig. 6.16.

Figure 6.17 shows a schematic flow diagram of a common natural circulation steam generator. Water at 170–260°C (depending on boiler pressure) from the high pressure feedwater heater enters the economiser and leaves saturated or as a two-phase mixture of low quality. It then enters the steam drum at midpoint (lengthwise). Water from the steam drum flows through insulated downcomers, which are situated outside the furnace, to headers. The headers are connected to the water tubes that line the furnace walls and act as risers. The water in the riser tubes receives heat mainly by radiation from the combustion gases and boils. The density differential between the water in the downcomers and that in the risers helps circulation. Steam is separated from the bubbling water in the drum and goes to the convective or primary superheater (CSH) where heat is absorbed by convective mode. From the convective superheater steam goes to the radiant superheater (RSH) installed at the top of the furnace, where heat is absorbed by radiation. Steam leaving the radiant superheater goes to the desuperheater where water of high purity is sprayed on to the steam to bring down the steam temperature to its desired value, if it exceeds the latter. From the desuperheater, steam finally goes to the pendent superheater (PSH) for further superheating before it leaves through the main stop valve to the high pressure (HP) turbine. Both RSH and PSH are often termed as secondary superheaters. Steam exiting the HP turbine goes back to the furnace for being re-superheated to the desired temperature in the reheater. Steam after being reheated flows to the intermediate pressure (IP) turbine for further expansion.

Atmospheric air fed by a forced draught (FD) fan is preheated by the outgoing flue gases in a heat exchanger called air preheater (APH). Air heated to the desired temperature, which varies from 250 °C to 400 °C depending on the type of coal burnt, is split into two streams, primary and secondary. Hot primary air (15–30% of total air) flows to the pulverizer to dry the coal for better grinding and also to help push the coal particles through the burners into the furnace. The hot secondary air flows into the windbox and gets distributed to the burners to help combustion. Fuel burns in the furnace with great intensity to some 1350 °C or even higher, depending on the quality of coal. The combustion gases impart portions of their energy to the water tubes (risers) and then to the superheaters, reheater, and economiser, and leave the latter at about 450 °C. From there they preheat the atmospheric air in the APH, leaving it at about 160°C. Dust collectors and electrostatic precipitators (ESP) collect the entrained solid particles (mostly ash) from the flue gas stream. An induced

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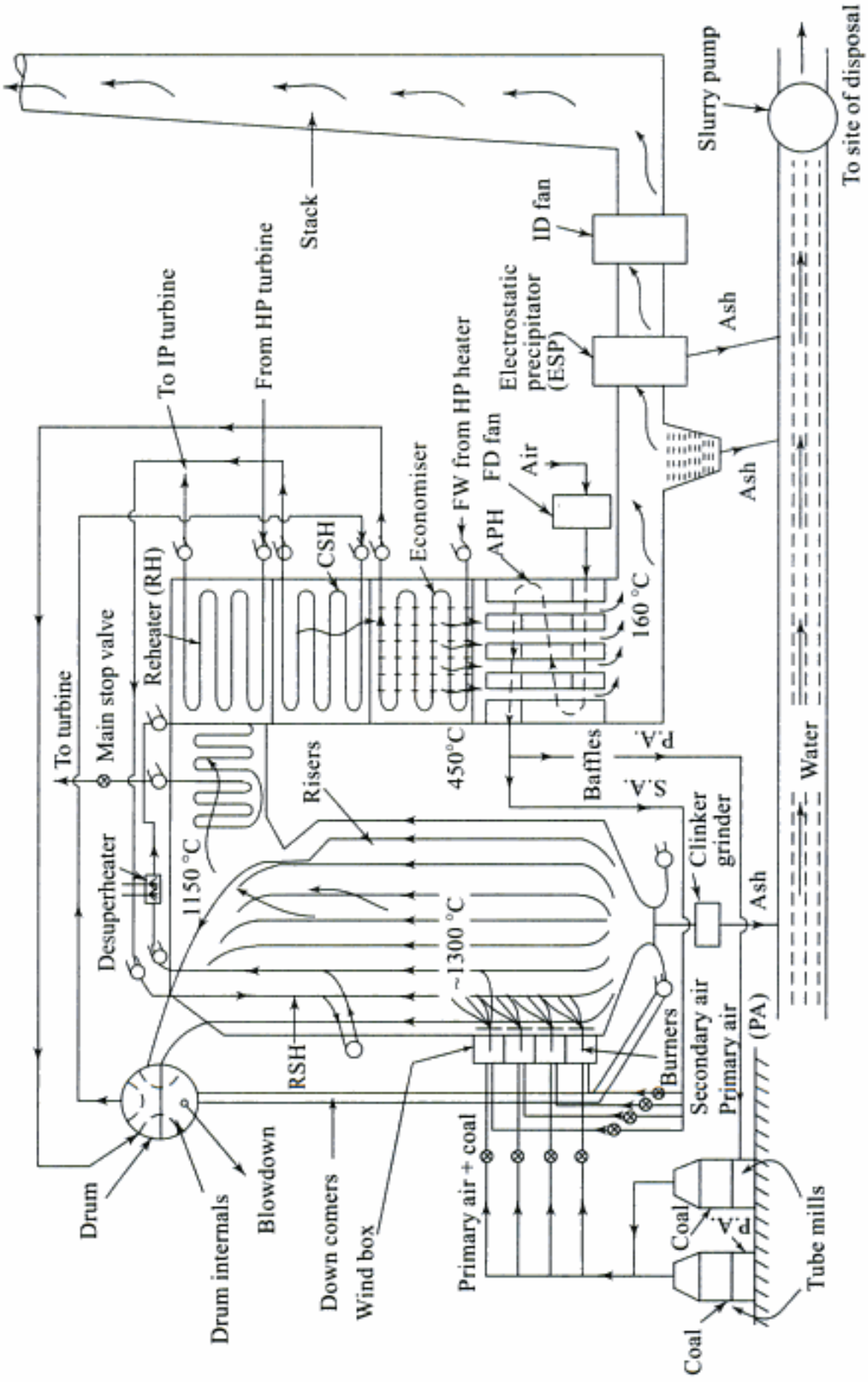

Fig. 6.17 *A typical natural circulation water-tube boiler*

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draught (ID) fan draws the flue gases from the furnace and sends them through the stack to the atmosphere. The exiting gas temperature (160 °C) represents an availability loss to the plant but it is accepted because of the following reasons.

(1) The gas temperature should be kept well above the dew point temperature of the water vapour in the gases (equal to saturation temperature at the partial pressure of water vapour) to prevent condensation which would form acids (with SO_2 or SO_3) that would corrode metal components in its path.

(2) The flue gases must have enough buoyancy to rise in a high plume above the stack for proper atmospheric dispersion.

From the dry bottom furnace the clinkers fall to the clinker grinder to be broken to pieces and then carried by water in a hydraulic sluice whereto also falls the ash from the dust collectors and the precipitator.

Another modern natural circulation two-drum boiler is shown in Fig. 6.18, where bulk of the steam is generated in the radiant boiler tubes along the furnace

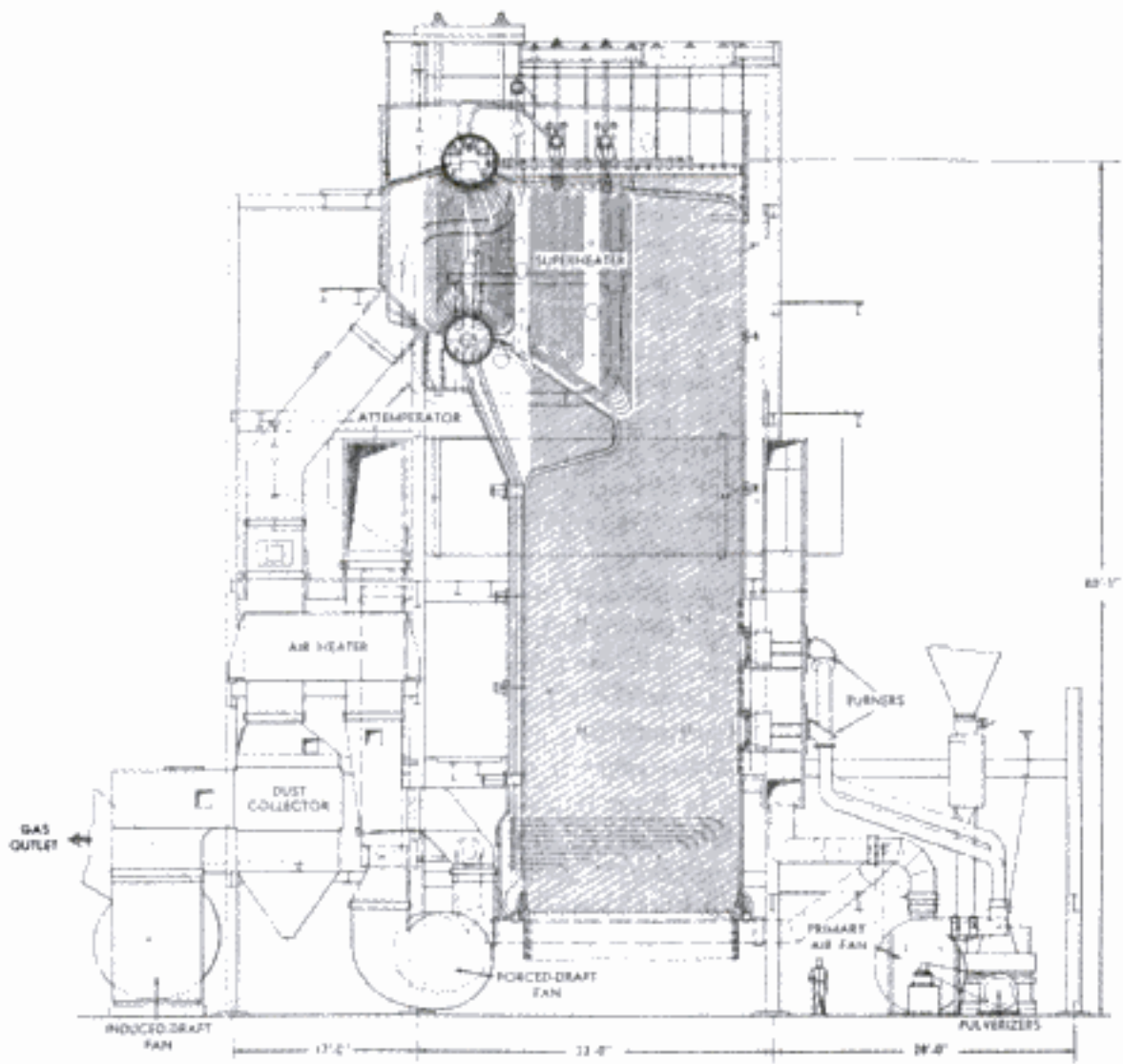

Fig. 6.18 *A two drum natural circulation steam generator*

walls. The steam water mixture flows upward to the main steam drum (upper drum), where a centrifugal separator separates the steam and water. After passing through a dryer, the steam proceeds to the two convective superheaters.

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In the superheater, the hot gases leaving the furnace superheat the steam to the desired temperature. Feedwater is fed to the drum from which water flows downward through the tubes at the far left to the lower drum. This drum feeds liquid to the radiant tubes all around the furnace, where it is partially vaporized. The left hand tube bank acts as an economiser. The exiting gases then proceed to an air heater, where they preheat the combustion air.

Figure 6.19 shows another typical single drum water tube boiler with natural circulation. For this type of boiler the capacity varies from 136 to 3200 t/h. The steam conditions are usually subcritical with throttle pressure 126–175 bar and the temperature around 540 °C. The boiler can use coal, lignites, oil and natural gas as the fuel.

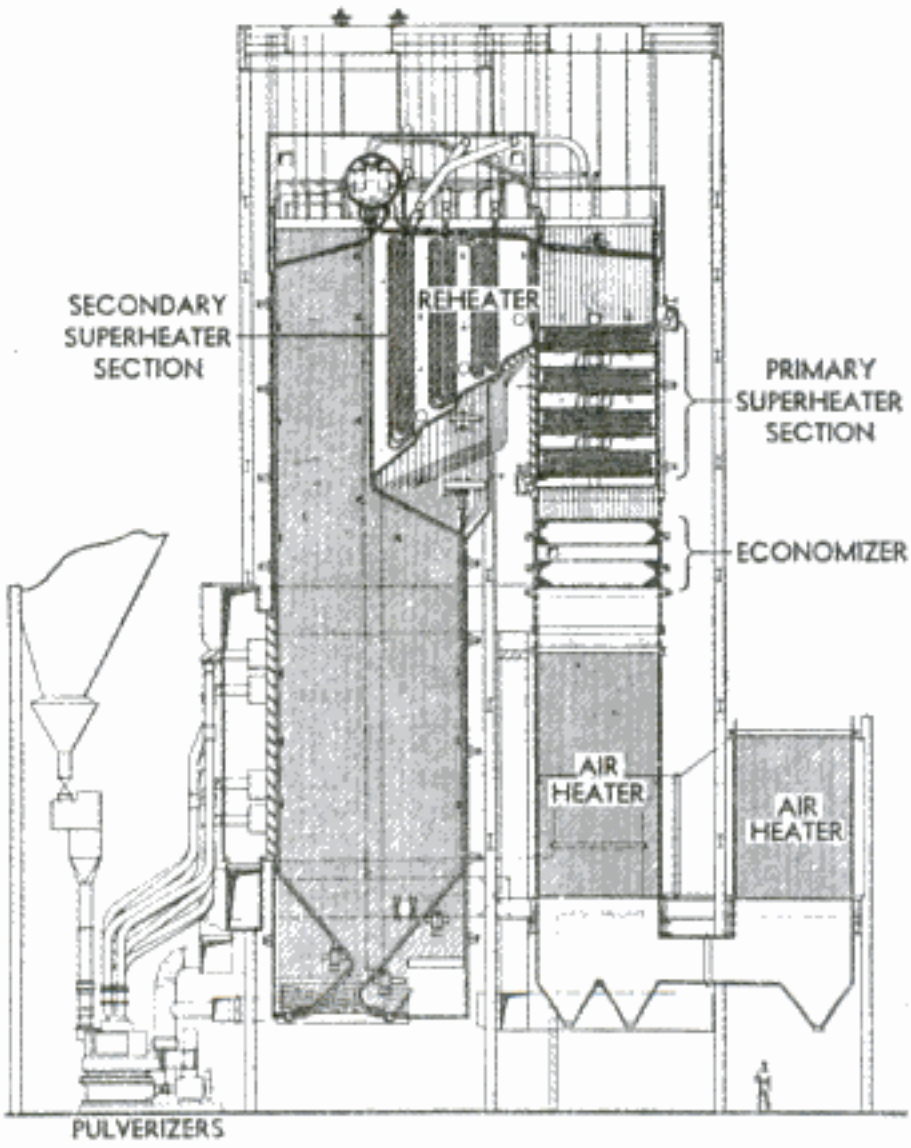

Fig. 6.19 *Radiant boiler for puliverized coal*

The general arrangement of a modern water-tube steam generator showing all the essential components is shown in Fig. 6.20 (a). Figure 6.20(b) shows the cross-section of the 210 MW Babcock downfired boiler installed in Bokaro "B" thermal power station (DVC), which burns low volatile high ash coal. The back pass is divided into two parallel sections: one consisting of horizontal reheater tubes and the other horizontal primary superheaters. The economiser is located below both the banks across the flow path.

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The 500 MW NTPC units have single drum natural circulation reheat units with steam condition at 170 bar, 540 °C/540 °C, while the 210 MW units have steam condition at 135 bar, 540 °C/540 °C. The CEGB (UK) has 660 MWe boilers operating at 165 bar outlet pressure with superheater and reheater temperatures of 568 °C, and designs are now being offered of 900 MWe capacity with steam conditions of 182 bar, 568 °C/568 °C. The superheater outlet pressure of these latter boilers represents the upper limit of drum-type natural circulation boilers. Figure 6.20(c) shows the schematic of a 660 MWe CEGB boiler, while Fig. 6.20(d) shows the gas and steam/water flow paths between the boiler components. The back pass is divided into two parallel gas paths, one containing primary superheater and the other convective reheater with an economiser across both. Figure 6.20(e) gives a comparison of fossil-fired boilers over about 25 years, from 1952 to 1974, with evaporation capacity increasing from 55 to 560 kg/s and the furnace height from 32 to 61.4 m.

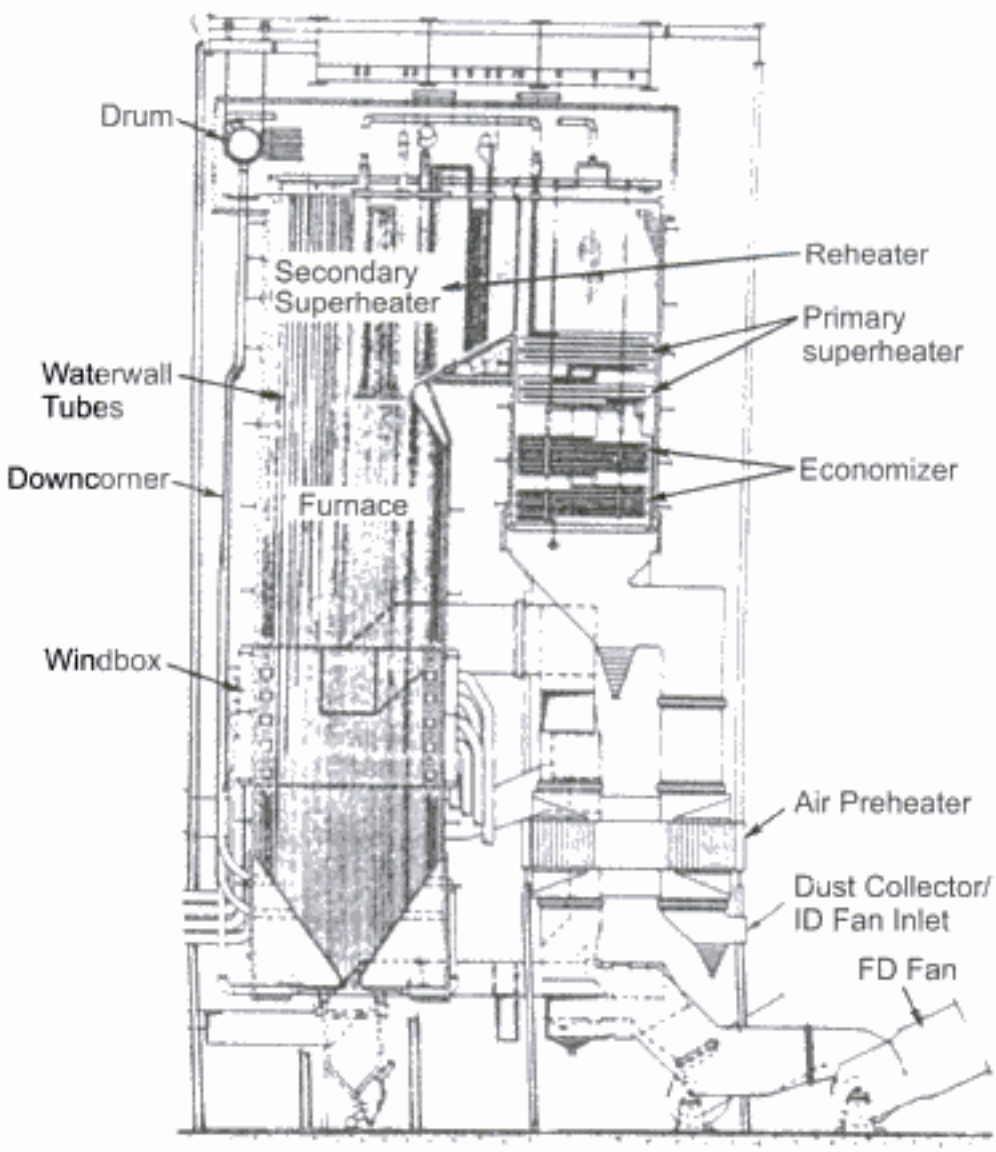

Fig. 6.20 (a) *General arrangement of a modern water-tube steam generator*

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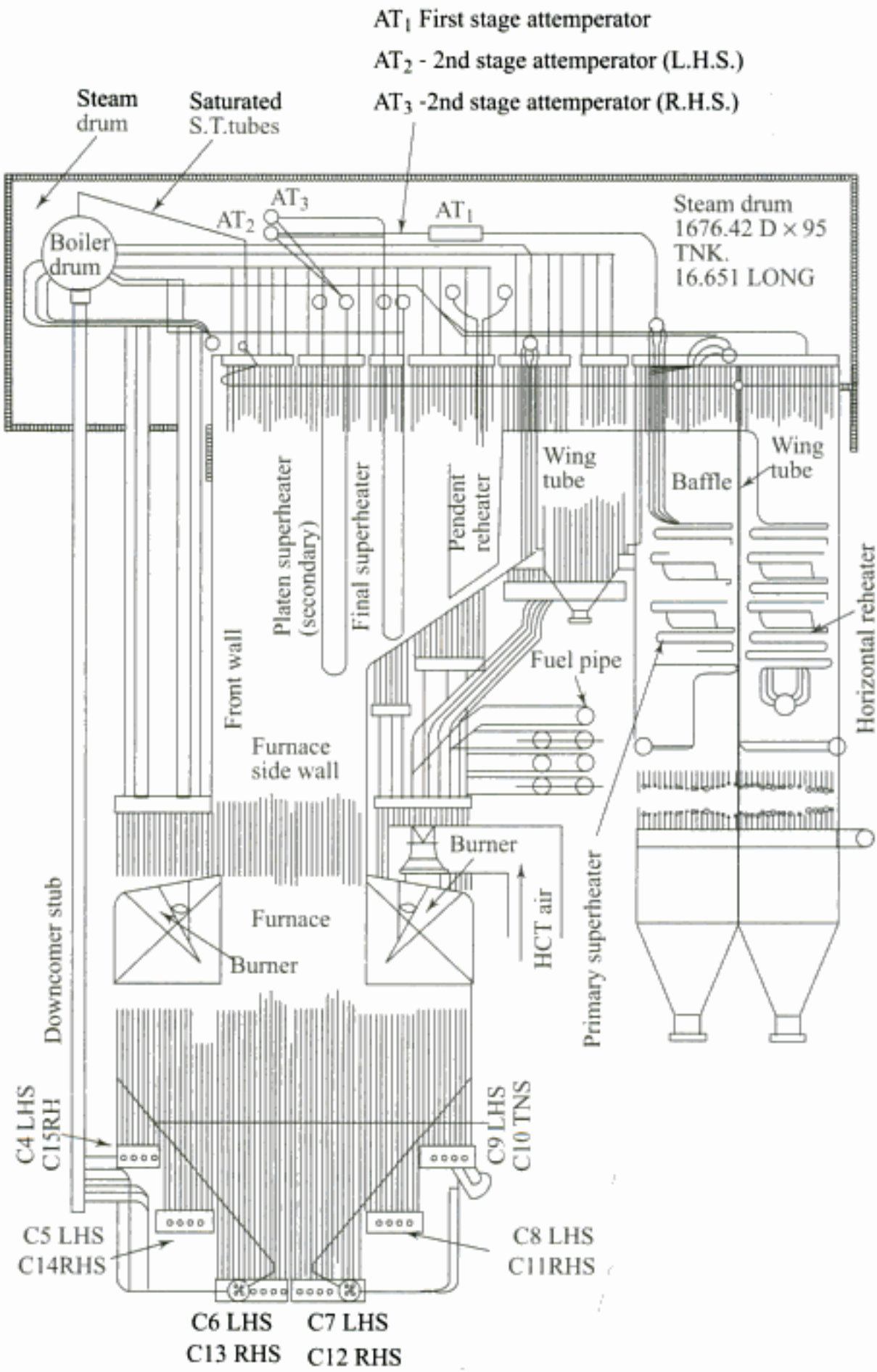

Fig. 6.20 *(b) Cross-section of a 210 MW Babcock down-fired steam generator (Bokaro 'B' thermal power plant, DVC)*

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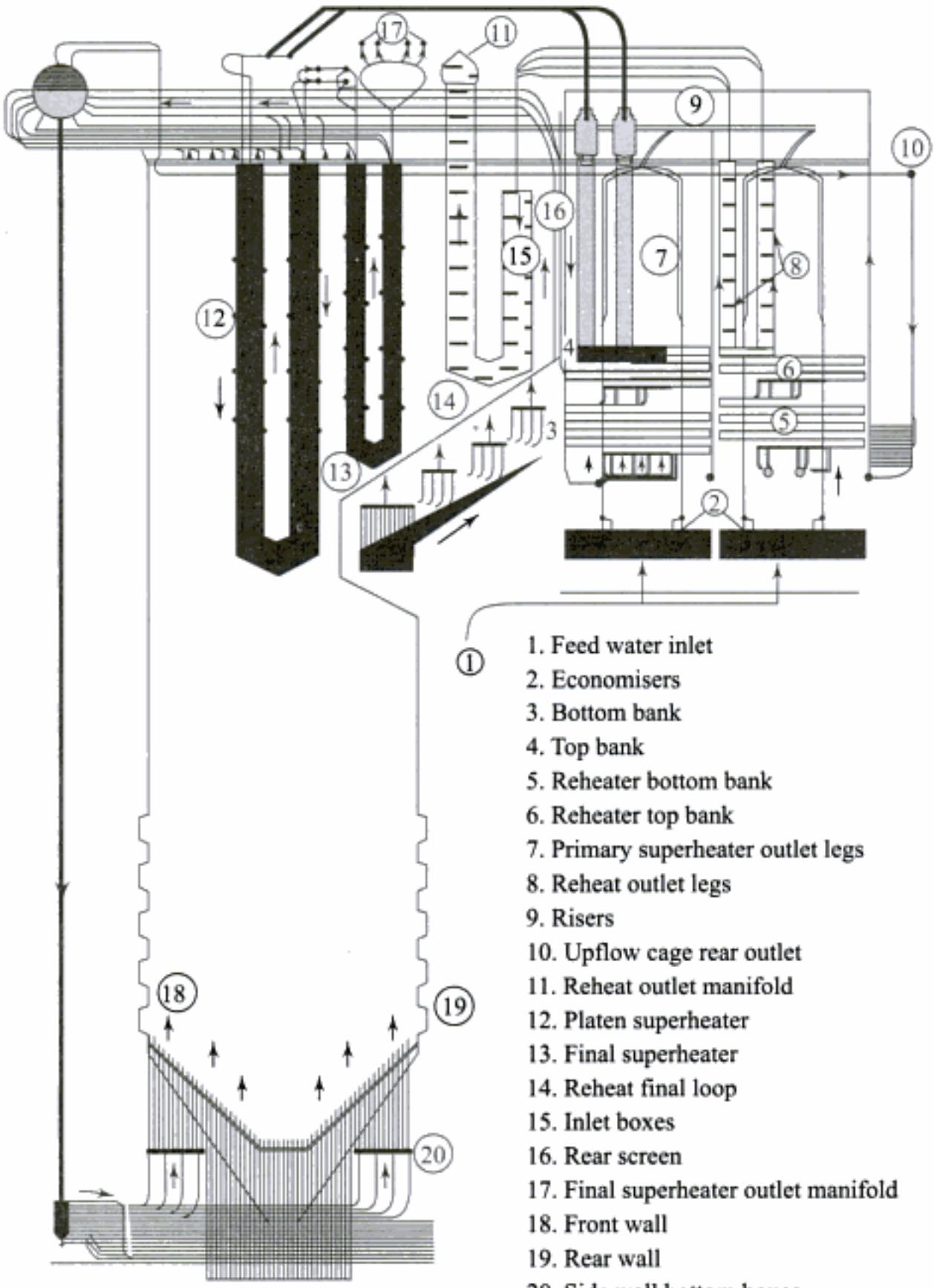

Fig. 6.20 *(c) Section through a 660 MW CEGB steam generator*

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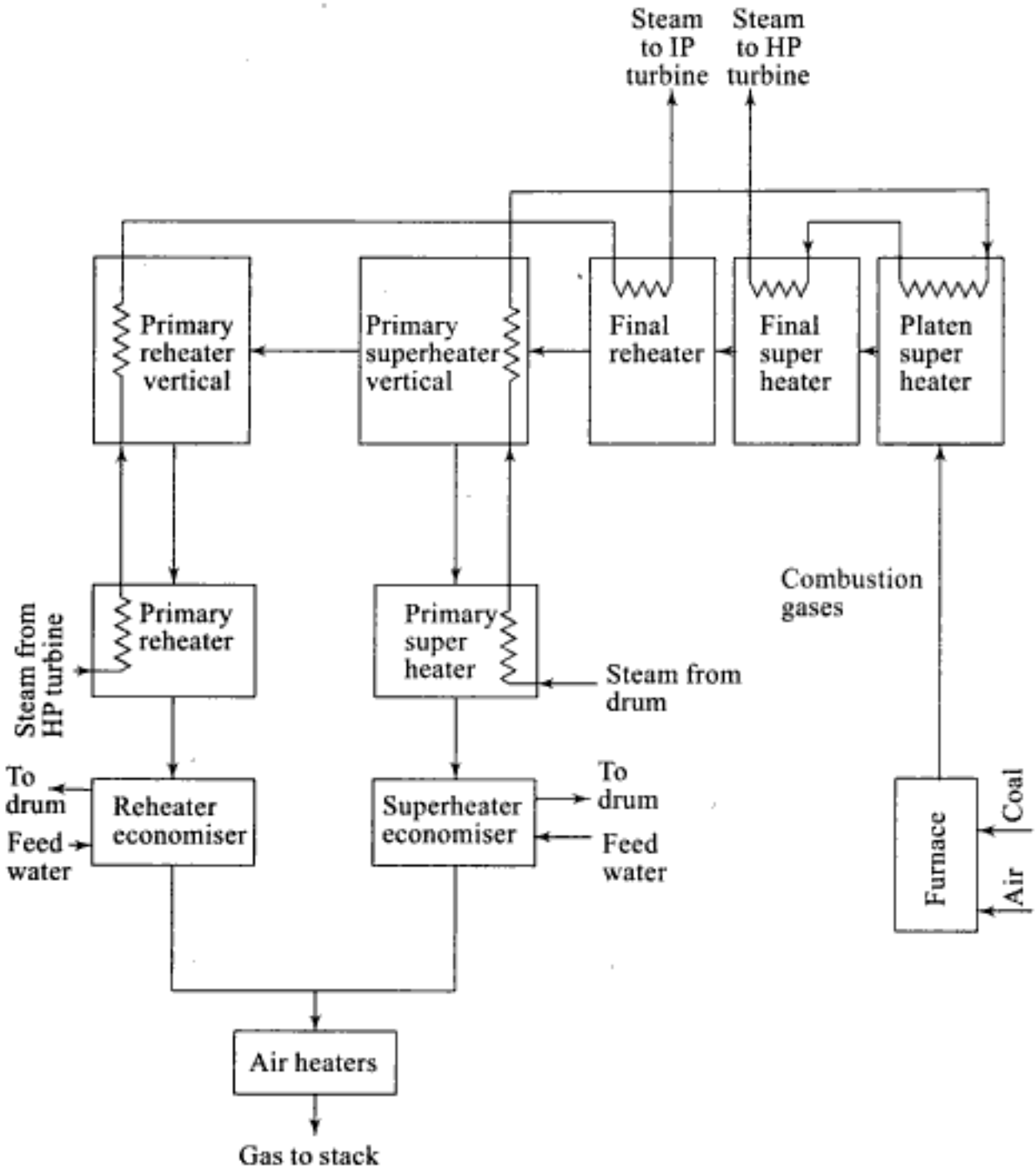

Fig. 6.20 *(d) Block diagram showing boiler elements and flowpaths in the 660 MW CEGB steam generator*

Boiler water wall The water wall consists of a number of water or evaporation tubes (risers) spaced all around the walls of the furnace and carries away the heat released in the furnace by the burning of fuel. The water tubes could be installed in various ways as shown in Fig. 6.21 such as (a) tangent tubes touching the refractory wall, (b) tubes embedded in the refractory, (c) studded tubes, and (d) membrane tubes. The membrane water wall is now more common. It consists of tubes spaced on centres having centre-to-centre distance varying from 1.2 to 1.3 times the tube diameter. The tubes are connected by bars or membranes welded to them at their centre lines. The membranes act as fins to increase the heat trasfer as well as to strengthen the

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furnace construction. Insulation is provided on the outer side of the wall, and then a metal lagging is given to protect the insulation. The tube diameter varies from 68 to 76.2 mm.

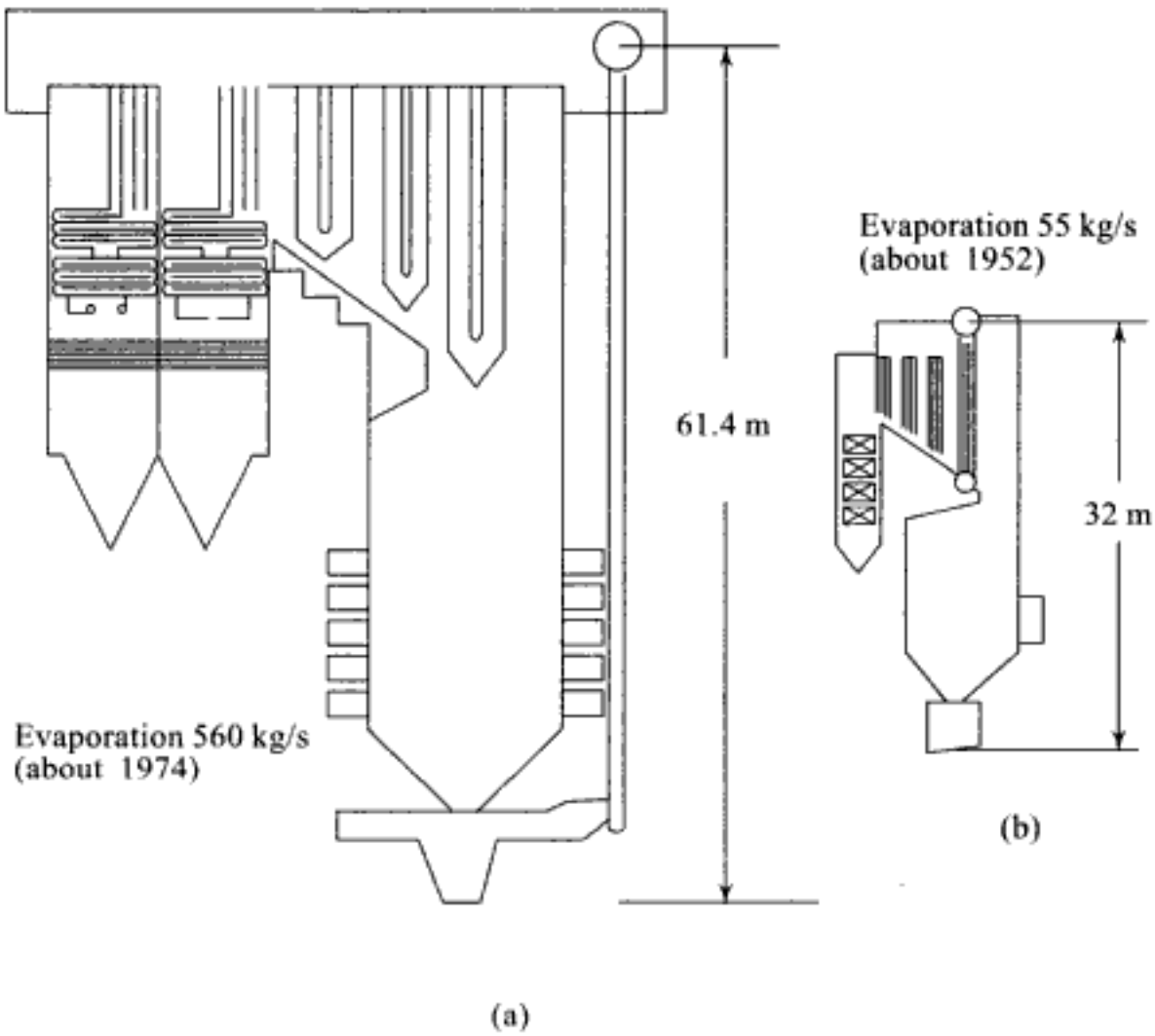

Fig. 6.20 *(e) Comparison of fossil fuel-fired steam generators over 25 years*

The water wall absorbs heat mainly by radiation. Heavier triatomic gases like CO_2, H_2O vapour, SO_2 etc. are selective radiators emitting and absorbing radiation in certain ranges of wavelengths, mostly outside the visible range. The radiant energy absorbed by the water wall depends on the $(T_g^4 - T_w^4)$, T_g and T_w being the gas and wall temperature (K) respectively, the partial pressures of the gases (p_{H_2O}, p_{CO_2} etc.) and the shape and size of the gases (view factor). This heat is conducted through the membranes and tube walls and is then convected to the two-phase mixture inside the tubes by nucleate boiling heat transfer. The heat transfer resistance of the latter is much smaller than the others and is often neglected in design calculations.

Forced circulation and once-through boilers A number of modern boilers use forced circulation rather than natural circulation. One scheme (La Mont type) is illustrated in Fig. 6.22. The operation is basically similar to a natural circulation unit, with a pump being used below the down-

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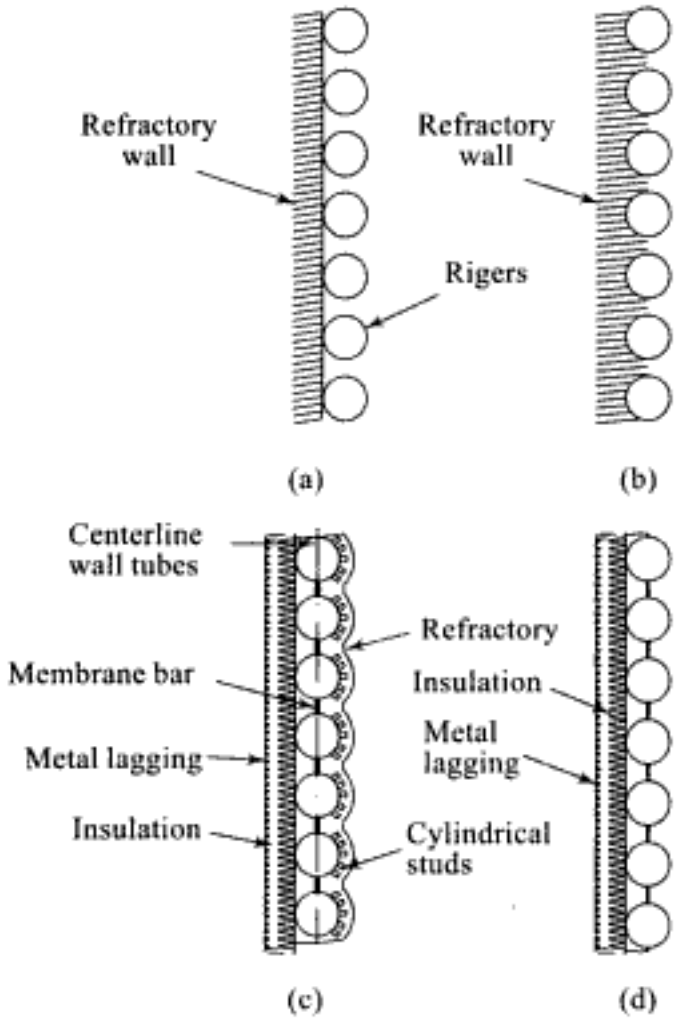

Fig. 6.21 *Different types of water-wall: (a) tangent tubes touching the refractory wall, (b) tangent tubes embedded in the refractory, (c) studded tubes, (d) membrane tubes*

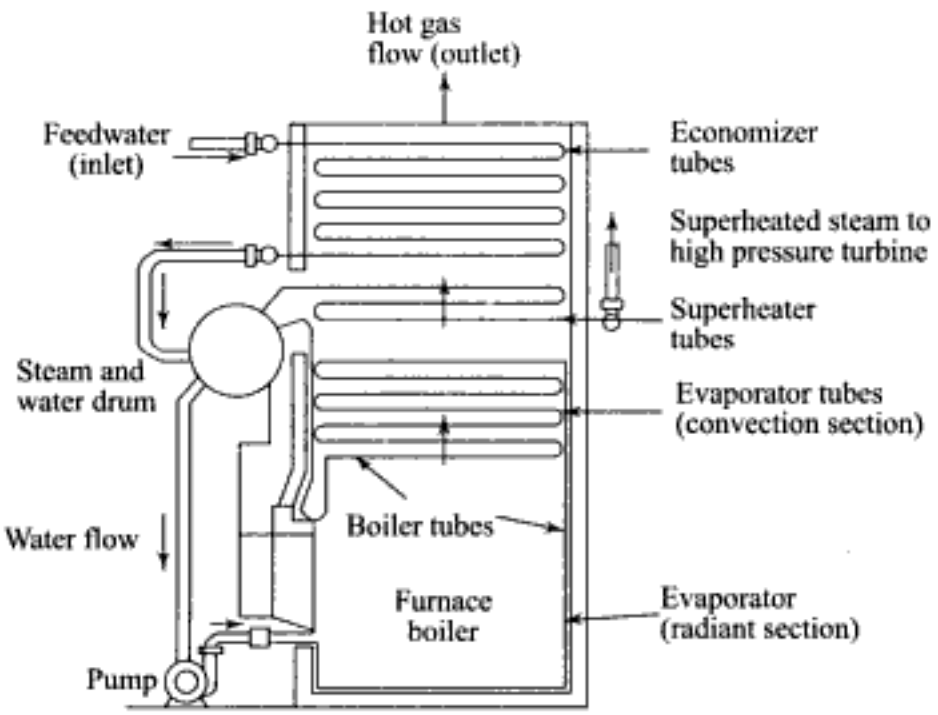

Fig. 6.22 *Modern forced circulation steam generator of* La *Mont type*

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comer for circulating water through the risers. These are used at very high pressures (often above 160 bar), where the reduced density difference between liquid and vapour limits the natural circulation. They are often called pump-assisted circulation units since the natural circulation head is still significant. Assisted circulation boilers can supply between 130 to 3000 t/h of steam, depending on the design, at pressures ranging from 115 to 165 bar.

Both natural circulation and assisted circulation steam generators are designated as recirculating units, since only a fraction of water is evaporated in the riser tubes, and the steam separated in the drum proceeds to the superheater (Fig. 6.23a). In an once-through (OT) boiler, the feedwater goes through the economiser, furnace walls, and superheater sections, changing sequentially to saturated water, saturated steam, and superheated steam in one continuous pass (Fig. 6.23b). No steam drum is required to separate saturated steam from boiling water and no water recirculation takes place. Reheating of steam after it is expanded in the h.p. turbine is performed in a reheater in the usual manner.

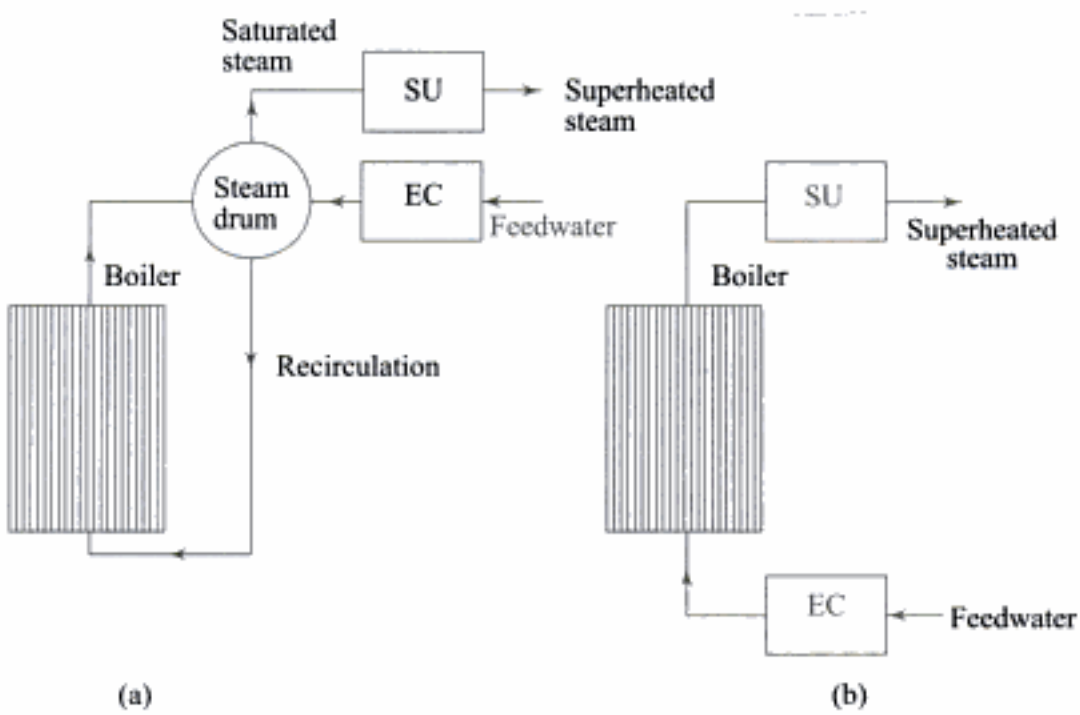

Fig. 6.23 *Schematic flow diagram of (a) drum type and (b) once-through steam generators. SU = superheater, EC = economiser*

The once-through boiler is the only type suited to supercritical pressure operation (above 221.2 bar) because there is no latent heat of vaporization involved, and water on being heated at constant (assumed) supercritical pressure suddenly flashes into vapour, there being no saturation temperature as such, and the vapour is further heated to the desired temperature in the superheaters. Although there is no drum, but there should be a transition section where the water is likely to flash in order to accommodate the large increase in volume. Water enters the boilers at one end of a number of parallel paths and emerges at the outlet as superheated steam. Movement of the water is forced by the boiler feed pump.

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A large number of pioneering super-critical pressure once-through boilers were built for the utility industry, many with double reheat, with steam pressures of the range 310–345 bar and steam temperatures 620–650 °C and cross compound (two-shaft) turbines. To reduce operational complexity and improve equipment reliability, moderate steam conditions of 240 bar and 540–565 °C were used. However, the use of higher steam conditions lowers the heat rate. Figure 6.24 shows a simplified view of a C.E. Sulzer Monotube Steam Generator with a triflux reheater where steam is reheated partly by steam and partly by the gas. The name derives from the feature of using continuous tubes from the inlet to the outlet of the evaporating section. This avoids the distribution problem inherent in a design employing intermediate headers.

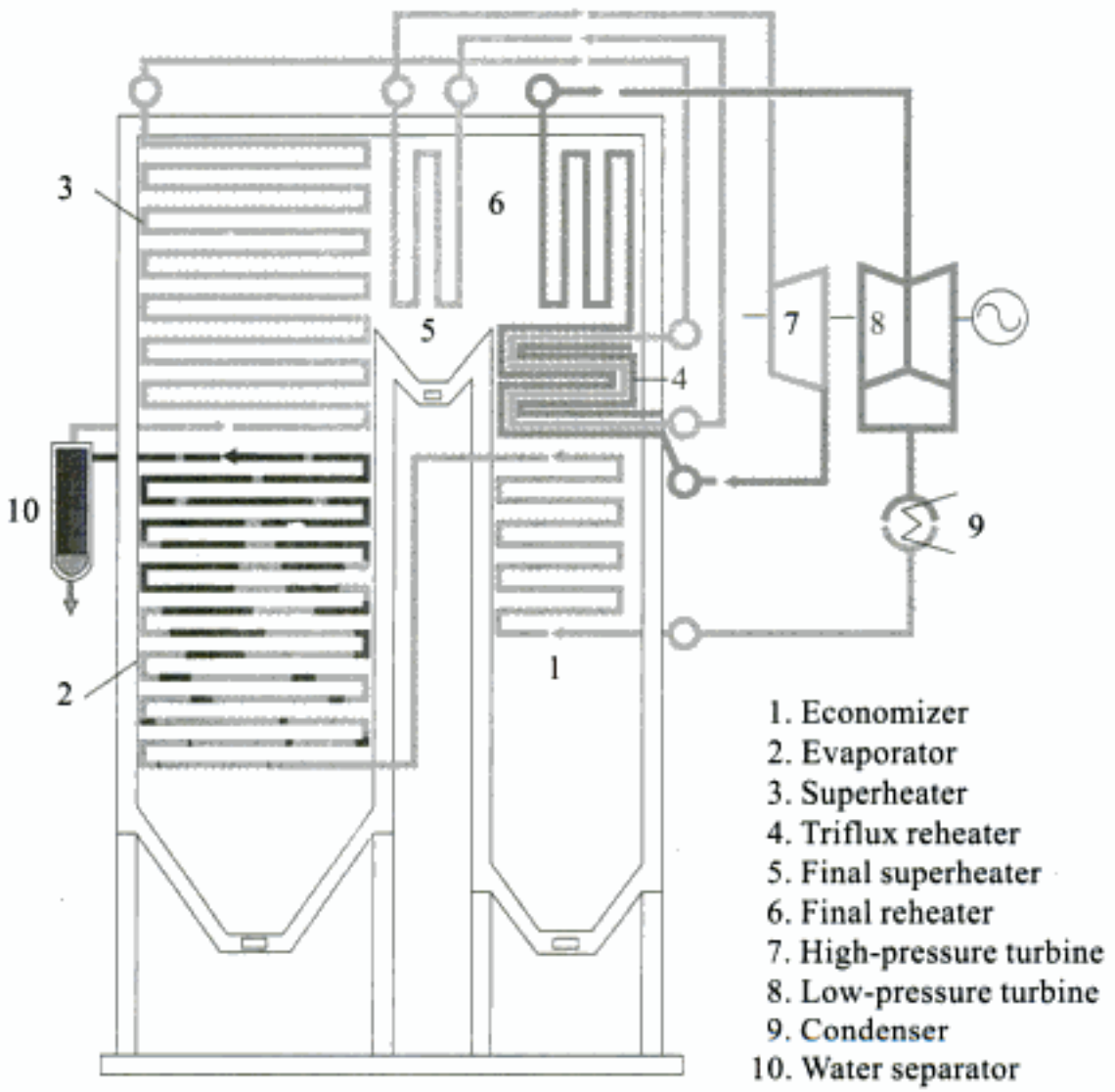

Fig. 6.24 *Simplified diagram of a sulzer monotube steam generator with triflux reheater*

Subcritical boilers may also use the once-through design, incorporating steam and water separation and removal of a small amount (about 5%) of the water to permit blowdown of concentrated solids. The economical range of pressure varies from 138 to 276 bar and the steam output varies from 3.8 to 1260 kg/s.

A principal advantage of the once-through boiler is that it does not require circulating pumps or drums. Energy required for circulation is provided by the feed pump. The design is well suited to quick starts and rapid load changes, and

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since it utilizes small diameter tubes, it has no drum, and provides positive circulation under all operating conditions. The prime disadvantage is the requirement of extremely pure boiler water, about 0.1 ppm TDS in the supercritical design, since all solids present are deposited in the tubes or carried along with steam to turbine. In the subcritical pressure range, the limitation is slightly less severe since there is some blowdown of boiler water. The capital costs for a supercritical steam generator are a few per cent higher than those of a drum type subcritical one of the same capacity, but because of the increased efficiency of the power plant the capital costs of the turbogenerator, as well as other components (condenser, cooling towers, feedwater heaters, etc.) are lower. The net effect is that the total production costs of electricity are lower for the supercritical cycle with double reheat.

Forced circulation is often adopted in low pressure waste heat boiler (HRSG) to enhance heat transfer (absorption) from waste gases since the heat transfer coefficient in forced circulation riser is more than that in natural circulation.

6.4 ECONOMISERS

An economiser is a heat exchanger which raises the temperature of the feedwater leaving the highest pressure feedwater heater to about the saturation temperature corresponding to the boiler pressure. This is done by the hot flue gases exiting the last superheater or reheater at a temperature varying from 370 °C to 540 °C. The term "economiser" was used historically because the throwing away of such high temperature gases involved a great deal of energy loss. By utilizing these gases in heating feedwater, higher efficiency and better economy were achieved, and hence the heat exchanger was called "economiser".

Modern economisers are often designed to allow some boiling of the feedwater in the outlet sections, up to 20 per cent quality at full power, less at part loads. They are often termed as "steaming economisers".

Economiser tubes are commonly 45–70 mm in outside diameter and are made in vertical coils of continuous tubes connected between inlet and outlet headers with each section formed into several horizontal paths connected by 180° vertical bends. The coils are installed at a pitch of 45 to 50 mm spacings, which depends on the type of fuel and ash characteristics.

Figure 6.25 shows an economiser coil and the temperature profiles of flue gas and water heated to saturation temperature.

The rate of heat transfer from the flue gases to feedwater is given by

$$\begin{aligned} Q_{\text{ECO}} &= \omega_g c_{pg} (t_{g1} - t_{g2}) \\ &= \omega_{fw} c_{pfw} (t_{sat} - t_{fw}) \\ &= U_o A_o \Delta t_{l.m} \end{aligned} \tag{6.25}$$

where $\Delta t_{l.m}$ = log–mean temperature difference

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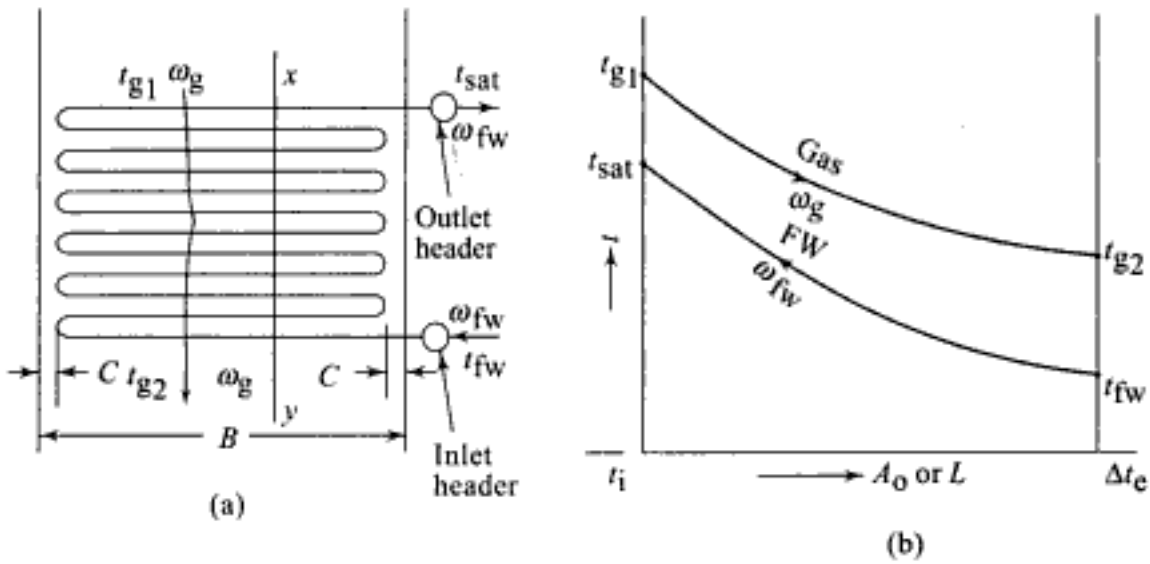

Fig. 6.25 *Economiser coil and the temperature profiles of flue gas and water*

$$\Delta t_{l.m.} = \frac{\Delta t_i - \Delta t_e}{\ln \dfrac{\Delta t_i}{\Delta t_e}}$$

$$\Delta t_i = t_{g1} - t_{sat}, \Delta t_e = t_{g2} - t_{fw} \quad \text{and}$$

$$\frac{1}{U_o\ A_o} = \frac{1}{h_{sc}\ A_i} + \frac{1}{h_i\ A_i} + \frac{x_w}{k_w\ A_{l.m}} + \frac{1}{h_{fo}\ A_o} + \frac{1}{h_o\ A_o} \tag{6.26}$$

where U_o = overall heat transfer coefficient, h_{sc} = heat transfer coefficient due to scale formed on the inside of the tubes, h_i = inside heat transfer coefficient (water side), k_w = thermal conductivity of wall material, $A_i = n\ \pi\ d_i\ l$, $A_o = n\ \pi\ d_o\ l$, l = length of one coil, n = number of coils, $A_{l.m}$ = the log-mean area $= (A_o - A_i)/(\ln A_o/A_i)$, x_w = thickness of tube wall, h_{fo} = heat transfer coefficient due to solids (ash and soot) on the outside surface of tubes, and h_o = heat transfer coefficient of the outside film (gas-side). If the resistanaces due to scale $(1/h_{sc} A_l)$ and fouling $(1/h_{fo} A_o)$ are neglected (considering them to be clean), and if the wall thickness is small, then the overall heat transfer coefficient is

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{x_w}{k_w} + \frac{1}{h_o} \tag{6.27}$$

The gas-side heat transfer coefficient is much less than the water-side heat transfer coefficient, i.e. $h_o << h_i$. To compensate this, the outer surface of the tubes may be finned to increase the surface area of heat transfer. Both gas and liquid flow under forced convection, for which

$$\text{Nu} = f(\text{Re}, \text{Pr})$$

where, Nu = Nusselt number = hd/k_f, Re = Reynolds number = $\overline{V}\ d/\nu_f$, ν_f = kinematic viscosity of fluid, Pr = Prandtl number = $(c_p\ \mu_f)/k_f$, μ_f = dynamic

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viscosity of fluid, and k_f = thermal conductivity of fluid film. If the flows are turbulent, Dittus-Boelter equation can be used to evaluate heat transfer coefficient as given below

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{n} \tag{6.28}$$

where $n = 0.4$ if the fluid is heated and $n = 0.3$, if the fluid is cooled.

With the help of Eqs (6.27) and (6.28), U_0 can be estimated, and from Eq. (6.25) the surface area A_0 needed can be ascertained, where $A_o = n\,\pi\,d_o\,l$.

Since feedwater is flowing through the tubes, at exit from the economiser, with water being saturated,

$$\omega_{fw} = \left(n\frac{\pi}{4}d_i^2\right)\frac{V_{fw}}{v_f} \tag{6.29}$$

where V_{fw} = velocity of fluid at exit and v_f = specific volume of saturated water.

Higher is the fluid velocity (or Reynolds number), less will be the film thickness and more will be the heat transfer coefficient. But higher velocity means more pumping power required, i.e. more power is required to be imparted to the fluid ($\dot{V}\,\Delta p$), where $\dot{V}$ is the volume flow of fluid ($= \omega_{fw}\,v_f = A_i\,V_{fw}$) the power required is thus proportional to the cube of fluid velocity (since $\Delta p\ \alpha\ V_{fw}^2$). These two conflicting considerations of higher heat transfer and more pumping power lead to the optimization of fluid velocity. On the gas-side, the velocity of flue gas is about 10–12 m/s and on the water-side, the water velocity is limited to 1–1.2 m/s. If the water velocity V_{fw} is prescribed, from Eq. (6.29), the number of parallel coil paths required to accommodate the given flow can be estimated, and since A_o has already been computed, the length of one coil (l) can be determined.

If the clearance C is given on two sides of the gas duct of width B (Fig. 6.25), then the number of turns, n_t, of one coil can be estimated from

$$n_t = \frac{l}{B - 2C} \tag{6.30}$$

Again, if the vertical pitch or the centre-to-centre distance of the consecutive horizontal tubes forming the coil, p, (Fig. 6.26) is assumed, then the height of duct occupied by the economiser is equal to

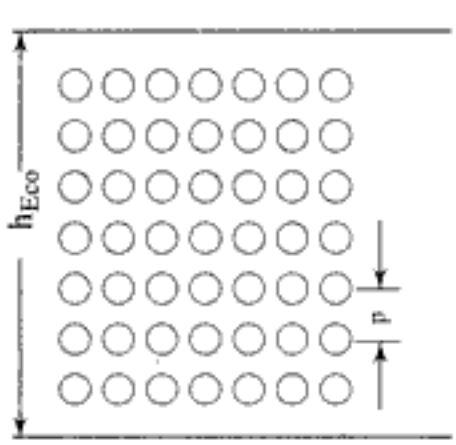

Fig. 6.26 *Section x–y in Fig. 6.25(a): height of the economiser duct*

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$$h_{ECO} = n_t \times p \tag{6.31}$$

The economiser is generally placed between the convective superheater and the air preheater.

6.5 SUPERHEATERS

The superheater is a heat exchanger in which heat is transferred to the saturated steam to increase its temperature. It raises the overall cycle efficiency. In addition, it reduces the moisture content in the last stages of the turbine and thus increases the turbine internal efficiency.

In modern utility high pressure boilers, more than 40% of the total heat absorbed in the generation of steam takes place in the superheaters. So, large surface area is required to be provided for superheating of steam.

Superheaters are commonly classified as either convective superheaters, radiant superheaters or combined superheaters, depending on how heat is transferred from the gases to steam. Convective superheaters are located in the convective zone of the furnace, usually ahead of the economiser. Earlier designs screened the convective superheater from the furnace radiation by a bank of water filled tubes which not only protected them from combustion flames and high temperature, but also intercepted the slag particles and reduced slagging problems. The convective superheaters are often termed as primary superheaters where saturated steam from the drum is admitted. Although the disposition of heating surface varies widely with the design of the steam generator, a typical arrangement is shown in Fig. 6.17. After convective superheater (CSH), the steam proceeds to the radiant superheater (RSH) which is placed in the radiant zone of the furnace near the water-wall to absorb heat by radiation. Steam leaving the radiant superheater goes to the desuperheater where highly pure water is directly sprayed on to steam in such a quantity that the temperature of steam after the last stage of superheating in the pendant superheater (PSH) does not exceed the rated value. The pendant superheater is a combined superheater in the sense that it receives heat partly by convection and partly by radiation. The radiant and combined superheaters together are often termed secondary superheaters.

Figure 6.27 shows the heat absorption in the three kinds of superheaters.

The energy balance of the convective superheater (Fig. 6.28) gives

$$\begin{aligned} Q_{CSH} &= \omega_g c_{pg} (t_{g_1} - t_{g_2}) = \omega_s (h_2 - h_1) \\ &= U_o A_o (\Delta t)_{\text{log-mean}} \end{aligned} \tag{6.32}$$

where $\dfrac{1}{U_o} = \dfrac{1}{h_i} + \dfrac{x_w}{k_w} + \dfrac{1}{h_o}$, $A_o = n\,\pi\, d_o\, l$, l = length of one coil

and n = number of coils. Since it is a gas-to-gas heat exchanger with low h_i (steam) and h_o (gas), the overall heat transfer coefficient U_o is also low. Now,

$$\omega_s = \left(n \frac{\pi}{4} d_i^2\right) \frac{V_s}{v_g} \tag{6.33}$$

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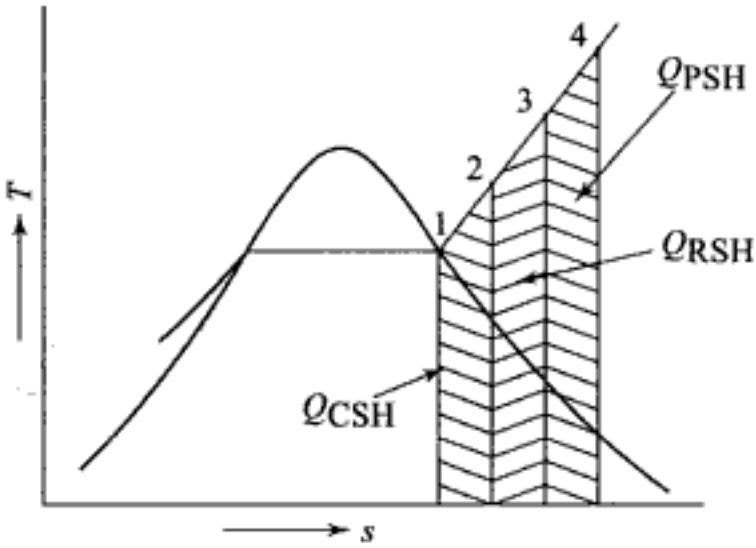

Fig. 6.27 *Heat absorption in the three kinds of superheater*

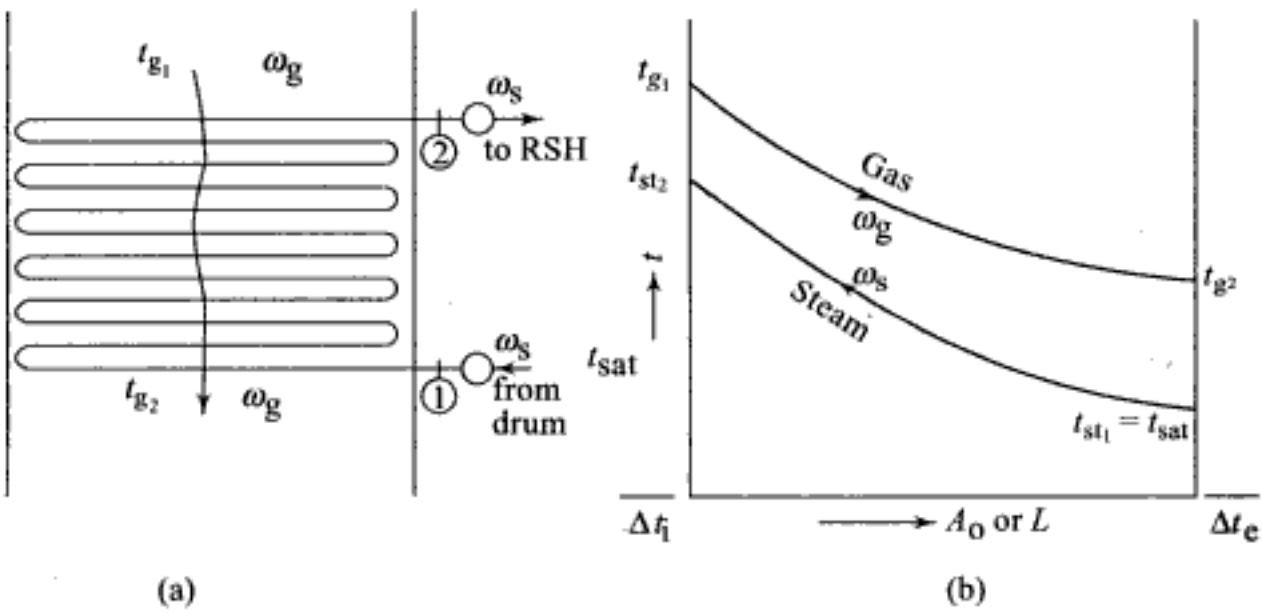

Fig. 6.28 *Convective superheater coil and the temperature profiles*

where V_s is the average velocity of steam in the superheater. For intermediate pressure boilers, the steam velocity may be about 22 m/s, for high pressure boilers it is about 12 m/s and for very high pressure boilers it is 10 m/s (approximate).

Superheaters and reheaters are made of tubes 50 to 75 mm OD in utility boilers. The smaller diameters have lower pressure stresses and withstand them better. The larger diameters have lower steam–flow pressure drops and are easier to align. The number of coils (n) needed for the flow is estimated from Eq. (6.33) and the length of one coil (l) is determined after A_o is computed from Eq. (6.32).

Higher is the gas velocity (V_g), higher will be the heat transfer coefficient on the gas side (h_o) and higher also will be the pressure drop of gas ($\alpha\ V_g^2$) and hence, the pumping power ($\alpha\ V_g^3$). Calculations involving the cost of heating surfaces, electrical power and so on in driving fans have given a figure of 20 m/s as an economic gas velocity (Kholodovskii, 1965). However, this

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economic gas velocity is not achieved in boiler plant because damage to heating surfaces by erosion from the fly ash in the flue gas increases as the cube of gas velocity, i.e.,

$$\text{Erosion rate } \alpha\ V_g^3 \tag{6.34}$$

Thus, the greater the ash content of the fuel, the less is the allowable gas velocity in the system. For low–ash coals, the gas velocity in the superheaters and economisers should not exceed 12 m/s, while in tubular air preheaters it should be below 16 m/s. For high–ash coals, the gas velocity should be limited to 8 m/s in superheaters and economisers, and in air preheaters it should be limited to 13 m/s.

As demand for steam increases with increase in load, fuel and air flow and hence, combustion gas flow are increased. The convective heat transfer coefficients (h_i and h_o) increase both inside and outside the tubes, increasing the overall heat transfer coefficient (U_o) between gas and steam faster than the increase in mass flow rate of steam alone. The combustion temperatures do not materially change with load. Thus, the steam receives greater heat transfer per unit mass flow rate, and its temperature increases with load (Fig. 6.29).

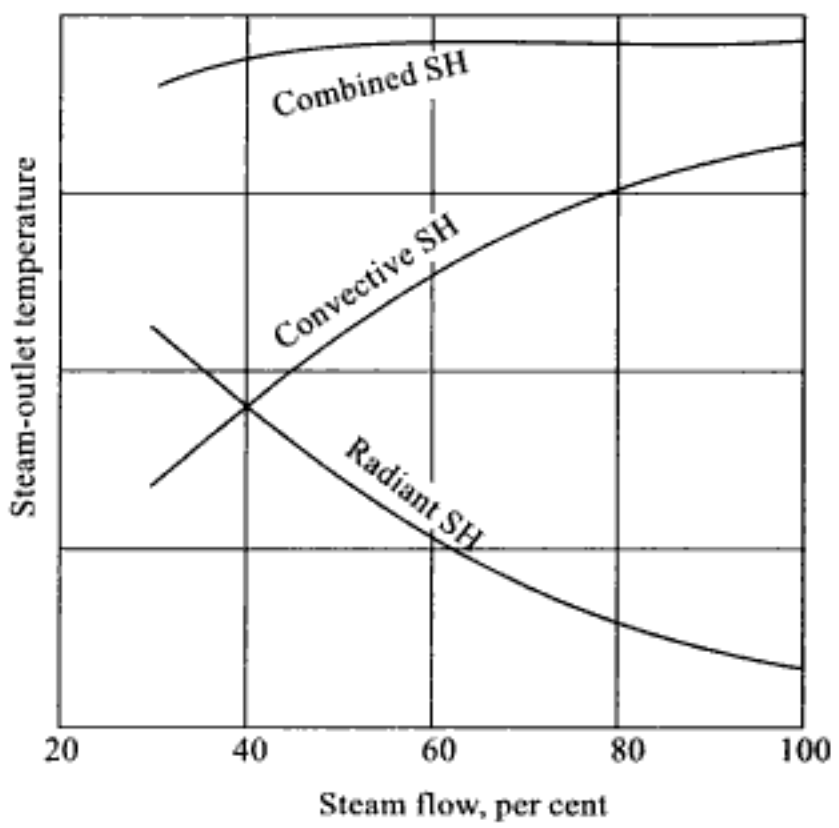

Fig. 6.29 *Steam-outlet temperature response of convective, radiant, and combined (in series) superheater*

The radiant superheater located in the radiant zone of the furnace permits greater heat absorption (Fig. 6.27) by the radiative mode of heat transfer.

$$Q_{RSH} = \sigma A_T F_{f\text{-}w} (T_f^4 - T_w^4) = w_s(h_3 - h_2) \tag{6.35}$$

where, σ = Stefan-Boltzmann constant = 5.67×10^{-8} W/m^{2}K^4, A_T = total surface area of superheater exposed to the flame, m^2, F_{fw} = view factor with respect to flame and wall, T_f = flame absolute temperature, K, and T_w = wall absolute temperature, K. Now, $t_w \simeq t_{sat}$ + 50 to 75°C.

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Since $T_f >> T_w$, $Q_{RSH} \alpha T_f^4$.

Now, T_f is not strongly dependent on load. As the steam flow increases with load, the heat transfer per unit mass flow decreases. Thus, an increase in steam flow due to an increased load demand would result in a reduction in exit steam temperature, the opposite effect of a convection superheater (Fig. 6.29). The combination of convective and radiative superheaters in series gives a relatively flat curve for steam outlet temperature and is thus insensitive to load change.

The steam velocity in radiant superheater is approximately 7.5 m/s, coefficient of heat transfer is about 3140 W/m^2 K and heat flux 116.28 kW/m^2.

Pendant type superheaters (PSH) are hung or suspended from above (Fig. 6.30). They have the advantage of firm structural support but the disadvantage of flow blockage by condensed steam after a cold shut down. It needs a slow restart to purge the water that accumulates in the bottom of each coil (by condensation, from previous run).

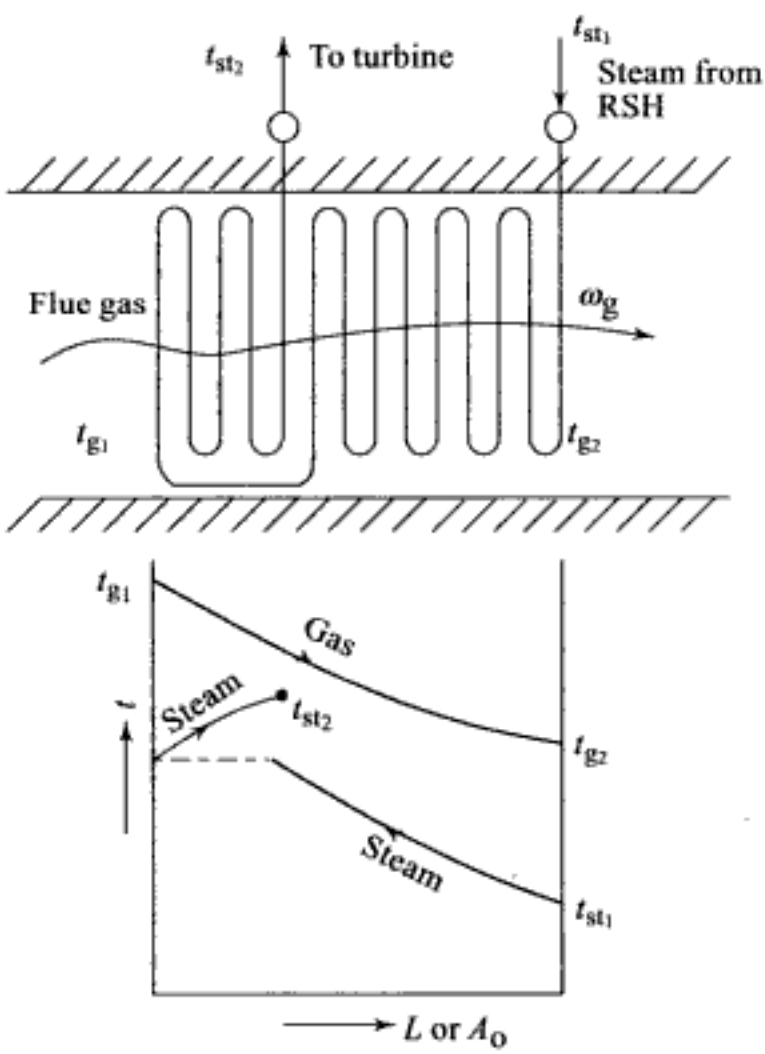

Fig. 6.30 *Pendant (mixed flow) superheater*

In a PSH, about 3/4th of the heating surface is made counter flow, while the remaining 1/4th acts in parallel flow (Fig. 6.30). If the entire coil were in counter flow, the exit end of the superheater coil will be subjected on the inside to the maximum steam temperature (say, 560 °C) and on the outside to the maximum gas temperature which may be about 1100 °C. This departure from

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the most economical arrangement (counter flow) is done by making a few turns act in parallel flow so as to avoid the exit end of the superheater being damaged by metal failure. The velocity of steam in the tubes must range from 15 to 25 m/s depending on the permissible pressure drop.

Radiant and pendant superheaters (RSH and PSH) can be in the form of coils or platens. Figure 6.31 shows different types of superheater coils. The bank of tubes becomes most compact when double-coiling is used (b and c). Platens are also hung from above and can be in various forms (Fig. 6.32). They are also compact heat exchangers, providing large surface area per unit volume.

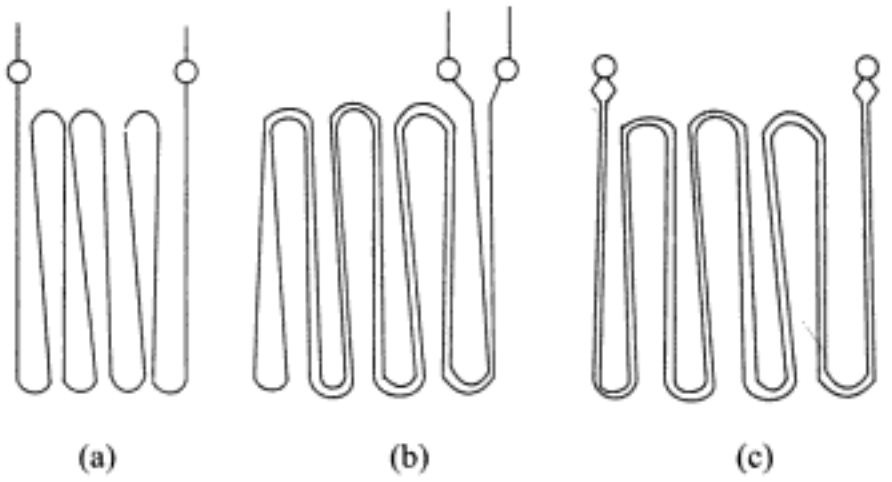

Fig. 6.31 *Types of superheater coils: (a) single coil, (b) double coil one tube, (c) double coil two-tube*

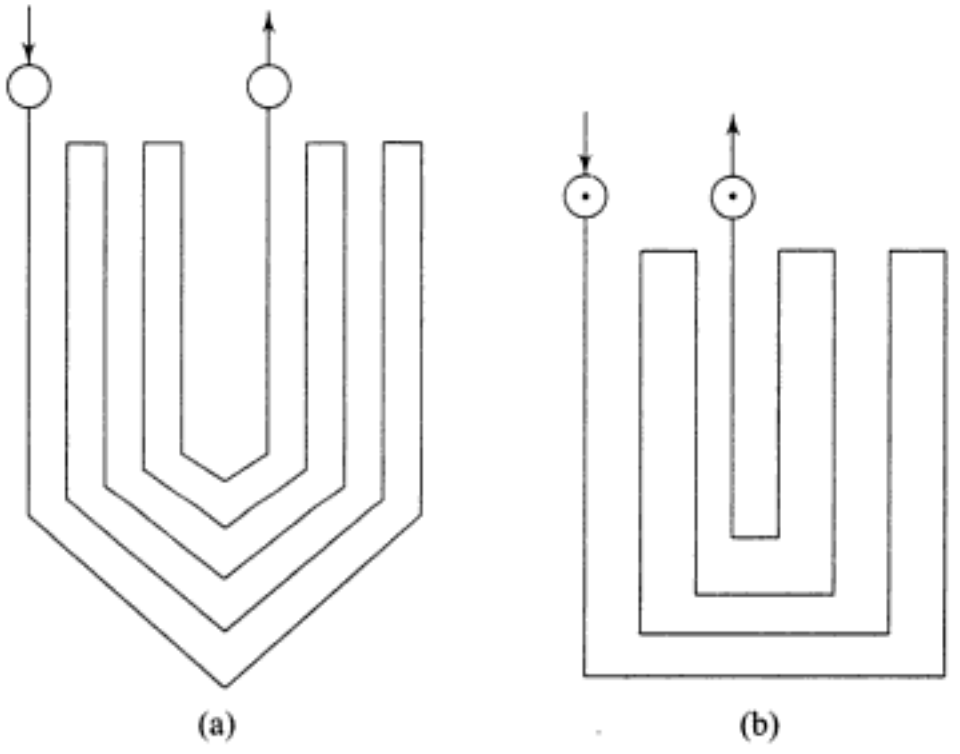

Fig. 6.32 *Types of platens (SH): (a) triangular, (b) rectangular*

Essentially, the modern boiler is designed with two parallel gas flow paths at the rear of the boiler prior to the air preheater (Fig. 6.20b). One gas path contains convective superheater surface, while the other contains convective

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reheater surface. A gas-tight baffle separates the two flow-paths. A common economiser is usually situated across both the two flow paths. The quantity of gas flowing down each of the two parallel flow paths is regulated by the gas dampers usually situated below the economiser surfaces. The reheat steam temperature control system is linked to control the damper position regulating the gas flow down the reheat gas pass.

6.6 REHEATERS

The design considerations for reheaters are similar to those for superheaters. Although the steam outlet temperatures are about the same, the overall temperatures are lower and the steam pressures are about 20–25% of those in the superheaters (Fig. 6.33). The pressure stresses are, therefore, lower and a lower grade steel alloy is tolerated. In addition, larger tube diameter with higher stresses may be used to reduce the pressure losses in the reheater.

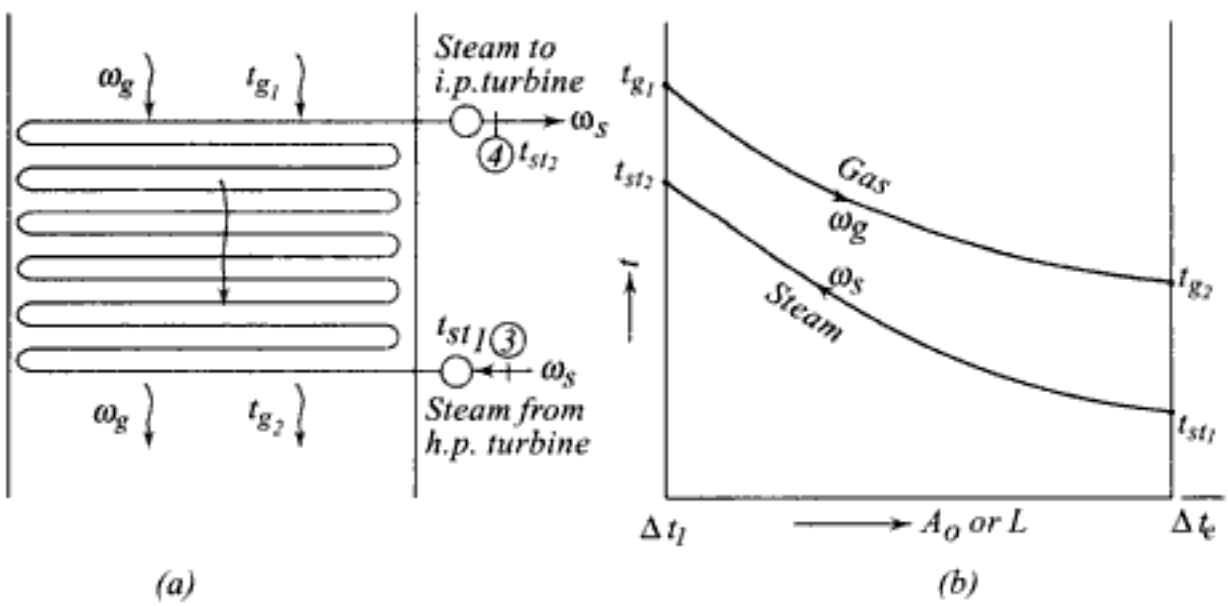

Fig. 6.33 *Convective reheater*

The rate of heat absorption in the reheater (Fig. 6.33) is

$$Q_{RH} = \omega_s(h_4 - h_3) = \omega_g\, c_{pg}\,(t_{g1} - t_{g2})$$

$$= U_o A_o \Delta t_{l.m} \tag{6.36}$$

where $A_o = n\,\pi\, d_o\, l$, n = number of coils, l = length of one coil, and

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{x_w}{k_w} + \frac{1}{h_o}$$

Again,

$$\omega_s = \left(n\frac{\pi}{4}\, d_i^2\right)\frac{V_s}{v_s} \tag{6.37}$$

where V_s is the steam velocity, say, at inlet, and v_s is the specific volume at that section (inlet).

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The reheater is usually located above the primary or convective superheater in the convective zone (Fig. 6.17) of utility boilers. It consists of a number of vertical coils of horizontal tubes connected between two headers, similar to the economiser and CSH. Steam comes from the h.p. turbine and after being reheated it goes to the i.p. turbine for further expansion.

In modern high pressure boilers, reheaters are normally in two sections. The primary section is placed in the convective zone of the backpass and the secondary section is placed just at the furnace exit hanging from the top, known as pendant reheater.

6.7 STEAM GENERATOR CONTROL

The object of steam generator control is to provide the steam flow required by the turbine at design pressure and temperature. The variables that are controlled are fuel firing rate, air flow, gas flow distribution, feedwater flow and turbine valve-setting. The key measurements that describe the plant performance are steam flow rate, steam pressure, steam temperature, primary and secondary air flow rates, fuel firing rate, feedwater flow rate and steam drum level, and electrical power output. The control system must act on the measurement of these plant parameters so as to maintain plant operation at the desired conditions.

Most modern control instrumentation employs closed loop control. In this mode, the actual output of the system is measured and compared to some demand signal (set point). The difference between the measurement and demand, called the *error signal*, is then used to reduce the difference between measurement and set point to zero. Proportional control is the simplest type of closed loop control. In this mode, the controller output is proportional to the error signal. The control signal will be either directly or inversely proportional to the error signal, depending on the control action required.

Only a few basic control systems related to feedwater and drum level, steam pressure and steam temperature are discussed below.

6.7.1 Feedwater and Drum Level Control

Feedwater and, therefore, steam flow is controlled to meet load demand by the turbine and at the same time maintain the level of water in the steam drum within relatively narrow limits. Normally, the water level in the drum is maintained half-full up to the diametral plane. A high steam consumption by the turbine, combined with low feedwater supply would lower the water level in the drum. A three-element automatic control system, of which the drum level is one element, is shown in Fig. 6.34. The drum level sensor responds to the error between actual drum level and its set point, such as in the case of high steam consumption and low feedwater supply, and acts on the controller to increase the feedwater valve opening to meet the steam flow demand. This action may be too slow and is supplemented by sensors for feedwater and steam flow. The

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difference between the signals from these two sensors anticipates changes in drum level and sends a signal to the controller to actuate the valve in the desired direction.

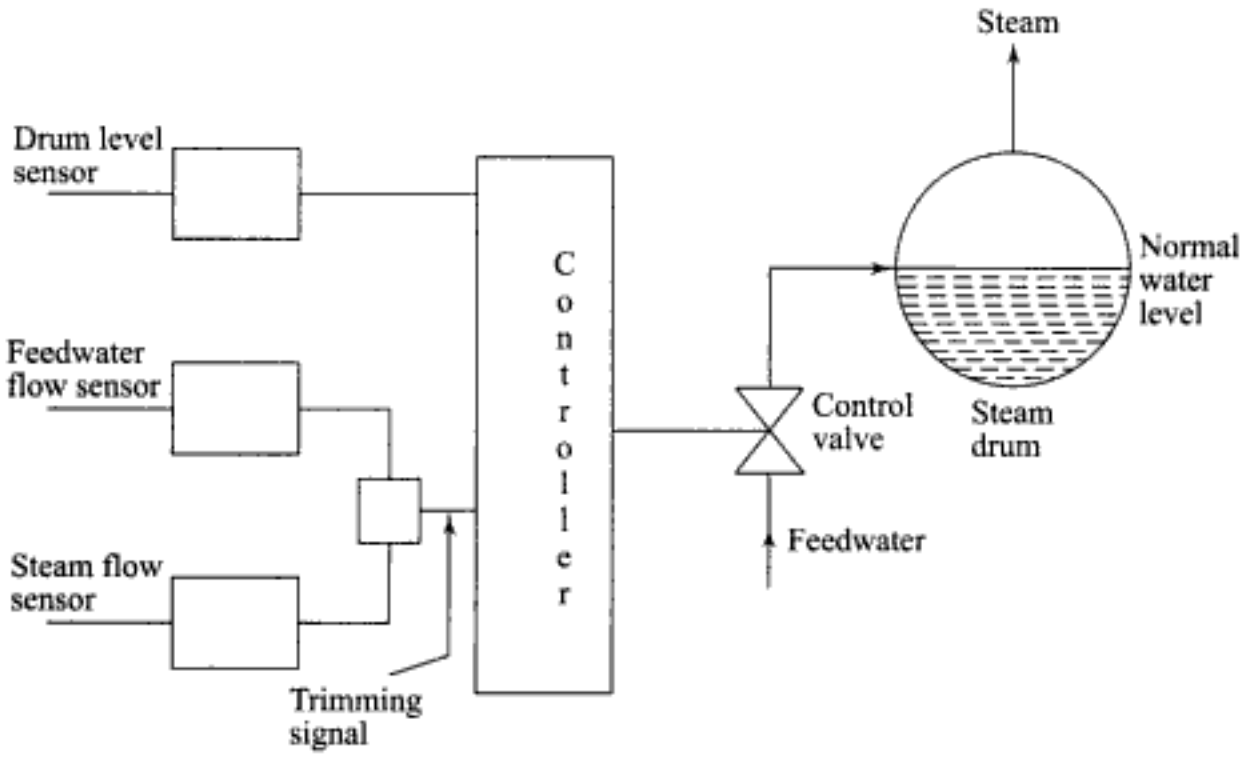

Fig. 6.34 *Schematic of a three-element feedwater control system*

6.7.2 Steam Pressure Control

The steam pressure control system, sometimes called the "boiler master", maintains steam pressure by adjusting fuel and combustion air flows to meet the desired pressure (Fig. 6.35). When pressure drops, the flows are increased. A steam pressure sensor acts directly on the fuel flow and air flow controls, such as the pulverized coal power drives and forced draught fan, to affect the desired changes. A trimming signal from fuel flow and air flow sensors maintains the proper fuel–air ratio. Since it is often difficult to obtain accurate fuel flows, a steam flow sensor is sometimes substituted for the fuel flow sensor. Usually, about a 5 – *s* delay is allowed when changing coal flow and air flow to ensure the prevention of a momentary rich mixture with high fuel–air ratio and thus assure smoke–free combustion.

6.7.3 Steam Temperature Control

An accurate control of superheat temperature is important for efficient power plant operation. The principal variables affecting superheat temperature are: (1) furnace temperature, (2) cleanliness of radiant and pendant superheaters, (3) temperature of gases entering the convective superheater, (4) cleanliness of convective superheater, (5) mass flow rate of gases through the convective superheater, (6) feedwater temperaure, and (7) variation of load on the unit. These variables are not unrelated but the inter-relationships among them are not

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simple. The last one, i.e. the effect of load variation on superheat temperature is, however, the most important.

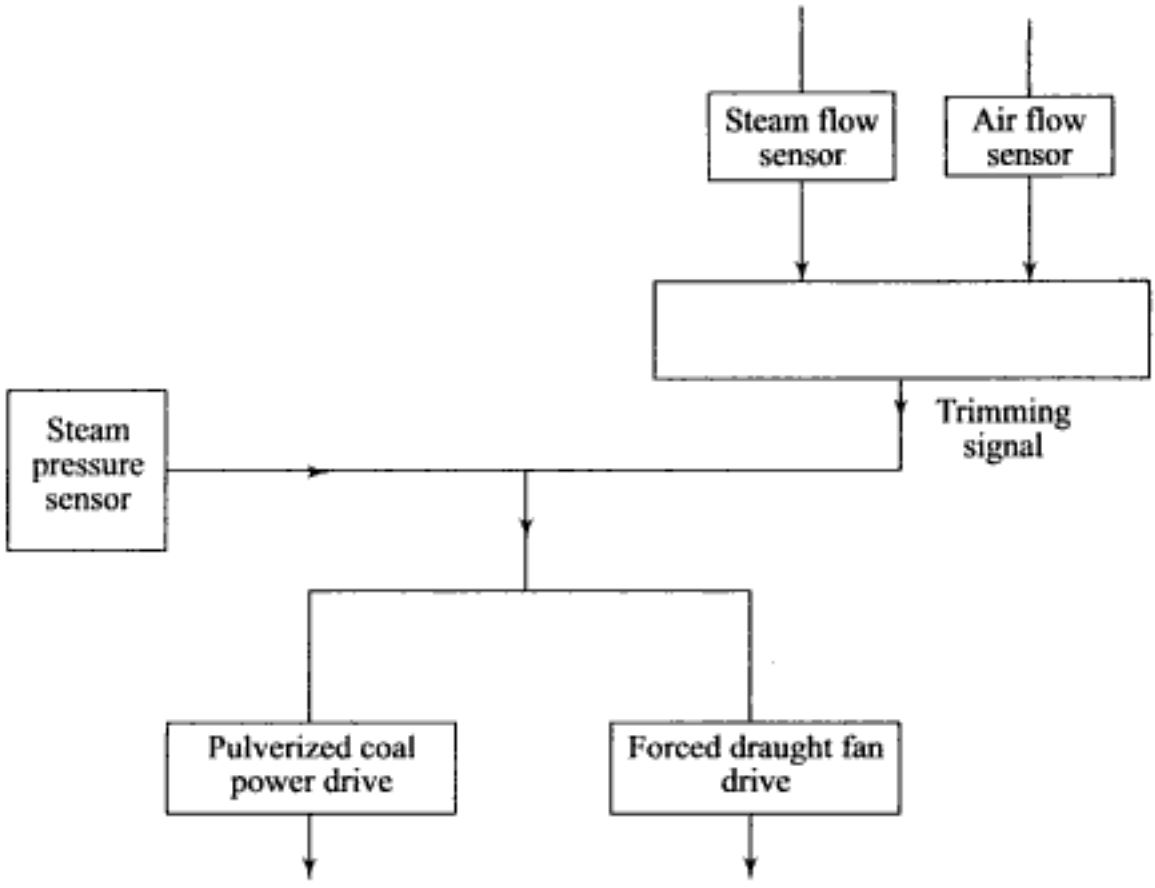

Fig. 6.35 *Schematic of a steam pressure control system*

A reduction in steam temperature results in loss in plant efficiency, e.g., a drop of about 20 °C in steam temperature results in about 1% increase in heat rate. On the other hand, a rise in steam temperature above design value may result in overheating and failure of superheater and reheater tubes and turbine blades.

The temperature of the saturated steam leaving the drum corresponds to the boiler pressure and remains constant if the steam-pressure controls are in working order. It is the superheater–reheater responses to load changes which need to be corrected. There are several ways of doing this as given below.

1. Combined radiant-convective superheaters Radiant and convectives superheaters are arranged in series to yield a relatively flat final steam temperature over a wide load range (Fig. 6.29), by taking advantage of their opposite steam temperature responses to load variation, as explained in Section 6.5.

2. Desuperheating and attemperation Control by attemperation means that the steam temperature is reduced by removing energy from the steam. In a tubular type, a portion of steam (w_1) taken out through tubes from a point between the primary and secondary superheaters by an automatic valve diverted to a shell-and-tube heat exchanger where boiled water from the

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drum may be circulated. The steam gives up some of its energy to that water and re-mixes with the primary steam (w_2) before entering the secondary superheater (Fig. 6.36).

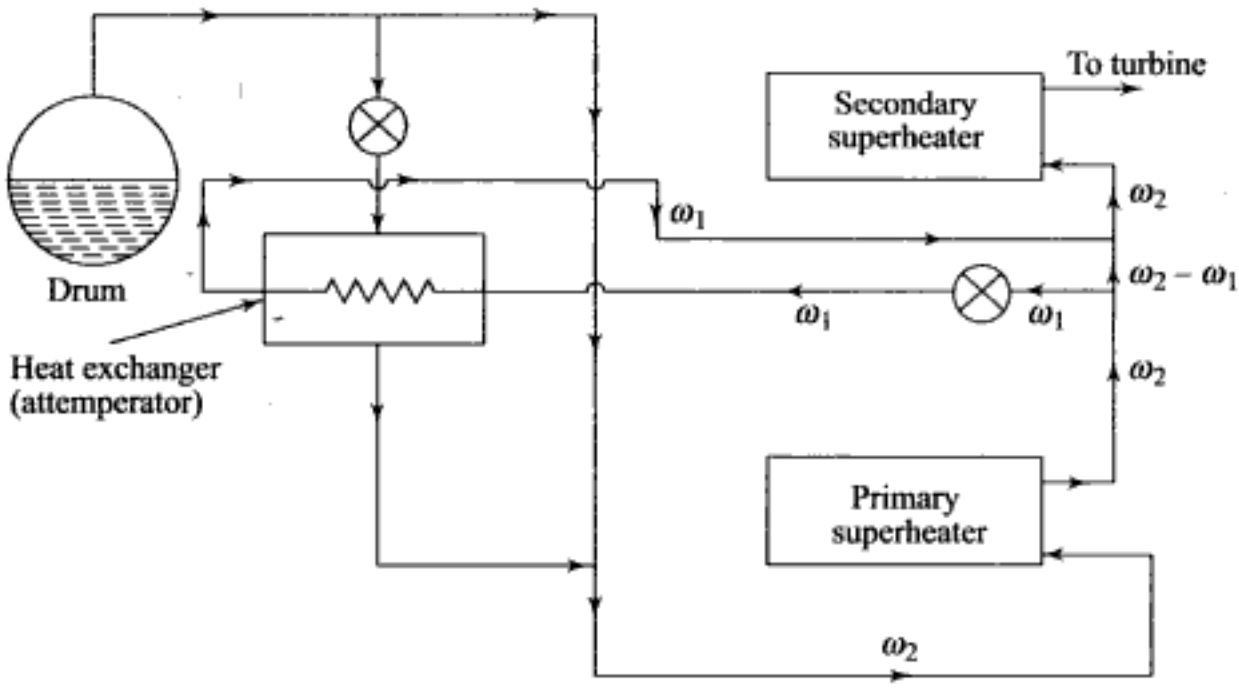

Fig. 6.36 *Control of steam temperature by attemperation*

A spray type attemperator, also called desuperheater (Fig. 6.37) reduces the steam temperature by spraying low temperature water from the boiler drum or

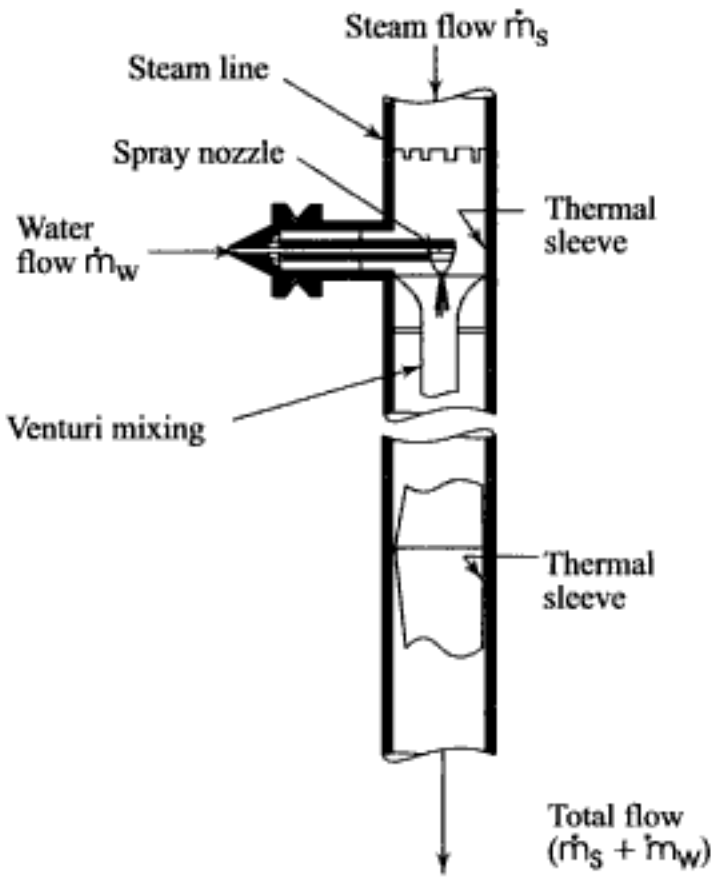

Fig. 6.37 *A spray attemperator for steam temperature control*

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economizer exit into the line between the primary and secondary superheaters (Fig. 6.38) or the radiant and pendant superheaters (Fig. 6.17). The spray nozzle injects water into the throat of a mixing venturi, where the water mixes with high velocity steam in the throat, vaporizes and cools the steam. The venturi and a thermal sleeve (a high chrome steel) protect the main steam pipe from thermal shock caused by any unvaporized water droplets that otherwise might impact on the pipe. The water used for spray must be of high purity so that no deposits (scale) are added on the superheater tubes, pipes, and turbine blades. The spray-type desuperheater or attemperator has been quite satisfactory in its service. It provides a rapid and sensitive means for temperature control. By regulating the amount of spray water, steam temperature is controlled to produce a flat temperature curve beyond point *a* (Fig. 6.39). Steam is heated from *m* to *n* in the primary superheater, cooled from *n* to *o* in the desuperheater by spray water, and again heated from *o* to *p* in the secondary superheater.

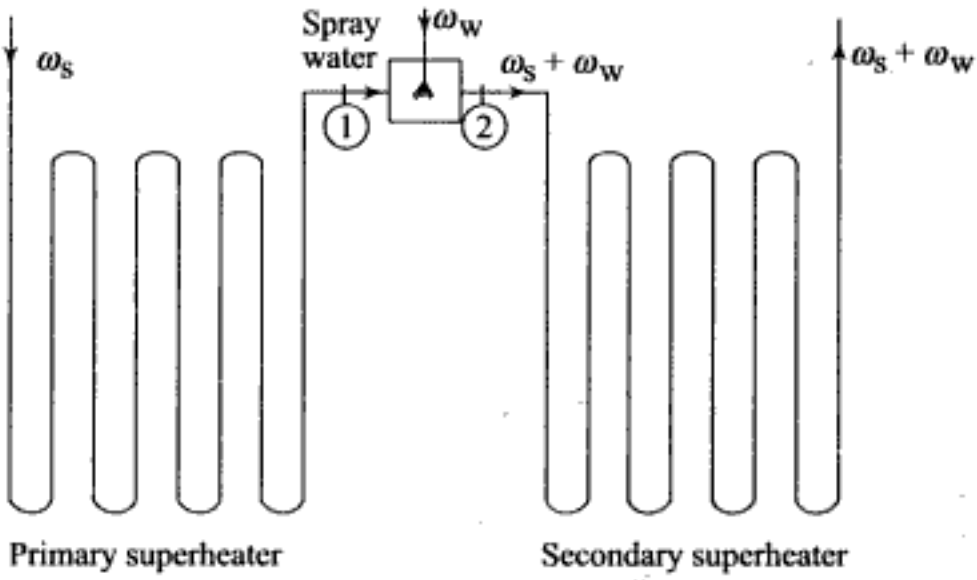

Fig. 6.38 *Spray-type desuperheater*

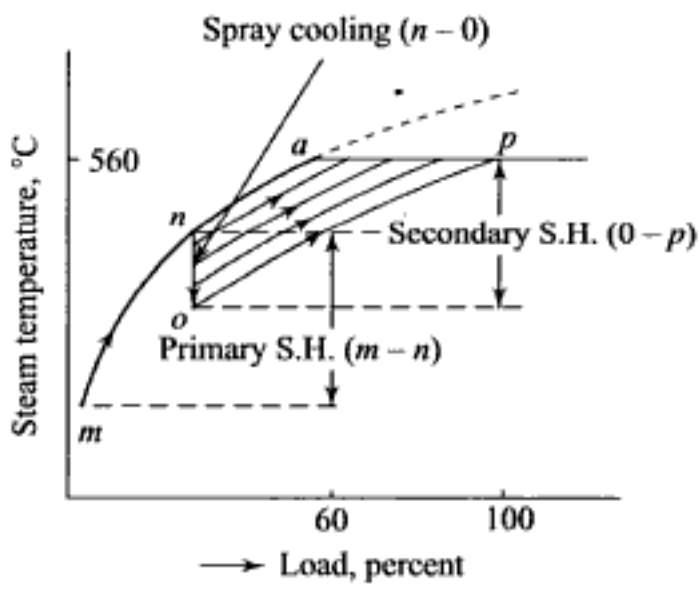

Fig. 6.39 *Regulation of spray water results in a flat temperature curve*

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By energy balance at steady state (Fig. 6.38),

$$\omega_s h_{s1} + \omega_w h_w = (\omega_s + \omega_w) h_{s2} \tag{6.38}$$

where,

ω_s and ω_w = mass flow rates of steam and water, respectively, kg/s

h_s and h_w = specific enthalpies of steam and water, kJ/kg.

If the desuperheater is located after the last stage superheater, the steam temperature exceeds the maximum desired temperature before attemperation which is harmful. For example, if the maximum permissible steam temperature is 560 °C and the desuperheater is after the last stage superheater, then at a reduced load the steam temperature may shoot up to 570 °C, which is brought down by water spray to 560 °C. However, the temperature has to be limited to 560 °C and cannot be allowed to exceed it. On the other hand, the desuperheater is located before the last stage superheater, then at rated load with no attemperation the steam temperature at the location should be, say, 510 °C, which increases to 560°C by absorption of heat in the last stage superheater. If at a reduced load the steam temperature at this location shoots up to 520 °C, then water will be sprayed to bring the temperature down to 510 °C, by which heat absorption in the last superheater rises to 560 °C before exiting to the steam turbine. The control is done one stage ahead and it is the preferred practice.

3. Gas by-pass or damper control If at any time the flow of gases through a convective superheater is reduced without changing the steam flow, then the final steam temperature will be reduced. Gas by-passing of the superheater or damper control utilizes this principle. If it is desired to have constant steam temperature for 3/4th steam flow and higher, then the superheater is designed to give the proper steam temperature at 3/4th steam flow. At greater steam flows, some of the flue gases are by passed around the superheater to maintain the desired steam temperature by partial closing of the dampers under the superheater (Fig. 6.40).

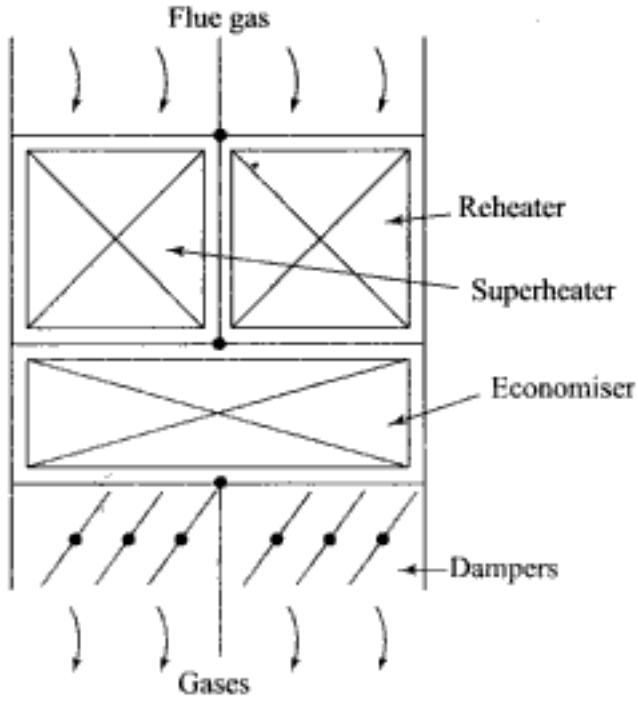

Fig. 6.40 *Steam superheat by gas by-pass or damper control*

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4. Gas recirculation In this system, gas from some point downstream of the superheater–reheaters, mostly from the economiser outlet, but sometimes from the air preheater outlet, is recirculated back to the furnace by means of a gas recirculation fan (Fig. 6.41). The function of the recirculated gas is to reduce furnace heat absorption by diluting the furnace gas and lowering the

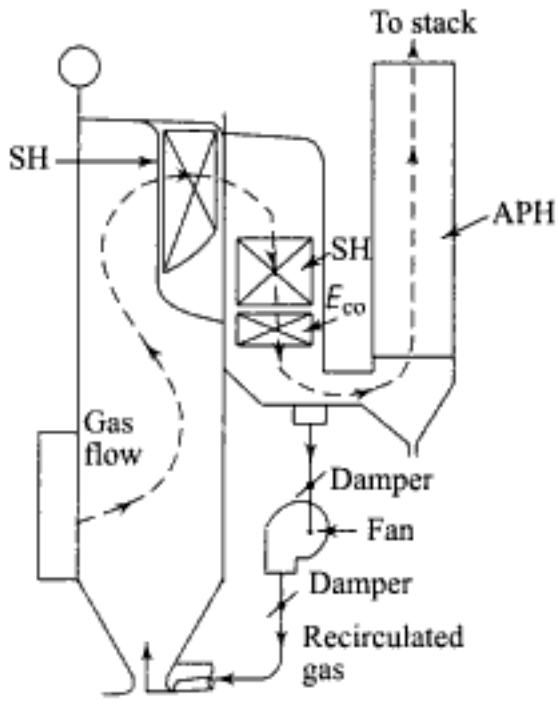

Fig. 6.41 *Gas recirculation for steam temperature control*

furnace zone temperature (radiant zone). The total gas enthalpy leaving the furnace tends to be increased, however, because of the greater gas mass. Thus the gas velocity and the heat transfer rate in the covective superheater are increased by recirculation. As load falls, a greater gas mass is recirculated to maintain full superheat. Figure 6.42 shows how the gas recirculation amount influences the heat absorption in the economiser, primary and secondary super-

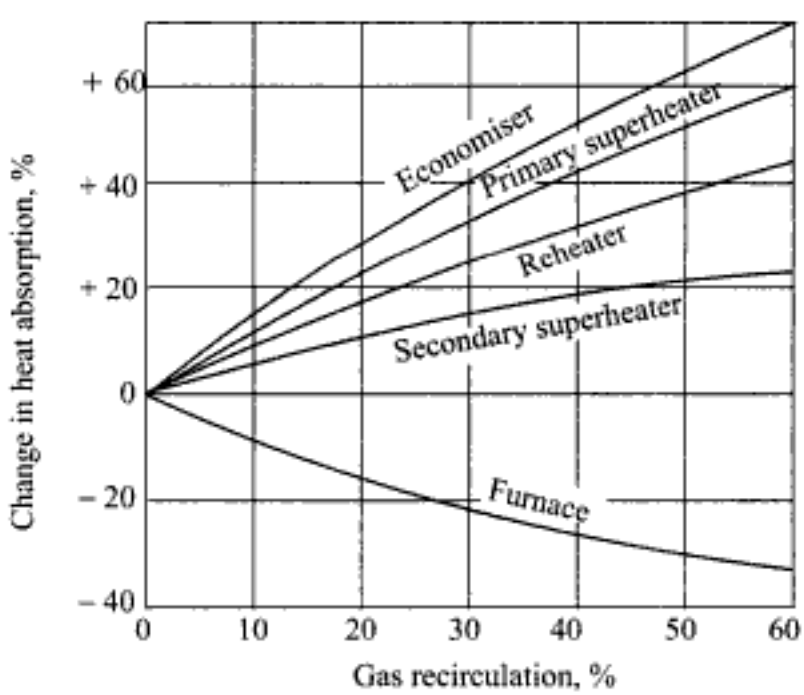

Fig. 6.42 *Effect of gas recirculation on the heat absorption pattern at a constant firing rate*

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heaters, reheater and the furnace. The range of control possible with recirculation is, however, limited largely by the power consumption of the recirculation fans. In some gas recirculation systems, recirculated gas is admitted near the furnace exit. This is called gas tempering. The furnace exit temperature is here reduced without affecting the furnace heat absorption.

Gas recirculation is sometimes used in series with desuperheating or attemperation for effective steam temperature control (Fig. 6.43).

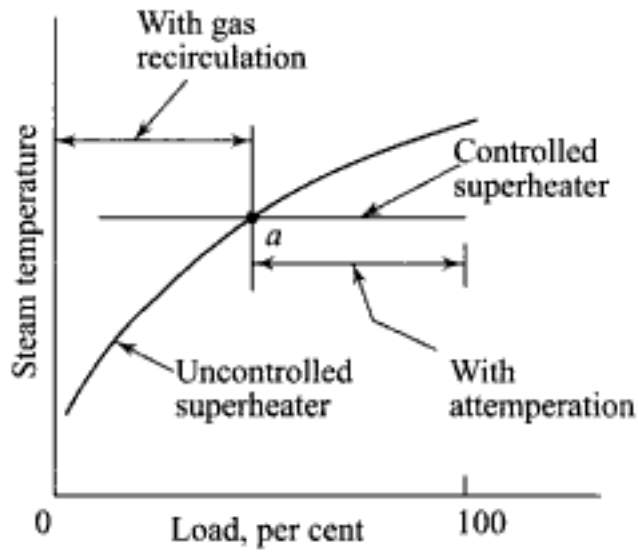

Fig. 6.43 *Gas recirculation and attemperation in series*

5. Excess air Increased excess combustion air has the effect of lowering furnace temperature. Since radiant heat transfer is a function of the difference in the fourth power of the temperatures, furnace heat absorption will be materially reduced when furnace temperatures are lowered by the use of excess air. Thus, the gas temperature entering the superheater is actually increased. This increase in gas temperature along with the increase in mass flow raises the steam temperature.

Although increased excess air is a means for steam temperature control, the increased mass of stack gas materially lowers the boiler efficiency.

6. Tilting burners Tilting or vertically adjustable burners (Fig. 6.44) change the gas temperature entering the superheater section by changing the elevation of the fireball within the furnace. At low loads (steam flow), the burners are tilted upward (the usual range $\pm 20°$) so that the lower portion of the furnace becomes less effective in absorbing energy. Thus, the gases enter the superheater at a higher temperature than if the burners were fixed, and the steam temperature leaving the convection superheater can be maintained constant from less than one-half (50%) load to full load (100%). This is a satisfactory and economical method of temperature control.

7. Burner selection Large steam generators have several burners that are arranged in horizontal rows. Since only some of the burners are needed at partial load, use of the upper row of burners will increase steam temperature in the same way as when the burners are tilted upward.

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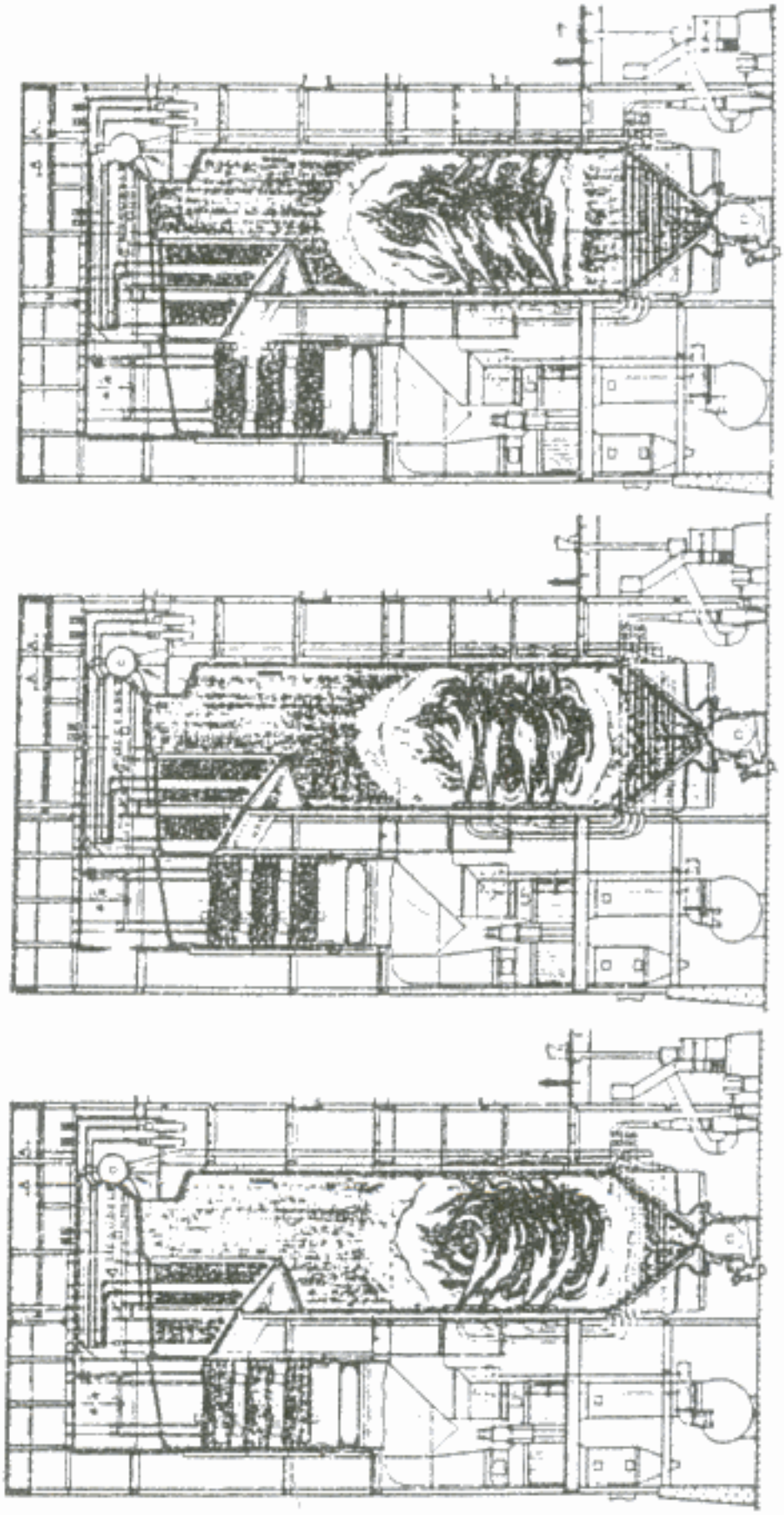

Fig. 6.44 *Use of tilting burners to vary effective furnace volume for steam temperature control*

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The effective furnace volume is reduced by not firing the upper burners at low loads (or by tilting the burners downward). The heat absorption at low loads is reduced and the gas temperature entering the convective superheater is kept near the desired level.

8. Separately fired superheater Separately fired furnaces for superheat regulation give a good degree of control over a wide range. The rate of firing is adjusted to yield a flat steam temperature load curve. This system is not generally economical for large utility boilers and is usually used in the chemical process industry.

Reheater outlet steam temperature is required to be controlled for the same reasons as those for regulating superheater outlet temperature, and the methods used are generally the same.

6.8 AIR PREHEATER

The temperature of feedwater in present day's power plants with five or more feedwater heaters goes up to 170–280 °C. It means that the temperature of flue gases leaving the economiser cannot be below 270–480 °C (because for gas-to-gas heat exchange, the temperature difference between gas and air is, say, of the order of 100 °C). At such a high temperature of exit gases, the loss of energy with stack exhaust is high, and so the efficiency of the boiler will be low.

It is possible to cool the exit gases in the heating of air supplied for the combustion of fuel. The use of hot air makes the combustion process more efficient by making it more stable and lessening the energy losses due to incomplete combustion and unburnt carbon.

The application of hot air also leads to the increase of theoretical flame temperature and to the intensification of radiant heat transfer in the furnace. As a result of the increased furnace exit temperature, gas temperatures throughout the steam generator are raised above the levels which would prevail in the absence of an air preheater. This also enhances convective heat transfer in downstream heat exchangers. The gas temperatures exiting the air preheater vary from 135 to 180 °C, not only for the prevention of acid corrosion problems (to remain above dew point temperature) but also for proper dispersion of exhaust gases from the stack to the atmosphere.

While flue gases enter the air preheater at temperatures varying between 315 to 450 °C and are cooled to 135–180 °C, air is heated from F D fan outlet temperature (40–60 °C) to 280–400 °C and sometimes still higher. For the same energy input to the furnace, the preheating of air saves fuel. This saving in fuel and the consequent increase in boiler efficiency are approximately proportional to the air temperature rise in the air preheater. Typical fuel savings are 4% for a 110 °C rise in air temperature and above 11% for a 280 °C rise in air temperture in the preheater.

Preheated air is also required for the operation of pulverized coal furnaces. Air (primary) of the temperature range 150 – 420 °C is needed for drying coal in the pulverizer. Air is also used for transporting pulverized coal to the furnace

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and burning it there. Small stoker fired units do not need preheated air. However, it is beneficial to use preheated air in large stoker fired boilers, but the air temperature should not exceed 180 °C to prevent damage to the stoker moving parts.

Air preheaters are in general divided into two types: the recuperative and regenerative. In recuperative air preheaters, heat is directly transferred from the hot gases to the air across the heat exchanging surface. They are commonly tubular, although some plate types are still in use. Tubular units are essentially counter-flow shell-and-tube heat exchangers in which the hot gases flow inside the vertical straight tubes and the air flows outside (Fig. 6.45 (a) and (b)). Baffles are provided to maximize air contact with the hot tubes. The tubes are mechanically expanded into top and bottom tube sheets. The rate of heat transfer from gas to air is given by

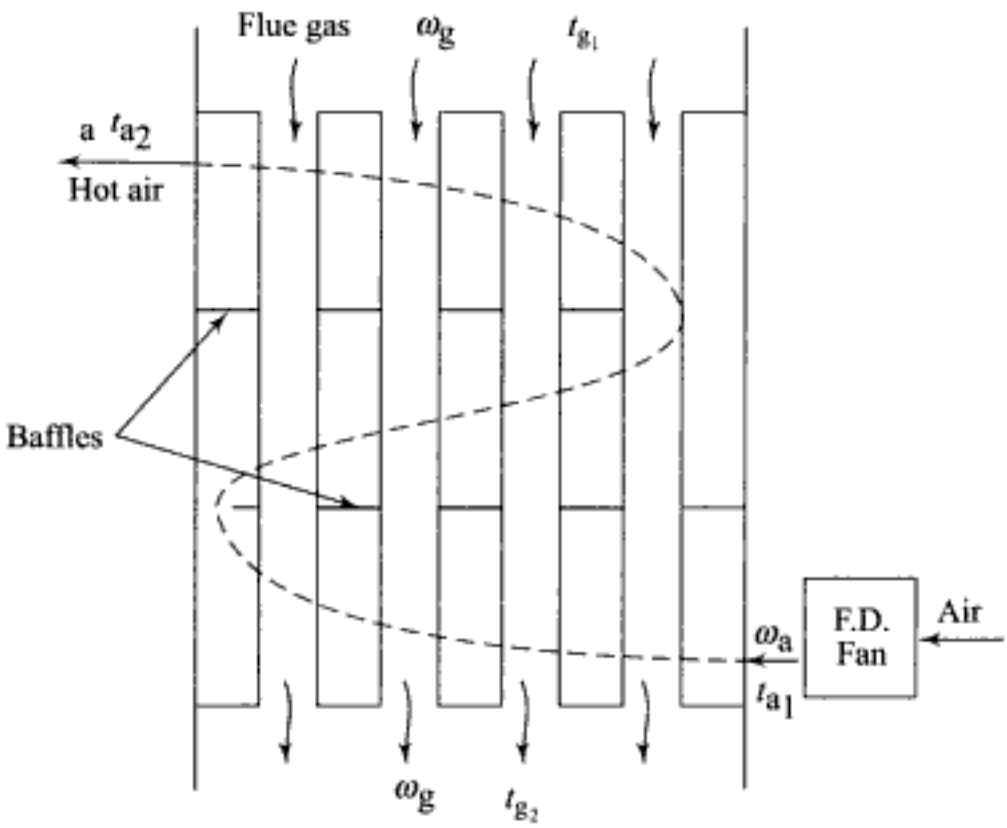

Fig. 6.45 *(a) Tubular air preheater*

$$Q_{APH} = \omega_g\, c_{p_g}\, (t_{g_1} - t_{g_2}) = \omega_a\, c_{p_a}\, (t_{a_2} - t_{a_1})$$
$$= U_o A_o (\Delta t)_{\text{log-mean}} \tag{6.39}$$

where
$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{1}{h_d A_i} + \frac{x_w}{k_w A_{l.m}} + \frac{1}{h_o A_o} \tag{6.40}$$

and h_d = heat transfer coefficient due to deposits on the inside heating surface, W/m^2 K. The temperature profiles of gas and air are shown in Fig. 6.46.

Since the gas is flowing through the tubes.

$$\omega_g = n \frac{\pi}{4} d_i^2\, \rho_i\, (V_g)_i \tag{6.41}$$

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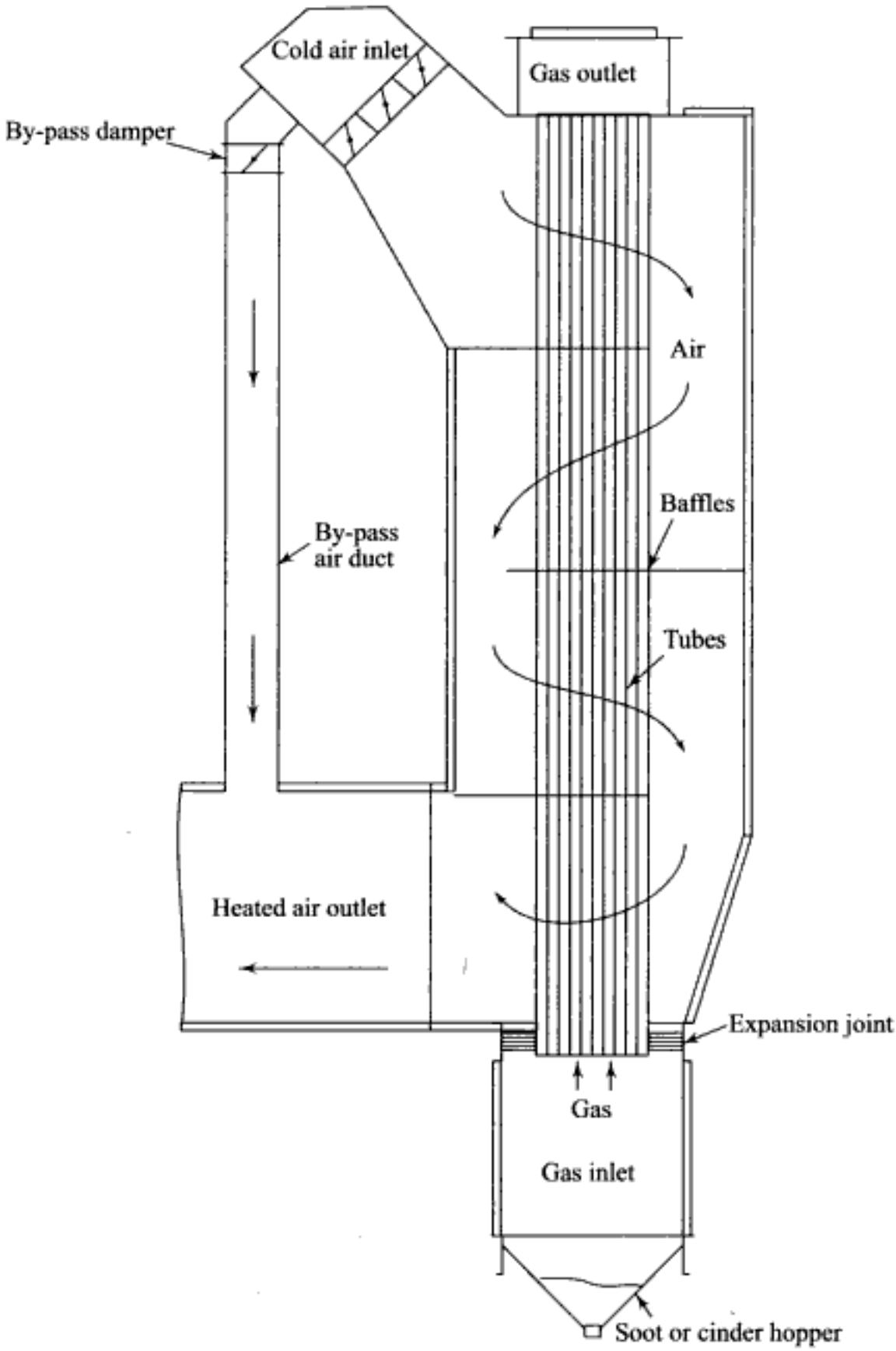

Fig. 6.45 *(b) Air heater arrangement*

where

n = number of tubes,

d_i = inside diameter of tubes, about 50 mm

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ρ_i = density of gas at inlet to the air preheater (may be approximately obtained from $\rho_i = \dfrac{p}{R_g (T_g)_i}$)

R_g = characteristic gas constant = 0.287 kJ/kg K

p = 1.03 bar

$(T_g)_i$ = inlet gas temperature (K)

$(V_g)_i$ = inlet gas velocity, which is about 12–14 m/s.

From Eq. (6.41), n can be determined.

If the fouling of the heating surface (inside) is not considered and the thickness of the tubes is small,

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{x_w}{k_w} + \frac{1}{h_o}$$

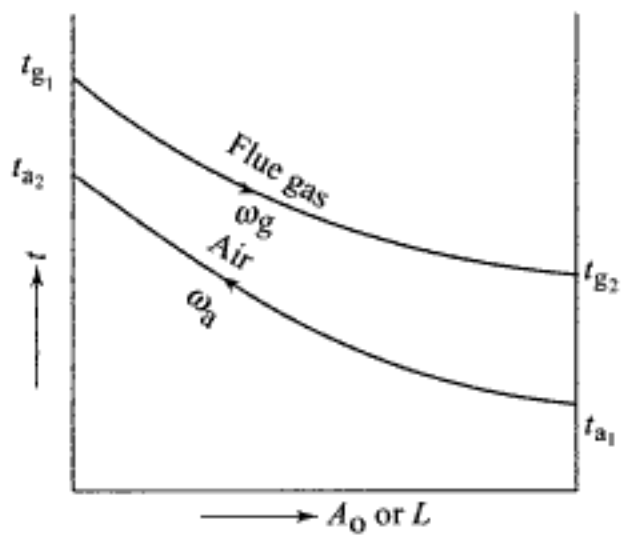

Fig. 6.46 *Temperature profiles of gas and air in APH*

Both h_i and h_o being small, the overall heat transfer coefficient U_o is low, ranging between 30 and 60 W/m^2 K. From Eq. 6.39, A_o can be estimated and since $A_o = n\pi d_o l$, d_o being the outside tube diameter, the length of the tubes, l, can be computed.

Cost, cleaning requirements and fuel type determine the diameter of the tubes used. Modern steam generators use 50–65 mm OD heater tubes.

Regenerative air preheaters, known as storage type heat exchangers, have an energy storage medium, called the matrix, which is alternately exposed to the hot and cold fluids. When the hot flue gases flow through the matrix in the first half of the cycle, the matrix gets heated and the gas is cooled. In the next half of the cycle when air flows through the matrix, air gets heated and the matrix is cooled. The cycle repeats itself. The heat transfer from gas to air is thus periodic, the amount depending on the number of heating–cooling cycles executed per second. The most common type is the rotary air preheater, known as *Ljungstrom air phreheater*, which consists of a rotor driven by an electric motor through reduction gearing at a low speed with 2–4 rpm. The rotor is

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divided into a number of radial sectors, 12 to 24 in number, each sector being filled with steel sheets, either flat or corrugated (Fig. 6.47). These sheets form the heat transfer surfaces. Two blank opposite sectors act as seals to prevent leakage of air to the gas path and vice-versa. As the rotating sectors enter the hot gas zone, they are progressively heated by the gas storing energy as sensible heat. When they enter the air zone, they progressively give up this heat to the air.

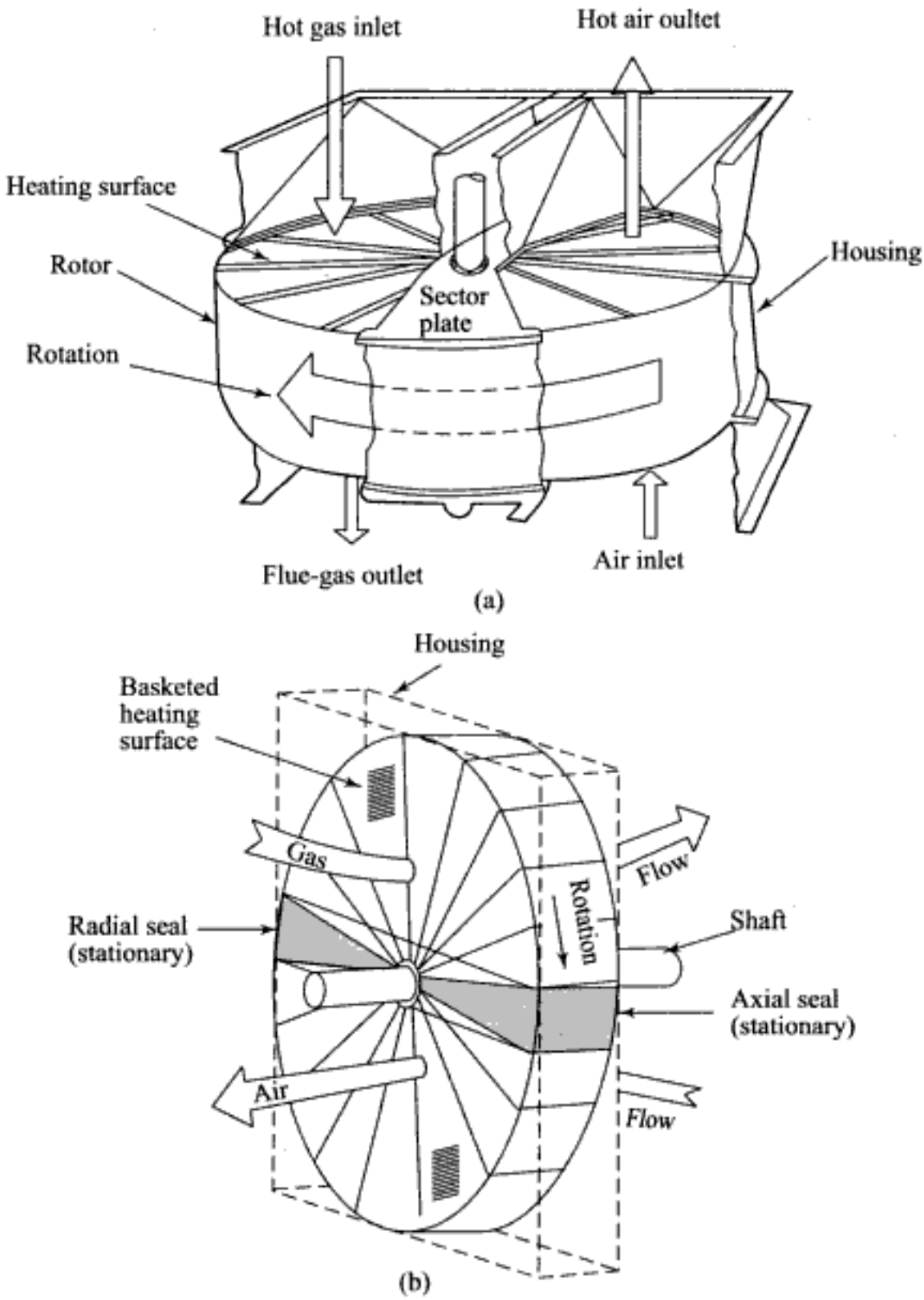

Fig. 6.47 *(a) Details of a regenerative air preheater (Ljungstrom APH) (b) Another view of a regenerative air preheater*

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Rotary air preheaters are compact heat exchangers with a large heat transfer surface being accommodated in a small volume. They are relatively cheap and have good heat transfer characteristics. They are almost as popular as tubular air preheaters in power station practice. However, there are certain drawbacks, *viz.*, (a) flow openings being plugged by fly ash, (b) sealing of gas to air path and vice-versa, and (c) large pressure drops for both gas and air.

If air is required to be heated to 400 °C or above, two stages of air preheating are used with the economiser installed in between the two air preheaters.The first air preheater (from the gas side) may be tubular, while the second one may be rotary.

6.9 FLUIDIZED BED BOILERS

Fluidized bed boilers produce steam from fossil and waste fuels by using a technique called fluidized bed combustion which has been discussed in the last chapter. These can be of two types:

1. Bubbling fluidized bed (BFB) boilers
2. Circulating fluidized bed (CFB) boilers

In BFB boilers, crushed coal (6–20 mm) is injected into the fluidized bed of limestone just above an air-distribution grid at the bottom of the bed (Fig. 6.48).

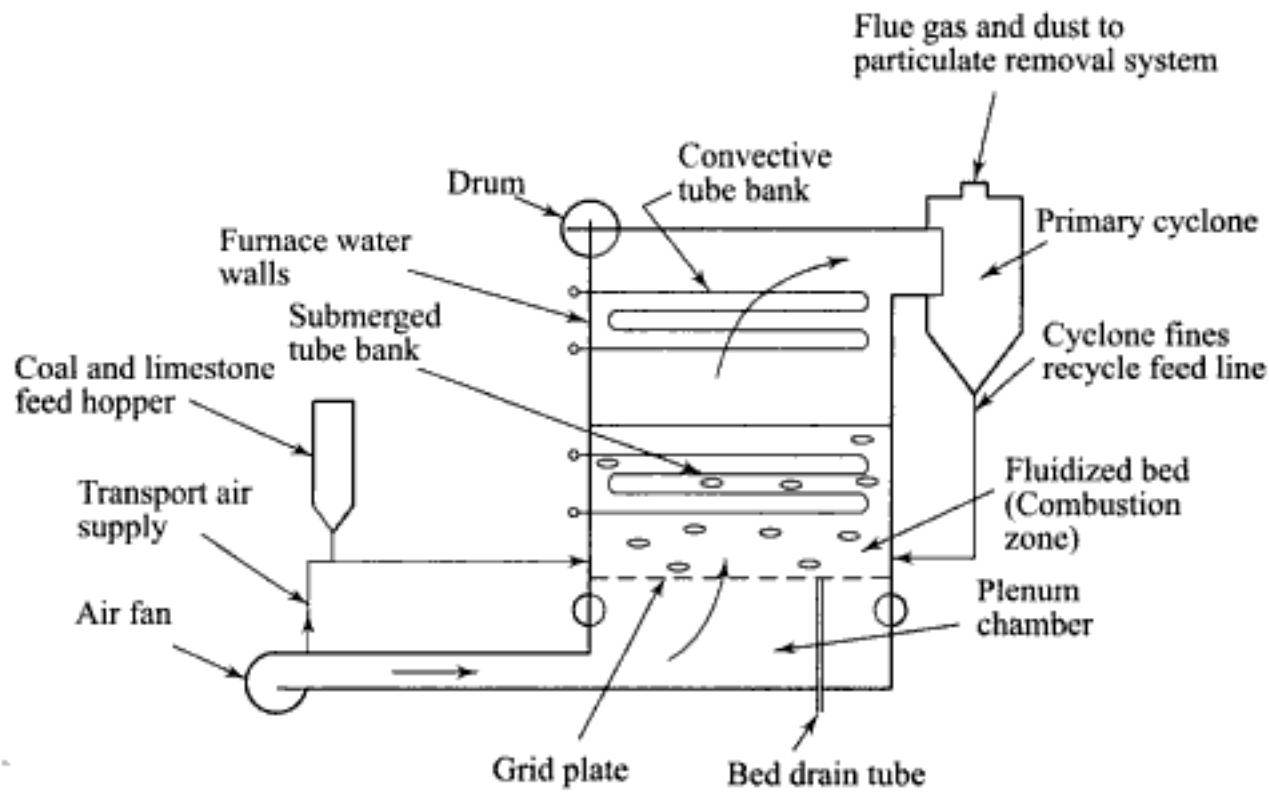

Fig. 6.48 *Schematic of bubbling fluidized bed boiler*

The air flows upwards through the grid from the air plenum into the bed, where combustion of coal occurs. The products of combustion leaving the bed contain a large proportion of unburnt carbon particles which are collected in cyclone separator and fed back to the bed. The boiler water tubes are located in the furnace.

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Since most of the sulphur in coal is retained in the bed by the bed material used (limestone), the gases can be cooled to a lower temperature before leaving the stack with less formation of acid (H_2SO_4). As a result of low combustion temperatures (800–900 °C), inferior grades of coal can be used without slagging problems and there is less formation of NO_x. Cheaper alloy materials can also be used, resulting in economy of construction. Further economies are achieved since no pulverizer is required. The volumetric heat release rates are 10 to 15 times higher and the surface heat transfer rates are 2 to 3 times higher than a conventional boiler. This makes the boiler more compact.

Figure 6.49 shows a bubbling fluid bed boiler system operating at atmospheric pressure, similar to the one of 160 MWe Tennessee Valley Authority (TVA) project at Shawnee, USA, recently installed (1993).

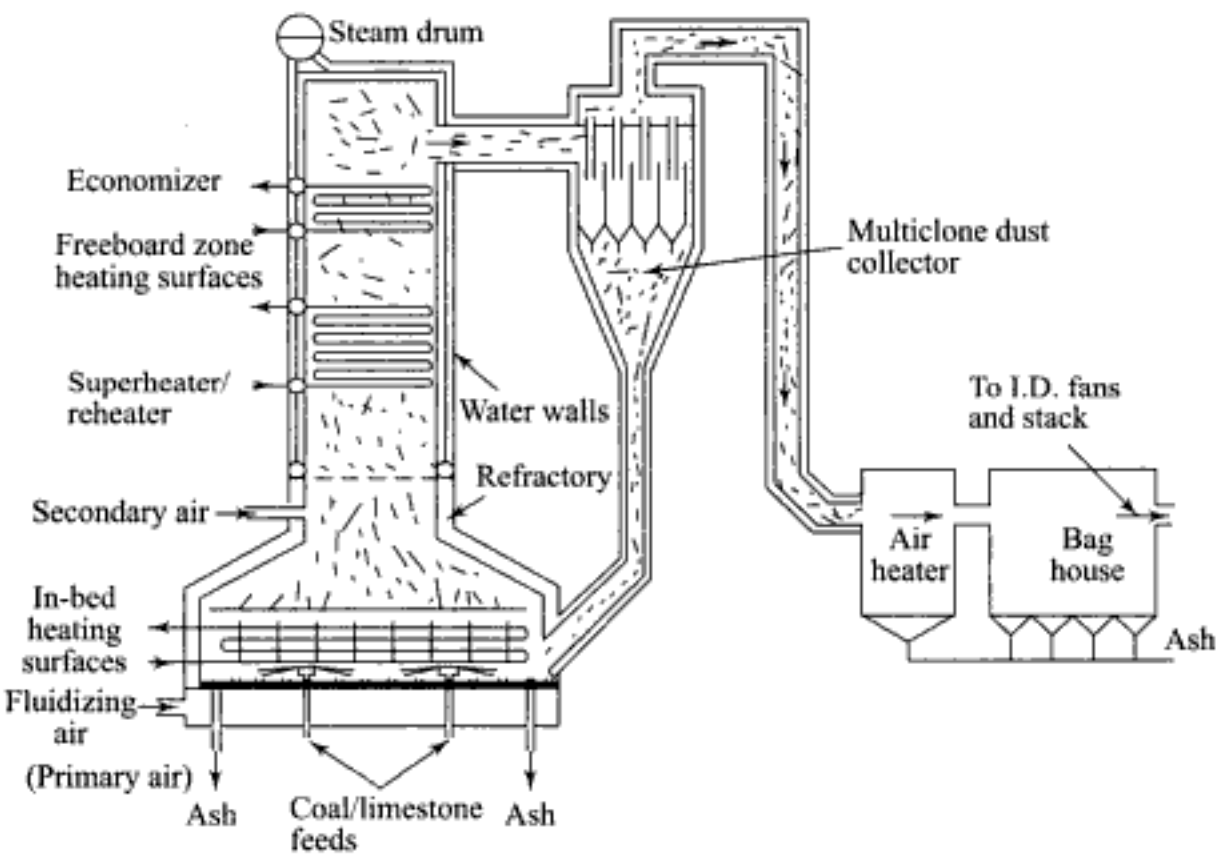

Fig. 6.49 *An atmospheric bubbling fluid bed boiler system*

The CFB boiler is said to be the second generation fluidized bed boiler (Fig. 6.50 (a)). It is divided into two sections. The first section consists of (a) furnace or fast fluidized bed, (b) gas–solid separator (cyclone), (c) solid recycle device (loop seal or L–valve), and (d) external heat exchanger (optional). These components form a solid circulation loop in which fuel is burned. The furnace enclosure of a CFB boiler is generally made of water tubes as in pulverized coal-fired (PC) boilers. A fraction of the generated heat is absorbed by these heat transferring tubes. The second section is the back-pass, where the remaining heat from the flue gas is absorbed by the reheater, superheater, economiser, and air preheater surfaces (as in a conventional PC boiler).

The lower part of the first section (furnace) is often tapered. Its walls are lined with refractory up to the level of secondary air entry. Beyond this the furnace walls are generally cooled by evaporative, superheater, or reheater

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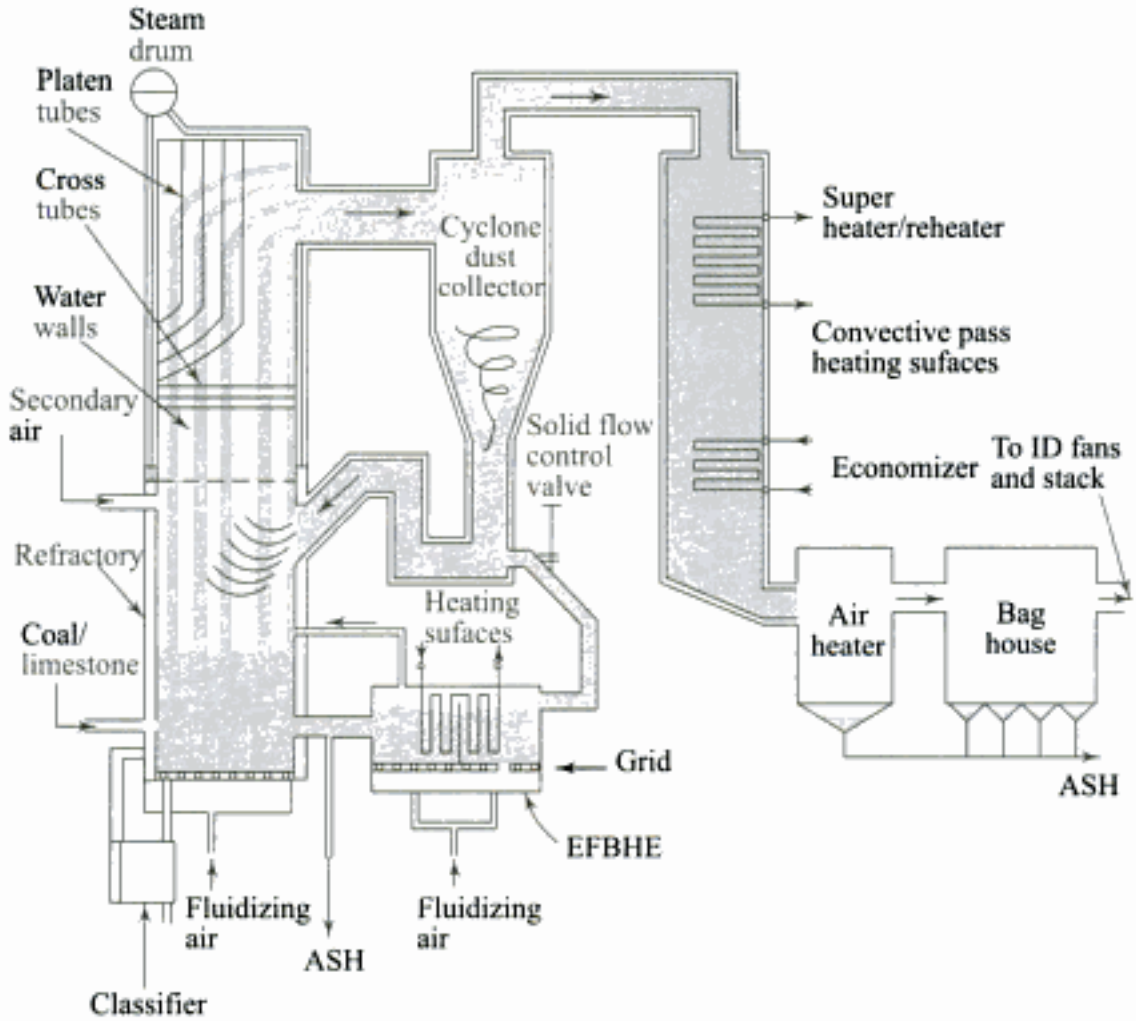

Fig. 6.50 *(a) Schematic of a circulating fluidized bed boiler*

surfaces. The gas–solid separator and the non-mechanical valve are also lined with refractory. In some designs, a part of the hot solids recycling between the cyclone and the furnace is diverted through an external heat exchanger, which is a bubbling fluidized bed with heat transfer surfaces immersed in it to remove heat from the hot solids.

Coal is generally injected into the lower section of the furnace. It is sometimes fed into the loop-seal, from which it enters the furnace along with returned hot solids. Limestone is fed into the bed in a similar manner. Coal burns when mixed with hot bed solids.

The primary combustion air enters the furnace through an air distributor or grate at the furnace floor. The secondary air is injected at some height above the grate to complete the combustion. Bed solids are well mixed throughout the height of the furnace. Thus, the bed temperature is nearly uniform in the range 800–900 °C, though heat is extracted along its height. Relatively coarse particles of sorbent (limestone) and unburned char, larger than the cyclone cut-off size, are captured in the cyclone and are recycled back near the base of the furnace. Finer solid residues (ash and spent sorbents) generated during combustion and desulphurization leave the furnace, escaping through the cyclones, but they are collected by a bag-house or electrostatic precipitator located further downstream.

Figure 6.50 (b) shows the CFB boiler of the 165 MWe Pt. Aconi Power Plant at Nova Scotia, Canada.

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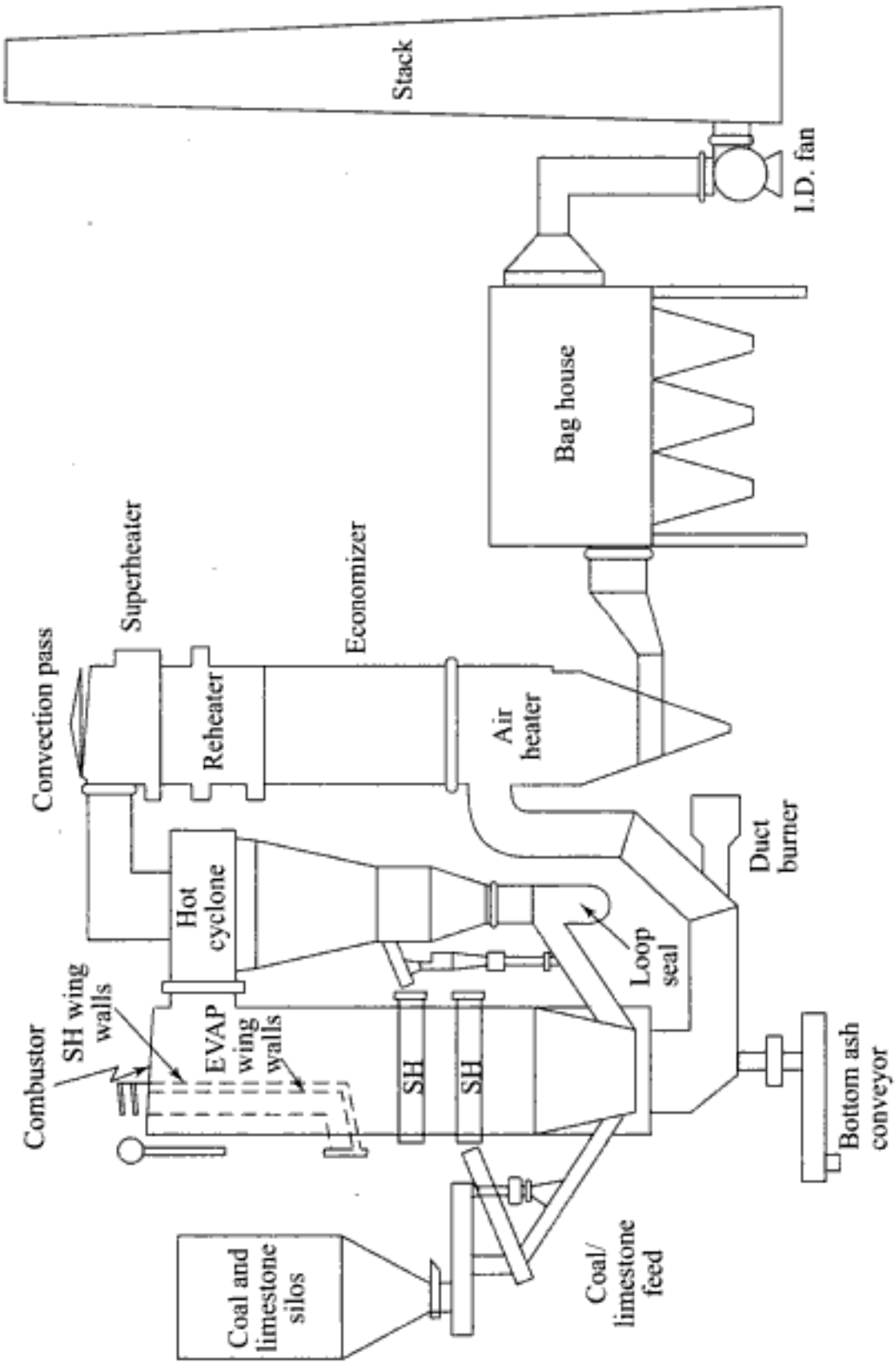

Fig. 6.50 *(b) CFB boiler of 165 MWe Point Aconi Power Plant at Nova Scotia, Canada*

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A schematic diagram of a CFB along with the flow regimes is shown in Fig. 6.51.

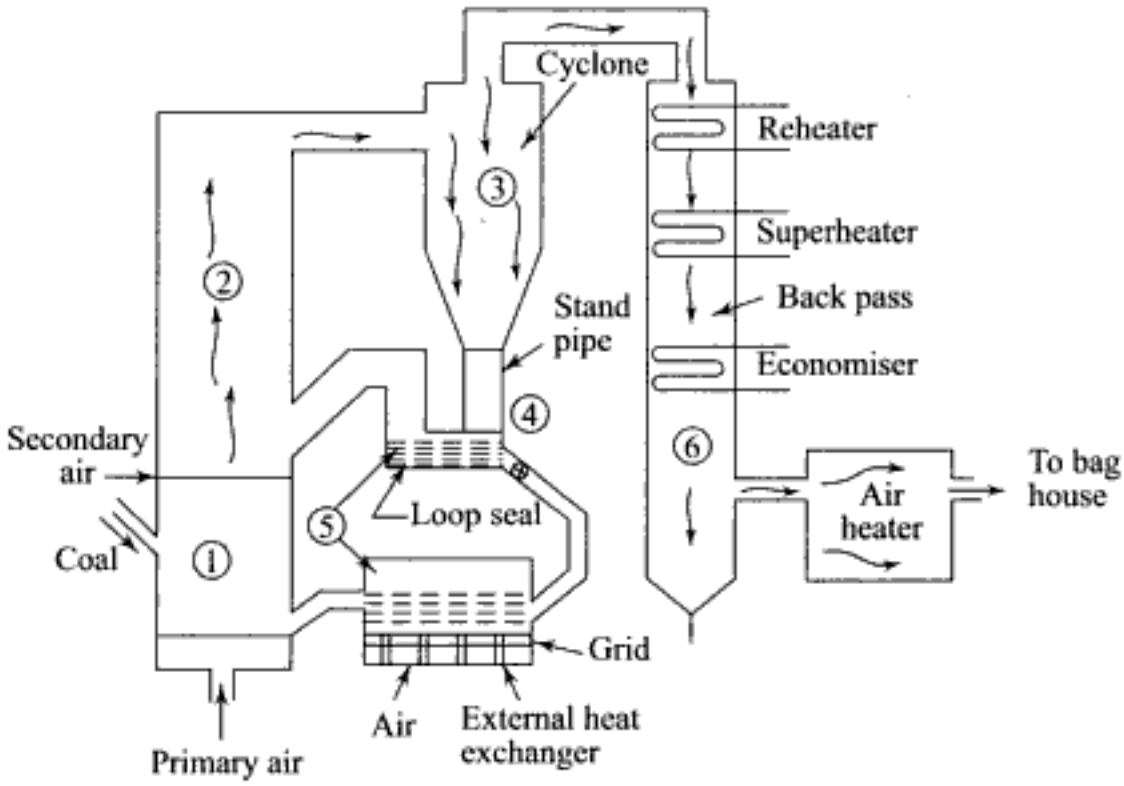

Fig. 6.51 *Flow regimes in a CFB boiler: 1. Turbulent/bubbling fluidized bed, 2. Fast fluidized bed 3. Swirl flow, 4. Moving packed bed, 5. Bubbling fluidized bed, 6. Pneumatic transport*

1. Furnace (below secondary air level)— turbulent/bubbling fluidized bed
2. Furnace (above secondary air level)— fast fluidized bed
3. Cyclone— Swirl flow
4. Return leg (stand pipe)— moving packed bed
5. Loop seal/external heat exchanger— bubbling fluidized bed
6. Back pass— pneumatic transport

6.9.1 Non-mechanical Valves

Non-mechanical valves are devices that allow the flow of solids between the return leg (stand pipe) and the furnace without any external mechanical force. Some of such valves (Fig. 6.52) are: (1) L-valve, (2) J-valve, (3) reverse seal valve, (4) loop seal, and (5) seal pot.

Solids in non-mechanical valves are moved by air. The air is added at a short height above the exit of the valve. The L-valve consists of a right-angled, bent (L-shaped) pipe connecting the two vessels between which the solids are to be transferred. It has a moving packed bed in the return leg and a less dense bed in the furnace into which the solid is fed. A small amount of air injected into the dense leg moves solids to the dilute bed. The L-valve can be designed from a pressure balance across the CFB loop.

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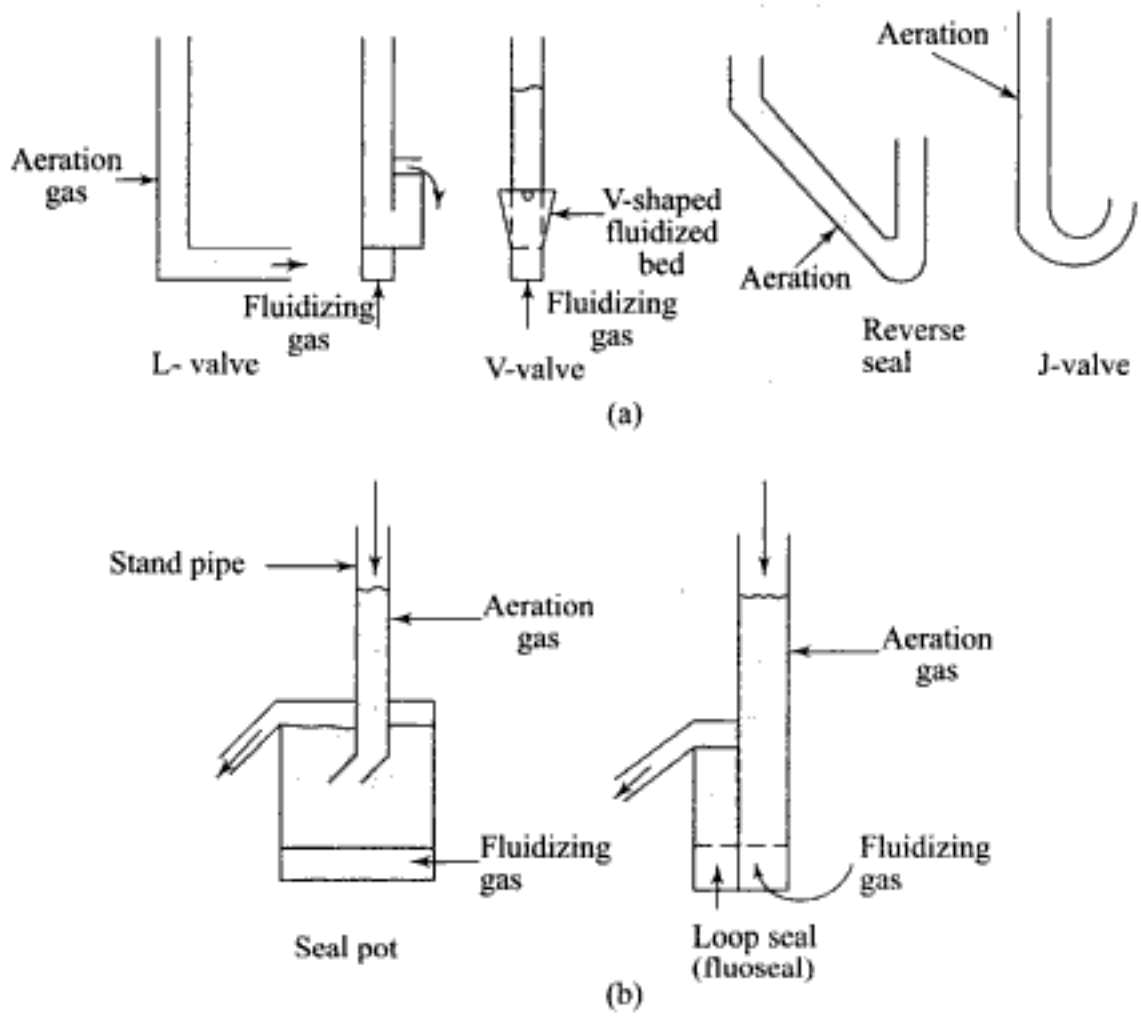

Fig. 6.52 *Non-mechanical valves: (a) controllable, (b) automatic*

The operation of J-valve and reverse seal is similar to the L-valve. The solid flow rate is controlled by changing the aeration rate.

A seal pot consists of a bubbling fluidized bed with an overflow pipe and the standpipe of the cyclone. Higher pressure in the seal pot prevents furnace air from entering the standpipe.

The loop seal is very similar to the seal pot, except that it is smaller and here solids move horizontally around a vertical partition separating the standpipe and the overflow part of the seal pot. It also requires less air than that in the seal pot.

6.9.2 Cyclone Separators

Cyclone separators are gas cleaning devices that utilize the centrifugal force created by a spinning gas stream to separate particles from a gas. A standard tangential inlet vertical cyclone separator is shown in Fig. 6.53. The gas flow is forced to follow the curved geometry of the cyclone, while the inertia of particles in the flow causes them to move toward the outer wall, where they collide and are collected. A particle of mass m_p moving in a circular path of radius r with a tangential velocity v_θ is acted upon by a centrifugal force, $F_c = \dfrac{m_p\, v_\theta^2}{r}$. At a typical velocity of $v_\theta = 10$ m/s, $r = 0.5$ m, $F_c = 200\ m_p$, whereas the gravity force $F_g = m_p \cdot g = 9.81\ m_p$. Thus, the centrifugal force is 20.4 times the gravity force on the same particle.

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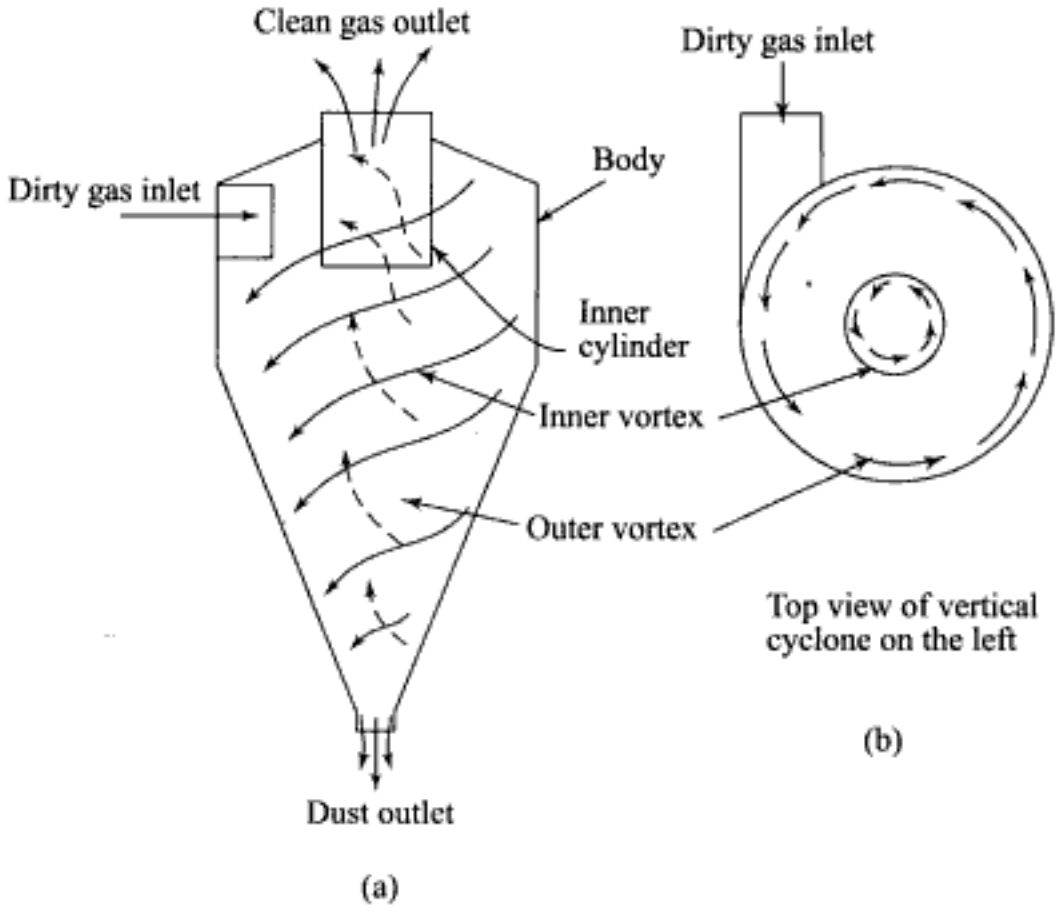

Fig. 6.53 *A tangential inlet vertical cyclone separator*

In a cyclone, the particles in the spinning gas stream move progressively closer to the outer wall as they flow through the device, The gas stream may execute several complete turns as it flows from one end of the device to the other.

The particle-laden gas is entered tangentially to the cyclone. The particles are forced to the wall by centrifugal force and then fall down the wall by gravity. At the bottom of the cyclone the gas flow reverses to form the inner core that leaves at the top of the unit.

Out of m kg of particles entering the cyclone with the gas stream, if m_1 kg (fines) leave with the gas at the top and is collected, say, in a bag filter, then $(m - m_1)$ kg of particles are separated by the cyclone from the gas–solid mixture, and the expression $\left(\frac{m - m_1}{m} \times 100\right)$ is called the *collection efficiency* of the cyclone. The cyclone collection efficiency increases with increasing (a) particle size, (b) particle density, (c) inlet gas velocity, (d) cyclone body length, (e) number of gas revolutions, and (f) smoothness of cyclone walls. The collection efficiency decreases with increasing (a) cyclone diameter, (b) gas outlet duct diameter, and (c) gas inlet area.

For any specific cyclone whose ratio of dimensions is fixed, the collection efficiency increases as the cyclone diameter is decreased. The design of a cyclone separator represents a compromise among collection efficiency, pressure drop, and size. Higher efficiency requires higher pressure drops (i.e. inlet gas velocity) and larger sizes (i.e. body length).

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Shepherd and Lapple determined optimum dimensions for cyclones (Fig. 6.54) (Perry and Chilton, 1973). All dimensions were related to the body diameter D_c.

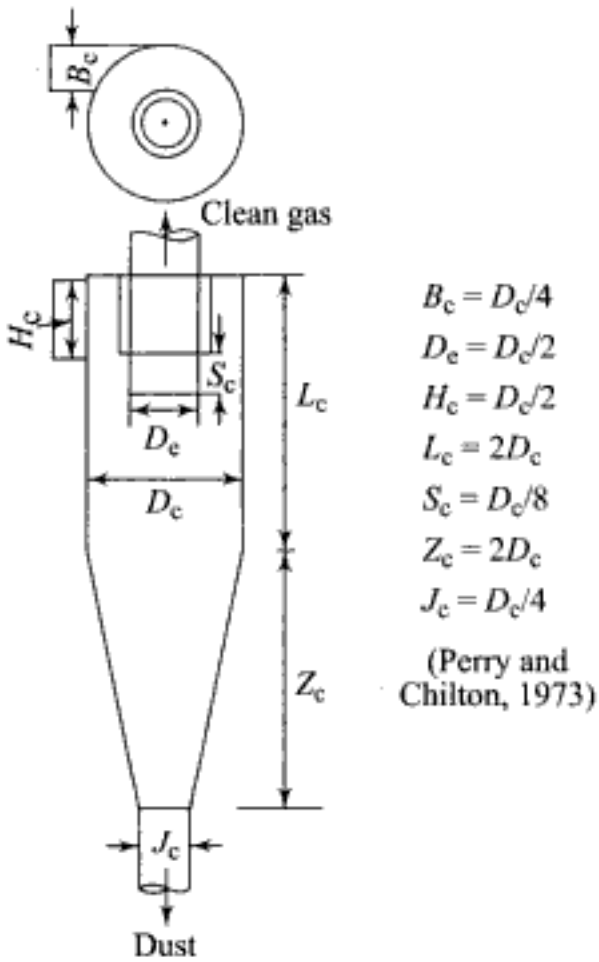

Fig. 6.54 *Optimum dimensions for cyclones*

The cyclone in CFB boilers handles a large volume of gas at high temperatures. Small diameter cyclones have higher collection efficiencies. Multiple numbers of cyclones of smaller size and higher efficiency can be used in parallel instead of a large cyclone of lower efficiency.

Cyclones used in CFB boilers often contain a large mass of refractory, which takes a long time to heat up and cool down. As a result, CFB boilers take a much longer time to start up, which is a drawback.

The outer skin temperature of the cyclone is relatively high, entailing high energy losses by natural convection and radiation. To overcome this problem, some cyclones are being made water or steam cooled. Typical performance parameters of a cyclone in a CFB boiler are given below:

Inlet gas velocity	: 25–43 m/s
Solid concentration at inlet	: 0.5–5 kg/m^3 gas
Particles size range	: < 1–1000 μm
Gas throughput	: < 250,000 Nm^3/h
Cut-off particle diameter for 50% efficiency	: ~ 10 μm
Pressure drop	: 100–200 mm water gauge

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6.9.3 Advantages of CFB Boilers

Circulating fluidized bed boilers have a number of unique features that make them more attractive than other solid fuel fired boilers as given below.

1. Fuel Flexibility This is one of the most attractive features of CFB boilers. Fuel particles constitute less than 1–3% by weight of all bed solids in the furnace of a typical CFB boiler. The rest of the solids are non-combustible: sorbents (limestone or dolomite) and fuel-ash. The special hydrodynamic condition in the CFB furnace allows an excellent gas–solid and solid–solid mixing. Thus fuel particles fed to the furnace are quickly dispersed into the large mass of bed solids, which rapidly heat the fuel paticles above their ignition temperature without any significant drop in the temperature of bed solids. This feature of a CFB furnace allows it to burn any fuel without the support of an auxiliary fuel. It can thus burn a wide variety of fuels very efficiently in the same boiler, either simultaneously or separately. High ash bituminous coals, anthracite, coal washery rejects, peat, wood chips, petroleum coke, oil shale and so on have been fired in CFB units.

To maintain the combustion temperature within an optimum range (800–900 °C), a certain portion of the generated heat is required to be absorbed from the combustion zone. This fraction varies from one fuel to another. It is controlled by adjusting the heat extraction from the recirculating solids outside the furnace by means of an external heat exchanger. If the external heat exchanger is absent, the hydrodynamic condition in the furnace is adjusted (say, by regulating the solid recirculation rate and superficial air velocity) so as to change the solid concentration (or suspension density) in the furnace, which in turn, alters the heat absorption in the furnace. The solid circulation rate can be adjusted by regulating the ratio of primary and secondary air. By increasing the primary air the solid circulation rate can be increased.

2. High combustion efficiency Superior mixing in the CFB, large reaction space (combustion zone extending up to the top of the furnace, about 40 m and above in large utility boilers, and further beyond into the hot cyclone) and consequently long residence time of combustion afford high combustion efficiencies, which can be as high as 99%, for a wide variety of fuels under different operating conditions.

3. Efficient sulphur removal When coal burns, sulphur is oxidized primarily to sulphur dioxide

$$S + O_2 = SO_2 + 9260 \text{ kJ/kg}$$

Some part of SO_2 may be converted to SO_3

$$SO_2 + \frac{1}{2}O_2 = SO_3$$

This SO_2 and SO_3 if thrown out to atmosphere undergo chemical reactions with moisture, catalyzed by sunlight, to form acids. These acids are precipitated

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on the earth through rain as acid rain. This is lethal to aquatic life, trees and plants and destroys the nutrients in soil.

Limestone ($CaCO_3$) and dolomite ($CaCO_3$, $MgCO_3$) are two principal sorbents used for the absorption of SO_2 in fluidized bed combustors. For limestone directly fed into the CFB furnace as bed material, there is calcination when the limestone is decomposed into CaO and CO_2 through an endothermic reaction

$$CaCO_3 = CaO + CO_2 - 180 \text{ kJ/kg } CaCO_3$$

In a sufficiently oxidizing environment, the porous calcined limestone reacts as

$$CaO + SO_2 + \frac{1}{2}O_2 = CaSO_4 + 15{,}141 \text{ kJ}$$

$CaSO_4$ retains the sulphur in the bed in the solid form, rather than allowing it to escape as gaseous SO_2.

Thermodynamically, the formation of $CaSO_4$ is favoured at lower temperatures. The rate of formation is maximized at about 850 °C. At temperatures exceeding 900 °C, the sulphur capture is greatly reduced. This is one of the reasons why fluidized bed combustion processes are constrained to the 800–900 °C temperature range.

Good contact between the gas and solid and the long residence time in the CFB combustor afford better sulphur capture at a given Ca/S ratio than in a bubbling fluidized bed. Industrial CFB boilers show 90% sulphur capture at Ca/S ratio of 1.5 to 2.0.

Limestone particles below 3 mm size are typically used. The SO_2 capture efficiency is a function of temperature, Ca/S ratio, stone structure and composition, FBC design and pressure. Representative results of the effects of Ca/S ratio and temperature for atmospheric bubbling beds are shown in Fig. 6.55 (a).

The effect to pressure on sulphur retention for limestone and dolomite is shown in Fig. 6.55 (b). Unlike atmospheric FBCS, the sulphur capture efficiency does not pass through a maximum, but continues to increase with combustor temperature.

4. Low NO_x emission Owing to low combustion temperature and the staged combustion process, NO_x emission in a CFB boiler can be kept low, in the range 50–150 ppm. Figure 6.55(c) shows the combustion zones together with the entry points of primary and secondary air. Primary air fed from below through the distributor contains 50–60% of the stoichiometric requirements. Thus, the fuel nitrogen, when released, does not find oxygen in the immediate vicinity to produce oxides. This strategy of low NO_x formation is more effective when the fuel contains a high level of volatile organic compounds. Secondary air is added above the reducing zone. Since the fuel nitrogen is already transformed into moleculer nitrogen, there is limited opportunity for the formation of NO_2 above this zone. The nitrogen in the combustion air is not normally oxidized to NO_2 at the relatively low temperatures (800–900 °C) of CFB combustors.

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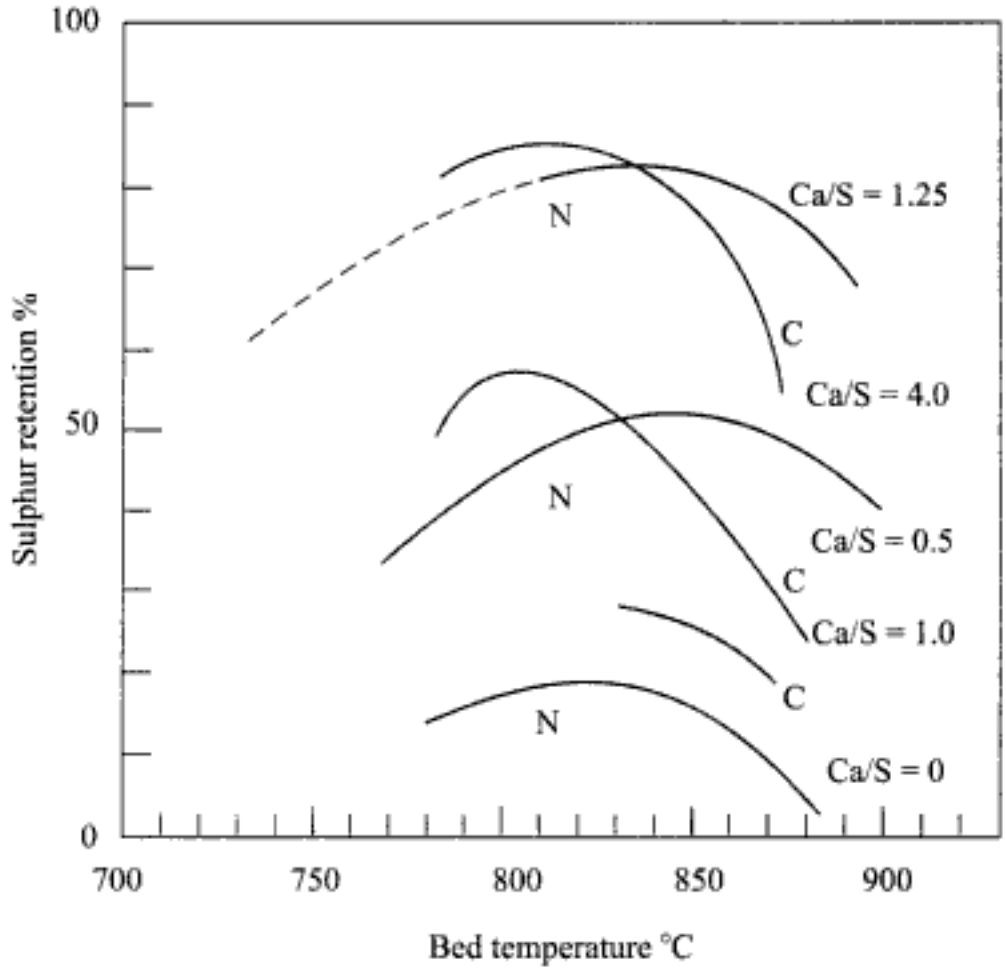

Fig. 6.55 (a) *Sulphur capture in atmospheric BFBC at various Ca/S molar ratios as a function of bed temperature*

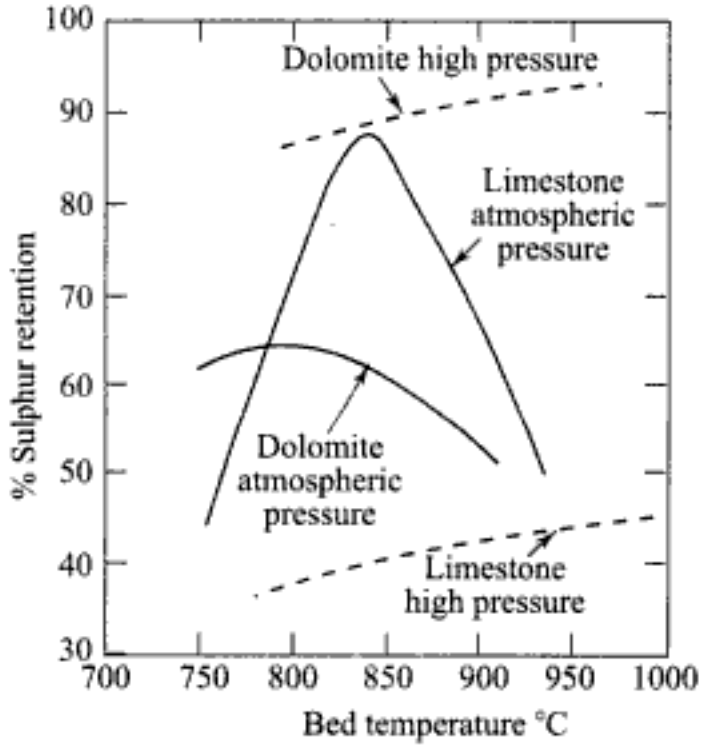

Fig. 6.55 (b) *Effect of pressure on sulphur retention for reactive limestone and dolomites sized 0-1600 μm, normalized to Ca/S = 2 and reaction time = 0.5 s*

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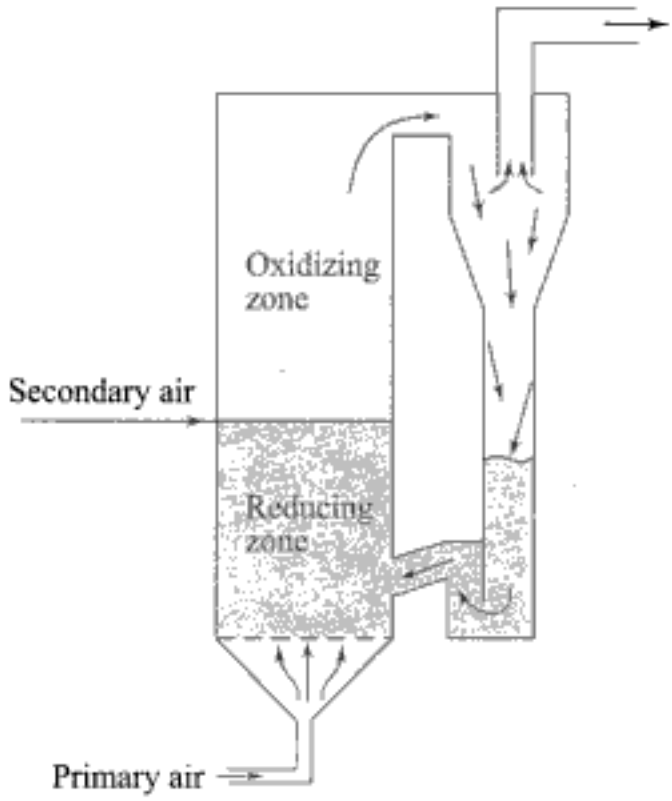

Fig. 6.55 (c) *Staged combustion in a CFB*

5. Simpler fuel handling and feed system In contrast to the conventional pulverized coal fired boiler, CFB boilers receive solid fuels in fairly coarse sizes. This simplifies the upstream feed preparation. But the most important advantage is the need for only one (industrial boiler) or few (utility boiler) feedpoints. The high degree of lateral solid mixing in turbulent zone at the bottom ensures uniform feed distribution within the bed. Besides, the large height corrects any maldistribution.

6. Small furnace cross-section A high heat release rate per unit furnace cross-section (about 5 MW/m^2) is a major advantage of the CFB boiler. It results in less floor area. A high superficial velocity (4–7 m/s) along with an intense gas–solid mixing promotes a high heat release and heat dispersion in the bed.

7. Good turndown and load following capability The relatively high fluidizing gas velocity and the use of staged combustion permit a fairly good turndown ratio by simply reducing the proportionate amounts of fuel and air. The turndown ratio of 5 : 1 has been achieved (20% load).

CFB boilers respond quickly to load changes. The given load can be readily adjusted by changing the ratio of primary and secondary air, and by controlling the solid circulation rate.

8. High availability The availability records of CFB boilers are highly impressive. It is more than 90%.

The major disadvantages of CFB boilers are, however, (i) erosion of reactor walls, (ii) attrition of particles, (iii) difficulty of immersing internals due to possible erosion, and (iv) complexities of the hydrodynamics.

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6.9.4 Status of CFB Boilers

The world's first commercial CFB boiler went into service at Pihlava, Finland in 1979. The pioneering company was Ahlstrom and the aim was to convert an oil-fired unit into a CFB boiler producing 20 t/h of steam from wood waste. Successful operation of this unit led to the start-up in 1981 of a 90t/h (steam) CFB boiler at Kauttua, Finland, that was designed to burn coal, wood waste and peat.

Lurgi built up in 1982 a 84 MW (thermal) CFB boiler at Leunen, Germany, burning coal washery rejects with ash content of up to 55%. In the relatively short span of its commercial life, the CFB boiler along with its "bubbling" counterpart, has established itself as a reliable and attractive option for industrial boilers to supply process steam. A few CFB utility boilers in the 100–165 MWe range are already in operation and some are in various stages of construction. The 165 MWe Point Aconi Power Station at Cape Breton in Nova Scotia, Canada, has now the largest CFB boiler in operation. The biggest CFB boiler in the erection stage is for a 250 MWe power unit at Lyon in France.

Table 6.1 gives a comparison of circulating fluidized bed boilers with other types of boilers (Basu and Fraser, 1991).

Table 6.1 *Comparison of boilers*

Characteristics	*Stoker*	*Bubbling*	*Circulating*	*Pulverized*
Height of bed or fuel burning zone (m)	0.2	1–2	15–40	27–45
Superficial velocity (m/s)	1.2	1.5–2.5	4–8	4–6
Excess air (%)	20–30	20–25	10–20	15–30
Grate heat release rate (MW/m^2)	0.5–1.5	0.5–1.5	3–5	4–6
Coal size (mm)	32–6	6–0	6–0	<0.0001
Turndown ratio	4 : 1	3 : 1	5 : 1	
Combustion efficiency (%)	80–85	90–96	95–99	99
NO_x emission (ppm)	400–600	300–400	50–200	400–600
SO_2 capture in furnace(%)	none	80–90	80–90	small

6.9.5 Pressurized Fluidized Bed Boiler

The combustion process takes place in a pressurized environment resulting in a compact furnace and improved combustion efficiency. There is *in situ* sulphur capture and low NO_x generation with the bed operating at about 850 °C like an atmospheric fluidized bed combustion (AFBC) furnace. The process allows the use of a gas turbine, driven by pressurized hot combustion gases. Thus, in

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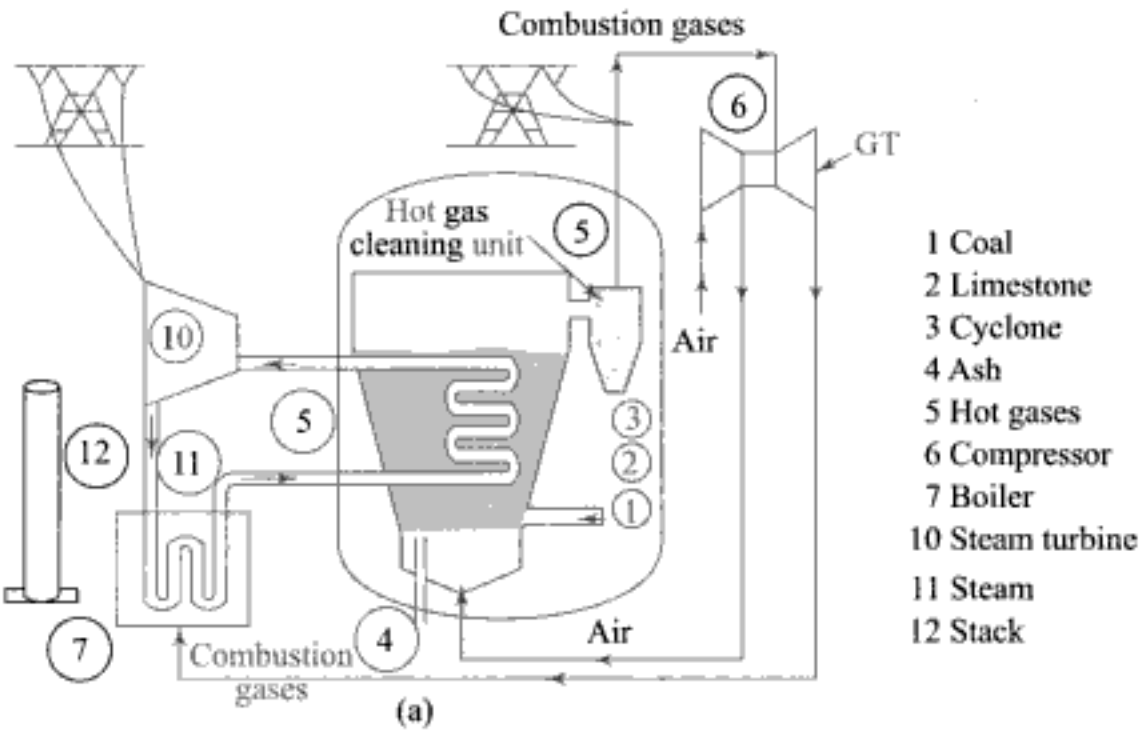
addition to having the ability to burn high sulphur and low grade coal in an environmentally acceptable manner and the compactness of the steam generator, PFBC also results in the ability to power a gas turbine by combusting inexpensive coal.

Table 6.2 lists the representative conditions in four major classes of fluidized bed combustors.

Table 6.2 *Representative conditions in major classes of FBCS*

	Atmospheric		*Pressurized (~ 17 atm)*	
	BFBC	*CFBC*	*BFBC*	*CFBC*
1. Gas velocity (m/s)	1.3–3.5	4–9	~ 1	~ 5
2. Bed height (m)	1.0–1.4	~ 35	3.5–4.0	40
3. Coal feed points/m^2	0.5–1.0	0.06–0.08	0.25–6	0.2–0.5
4. Bed residence time (s)	~ 0.5	~ 5	~ 4	~ 8

Figure 6.56 shows three schemes of turbo-charged, combined and hybrid advanced cycles, each equipped with a PFBC unit.



(a)

Like AFBC units, there are two types of pressurized combustors:

1. Pressurized bubbling fluidized bed combustor (PBFBC).
2. Pressurized circulating fluidized bed combustor (PCFBC).

In a pressurized bubbling fluidized bed combustor (PFBC), the granular solids are fluidized in bubbling fluidized mode. Here, the superficial gas velocity generally lies between the minimum bubbling velocity of coarser particles and terminal velocity of finer particles. A part of the heat generated in the fluidized bed is extracted by water carrying tubes in the bed. The hot gas is cleaned and then expanded through the gas turbine (Fig. 6.57). The steam produced in the

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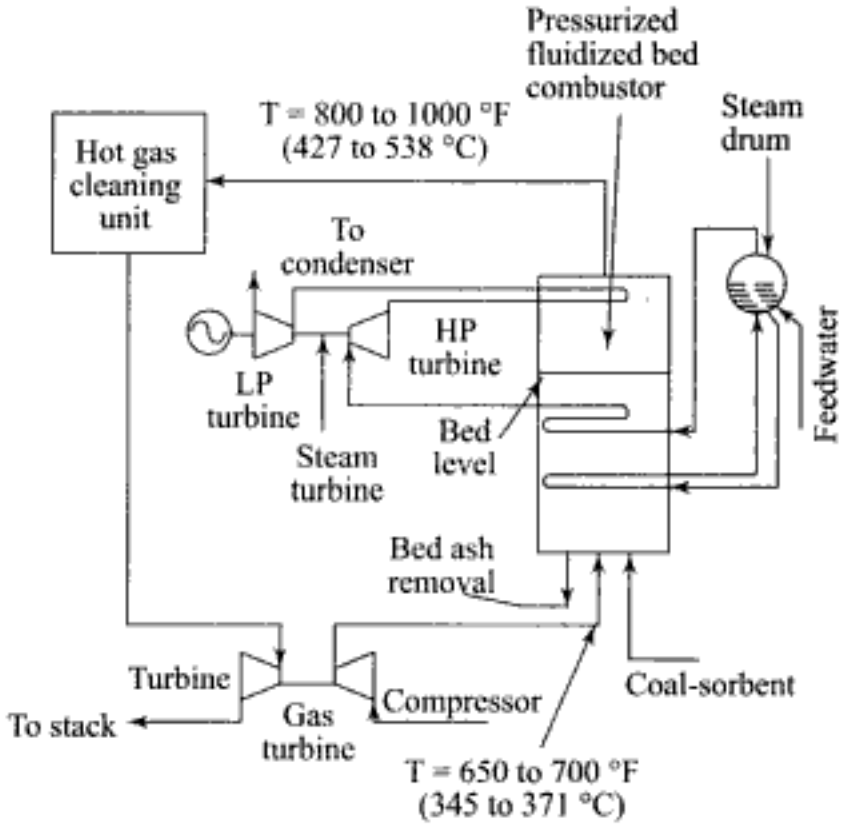
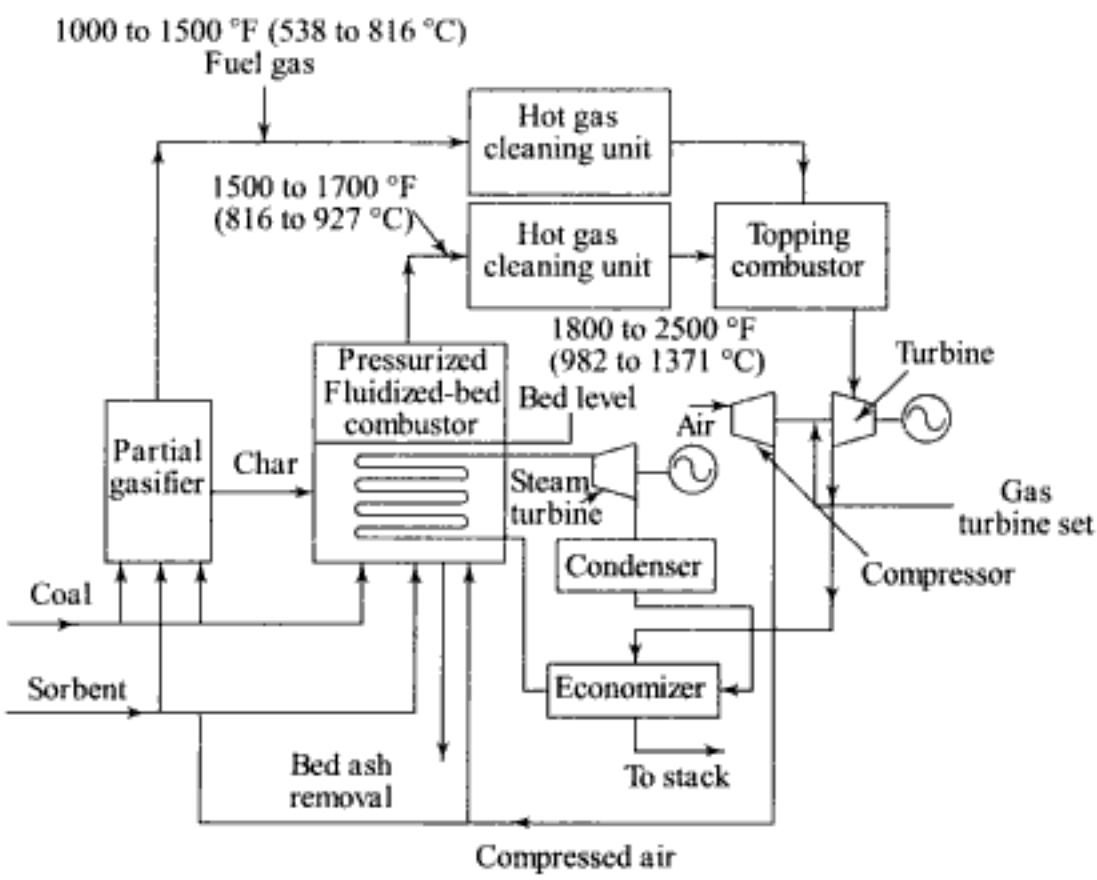


(b)

combustor or the waste heat recovery heat exchanger at the gas turbine exit produces further power by expanding through a steam turbine.



(c)

Fig. 6.56 *PFBC units in (a) combined cycle, (b) turbocharged cycles, and (c) hybrid advanced cycle*

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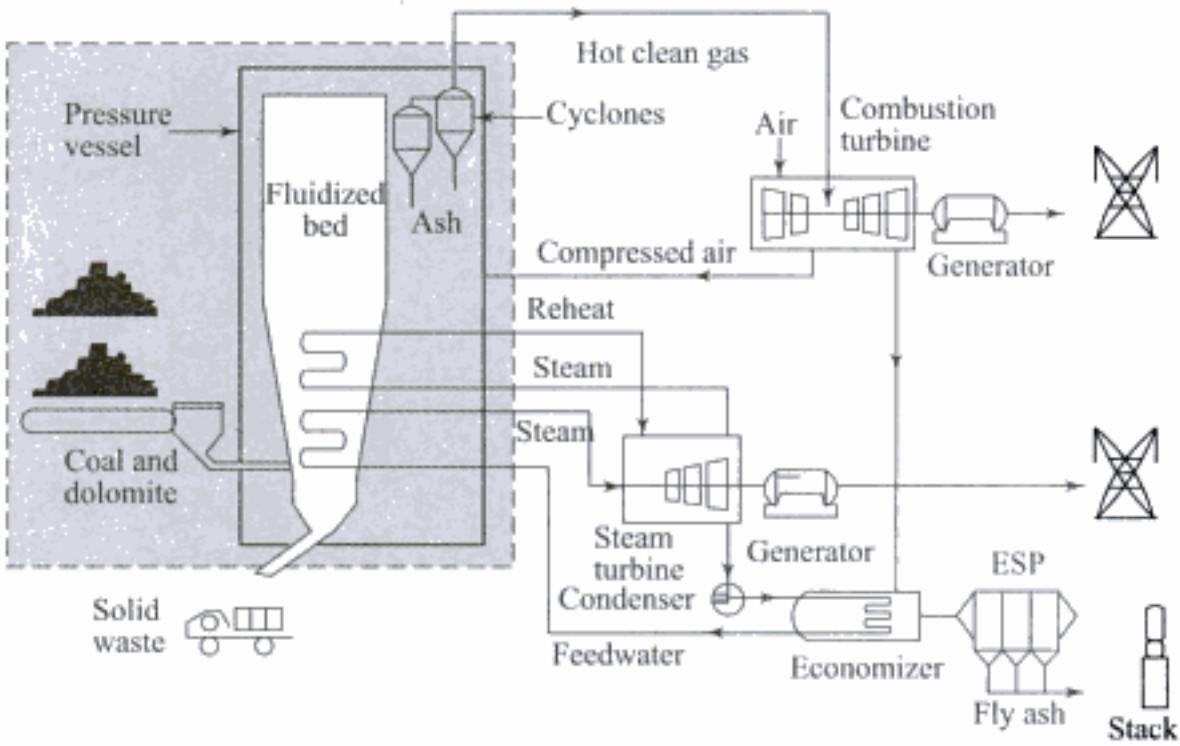

Fig. 6.57 *Schematic of a pressurized bubbling fluidized bed combustor*

The advantages of PBFBC are as follows:

1. There is an increase in its specific power output and hence, a potential reduction in capital cost. At a typical fluidizing velocity of 2 m/s the bed area of AFBC at 1 bar is 2 m^2/MW, while it is 0.2 m^2/MW at 10 bar.
2. By combining a gas turbine with a steam turbine, the overall efficiency of power generation system can be increased from 33% for conventional power plants to more than 40% in PBFBC plants.
3. The emission of nitrogen oxides can be substantially reduced.
4. The PBFBC has a higher combustion efficiency than an AFBC.
5. The gas residence time in an AFBC is about 0.5 s, whereas it is about 5 s in a PBFBC (due to low fluidizing velocity of about 1 m/s for a higher gas density). Therefore, the sulphur capture is more.

The solids in a pressurized circulating fluidized bed (PCFB) are kept in "fast fluidized" condition (Fig. 6.58). A major feature of PCFB is that it approaches a condition where solids are fully mixed throughout the combustor, but there is very little mixing of gas in the axial direction. The gas flows upward with negligible back mixing. Primary air, less than stoichiometric amount, enters the combustor through the grid at the bottom of the combustor. The secondary air enters at some distance above the grid. Due to the absence of back mixing of the air, a substoichiometric condition prevails in the lower section of the bed. The boiler tubes are generally located above this zone. This staging of air reduces NO_x emission and also the chance of corrosion of the tubes.

As compared to PBFB, PCFB offers the following advantages.

1. *Reduced space requirement:* PCFB has a higher velocity, higher heat release per unit area and fewer components inside the pressure vessel. So, the pressure vessel diameter is smaller.

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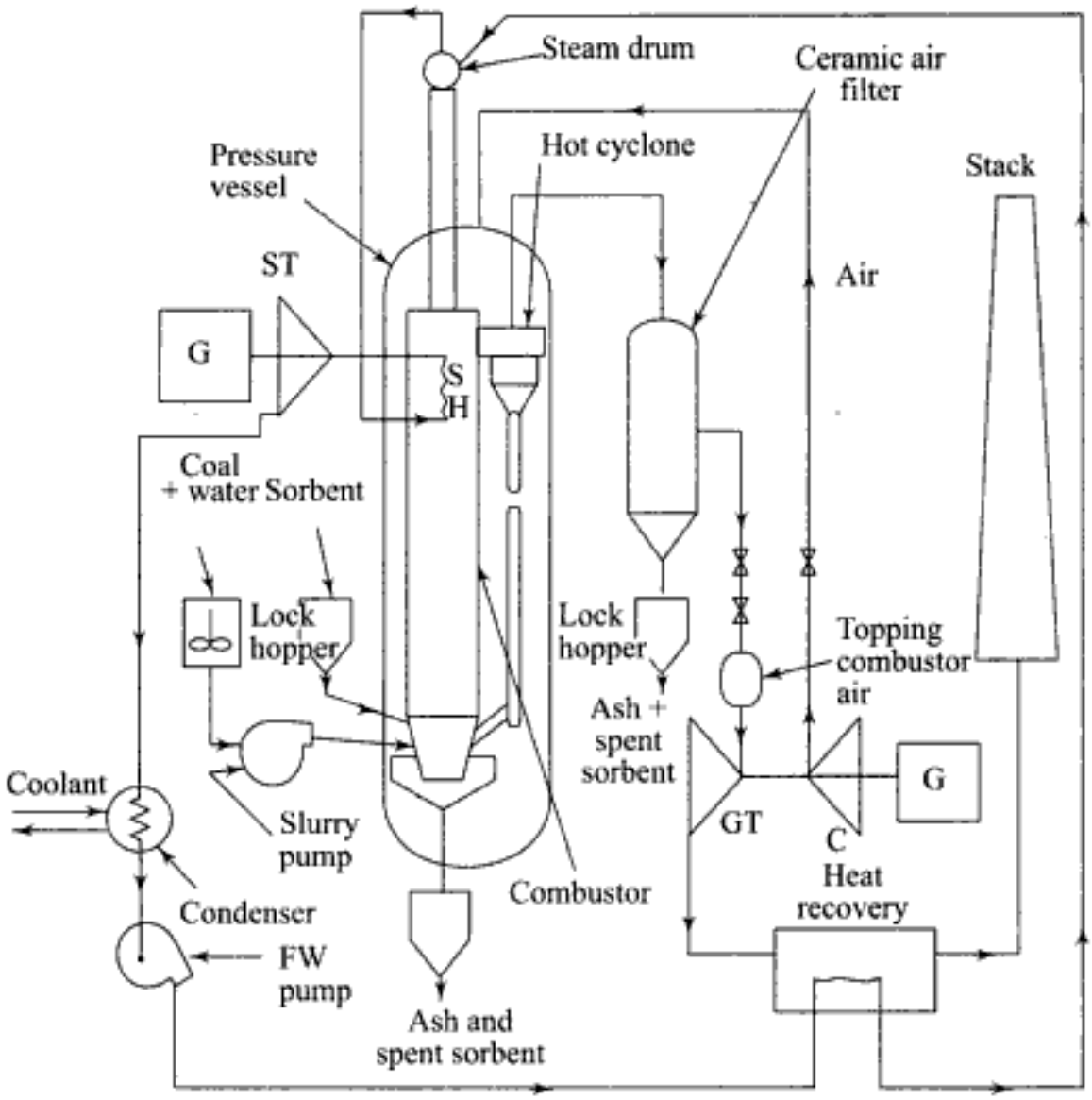

Fig. 6.58 *A pressurized circulating fluidized bed combustor*

2. *Less erosion and more accessibility:* PCFB has a low particle density fluidized bed with fine particles distributed throughout the furnace. In PBFB, submerged heat transfer surfaces are used. So in PCFB (a) the potential for erosion is greatly reduced with no immersed surfaces, (b) platens for superheating are more accessible for inspection and maintenance.
3. PCFB has *more vigorous mixing* in the furnace which results in fewer feedpoints and results in a simpler, more reliable system.
4. PCFB *does not require a bed material handling system* to control the bed temperature.
5. The *exit gas temperature of PCFB is higher* since there is no freeboard where the flue gas may be cooled down. This is favourable to the gas turbine.
6. PCFB has *lower NO_x emission* due to enhanced staged combustion.
7. *Load following is more effectively accomplished* by varying the fuel and air flow and the proportion of primary and secondary air.

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8. PCFB has *less bed inventory and less total metal and refractory*. This results in a shorter start-up and cool-down period and a quicker response to load changes.
9. At higher pressures, the partial pressure of carbon dioxide, P_{CO2}, in a bubbling bed is higher, which requires higher temperature for calcination of $CaCO_3$. One *has to use dolomite instead of limestone in PBFBC.*

In PCFB air is added in stages. In substoichiometric zone P_{CO_2} is lower. So, $CaCO_3$ in limestone is easily calcined increasing pore volumes for sulfation reaction. So, PCFB units *can use more readily available limestone than PBFBC.*

The boiler, which is integral with combustor could use either of the three types of circulation (Fig. 6.59).

(a) Once-through circulation: The feed pump forces water through the entire steam-water circuit. It is suitable at high load. When water-flow rate decreases, the evaporator surface can perform superheater duty. This increases the metal temperature, which, in turn, reduces the heat absorption—a requirement at low load. A vertical steam–water separator feeds saturated water to a circulation pump below, which maintains sufficient flow through the boiler.

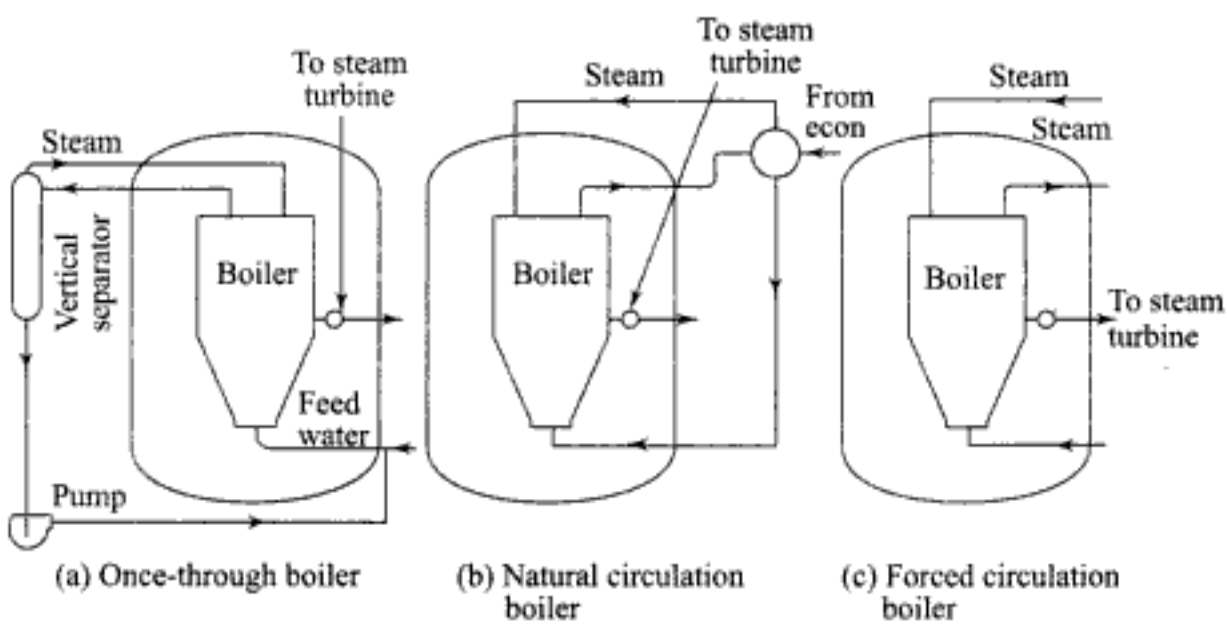

Fig. 6.59 *Different steam–water circulation systems in PFBC boilers*

(b) Natural circulation: This is caused by density difference, as explained earlier. It is convenient at low load. The steam drum is installed outside to provide adequate hydrostatic head for natural circulation.

(c) Forced circulation: Water from the economiser is fed by the boiler feed pump into the boiler drum. A forced circulation pump circulates the water from the drum through the downcomer–riser circuits back to the drum. Superheater and reheater steams flow under their own pressure.

Two–stage cyclones are used to clean the hot flue gas adequately for expansion through commercial gas turbines. The entry velocities of gas at entry to the first and second stage cyclones are less than 30 m/s and 42 m/s respectively. The dust loading in the cleaned gas is about 250 ppm. The gas requires to be further cleaned by passing it through ceramic filters.

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6.10 ELECTROSTATIC PRECIPITATOR

In 1905, Dr. F.G. Cottrell, Professor of Physical Chemistry at the University of California, conducted a series of laboratory experiments that resulted in the development of the first commercial electrostatic precipitator. It was an immediate success and the precipitator soon came to be widely used in power plants, smelters, steel plants, paper mills and many other industries.

The principal components of an electrostatic precipitator (ESP) are two sets of electrodes insulated from each other. The first set is composed of rows of electrically grounded vertical parallel plates, called the *collection electrodes*, between which the dust-laden gas flows. The second set of electrodes consists of wires, called the *discharge* or *emitting electrodes* that are centrally located between each pair of parallel plates (Fig. 6.60). The wires carry a unidirectional negatively charged high-voltage (between 20 and 100 kV) current from an external dc source. The applied high voltage generates a unidirectional, non-uniform electrical field whose magnitude is greatest near the discharge electrodes. When that voltage is high enough, a blue luminous glow, called a *corona*, is produced around them. Electrical forces in the corona accelerate the free electrons present in the gas so that they ionize the gas molecules, thus forming more electrons and positive gas ions. The new electrons create again more free electrons and ions, which result in a chain reaction.

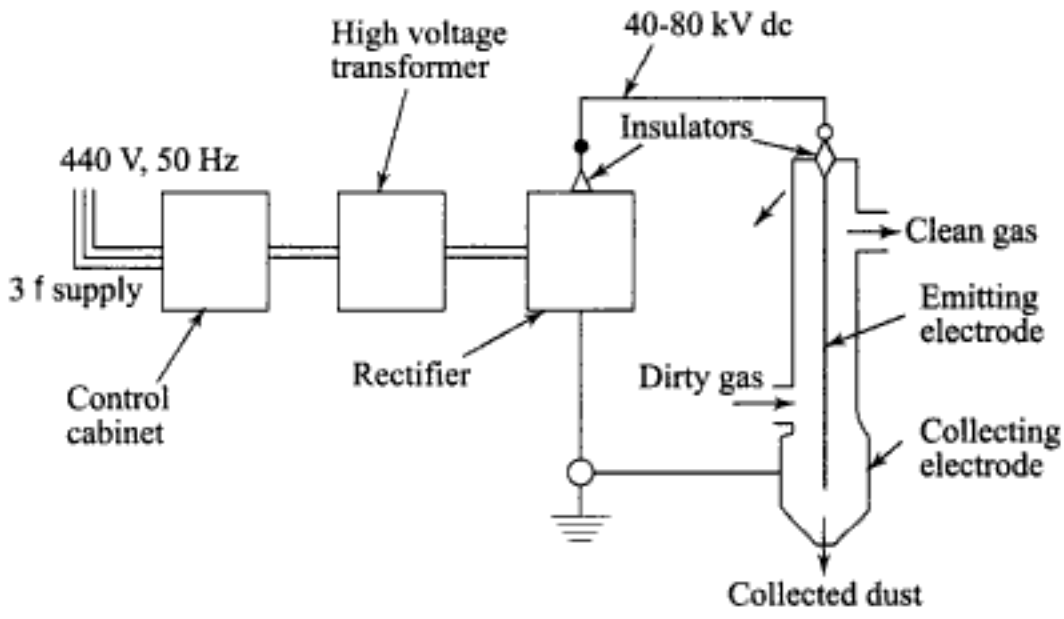

Fig. 6.60 *Basic elements of an electrostatic precipitator*

The positive ions travel to the negatively charged wire electrodes. The electrons follow the electrical field toward the grounded electrodes, but their velocity decreases as they move away from the corona region around the wire electrodes toward the grounded plates. Gas molecules capture the low velocity electrons and become negative ions. As these ions move to the collecting electrode, they collide with the fly ash particles in the gas stream and give them

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negative charge. The negatively charged fly ash particles are driven to the collecting plate by the force which is proportional to the product of this charge and the strength of the electric field (Fig. 6.61).

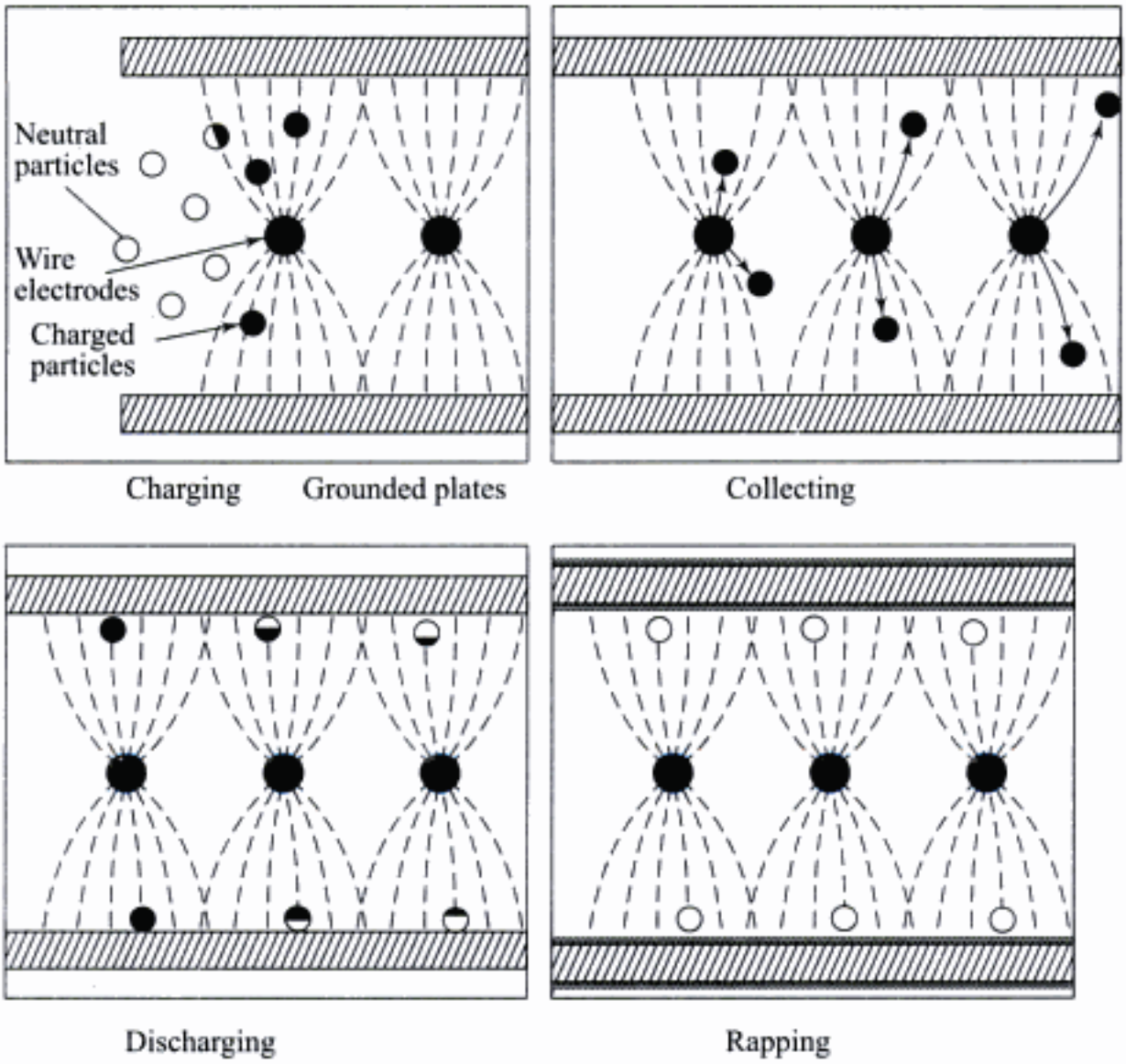

Fig. 6.61 *Vertical electrodes and grounded plates in an ESP showing four basic operations*

When the particles collect on the grounded plates, they lose their charge on the ground. The electrical resistivity of the particles, however, cause only partial discharging, and the retained charge tends to hold the particles to the plates. High resistivity causes retention of most of the charge, which increases the forces holding the particles to the plates and makes removal more difficult. This can be rectified either by operating at high gas temperatures (before APH) or by superimposing a high voltage pulse on the base voltage to enhance ESP performance during operation under high-resistivity conditions.

Collected particulate matter must be removed from the collecting plates on a regular schedule to ensure efficient collector operation. Removal is usually accomplished by a mechanical hammer scrapping system. The vibration knocks the particulate matter off the collecting plates and into a hopper at the bottom of the precipitator (Fig. 6.62).

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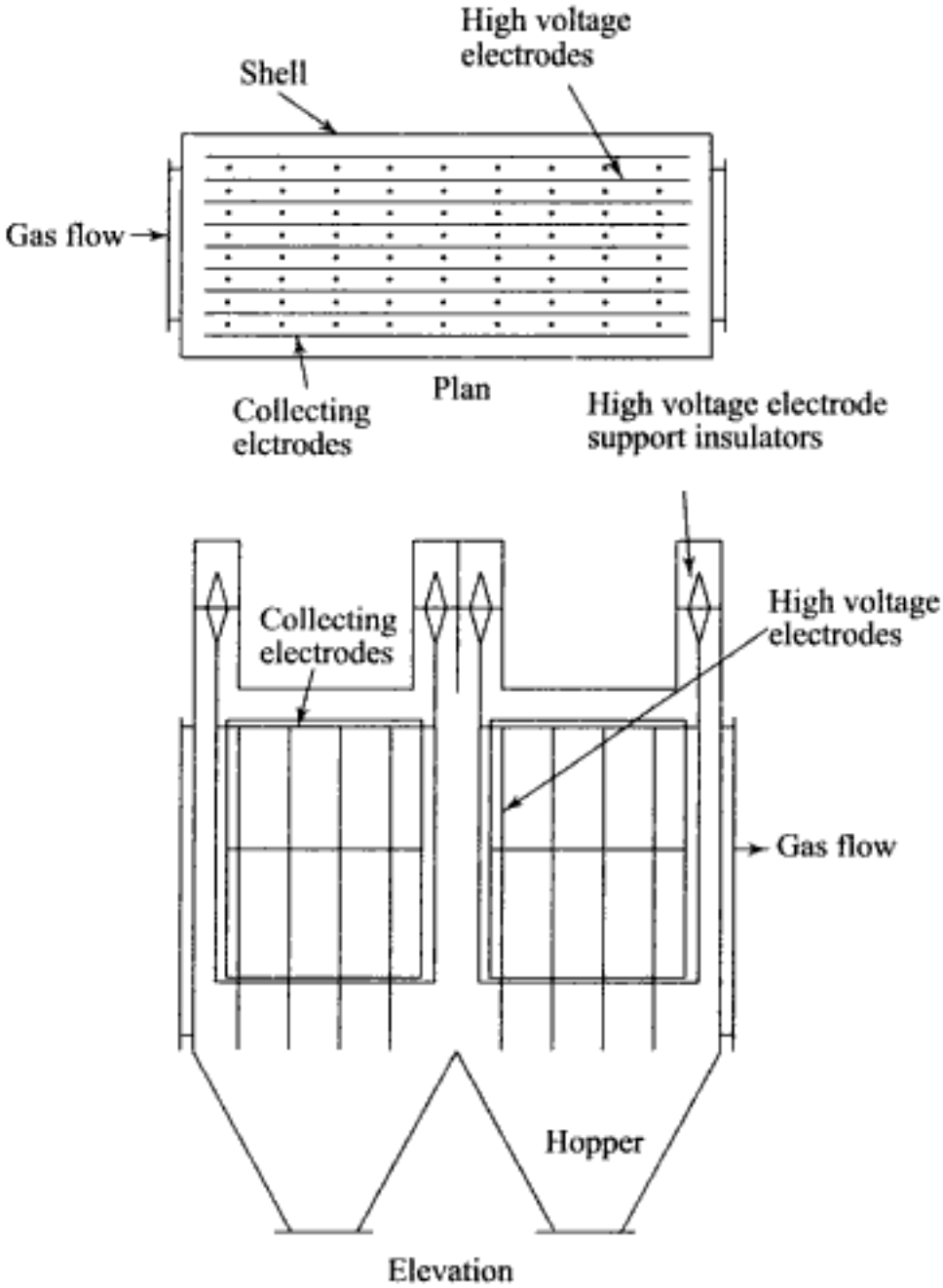

Fig. 6.62 *Arrangement of an electrostatic precipitator*

An electrostatic precipitator, like a cyclone separator, has an overall collection efficiency, η_0, defined by

$$\eta_0 = \frac{\text{mass of all particles retained by collector (ESP)}}{\text{mass of all particles entering collector}}$$

It is easier to collect large particles than smaller particles. It can be approximately estimated by the *Deutsch equation* as given below

$$\eta_0 = 1 - \exp(-AV_{mo}/Q) \tag{7.39}$$

where A = area of the collector plate, m^2,

V_{mo} = effective migration velocity of particles, m/s,

Q = flue gas volume flow rate for each plate, m^3/s.

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6.11 FABRIC FILTERS AND BAGHOUSES

Fabric filters are used in power plants to remove dust particles from a gas stream. They are made of porous material which retains particulate matter while the carrier gas flows out through the voids.

A fabric filter element is usually made in the form of a long, hollow cylindrical tube that provides a large surface area per unit of gas volumetric flow rate (m^2/m^3 gas flow). The reciprocal of this parameter, called the *air-to-cloth ratio* (A/C value). It is a fundamental fabric filter descriptor denoting the ratio of the flue gas flow to the amount of fabric, or filtering surface area. It ranges typically between 5 to 40 mm/s.

A fabric filter system usually contains a large number of fabric filter elements arranged in parallel rows. It is called a *baghouse* (Fig. 6.63). A power plant baghouse might contain several thousand vertical hollow cylinders, each ranging in diameter from 125 to 350 mm and up to 12.5 m in height. The elements have, in general, an open bottom and closed top. They rest on a tube sheet above a dirty air plenum.

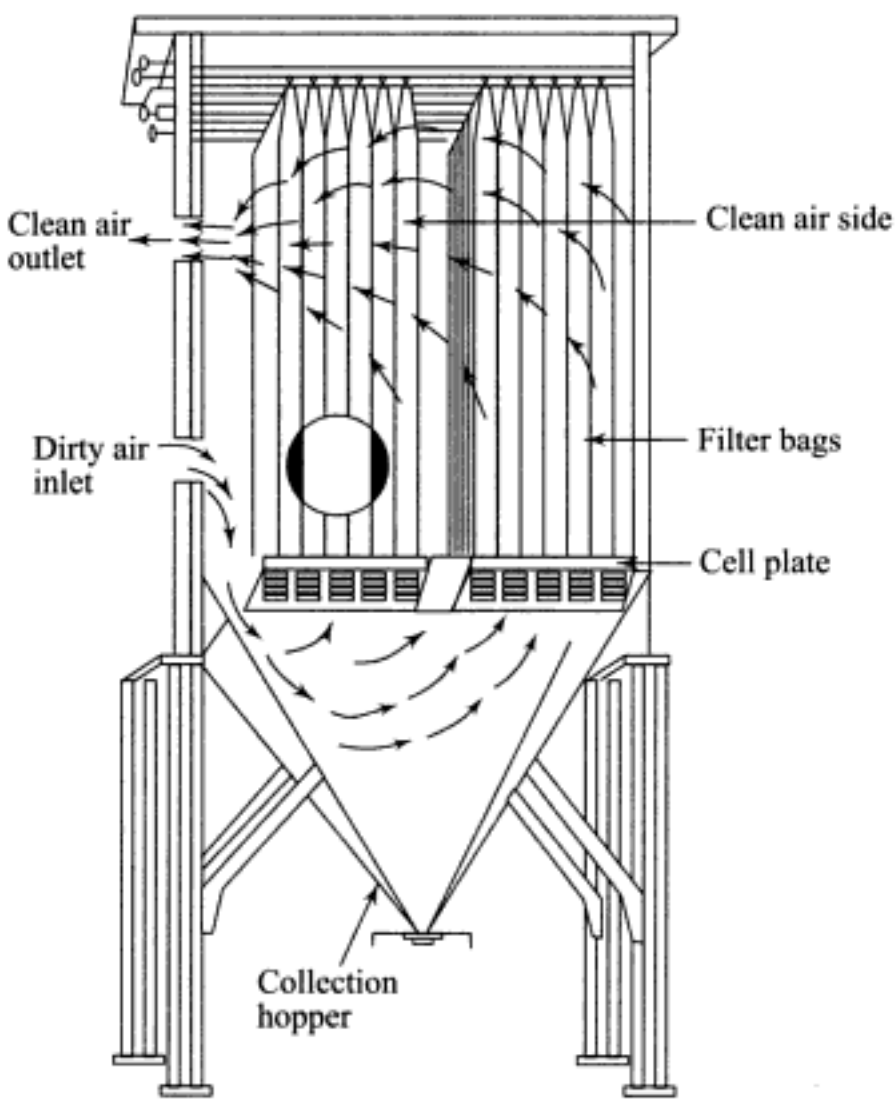

Fig. 6.63 *Typical baghouse with mechanical shakers*

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Fabric filters come in three basic types: reverse-air, shake-deflate, and pulse-jet units (Fig. 6.64). Reverse air fabric filters are characterized by low A/C values and by the flow in which dirty gas flows from inside the bags to outside. Clean air is then forced through the filter elements in a reverse direction, from outside into the bag, which collapses the bag and fractures the dust cake. When the bag is brought back on line with dirty gas flowing in to out, it re-inflates and removes the broken dust cake into the hopper below (Fig. 6.64(a)). This type of baghouse is found to be economical for large power plants.

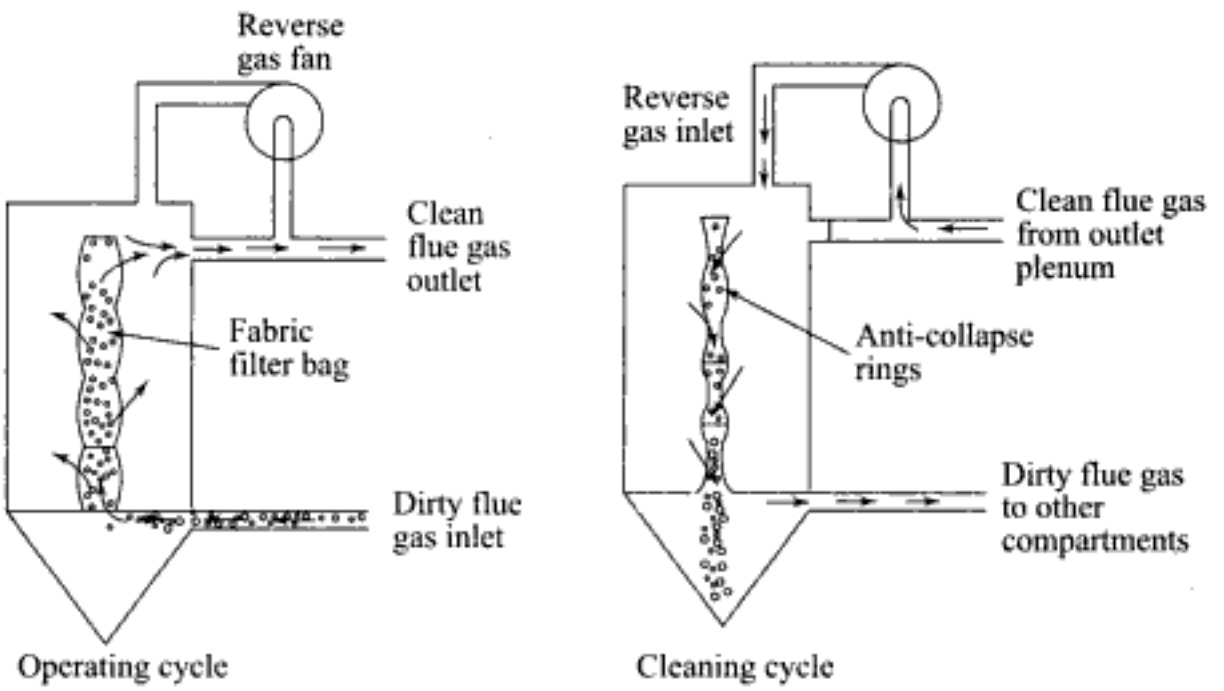

Fig. 6.64 (a) *Operating Schematic Reverse Air Fabric Filter*

A shake–deflate unit is also of low A/C value and collects dust on the inside of the bags as in the reverse-air design. To clean the bags, the top end is shaken by a drive linkage.

In pulse-jet units, gas flow is from outside the bags inward. Usually A/C ratios are much higher than the previous two types, Cleaning is accomplished with a high-pressure burst of air into the open end of the bag at the top. Because of compactness, these are preferred for industrial boilers (Fig. 6.64(b)).

Pressure drop and bag life are the criteria for selecting the bag filter material, which can fail as a result of high temperature burning, caking, erosion, chemical attack and aging. Fabrics considered include wool, cotton, nylon, glass fibre, polyester and aromatic polyamides. Wool and cotton can be used only up to about 80 or 95 °C. Over 90% of the bags in the USA utility service are the woven fibreglass variety. Bag performance is rated in terms of permeability, cleanability and durability. A baghouse is normally placed after the air preheater.

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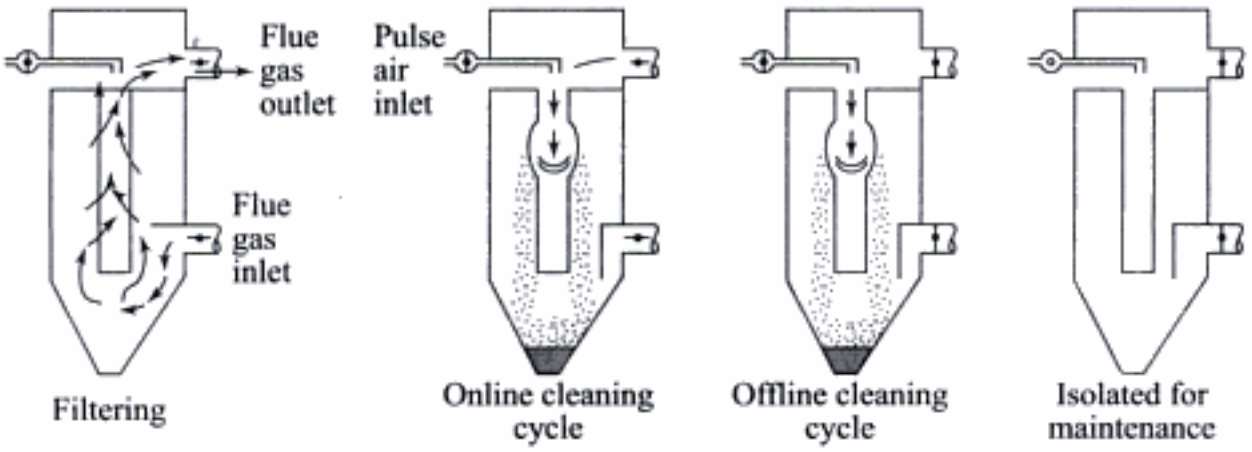

Fig. 6.64 (b) *Operating schematic pulse jet fabric filter*

6.12 ASH HANDLING SYSTEM

Boilers burning pulverized coal (PC) have dry bottom furnaces. The large ash particles are collected under the furnace in a water- filled ash hopper. Fly ash is collected in dust collectors with either an electrostatic precipitator or a baghouse. A PC boiler generates approximately 80% fly ash and 20% bottom ash. Ash must be collected and transported from various points of the plants as shown in Fig. 6.65. Pyrites, which are the rejects from the pulverizers, are disposed with the bottom ash system. Three major factors should be considered for ash disposal systems.

1. Plant site
2. Fuel source
3. Environmental regulation

Needs for water and land are important considerations for many ash handling systems. Ash quantities to be disposed depend on the kind of fuel source. Ash storage and disposal sites are guided by environmental regulations.

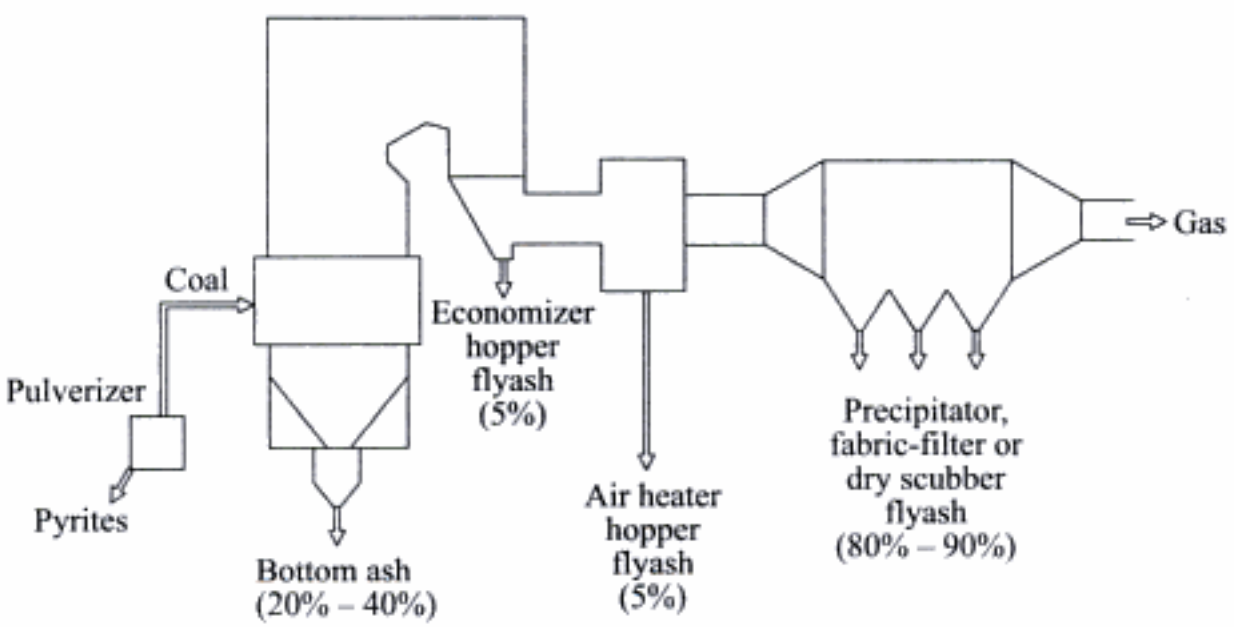

Fig. 6.65 *Ash collection and transportation*

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The sluice conveyor system (Fig. 6.66a) is the most widely used for bottom ash handling, while the hydraulic vacuum conveyor (Fig. 6.66b) is the most frequently used for fly ash systems.

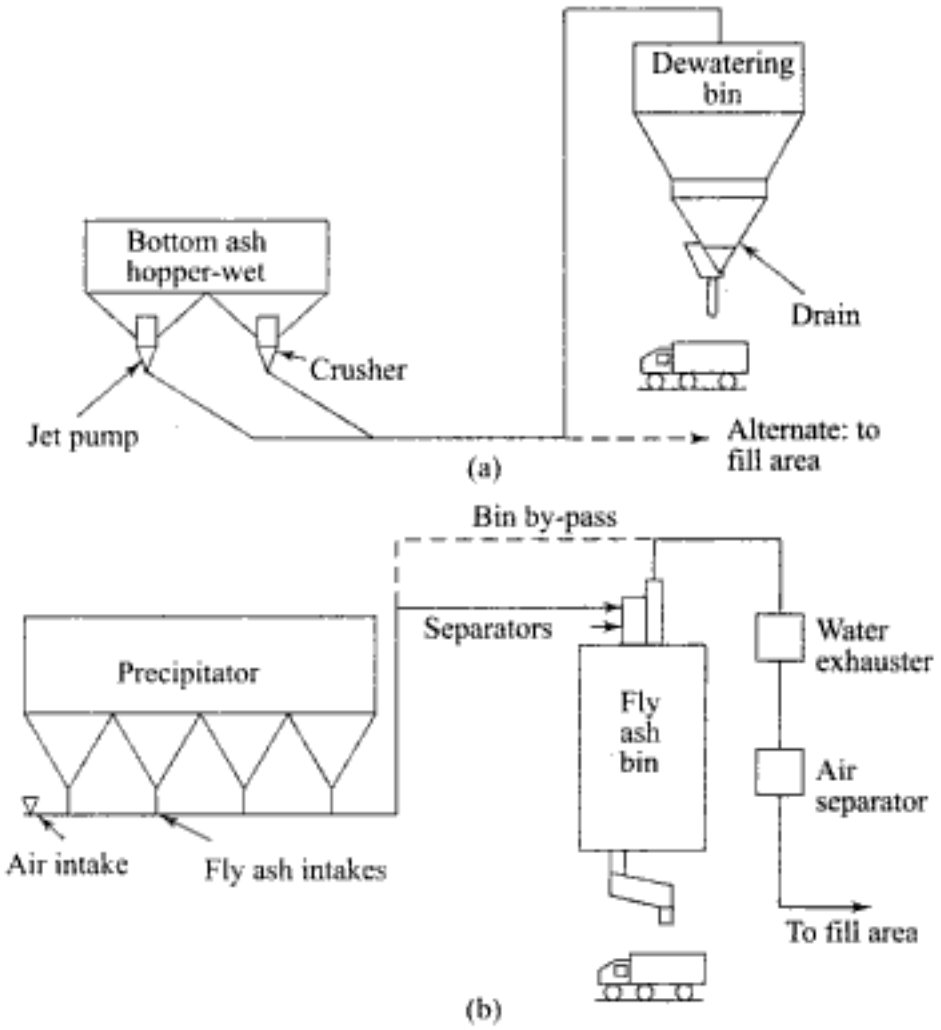

Fig. 6.66 *(a) Bottom ash sluice conveyor, (b) Fly ash hydraulic vacuum conveyor*

Bottom ash and slag may be used as filling material for road construction. Fly ash can partly replace cement for making concrete. Bricks can be made with fly ash. These are durable and strong.

6.13 FEEDWATER TREATMENT

Boiler make-up water to the extent of 1.5–2 per cent of the total flow rate is required to replenish the losses of water through leakage from fittings and bearings, boiler blowdown, escape with non-condensable gases in the deaerator, turbine glands, and other causes. This make-up water needs to be treated prior to feeding it to the boiler for

1. Prevention of hard scale formation on the heating surfaces
2. Elimination of corrosion,
3. Control of carry-over to eliminate deposition on superheater tubes, and
4. Prevention of silica deposition and corrosion damage to turbine blades.

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Raw water is, therefore, first pre-treated and then demineralized. For once-through boilers and boiling water nuclear reactors, which require high water-purity, a condensate polishing system is used to further polish the water. Raw water contains a variety of impurities, such as (a) suspended solids and turbidity, (b) organics, (c) hardness (salts of calcium and magnesium), (d) alkalinity (bicarbonates, carbonates, hydrates), (e) other dissolved ions (sodium, sulphate, chloride, etc.), (f) silica, and (g) dissolved gas (O_2, CO_2). The extent of pre-treatment depends on the source of raw water.

6.13.1 External Treatment

The first step of pre-treatment of boiler feedwater is clarification, in which the water is chlorinated to prevent biofouling of the equipment. The suspended solids and turbidity are coagulated by adding special chemicals (like aluminium sulphate, $Al_2(SO_4)_3$) and agitated. The coagulated matter settles at the bottom of the clarifier and is removed.

If the turbidity of clarified effluent is high, positive filtration is needed. Both gravity filters and pressure-type filters are used, but the latter is preferred. A granular medium like sand is commonly used for filtration. The pressure difference across the filtering medium is an indication of solid accumulation. When it reaches a given limit, the solids are removed from the bed by backwashing. Further filtration by activated carbon can absorb organics and remove residual chlorine from the chlorination process.

The dissolved salts of calcium and magnesium give to water a quality called *hardness*. Hardness is characterized by the formation of insoluble precipitates or curds with soaps, and is usually measured with a standard soap test. All natural waters are hard and contain scale-forming impurities which are mainly the salts of calcium and magnesium in the form of carbonates, bicarbonates, chlorides and sulphates. The hardness is expressed in ppm of dissolved salts. Softening of water, i.e. removal of hardness from water, can be done by lime-soda process, phosphate process, zeolite process and demineralization.

6.13.2 Lime–Soda Process

In lime–soda softening, calcium and magnesium salts are removed using lime (calcium hydroxide) and soda ash (sodium carbonate). When this process is carried out at normal raw-water temperature, it is called a "cold process" softening; and when carried out at or near the boiling point, it is referred to as a "hot process" softening. Since heating greatly accelerates the necessary reactions, the hot process is preferred for boiler water treatment, where most of the energy used in heating the water may be retained in the cycle. The representative reactions are given below:

$$Ca(HCO_3)_2 + Ca(OH)_2 = 2CaCO_3 + 2H_2O$$

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$$MgCl_2 + Ca(OH)_2 = Mg(OH)_2 + CaCl_2$$

$$MgSO_4 + Ca(OH)_2 = Mg(OH)_2 + CaSO_4$$

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4$$

$$CaCl_2 + Na_2CO_3 = CaCO_3 + 2NaCl$$

The products, calcium carbonate and magnesium hydroxide are insoluble in water and settle to the bottom of the vessel. The softened effluent is then passed through sand or charcoal filters before usage.

6.13.3 Hot Phosphate Softening

In the hot phosphate softening process, calcium and magnesium hardness is removed using phosphate and caustic soda, Tricalcium phosphate ($Ca_3(PO_4)_2$) and magnesium hydroxide are precipitated. The process is carried out at a temperature of 100 °C or above. Since the hot phosphate process requires more expensive chemicals than the lime–soda process, it is used where the initial water hardness is 60 ppm or less. Where hardness is greater than this, a lime-soda process may be used first, followed by a phosphate clean-up. The representative reactions are given below:

$$3Ca(HCO_3)_2 + 6NaOH = 2CaCO_3 + 2Na_2CO_3 + 6H_2O$$

$$CaCO_3 + 2Na_3PO_4 = Ca_3(PO_4)_2 + 2Na_2CO_3$$

$$Mg(HCO_3)_2 + 4NaOH = Mg(OH)_2 + 2Na_2CO_3 + 2H_2O$$

$$3CaSO_4 + 2Na_3PO_4 = Ca_3(PO_4)_2 + 3Na_2SO_4$$

6.13.4 Sodium Zeolite Softening

Water can be softened by passing it through a bed of sodium zeolite, which may be natural compounds of sodium aluminium silicate, with the cations of calcium and magnesium removed in the process.

$$CaCO_3 + Na_2Z = CaZ + Na_2CO_3$$

$$MgSO_4 + Na_2Z = MgZ + Na_2SO_4$$

$$Ca(HCO_3)_2 + Na_2Z = CaZ + Na_2(HCO_3)_2$$

The softening capacity of the bed gets exhausted in course of time, and the bed can be regenerated by flushing it with brine (NaCl),

$$CaZ + 2NaCl = Na_2Z + CaCl_2$$

$$MgZ + 2NaCl = Na_2Z + MgCl_2$$

Zeolite softening is not ideal for the following reasons: (i) Water of high or low pH have a deleterious effect on zeolites, (ii) high temperatures also have a bad effect, (iii) turbid waters coat the zeolite material, reducing its efficiency, (iv) there is no reduction in alkalinity or total solids, (v) there can be silica gain

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in water from the zeolite, (vi) with low content of calcium, the water can be corrosive. Sodium zeolite softening in conjunction with the use of evaporators may be more effective.

6.13.5 Hydrogen Zeolite Softening

When water containing calcium, magnesium, and sodium ions is passed through a hydrogen zeolite, these ions are exchanged for hydrogen and the bicarbonate, sulphate, chloride and nitrate radicals are converted to their respective acids. Typical reactions are:

$$\begin{matrix} (Ca) & & (Ca) \\ (Mg)\ (HCO_3)_2 + H_2Z & = & (Mg)\ Z + 2H_2CO_3 \\ (Na_2) & & (Na_2) \end{matrix}$$

$$\begin{matrix} (Ca) & & (Ca) \\ (Mg)\ SO_4 + H_2Z & = & (Mg)\ Z + H_2SO_4 \\ (Na_2) & & (Na_2) \end{matrix}$$

When the hydrogen zeolite becomes exhausted, it is backwashed and regenerated with acid. After being rinsed, it is ready for use again. Sulphuric acid is generally used for regeneration because of its relatively low cost.

$$MgZ + H_2SO_4 = H_2Z + MgSO_4$$

$$CaZ + H_2SO_4 = H_2Z + CaSO_4$$

6.13.6 Anion Exchangers

Anion exchangers can remove the anions like chlorides, sulphates and nitrates (acid forms) present in hydrogen zeolite effluent by resinous materials which adsorb them. Typical reactions are:

$$2HCl + RCO_3 = RCl_2 + H_2CO_3$$

$$H_2SO_4 + RCO_3 = RSO_4 + H_2CO_3$$

$$2HNO_3 + RCO_3 = R(NO_3)_2 + H_2CO_3$$

Carbonic acid is removed by aeration. When the acidic water is sprayed in a shower to expose large surface area, the carbon dioxide gas is released.

$$H_2CO_3 \rightarrow CO_2 + H_2O$$

When the anion exchanger is exhausted, it is regenerated by backwashing with soda ash

$$RCl_2 + Na_2CO_3 = RCO_3 + 2NaCl$$

$$RSO_4 + Na_2CO_3 = RCO_3 + Na_2SO_4$$

6.13.7 Demineralizing Plant

The process of removing dissolved solids in water by *ion exchange* is called *demineralization*. Two types of resins, cation and anion, are used. The cation

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resin is the hydrogen zeolite where the hydrogen ion is exchanged for the cations calcium, magnesium and sodium, and the anion resin adsorbs the anions chlorides, nitrates and sulphates, as discussed above. Both ion-exchange processes are reversible, and the resins are restored to their original form by regeneration,

A typical dimineralizing plant consisting of a cation exchanger, an anion exchanger, a degasifier and a silica adsorber in series is shown in Fig. 6.67. In the degasifier, carbon dioxide gas is removed by aeration. Silica in water is very detrimental at high pressure. It vaporizes at high pressure and flows with steam, condenses on turbine blades in the form of hard glassy scales which are difficult to remove. Magnesium hydroxide is often used to adsorb silica from water.

The membrane treatment for removing the total dissolved solids from make-up water is also an energy efficient process and is gradually gaining more acceptance. It uses the principle of either *reverse osmosis* or *electrodialysis*. The driving force for reverse osmosis is the application of counter pressure to normal osmotic pressure, driving water molecules through the membranes in preference to dissolved salts. The basis of electrodialysis is the reverse of that of reverse osmosis in that it moves dissolved salts away from the water accruing a practical advantage because the quantity of salts is far less than the water volume. The membranes used in either process are expensive, and coagulation, settling and filtration are used first to protect them.

6.13.8 Condensate Polishing

A high quality make-up water can be produced for the plant by using demineralizing systems as discussed above. However, this treated water while flowing through the cycle can pick up impurities due to condenser leakage from the circulating water through the tubes as well as metallic ions, such as iron and copper, from pipelines. Condensate polishing is accomplished by passing the condensate through large demineralizing vessels, called *mixed bed units*, which contain both cation and anion resins. The resins not only remove dissolved salts in the above manner, but also act as filters for impurities or suspended solids. Power plants using once-through boilers and nuclear reactors generally require high quality water and use condensate polishing systems.

6.14 DEAERATION

Deaeration (degasification) is one of the most important steps in boiler water treatment. It depends on the decrease in solubility of dissolved gases, notably O_2 and CO_2 as the water temperature is increased. The deaerator is a direct contact feedwater heater which has been described in Section 2.14.

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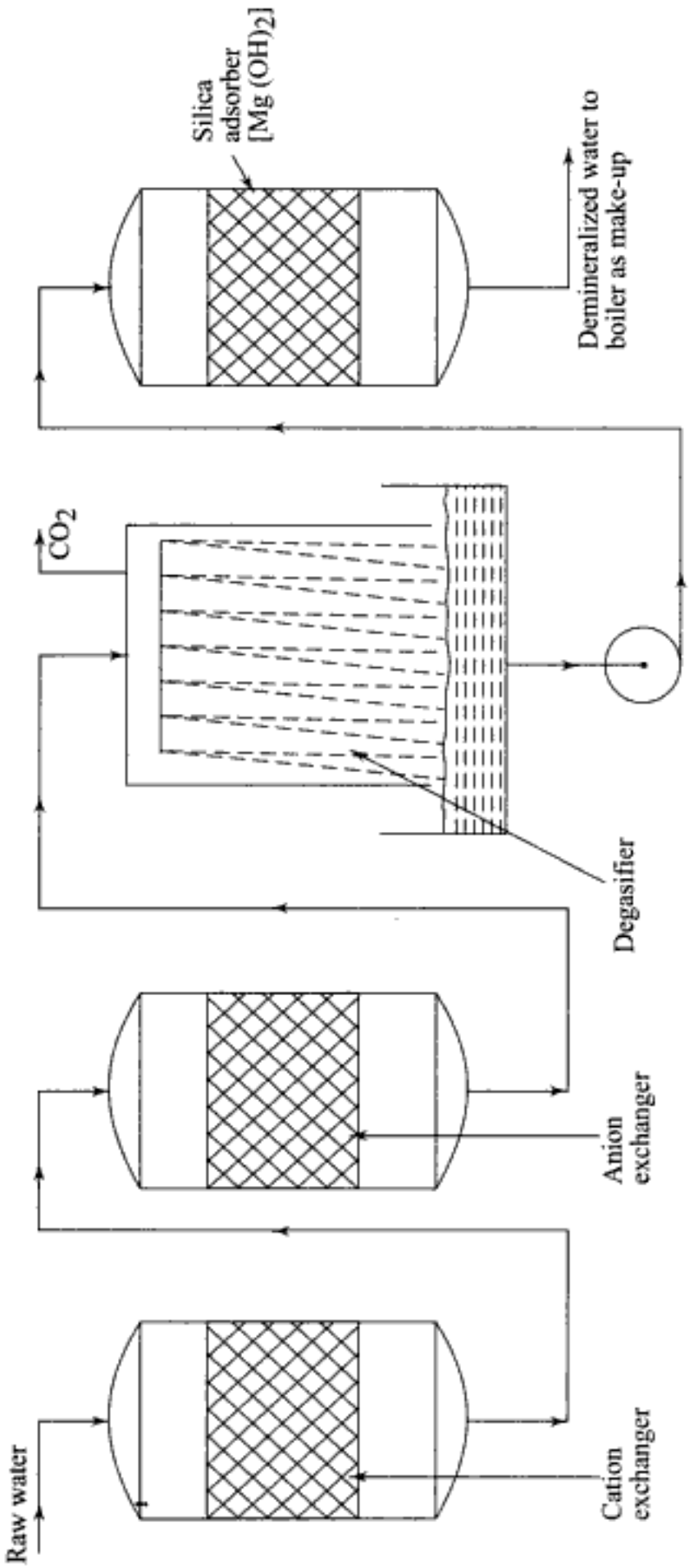

Fig. 6.67 *A typical demineralizing plant*

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6.15 EVAPORATION

Evaporation of pre-treated feedwater (w) by motive steam (m) bled from the turbine is often used where the boiler make-up water requirement is not large. Both evaporated water (v) and condensed steam (c) exiting the evaporator are fed back as make-up into the plant, normally the deaerator (Fig. 6.68). Prior to the evaporator, the boiler feedwater is softened by passing it through a sodium zeolite bed or by lime-soda process.

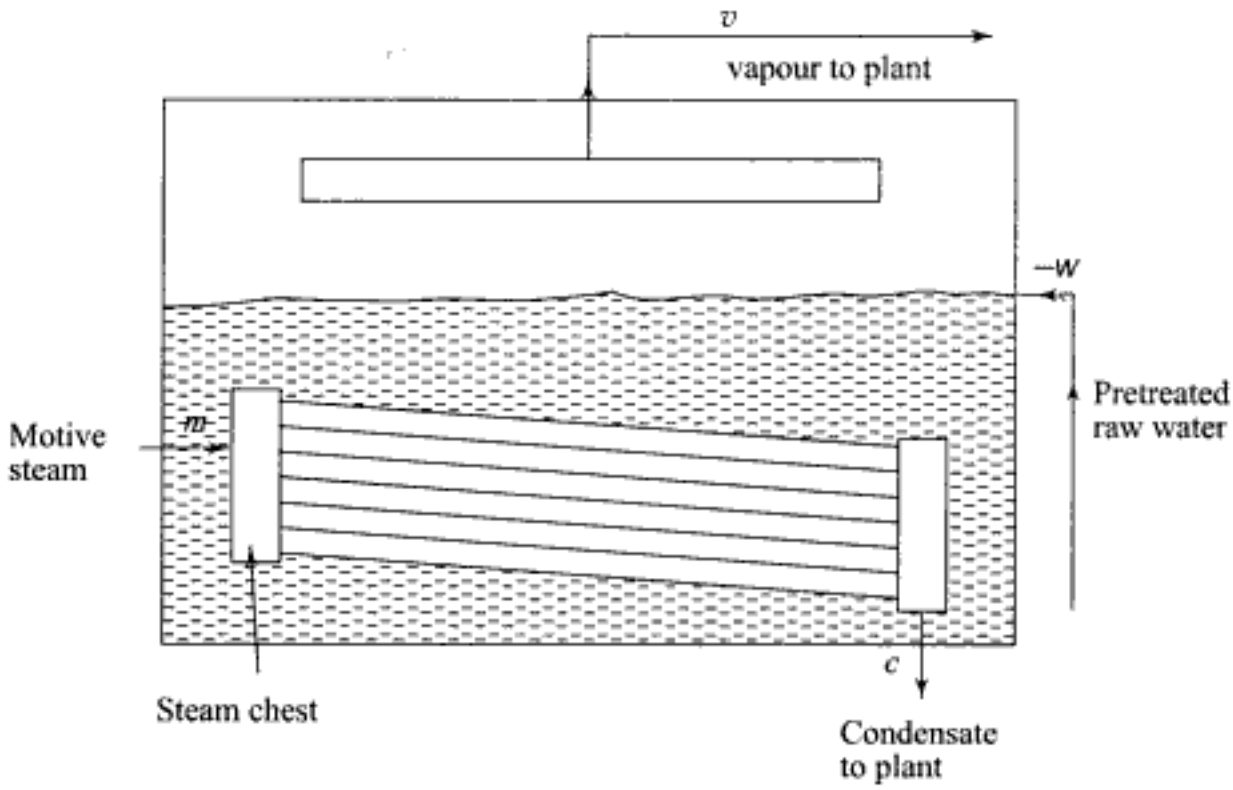

Fig. 6.68 *Schematic of a single-effect submerged evaporator*

6.16 INTERNAL TREATMENT

The dissociation and recombination of water go on till equilibrium is reached,

$$HOH = H^+ + (OH)^-$$

The presence of excess hydrogen ions makes the water acidic and hence corrosive. The pH is the logarithm of the reciprocal of the hydrogen ion concentration. If the hydrogen ion concentration in water is 10^{-7}, the pH value of water would be $\log\left(\frac{1}{10^{-7}}\right)$ or 7. The pH of the water varies with temperature (Fig. 6.69). At 15 °C, the pH of neutral water is 7. If pH > 7, water is alkaline and if pH < 7, it is acidic. Trisodium phosphate, Na_3PO_4, is injected to water in suitable doses to increase alkalinity. Monosodium phosphate, NaH_2PO_4, is used to decrease alkalinity. A pH meter can be used to measure the pH of water.

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The essential factors required for corrosion are hydrogen ions (low pH) and dissolved oxygen. Scale formation and corrosion are reciprocal phenomena: conditions for one preclude the other, and the pH value of water determines it. A pH 10.5 is usually maintained for boiler water in order to minimize corrosion. It will, however, cause scale formation. Two principal steps in scale prevention are (i) periodic or continuous blowdown, and (ii) external or internal treatment to eliminate the scale producers by chemical means. The external treatments have already been discussed.

Internal treatment to prevent scale deposition is usually carried out with some form of sodium phosphate (trisodium, disodium or monosodium). Once in the boiler, all these phosphates react with the calcium salts to form tricalcium phosphate, which is rather a flocculent precipitate. The magnesium salts are also

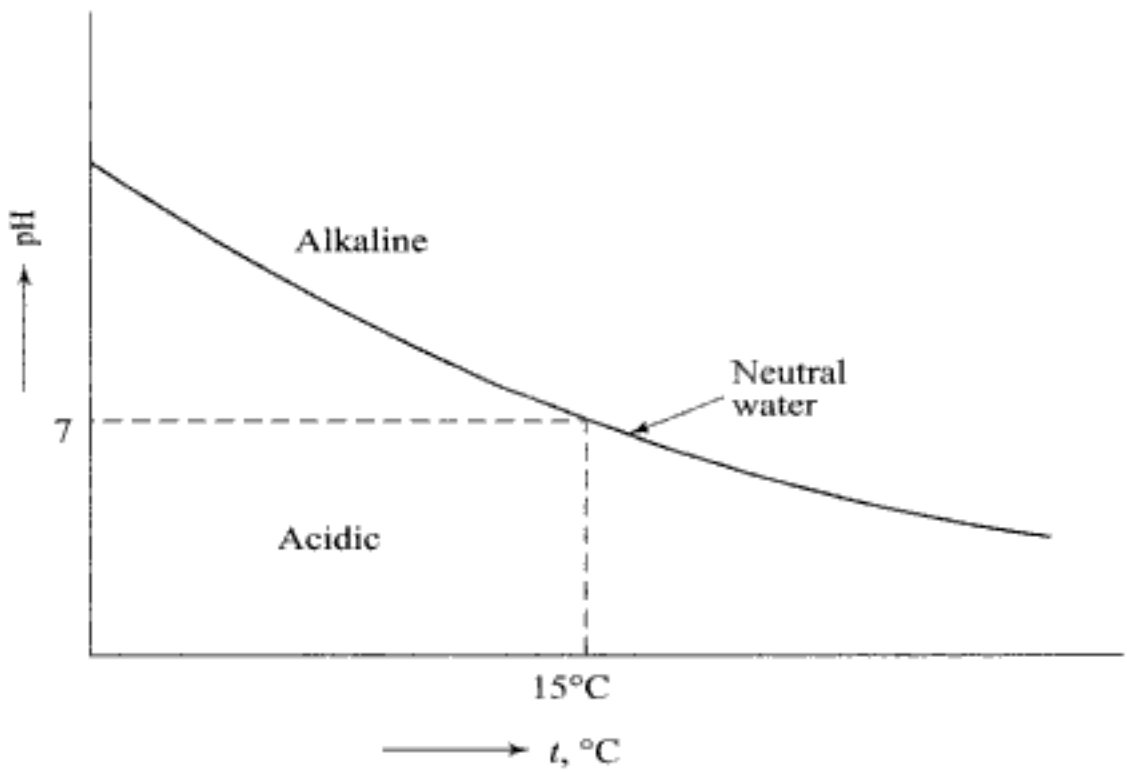

Fig. 6.69 *pH of water varying with temperature*

readily precipitated due to alkalinity of boiler water. Tannins or starches are sometimes used to prevent the precipitates from adhering to the boiler surfaces.

6.17 BOILER BLOWDOWN

Concentrated solid precipitates at drum bottom (by internal treatment) are removed through blowdown, which may be either intermittent or continuous (see section 6.3.5), the amount of blowdown being governed by the allowable solids concentration and the purity of feedwater. It is usually expressed as a percentage. Thus,

$$\% \text{ blowdown} = \frac{\text{quantity of water blown down}}{\text{quantity of feed water}} \times 100$$

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For instance, 3% blowdown means that 3 per cent of the total amount of feedwater supplied is removed as liquid by blowdown of the drum. The continuous blowdown has the advantage of closer control of concentration and the recovery of part of the energy of the blowdown.

Since water purity is controlled by blowdown in drum type boilers, condensate polishing is not required in such boilers.

6.18 STEAM PURITY

The purity of steam is determined by the quality of the steam delivered by the boiler and the boiler salt concentration. Solids in the steam tend to cause scale deposits on superheater tubes or on turbine blades. Scrubbers and steam washers in the drum (see Section 6.3.5) help achieving the desired steam purity. By measuring the electrical conductivity of condensed steam samples, the steam purity can be determined. Pure water has a high electrical resistance. Presence of solids in water decreases the resistance. So the electrical conductivity is proportional to the solids in the water.

Foaming and priming often cause excessive carry-over of water with steam leaving the drum. *Foaming* is the condition that results from the formation of bubbles on the drum water surface by the saponification agents like oil, high alkalinity and so on. *Priming* is a violent, spasmodic action which results in the throwing of slugs of water with steam. It is caused by too high a water level, uneven water circulation and rapid changes in steaming rate. For the desired steam purity, both foaming and priming have to be adequately controlled.

Example 6.1 A furnace wall riser, 18 m long, 76.2 mm OD and 6.1 mm thick receives saturated water at 80 bar and 1.5 m/s velocity. Assuming a circulation ratio of 12.5 and a slip ratio of 1.2, determine (a) the pressure head developed, (b) the void fraction at riser exit, and (c) the heat transfer rate per unit projected area of the riser tube.

Solution The pressure head developed due to natural circulation (Fig. E6.1),

$$\Delta p = Hg(\rho_f - \rho_m)$$

where ρ_m = average density of liquid–vapour mixture in the riser tube

$$x_{top} = \text{TDF} = \frac{1}{\text{CR}} = \frac{1}{12.5} = 0.08$$

The quality of steam at the top of the riser is 8%.

At 80 bar, $v_f = 0.001384\ \text{m}^3/\text{kg}$

$v_{fg} = 0.02214\ \text{m}^3/\text{kg}$

$\rho_f = 1/v_f = 722.543\ \text{kg/m}^3$

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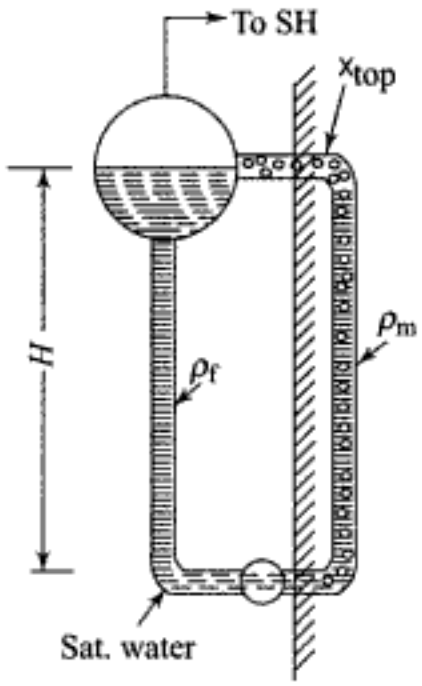

Fig. E6.1

$$v_{top} = v_f + x_{top}\, v_{fg} = 0.00138 + 0.08 \times 0.02214$$
$$= 0.0031552\ \text{m}^3/\text{kg}$$

$$\rho_{top} = \frac{1}{v_{top}} = 316.937\ \text{kg/m}^3$$

$$\rho_m = \frac{\rho_{top} + \rho_f}{2} = \frac{316.937 + 722.543}{2} = 519.74\ \text{kg/m}^3$$

(a) $$\Delta p = Hg(\rho_f - \rho_m)$$

$$= 18\ \text{m} \times 9.806\ \frac{m}{s^2}\ (722.54 - 519.74)\ \text{kg/m}^3$$

$$= 35.8\ \text{kPa} \qquad \textit{Ans.}$$

(b) $$\psi = \frac{v_f}{v_g} S = \frac{0.001384}{0.02352} \times 1.2 = 0.0706$$

$$\alpha = \frac{1}{1 + \frac{1-x}{x}\psi} = \frac{1}{1 + \frac{0.92}{0.08} \times 0.0706} = 0.5519 \qquad \textit{Ans.}$$

(c) $$h_{fg} = 1441.3\ \text{kJ/kg} \quad \text{and} \quad d_i = 76.2 - 12.2 = 64.0\ \text{mm}$$

Mass flow rate of saturated water entering the riser,

$$\omega = \rho_f A_i V = 722.543 \times \frac{\pi}{4} \times (0.064)^2 \times 1.5$$
$$= 3.4866\ \text{kg/s}$$

Rate of steam formation in the riser tube,

$$\omega_s = 0.08 \times 3.4866 = 0.2789\ \text{kg/s}$$

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Heat transfer rate per unit projected area

$$= \frac{\omega_s h_{fg}}{D_o H} = \frac{0.2789 \times 1441.3}{0.0762 \times 18} = 293.1 \text{ kW/m}^2 \qquad \textit{Ans.}$$

Example 6.2 A spray type desuperheater is supplied with water at 60 °C. It is connected in a steam line carrying 200 t/h of steam at 35 bar. Calculate the amount of water that must be sprayed per hour to maintain steam at 400 °C when the boiler load causes steam to leave at 450 °C.

Solution Let ω be the rate of water spray required to bring down the steam temperature from 450 °C to 400 °C at 35 bar. We would neglect any pressure drop due to mixing and heat loss to the surroundings. By energy balance of the control volume containing the disuperheater (Fig. E6.2),

$$\omega_s h_1 + \omega h_2 = (\omega_s + \omega) h_3$$

$$\omega_s (h_1 - h_3) = \omega (h_3 - h_2)$$

or

$$\omega = \frac{\omega_s (h_1 - h_3)}{h_3 - h_2}$$

Now, $h_1 = 3337.2$ kJ/kg, $h_2 = 252$ kJ/kg and $h_3 = 3222.3$ kJ/kg

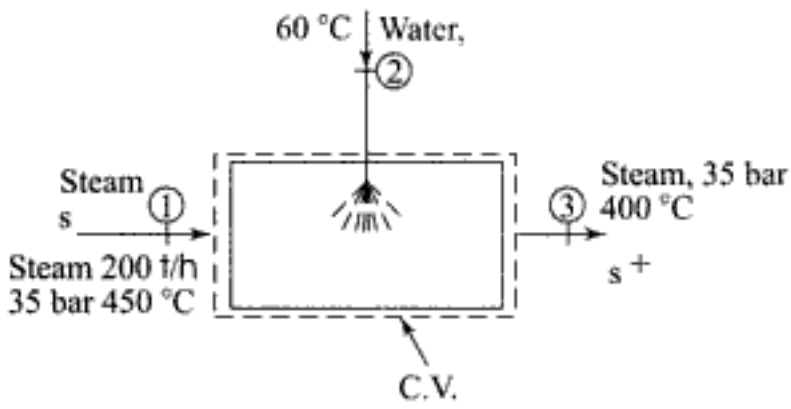

Fig. E6.2

$$\omega = \frac{200(3337.2 - 3222.3)}{3222.3 - 252} = \frac{200 \times 114.9}{2970.3} = 7.733 \text{ t/h}$$

$$= 2.148 \text{ kg/s} \qquad \textit{Ans.}$$

Example 6.3 A 15 m high downcomer–riser circuit operates at 160 bar. The riser receives uniform heat flux and saturated water. The exit quality is 50 per cent. Calculate the pressure head developed due to natural circulation. Take the slip factor as 1.2.

Solution Using Eq. (6.17),

$$\psi = \frac{v_f}{v_g} S = \frac{0.001711}{0.009306} \times 1.2 = 0.2206$$

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Using Eq. (6.16),

$$\alpha_e = \frac{1}{1+\frac{1-x_e}{x_e}\psi} = \frac{1}{1+\frac{0.5}{0.5}\times 0.2206} = 0.8193$$

Using Eq. (6.21) for uniform axial heating,

$$\rho_m = \rho_f - \frac{\rho_f - \rho_g}{1-\psi}\left[1-\left\{\frac{1}{\alpha_e(1-\psi)}-1\right\}\ln\frac{1}{1-\alpha_e(1-\psi)}\right]$$

$$= 584.45 - \frac{584.45-107.45}{1-0.2206}\left[1-\left\{\frac{1}{0.8193(1-0.2206)}-1\right\}\right.$$

$$\left.\ln\frac{1}{1-0.8193(1-0.2206)}\right]$$

$$= 584.45 - 612\,\{1-0.566\times 1.0178\}$$
$$= 325\ \text{kg/m}^3$$

Pressure head developed due to natural circulation,

$$\Delta p = g(\rho_f - \rho_m)H = 9.806\,(584.45-325)15$$
$$= 38162\ \text{N/m}^2 = 38.162\ \text{kPa} \qquad \textit{Ans.}$$

Example 6.4 A power plant producing 120 MW of electricity has steam condition at boiler outlet as 100 bar, 500 °C and the condenser pressure is 0.1 bar. The boiler efficiency is 90%. The boiler consumes coal of calorific value 25.7 MJ/kg. The feedwater temperature at boiler inlet is 160 °C. The steam generator has risers in the furnace wall 40 m high and unheated downcomers. The quality at the top of the riser is 8% and a minimum exit velocity of mixture leaving the riser and entering the drum is required to be 2 m/s. The risers have 60 mm o.d. and 3 mm wall thickness. Neglecting any pressure drop and heat loss, as well as the pump work, estimate (a) the steam generation rate, (b) the fuel burning rate, (c) the evaporation factor, (d) the pressure head available for natural circulation, (e) the circulation ratio, (f) the number of risers required, and (g) the heat absorption rate per unit projected area of the riser.

Solution With reference to Fig. E6.4,

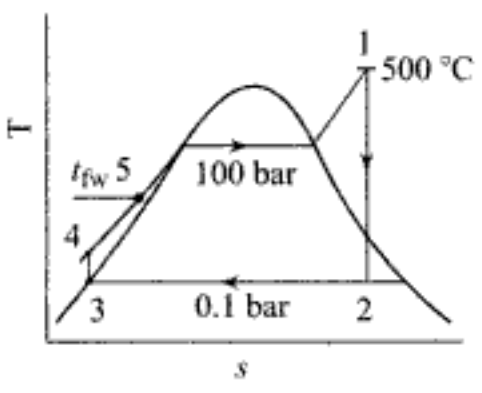

Fig. E6.4

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$$h_1 = 3374.8 \text{ kJ/kg}$$
$$s_1 = 6.6011 \text{ kJ/kg K} = 0.6479 + x_2 (7.5055)$$
$$x_2 = 0.7932$$
$$h_2 = 191.46 + 0.7932 \times 2393.29 = 2089.82 \text{ kJ/kg}$$
$$h_3 = 191.46 \text{ kJ/kg} = h_4, \qquad h_5 = 675.5 \text{ kJ/kg}$$
$$\omega_s(h_1 - h_2) = 120 \times 10^3 \text{ kW}$$
$$\omega_s = \frac{120 \times 10^3}{1285} = 93.385 \text{ kg/s} \qquad \textit{Ans.}(a)$$
$$\eta_{st.gen} = 0.90 = \frac{93.385 \times (3374.8 - 675.5)}{\omega_f \times 25700}$$
$$\omega_f = 10.898 \text{ kg/s} \qquad \textit{Ans.}(b)$$
$$\text{Evaporation factor} = \omega_s/\omega_f = 93.385/10.898 = 8.57 \qquad \textit{Ans.}(c)$$

Pressure head available for natural circulation,

$$\Delta p = gH(\rho_f - \rho_m)$$
$$v_{top} = 0.0014523 + 0.08 \times 0.0165884 = 0.002779 \text{ m}^3/\text{kg}$$
$$\rho_{top} = 359.8 \text{ kg/m}^3, \qquad \rho_f = 1/v_f = 688 \text{ kg/m}^3$$
$$\rho_m = \frac{\rho_f + \rho_{top}}{2} = 523.9 \text{ kg/m}^3$$
$$\Delta p = 40 \times 9.81\,(688 - 523.9) = 64.327 \text{ kPa}$$
$$= 0.6433 \text{ bar} \qquad \textit{Ans.}(d)$$
$$\text{Circulation ratio (CR)} = 1/x_{top} = 1/0.08 = 12.5 \qquad \textit{Ans.}(e)$$
$$d_i = 60 - 6 = 54 \text{ mm}$$

Rate of steam formation in a riser,

$$\omega = \left[\left(\frac{\pi}{4} d_i^2\right)\rho_{top} V_i\right] x_{top} = \frac{\pi}{4}(54 \times 10^{-3})^2 \times 2 \times 359.8 \times 0.08$$
$$= 0.131 \text{ kg/s}$$
$$\text{Number of risers} = 93.385/0.131 = 712 \qquad \textit{Ans.}(f)$$

Heat absorption rate per unit projected area of the riser

$$= \frac{\omega\, h_{fg}}{D_o\, H} = \frac{0.131 \times 1319.8}{0.06 \times 40} = 72.47 \text{ kW/m}^2 \qquad \textit{Ans.}(g)$$

Example 6.5 Steam leaves the drum of a boiler at the rate of 64 kg/s at 60 bar, with 2% moisture. The feedwater from the economiser enters the drum at the rate of 62 kg/s and has 3 ppm. The makeup water at the rate of 2 kg/s is fed into the drum and it has 50 ppm. Effective steam washing reduces the solids in the moisture, leaving the drum with steam, to 5 ppm. If the solid concentration in the drum water is to be maintained at 1000 ppm, find (a) the blowdown required in kg/s, (b) heat loss in blowdown as a percentage of total heat released in the furnace, if blowdown heat recovery is not used, when the

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fuel burning rate is 7 kg/s and its heating value 23 MJ/kg, and (c) the deposition of scale in superheater tube in kg/day. Take the room temperature as 30 °C.

Solution Making an impurity balance of the drum at steady state (Fig. E6.5).

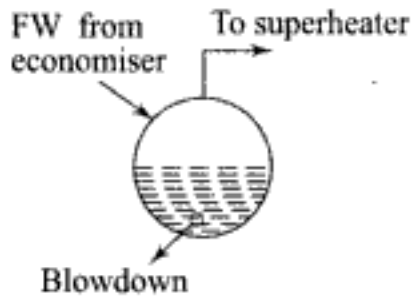

Fig. E6.5

(Impurities entering with feedwater + impurities entering with make-up = impurities leaving with blowdown + impurities leaving with moisture carry-over with steam)

$$62 \times 3 \times 10^{-6} + 2 \times 50 \times 10^{-6} = \text{BD} \times 1000 \times 10^{-6} + 0.02 \times 64 \times 5 \times 10^{-6}$$

The rate of blowdown is

$$\text{BD} = \frac{62 \times 3 + 2 \times 50 - 0.02 \times 64 \times 5}{1000} = 0.2796 \text{ kg/s} \quad \textit{Ans.}(a)$$

The energy loss in blowdown as a percentage of total heat released in furnace

$$= \frac{\text{BD}(h_{f\,60\,\text{bar}} - h_{\text{ambient}})}{\omega_f \times \text{HHV}} \times 100$$

$$= \frac{0.2796(1213.35 - 125.79)}{7 \times 23000} \times 100 = 0.19\,\% \quad \textit{Ans.}(b)$$

Scale deposition in superheater tubes

$$= 0.02 \times 64 \times 5 \times 10^{-6} \times 3600 \times 24$$

$$= 0.553 \text{ kg/day} \quad \textit{Ans.}(c)$$

Example 6.6 Feedwater from the high pressure heater enters the inlet header of the economiser at the rate of 600 kg/s and at 140 bar, 170 °C. It is heated by flue gases till it becomes saturated liquid at that pressure and leaves the outlet header to flow into the drum. Flue gases flow over the economiser coils at the rate of 1250 kg/s and leave at 450 °C. To restrict the erosion rate by fly ash, the flue gas velocity should not exceed 12 m/s, while the optimum water velocity leaving the coils is 1.2 m/s. The tubes are of 70 mm od and 60 mm id. The overall heat transfer coefficient may be taken as 70 W/m^{2}K. Determine the number of coils needed in the economiser and the length of one coil. Take c_p of flue gases as 1.12 kJ/kg K. If the vertical pitch of the coil is 80 mm, and the clearance on the two sides of the duct of width 4.8 m is 5 mm, find the vertical height of the economiser coils.

Solution At 140 bar, $h_f = 1571.1$ kJ/kg,

$$t_{sat} = 336.75\,°\text{C}, \qquad v_f = 0.001611 \text{ m}^3/\text{kg}$$

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At 170 °C, $h_f = 719.21$ kJ/kg

$v_f = 0.001114\ m^3/kg$

The rate of heat transfer in the economiser (Fig. E6.6)

$$Q = \omega_s(h_f - h_{170\,°C}) = \omega_g C_{pg}(t_{g1} - t_{g2}) = U_o A_o \Delta t_{1.m.}$$

or $600(1571.1 - 719.21) = 1250 \times 1.12\,(t_{g1} - 450)$

or $511134 = 1400(t_{g1} - 450)$

$$t_{g1} = 815\,°C$$

$$\Delta t_{1.m} = \frac{478.25 - 280}{\ln \dfrac{478.25}{280}} = 370.32\,°C$$

$$A_o = \frac{Q}{U_o\ \Delta t_{1.m}} = \frac{511134\ kW}{70 \times 10^{-3} \times 370.32\ kW/m^2}$$

$$= 19.72 \times 10^3\ m^2$$

$$\omega_s = \left(n \frac{\pi}{4} d_i^2\right) \frac{V_w}{v_f} = 600\ kg/s$$

$$n \times \frac{\pi}{4} \times (60 \times 10^{-3})^2 \times \frac{1.2}{0.001611} = 600$$

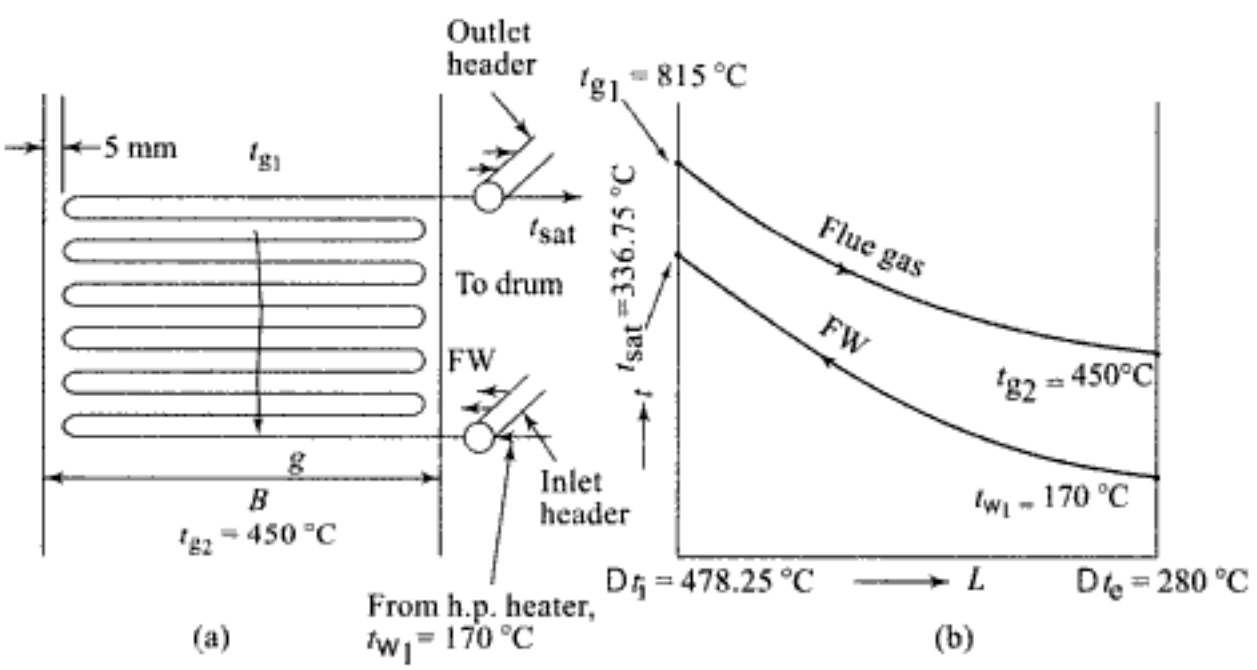

Fig. E6.6

$$n = \frac{600 \times 4 \times 0.001611}{\pi \times 3600 \times 10^{-6} \times 1.2} = \frac{4 \times 1611}{\pi \times 6 \times 1.2} = 285 \qquad \textit{Ans.}$$

$$A_o = n\,\pi d_o l = 19.72 \times 10^3\ m^2$$

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$$l = \frac{19.72 \times 10^3}{285 \times \pi \times 70 \times 10^{-3}} = 314.6 \text{ m} \qquad \textit{Ans.}$$

Number of turns of one coil,

$$n_t = \frac{1}{B - 2C} = \frac{314.6}{4.8 - 2 \times 0.005} = 65.6 = 66$$

The vertical height of the duct occupied by the economiser coils

$$= 66 \times 0.08 = 5.28 \text{ m} \qquad \textit{Ans.}$$

Example 6.7 For the above steam generator there is a tubular type air preheater following the economiser, where the flue gases flow through the tubes and are cooled to 160 °C and air entering at 35°C flows outside the tubes at the rate of 1167 kg/s. The inlet velocity of flue gases is 13 m/s and the tubes are of 60/65 mm. If the overall heat transfer coefficient is 30 W/m^2 K, find the length and the number of tubes. Assume that the flue gases behave as an ideal gas having $c_p = 1.10$ and $R = 0.287$ kJ/ kg K.

Solution For the air preheater (Fig. E6.7), the given particulars are:

$$\omega_g = 1250 \text{ kg/s}, \; t_{g_1} = 450 \text{ °C}, \; t_{g_2} = 160 \text{ °C}, \; t_{a_1} = 35 \text{ °C},$$
$$t_{a_2} = 375 \text{ °C}, \quad U_o = 30 \text{ W/m}^2\text{ K}, \quad V_g = 13 \text{ m/s}$$

$$v_{g_1} = \frac{R\,T_{g_1}}{p} = \frac{0.287 \times (450 + 273)}{101.325} = 2.048 \text{ m}^3\text{/kg}$$

$$Q = \omega_g\, c_{pg}\,(t_{g_1} - t_{g_2}) = \omega_a\, c_{pa}\,(t_{a_2} - t_{a_1}) = U_o\, A_o\, \Delta t_{1.m}$$

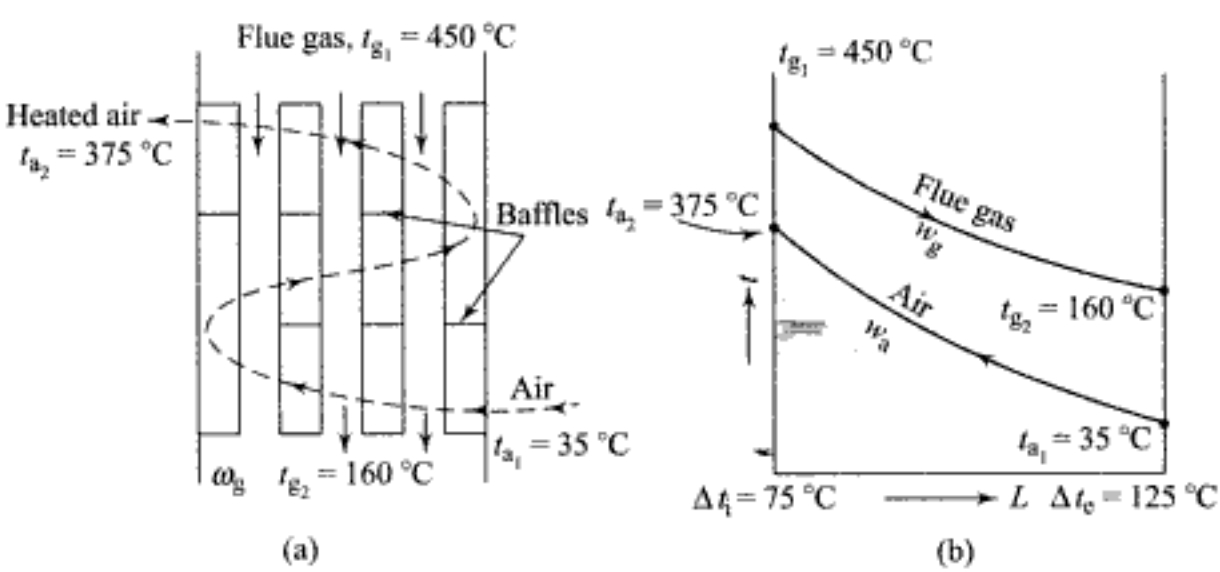

Fig. E6.7

$$1250 \times 1.1\,(450 - 160) = 1167 \times 1.005\,(t_{a2} - 35)$$

$$t_{a_2} = 375 \text{ °C}.$$

$$\therefore \quad \Delta t_{1.m} = \frac{75 - 125}{\ln \dfrac{75}{125}} = 97.5 \text{ °C}$$

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$$1250 \times 1.1 \times 290 = 0.03 \times A_o \times 97.5 \quad \text{or} \quad A_o = 136325\ \text{m}^2$$

$$\omega_g = \left(n\frac{\pi}{4}d_i^2\right)\frac{V_{g_1}}{v_{g_1}} = 1250\ \text{kg/s}$$

$$n = \frac{1250 \times 4 \times 2.048}{\pi \times (0.06)^2 \times 13} = 69{,}647 \qquad \textit{Ans.}$$

$$A_o = n\,\pi\,d_o\,l = 69647 \times \pi \times 0.065 \times l = 136325$$

$$l = \frac{136325}{69{,}647 \times \pi \times 0.065} = 9.59\ \text{m} \qquad \textit{Ans.}$$

Example 6.8 Find the number and length of superheater coils of 50 mm id and 5 mm thickness to be provided if steam at exit is at 60 bar, 500 °C and flows with a velocity of 10 m/s and mass flow of 80 kg/s. Due to restriction by materials, the heat flux in the superheater coils is to be limited to 140 kW/m^2.

Solution From steam tables, at state points shown in Fig. E6.8,

$$h_1 = h_g = 2784.3\ \text{kJ/kg}$$
$$h_2 = 3422.2\ \text{kJ/kg}, \quad v_2 = 0.05665\ \text{m}^3/\text{kg}$$

Heat absorption rate in superheater coils

$$= \omega_s(h_2 - h_1) = 80\,(3422.2 - 2784.3) = 51032\ \text{kW}$$

$$\text{Surface area required} = \frac{51032}{140} = 364.5\ \text{m}^2$$

Now,
$$\omega_s = \rho_1 A_1 V_1 = \rho_2 A_2 V_2 = \left(n\frac{\pi}{4}d_i^2\right)\frac{V_2}{v_2} = 80\ \text{kg/s}$$

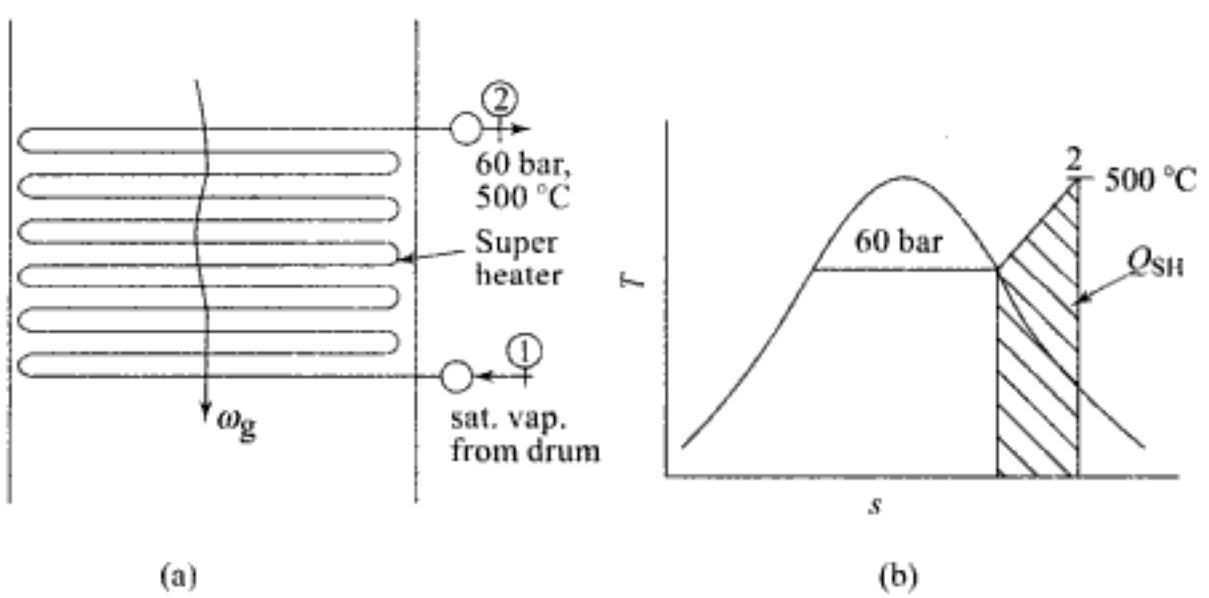

Fig. E6.8

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Number of coils, $n = \dfrac{80 \times 4 \times 0.05665}{\pi \times (0.05)^2 \times 10} = 231$ *Ans.*

Now, $A_o = 364.5 \text{ m}^2 = n\,\pi d_o l$

Length of one coil, $l = \dfrac{364.5}{231 \times \pi \times 0.06} = 8.37 \text{ m}$ *Ans.*

Example 6.9 A boiler plant incorporates an economiser and air preheater and generates steam at 40 bar and 300°C with fuel of heating value 33,000 kJ/kg burned at a rate of 500 kg/h. The temperature of feed water is raised from 40°C to 125°C in the economiser and the flue gases are cooled at the same time from 395°C to 225°C. The flue gases then enter the air preheater in which the temperature of combustion air is raised by 75°C. A forced draught fan delivers the air to the air preheater at a pressure of 1.02 bar and a temperature of 16°C with a pressure rise across the fan of 180 mm of water. The power input to the fan is 5 kW and it has a mechanical efficiency of 78%. Neglecting heat losses and taking C_p as 1.01 kJ/kg K for flue gases, calculate (a) the mass flow rate of air, (b) the temperature of flue gases leaving the plant, (c) the mass flow rate of steam and (d) the efficiency of the boiler.

Solution Power input to the *FD* fan, $P = \dfrac{\dot{V} \times \Delta p}{\eta_{mech}}$

$$\therefore \quad \dot{V} = \frac{P \times \eta_{mech}}{\rho_w g h} = \frac{5 \times 10^3 \times 0.78}{1000 \times 9.81 \times 0.180} = 2.21 \text{ m}^3/\text{s}$$

$$\text{Density of air, } \rho_a = \frac{p}{RT} = \frac{1.02 \times 10^5}{287 \times (273 + 16)} = 1.23 \text{ kg/m}^3$$

Mass flow rate of air, $\dot{m}_a = \rho_a \times \dot{V} = 1.23 \times 2.21$

$= 2.72$ kg/s *Ans.* (a)

Assuming complete combustion of fuel, the mass flow rate of flue gases,

$$\dot{m}_g = \dot{m}_a + \dot{m}_f = 2.72 + \frac{500}{3600} = 2.855 \text{ kg/s}$$

Energy lost by flue gas = energy gained by air in air preheater

$2.855 \times 1.01 \times \Delta T = 2.72 \times 1.005 \times 75$

$\therefore$ ΔT = temperature drop of flue gases

$= 71$°C

$\therefore$ Temperature of flue gases leaving the plant

$= 225 - 71 = 154$°C *Ans.* (b)

Making an energy balance for the economiser

$$\dot{m}_w\, c_w (125 - 40) = \dot{m}_g\, c_{p_g} \times (395 - 225)$$

$$\therefore \quad m_w = \frac{2.855 \times 1.01 \times 170}{4.182 \times 85} = 1.379 \text{ kg/s}$$ *Ans.* (c)

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Water enters the boiler as feedwater at 40°C

$$h_f = 167.4 \text{ kJ/kg}$$

Steam leaves the boiler at 40 bar, 300°C.

$$h = 2962.3 \text{ kJ/kg}$$

$$\therefore \quad \text{Boiler efficiency} = \frac{(2962.3 - 167.4) \times 1.379}{33000 \times 500/3600} = 0.84 \text{ or, } 84\% \qquad \textit{Ans.} \text{ (d)}$$

SHORT-ANSWER QUESTIONS

6.1 What is the function of a steam generator? Enlist its basic components.
6.2 How does an industrial steam generator differ from a utility boiler?
6.3 Explain the characteristic features of a fire tube boiler. How is it different from a water-tube boiler.
6.4 What is a fusible plug? Why is it used?
6.5 How does a spring-loaded safety valve operate?
6.6 What are the characteristic features of a package boiler?
6.7 Mention the merits and demerits of fire-tube boilers.
6.8 How does a cross-drum straight tube boiler differ from a longitudinal drum boiler?
6.9 What are the shortcomings of straight tube boilers?
6.10 What is a Stirling boiler? What are its main features?
6.11 What is circulation? What is the difference between natural circulation and forced circulation?
6.12 What is the pressure head developed for natural circulation?
6.13 Why and when are downcomers placed outside the furnace?
6.14 Under what situation is natural circulation not relied upon and forced circulation adopted?
6.15 What is circulation ratio? Mention the range in which the circulation ratio should vary.
6.16 What is TDF?
6.17 Why is too much steaming not desired in a riser tube?
6.18 When does a riser tube become vulnerable to rupture?
6.19 Why are downcomers fewer in number and bigger in diameter, while risers are more in number and smaller in diameter?
6.20 What is DNB? Why is it avoided?
6.21 What is void fraction? What is slip ratio? How are these two related to quality of a liquid–vapour mixture?
6.22 Why should no moisture flow along with steam from the drum to the superheater?
6.23 Why is no vapour bubble desired to flow along with saturated water from the drum to the downcomers?
6.24 What is blowdown? Why is it needed?
6.25 Why is trisodium phosphate injected into the drum?
6.26 What are the functions of the steam drum in a water tube boiler?
6.27 What are drum internals? Why are they required?

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6.28 What are primary and secondary separators inside the drum?
6.29 What is steam-washing?
6.30 What do you mean by priming and foaming?
6.31 What are "courses"?
6.32 What is a water wall? What do you mean by a membrane water wall?
6.33 What are once-through boilers? How do they differ from drum boilers?
6.34 Explain the functions of primary air and secondary air.
6.35 What is an assisted circulation steam generator?
6.36 What is a La Mont boiler?
6.37 What is a supercritical boiler? What are its merits and demerits?
6.38 What is the function of an economiser? What are steaming and non-steaming economisers?
6.39 Why are the economiser tubes often finned or gilled on the gas-side?
6.40 How is the water velocity inside the economiser tube optimised?
6.41 What are primary and secondary superheaters?
6.42 What is a pendant superheater?
6.43 How are the gas velocity on the outside and steam velocity on the inside of a superheater coil decided?
6.44 How is the erosion rate of fly ash related to gas velocity?
6.45 Why does the exit steam temperature increase with the increase in load in a convective superheater?
6.46 Why does the exit steam temperature decrease with the increase in load in a radiant superheater?
6.47 What are platens? Where are they used?
6.48 Where is reheater located?
6.49 What is a closed loop control? What is proportional control?
6.50 How is the water level in steam drum controlled?
6.51 What is a "boiler master"? How is steam pressure controlled?
6.52 What are the different methods of controlling the superheat (and reheat) temperature of steam?
6.53 Explain the operation of a spray-type desuperheater.
6.54 Why is the desuperheater located before the last stage of superheating?
6.55 Explain the steam temperature control by gas recirculation. What is gas tempering?
6.56 What are tilting burners?
6.57 What is the function of air preheater? How does air preheating save fuel?
6.58 What are recuperative air preheaters? Why are baffles provided? Why is the overall heat transfer coefficient in APH low?
6.59 Explain the operation of Ljungstrom (regenerative) air preheaters. What are their advantages and disadvantages?
6.60 When are two stages of air-preheating required?
6.61 What is a bubbling fluidized bed combustor? What are its merits?
6.62 Explain the characteristics of a circulating fludized bed boiler. What is the backpass?
6.63 What is the function of the external fluid bed heat exchanger?
6.64 What is a non-mechanical valve? Name some such valves.

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6.65 How does the L-valve operate?
6.66 What is a loop seal? How is it different from a seal pot?
6.67 What are cyclone separators? Explain the principle of gas solid separation in a cyclone.
6.68 Define the collection efficiency of a cyclone. What is the cut-off size?
6.69 On what factors does the collection efficiency of a cyclone depend?
6.70 Enlist the advantages and disadvantages of CFB boilers.
6.71 Explain the aspect of fuel flexibility in a CFB boiler. How is the combustion temperature (800–900 °C) maintained in the furnace at varying loads?
6.72 What is acid rain? How is sulphur removed in a fluidized bed combustor? At what temperature is sulphur capture maximized? What should be the Ca/S ratio for high sulphur capture?
6.73 How is low NO_x emission possible in a CFB combustor? Explain the staged combustion process.
6.74 What do you mean by turndown ratio? What is the turndown ratio achieved in a CFB boiler?
6.75 Which is the world's first CFB boiler? Which is now the biggest CFB boiler in operation? Which is the biggest boiler in the erection stage?
6.76 What is a pressurized fluidized bed boiler? What are its merits and demerits?
6.77 How does a PFBC differ from a PCFB?
6.78 Explain briefly the characteristic features of a PCFB boiler. How does it fit into the combined cycle power generation?
6.79 What are the circulation systems in a CFB boiler?
6.80 Explain the operation of an electrostatic precipitator.
6.81 What is a fabric filter? What do you mean by the term 'air-to-cloth ratio'?
6.82 What are the different types of fabric filters?
6.83 What is a baghouse?
6.84 What are the factors to be considered for an ash disposal system.
6.85 What are the fractions of fly ash and bottom ash in a pc boiler?
6.86 What are the uses of fly ash?
6.87 What are the basic objectives of feedwater treatment?
6.88 What are the impurities in raw water?
6.89 What do you mean by clarification?
6.90 How are gravity filters different from pressure filters?
6.91 What are the basic processes of water softening?
6.92 Explain the lime–soda process for softening of feedwater.
6.93 What do you understand by hot phosphate softening?
6.94 Explain how a bed of sodium zeolite softens feedwater. How is the bed regenerated?
6.95 What are the basic components of a demineralization plant?
6.96 What do you mean by ion exchange?
6.97 What is the harmful effect of silica in water? How is it removed?
6.98 What is reverse osmosis? What is electrodialysis?
6.99 What do you understand by condensate polishing? Where is it required?
6.100 What is pH value? How is it controlled in water?
6.101 What is internal treatment of feedwater? How is it done?
6.102 What do you mean by steam purity?

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PROBLEMS

6.1 A boiler plant comprises a boiler, superheater, economiser and air preheater. The feedwater enters the economiser at 45 °C. Air is preheated from a temperature of 15 °C to 150 °C. The steam is generated in the boiler at a pressure of 35 bar, 0.98 dry and leaves the superheater at 350 °C. When using oil of calorific value 42 MJ/kg, the evaporation rate is 10 kg steam per kg fuel and the air to fuel ratio is 20 : 1 by mass. Neglecting heat losses and pressure drops, estimate the heat transfer per kg fuel in each component and the plant efficiency. Assume c_p for air and flue gas as 1.005 and 1.045 kJ/ kg K, respectively. [*Ans.* $\eta_{plant} = 0.696$]

6.2 A boiler drum has a concentration of 1250 ppm. What percentage of water in the boiler must be wasted through the blowdown to reduce the concentration to 1000 ppm when the feedwater concentration is 11 ppm? If it is desired to have not more than 1 ppm in the steam from the boiler having a concentration of 1000 ppm, what will be the maximum moisture in the steam? [*Ans.* 20.2%, 0.1%]

6.3 The following data apply to a steam generator with economiser and air preheater. Steam output; 100 t/h: blowdown 3%; gas flow: 130 t/h; gas temperature to economiser: 410°C; gas temperature leaving air preheater: 150 °C; air flow 100 t/h; water temperature to economiser: 210 °C; water temperature from economiser: 225 °C; air temperature to air preheater: 44 °C; heat transfer surface area of economiser: 380 m^2; heat transfer surface area of air preheater: 5000 m^2. Calculate, making needed assumptions, (a) the gas temperature leaving economiser, (b) the air temperature leaving air preheater, (c) the overall heat transfer coefficients for both economiser and air preheater. Take c_p for air, gas and feedwater as 1.005, 1.05 and 4.2 kJ/kg K, respectively. [*Ans.* (a) 362.5 °C, (b) 332.62 °C, (c) 0.1015, 0.0965 kW/m^2 K]

6.4 A drum type steam generator operates at 160 bar. Water from the drum, subcooled at 7.3 °C flows down the downcomer at the rate of 1250 kg/s. The downcomers and risers are all 15 m high. The average density of water–steam mixture in the risers is 350 kg/m^3. The pressure losses in the downcomer and risers is 0.52 bar. Calculate the power needed to drive a forced circulation pump of 70% efficiency.

6.5 A 15 m long, 75 mm diameter riser tube receives saturated water at 160 bar and at a velocity of 0.7 m/s. Heat is added to it uniformly. The slip ratio is 1.7. Estimate the maximum heat added to the tube in kJ/m if the exit void fraction is not to exceed 0.80.

6.6 Saturated steam at 160 bar enters a convective superheater. At a given load it leaves at 480 °C. The convective heat transfer coefficients h_i and h_o are proportional to $\omega^{0.8}_s$ and $\omega^{0.6}_g$, respectively, where ω_s and ω_g are the mass flow rates of steam and gas. Find the exit temperature of steam if both steam and gas flows are doubled. The gas temperature T_g remains constant at 1500 °C. Assume for simplicity that the heat transferred Q is proportional to U $(T_g - T_s)$, where T_s is the average of the exit and inlet steam temperature.

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6.7 Steam enters a radiant superheater in a dry saturated state at 160 bar. The convective heat transfer coefficient inside the tubes is proportional to $\omega^{0.8}_s$, where ω_s is the steam flow rate. The radiative heat transfer between the combustion gases and the tube surface is proportional to $(T^4_f - T^4_w)$, where T_f and T_w are the flame and tube wall temperatures, respectively. At a given load, the superheater outlet temperature is 560 °C, and h_o is 5-times h_i. Neglecting the conductive resistance of the wall, calculate the exit temperature of steam, if the steam mass flow is doubled.

6.8 A 24-section Ljungstrom air preheater contains 1.6 mm thick corrugated steel sheet and rotates at 2 rpm. The hot gases enter at 315 °C and leave at 175 °C. The air enters at 25 °C. For simplicity, assume that both gases and air have equal mass flow rates of 12.8 kg/s and the same specific heat of 1.008 kJ/kg K. The preheater has an effectiveness of 0.8 (ratio of actual to maximum temperature rise of air). Steel has a density of 7850 kg/m^3 and a specific heat of 0.418 kJ/ kg K. Calculate (a) the air exit temperature, and (b) the total area provided by the steel sheets.

6.9 A certain steam generator, operated at 40 bar, has a horizontal 1.5 m id steam drum which is 6.2 m long. The normal water level is at the mid-point. If the water level falls more than 0.30 m below this point, serious "starvation" of some of the downcomers will result. When the steaming rate is 12 kg/s, how long an interruption in feedwater supply could be tolerated?

6.10 A forced circulation boiler delivering 36 kg/s at 130 bar is operated with a circulation ratio of 5 : 1. The circulation pumps impart a head rise of 2.8 bar, with suction conditions of 350 °C and 140 bar. What would the ideal pump work amount to per kg of steam delivered?

6.11 A certain steam generator is to deliver 84 kg/s at 500 °C and 100 bar. The condition of steam after the primary superheater is expected to be 495 °C and 101 bar. It will be necessary to reduce the temperature to 450 °C prior to entering the secondary superheater if the desired final temperature is to result. What quantity of desuperheater water at 115 bar, 175 °C must be supplied per second to the desuperheater spray nozzles?

6.12 Calculate the efficiency of a marine boiler operating under the following conditions: feedwater at 50 bar and 120 °C is supplied to the boiler at the rate of 30 kg/s. Of this, 25 kg/s leaves the superheater outlet at 40 bar and 450 °C and the remainder leaves the steam drum as dry saturated steam at 42 bar for auxiliary purposes. The fuel used is oil having a calorific value of 40,414 MJ/ m^3 and it is supplied at the rate of 7.2 m^3/h.

6.13 The boiler in Prob. 6.12 has an integral non-steaming economiser from which water leaves at a temperature of 175 °C. Of all the heat absorbed by the water and steam, determined (a) the percentage absorbed in the economiser, (b) the percentage absorbed in the evaporator, and (c) the percentage absorbed in the superheater.

6.14 It is proposed to install an economiser on a steam generator which at present generates saturated steam at 20 bar while consuming 3 tonnes of coal per hour. The coal has a HHV of 31.4 MJ/kg and contains a total of 5% H_2 as fired. Addition of the proposed economiser surface would result in a final gas

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temperature of 195 °C entering the stack compared to the present temperature of 260 °C. There is no reason to believe that addition of the economiser would change appreciably the losses due to unburnt carbon, CO or radiation. At present, these losses total 6.5% of the HHV. Combustion air is supplied at 30 °C and may be considered to be dry. There are 14 kg of dry flue gas per kg of fuel. (a) Calculate the new steam generator efficiency (after addition of the economiser). (b) Calculate the reduction in fuel consumption due to this addition.

6.15 A boiler plant incorporates an economiser and an air preheater, and generates steam at 40 bar, 300 °C with fuel of calorific value 33 MJ/kg burned at a rate of 0.111 kg/s. In the economiser, the feedwater is heated from 40 °C to 125 °C by cooling flue gases from 395 °C to 225 °C. The flue gases then enter the air preheater in which air is heated and its temperature rises by 75 °C. An FD fan delivers air at 1.02 bar and 16 °C producing a pressure head of 180 mm of water. The power input to the fan is 5 kW and it has a mechanical efficiency of 76%. Estimate (a) the mass flow rate of air, (b) the temperature of flue gases leaving the plant, (c) the mass flow rate of steam, and (d) the efficiency of the boiler plant. Neglect heat losses and take c_p = 1.01 kJ/kg K for the flue gases.

[*Ans*. (a) 2.72 kg/s, (b) 150.4 °C, (c) 1.25 kg/s, (d) 83%]

6.16 A 210 MW power plant has steam condition at boiler outlet as 150 bar, 550°C and the condenser pressure is 0.1 bar. The boiler efficiency is 90 per cent. The boiler consumes coal of calorific value 25 MJ/kg. The feedwater temperature at boiler inlet is 170°C. The steam generator has risers in the furnace wall 43 m high and unheated downcomers. The boiler operates on natural circulation and to ensure wetted surface on the riser tubes it has a circulation ratio of 16. A minimum exit velocity of water-steam mixture leaving a riser and entering the drum is required to be 1.8 m/s. The risers have 60 mm outer diameter and 3 mm thickness. Taking η_{Gen} = 0.94 and η_T = 0.92 and neglecting any heat loss and pressure drop, as well as the pump work, estimate (a) the steam generation rate, (b) the fuel burning rate, (c) the evaporation factor, (d) the pressure head developed due to natural circulation, (e) the quality of steam at the top of the riser, (f) the void fraction of steam at the riser exit, if slip ratio is 1.2, (g) the number of risers required, (h) the heat absorption rate per unit projected area of a riser, (i) the overall plant efficiency, if 5 per cent of the total power is used to drive the auxiliaries.

6.17 In continuation of the above problem, feedwater enters the economiser inlet header at 150 bar, 170°C and leaves in the saturated vapour state. The economiser consists of tubes 70 mm outer diameter and 60 mm inner diameter made in the form of vertical coils. To restrict the erosion relate by flyash, the flue gas velocity should not exceed 12 m/s, while the velocity of water leaving the outlet header is 1.2 m/s. The ash content of coal is 30 per cent by mass and 9.8 kg air is supplied per kg of coal. Flue gases leave the economiser coils at 450°C. Taking the overall heat transfer coefficient as 80 W/m^{2}K, estimate the number of coils needed in the economiser and the length of a coil. If the vertical pitch of the coils is 80 mm and the clearance on the two sides of the duct having width 4.8 m, is 5 mm, find the vertical height of the economiser.

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6.18 For the above steam generator there is a tubular type air preheater following the economiser, where the flue gases flow through the tubes and are cooled to 150°C. Air entering at 35°C flows outside the tubes. The outlet velocity of flue gases is 14 m/s and the tubes are of 60/65 mm. If $U_0 = 30$ W/m^{2}K, find the number of tubes of the air preheater and the length of tubes. Assume that the flue gases behave as an ideal gas having $c_p = 1.10$ and R = 0.286 kJ/kgK.

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7

Steam Turbines

7.1 INTRODUCTION

A steam turbine is a prime mover which continuously converts the energy of high-pressure, high temperature steam supplied by a steam generator into shaft work with the low temperature steam exhausted to a condenser. This energy conversion essentially occurs in two steps:

1. The high-pressure, high-temperature steam first expands in nozzles and comes out at a high velocity.
2. The high velocity jets of steam coming out of the nozzles, impinge on the blades mounted on a wheel, get deflected by an angle and suffer a loss of momentum which is absorbed by the rotating wheel in producing torque.

A steam turbine is basically an assemblage of nozzles and blades.

The Greek inventor Hero of Alexandria built the first prototype of a steam turbine in 120 BC which operated on the reaction principles. A simple closed spherical vessel, mounted on bearings, carrying steam from a cauldron or boiler with four tangential pipes discharging steam is driven around by the reaction of the steam jets (Fig. 7.1 a). Many centuries later, Giovanni Branca made the prototype of the impulse steam turbine in 1629, which is often referred to as the Branca's wheel.

In 1878, a Swedish engineer, Carl Gustav Patrik de Laval (1845-1913), developed a simple impulse turbine to separate cream from milk, using a convergent-divergent (supersonic) nozzle which ran the turbine to a maximum speed of 100,000 rpm. He constructed in 1897 a velocity-compounded impulse turbine (a two-row axial turbine with a row of guide vane stators between them, all fed by a single set of high velocity nozzles).

In France, Auguste Rateau (1863-1930) experimented with a de Laval turbine in 1894, and developed the pressure compounded impulse turbine by 1900.

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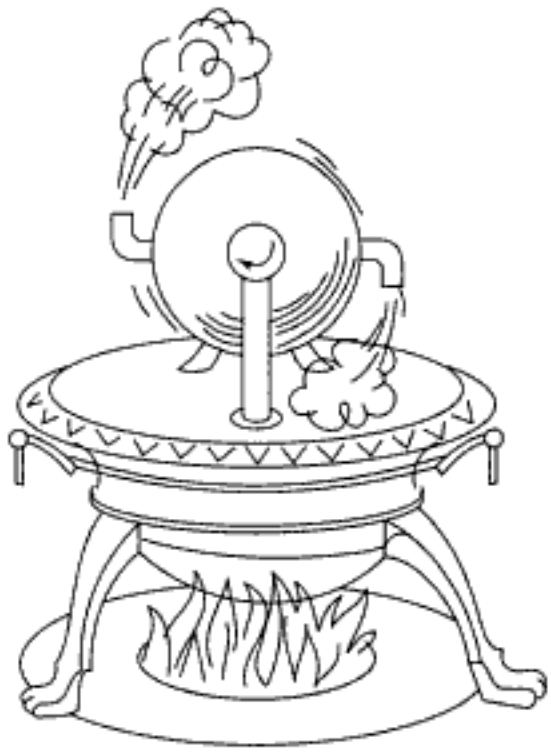

Fig. 7.1 (a) *Hero's turbine*

In the USA, Charles G. Curtis (1860-1953) patented in 1896 the velocity-compounded turbine similar to a two-stage de Laval turbine, but after further development he sold all his rights to General Electric in 1901.

In England, Charles A, Parsons (1854-1931) made in 1884 a multi-stage axial flow reaction turbine using brass blades on a steel wheel. Berger Ljungstrom (Sweden) became famous for his development of a double flow radial reaction turbine in 1912.

Steam turbines are not only employed to operate electric generators in thermal and nuclear power plants to produce electricity, but they are also used (a) to propel large ships, ocean liners, submarines and so on, and (b) to drive power absorbing machines like large compressors, blowers, fans and pumps.

Turbines can be condensing or non-condensing, depending of whether the back pressure is below or equal to the atmospheric pressure. Figure 7.1(b) shows different types of turbines, in regard to steam flow through the units.

For small units without reheat, the steam turbine may consist of a single turbine when the steam expanding through the turbine exhausts to a condenser or to a process line. This is a straight-flow turbine as shown in Fig. 7.1 (b) (A) and (B). For a large unit without reheat, the steam may expand through an initial section and then exhaust to another turbine. This later turbine may then exhaust to a condenser or to a process. The initial turbine is designated as the high-pressure (HP) turbine and the second turbine the low-pressure (LP) turbine, as shown in Fig. 7.1(b) (C).

For a single reheat cycle, the steam from the boiler flows to the HP turbine where it expands and is exhausted back to the boiler for reheating. The reheat steam coming from the boiler flows to the intermediate-pressure (IP) or reheat turbine where it expands and exhausts into a crossover line that supplies steam

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to the double-flow LP turbine. The steam expands through the LP turbine and exhausts to a condenser as shown in Fig. 7.1(b) (D).

Figure 7.1(b) (E) shows double-reheat cycle where steam is reheated twice, having four turbine sections, viz. HP, IP, reheat and double-flow LP. Flow designations of triple-flow, four-flow, six-flow and eight-flow are also used. A triple-flow designation indicates the use of one double-flow LP turbine and a single-flow LP turbine, as shown in Fig. 7.1(b) (F). Four-flow designation indicates the use of two double-flow LP turbines (Fig. 7.1(b)(G)).

Turbines are also designated by the type of extraction involved, if any. Steam extracted for feedwater heating varies in pressure and flow as a function of load. This variation is acceptable and so no effort is made to regulate the pressure. This is called uncontrolled, simple or nonautomatic extraction, as shown in Fig. 7.1(b)(H). However, turbines supplying steam for process applications must supply steam at a constant pressure. Since the pressure varies with load, regulating valve is used to control the pressure. This is called a controlled or automatic extraction turbine, as shown in Fig. 7.1(b)(I).

The overall steam turbine generator arrangement of a power plant is designated as *tandem-compound* or *cross-compound* on the basis of shaft orientation, as shown in Fig. 7.1 (c). The tandem-compound unit has all turbines and the generator in-line, connected to the same shaft and thus operate at the same speed. The cross-compound arrangement typically consists of HP and IP turbines operating at 3600 rpm (for frequency of 60 Hz and a bipolar generator, rpm = $(120 \times 60)/2$ = 3600 rpm as in the USA, while in India with 50 Hz frequency, it is 3000 rpm) driving a generator. The exhaust steam from the IP turbine crosses over to a LP turbine that operates at 1800 rpm (1500 rpm for 50 Hz and 4 poles), driving a separate generator. The LP turbine operating at slower speed allows the use of longer last-stage turbine blades with expansion to higher moisture percentages and less exhaust losses, resulting in higher turbine efficiencies.

Large cross-compound units in the 1300 MW range with both shaft orientations operating at 3600 rpm have been built. The dual-shaft arrangement was used to minimize shaft length and any single generator size.

Power plant steam turbines are thus typically designated by shaft orientation, number of LP turbine steam flow paths and the last stage blade length of the LP turbine. A turbine designated as TC4F30, for example, indicates a unit that is tandem-compound (TC) having two double-flow (4F) LP turbines with 30 in. (76.2 cm) last stage blade length. A CC2F23 indicates a unit that is cross-compound (CC) having one double-flow (2F) LP turbine with 23 in. (58.4 cm) last stage blade length.

7.2 FLOW THROUGH NOZZLES

A *nozzle* is a duct by flowing through which the velocity of a fluid increases at the expense of pressure drop. A duct which decreases the velocity of a fluid and causes a corresponding increase in pressure is called a *diffuser*. The same duct

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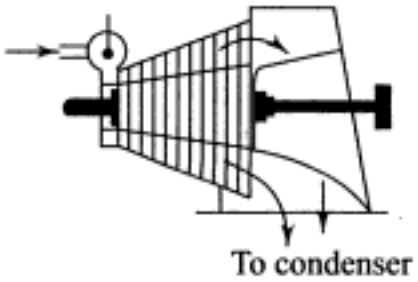
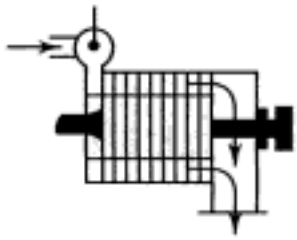
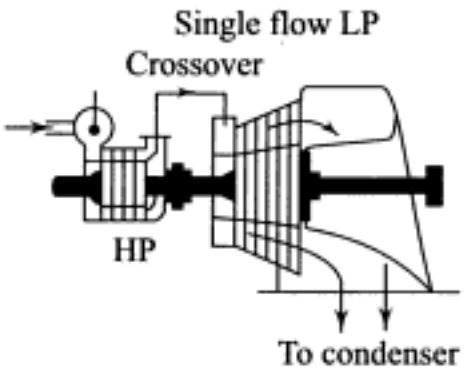


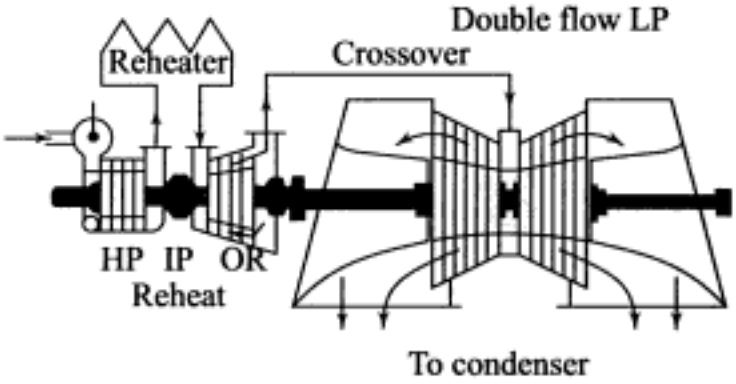
(A) Straight flow condensing



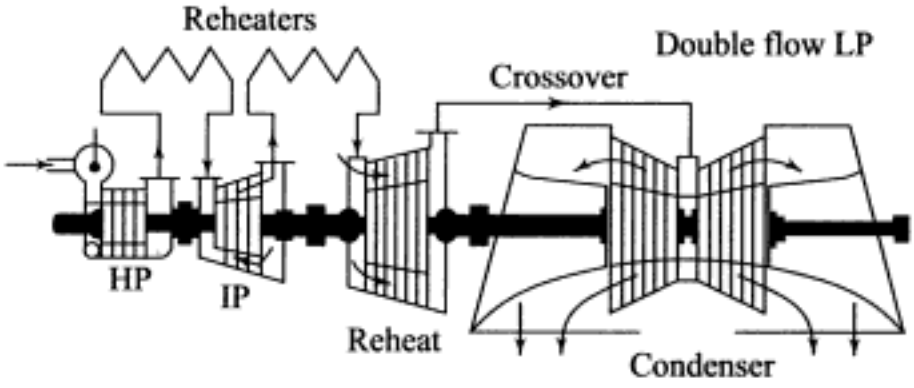
(B) Straight flow noncondensing



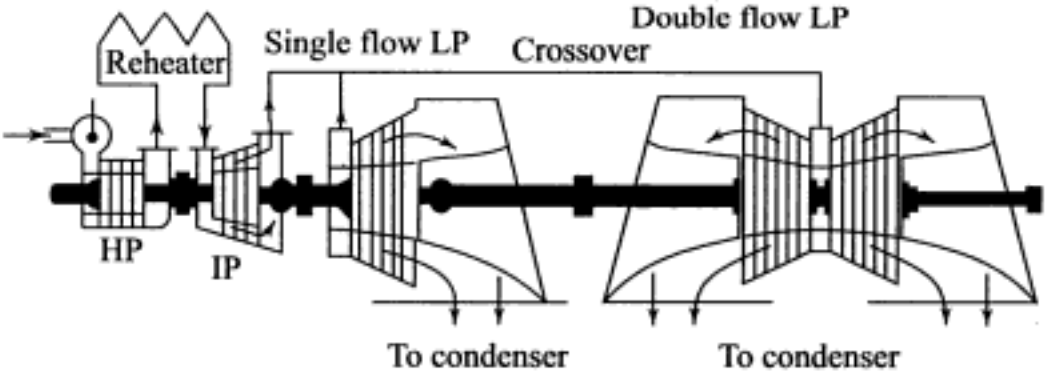
(C) Two section condensing



(D) Single reheat cycle



(E) Double reheat cycle



(F) Triple flow low pressure tubine cycle

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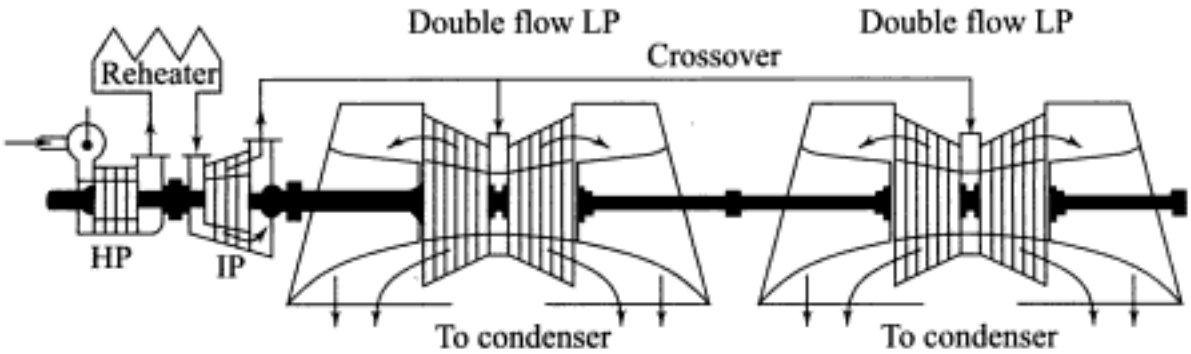
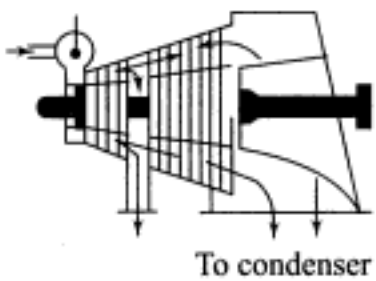
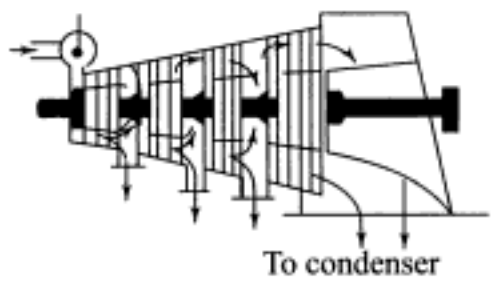


(G) Four flow low pressure turbine cycle

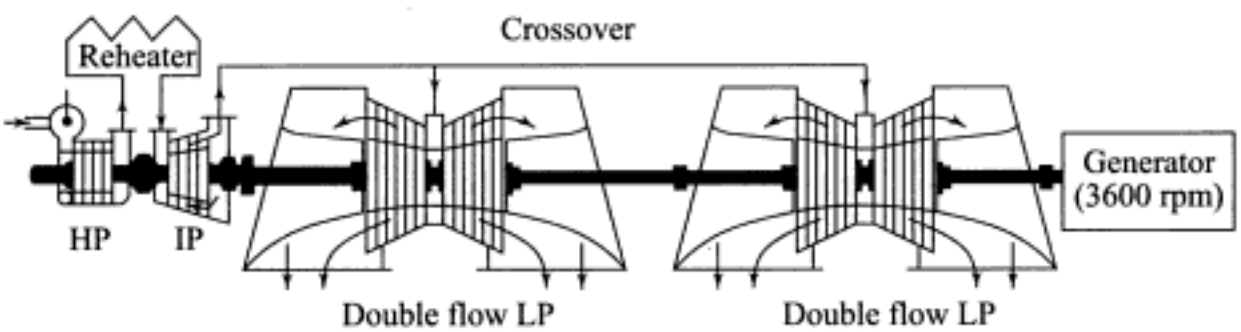


(H) Single nonautomatic extraction condensing

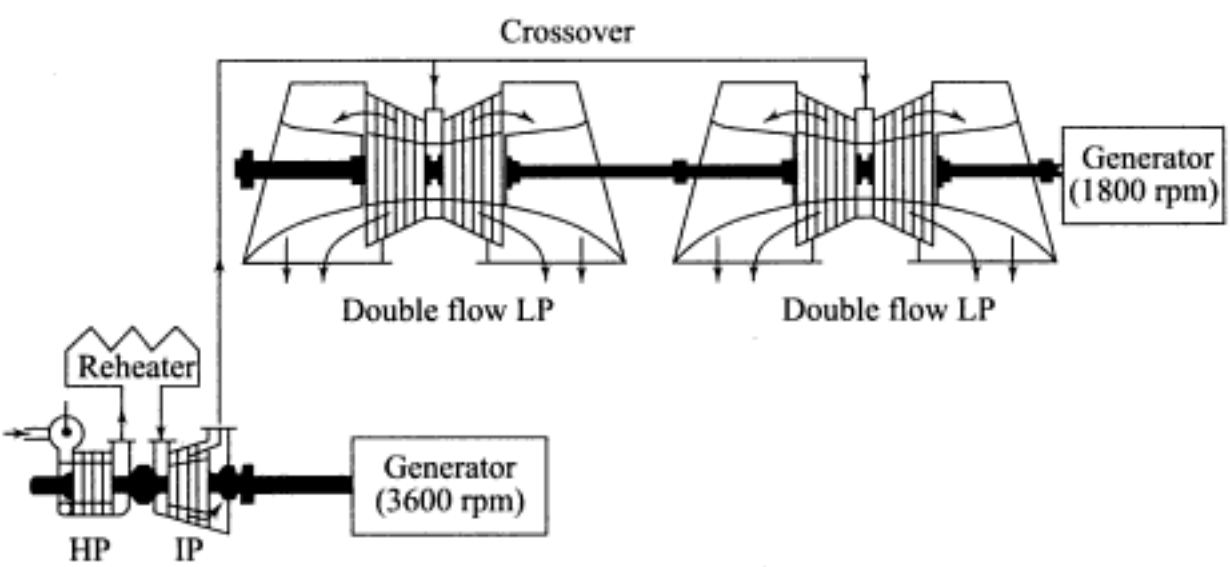


(I) Triple automatic extraction condensing

Fig. 7.1 (b) *Turbine Types*



(A) Tandem compound unit



(B) Cross compound unit

Fig. 7.1 (c) *Turbine shaft orientations*

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may be either a nozzle or a diffuser depending upon the end conditions across it. If the cross-section of a duct decreases continuously from inlet to exit, the duct is said to be convergent, and if it increases from inlet to exit, the duct is said to be divergent. If the cross-section initially decreases and then increases, the duct is called convergent-divergent. The minimum crosssection of this duct is referred to as throat. A fluid is said to be *compressible* if its density changes with a change in pressure (or temperature). If the density does not changes or changes very little, the fluid is said to be incompressible. Gases and vapours are compressible, whereas liquids are *incompressible*.

7.2.1 Velocity of Pressure Pulse in a Fluid

Let us consider an infinitesimal pressure wave initiated by a slight movement of a piston to the right (Fig. 7.2) in a pipe of uniform cross-section. The pressure wave front propagates steadily with a velocity c, which is known as the velocity of sound, sonic velocity or acoustic velocity. The fluid near the piston will have a slightly increased pressure and will be slightly more dense than the fluid, away from the piston.

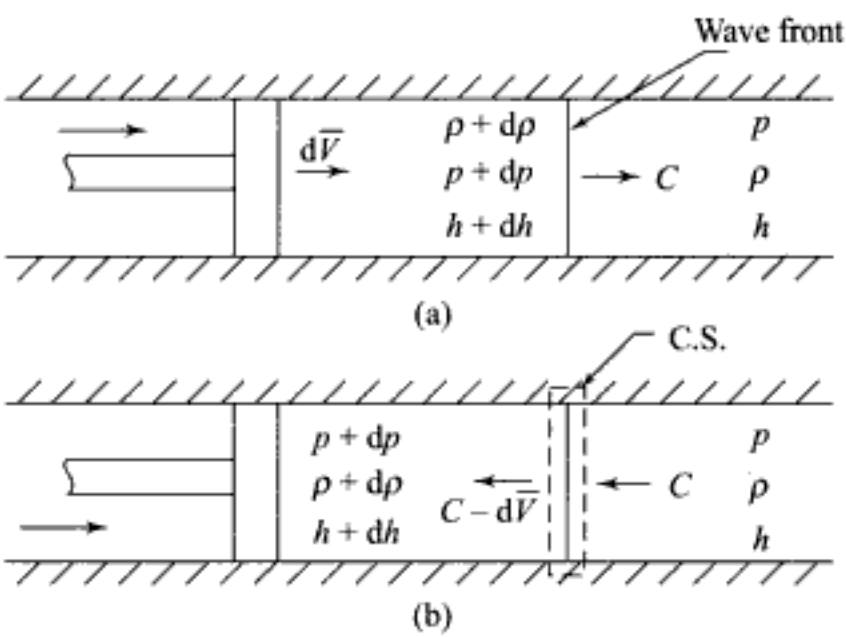

Fig. 7.2 *Diagram illustrating sonic velocity: (a) stationary observer, (b) observer travelling with wave-front*

To simplify the analysis, let an observer be assumed to travel with the wave front to the right with the velocity c. Fluid flows steadily from right to left as it passes through the wave front, the velocity is reduced from c to $c - dV$. At the same time, the pressure rises from p to $p + dp$ and the density from ρ to $\rho + d\rho$. The continuity equation for the control volume gives

$$\rho A c = (\rho + d\rho)\, A\, (c - d\bar{V})$$

or

$$\rho c = \rho c - \rho d\bar{V} + c d\rho - d\rho\, d\bar{V}$$

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Neglecting the product $d\rho . d\overline{V}$, both being very small,

$$\rho d\overline{V} = c\, d\rho \tag{7.1}$$

The momentum equation for the control volume gives

$$[p - (p + dp)]A = \rho Ac\, [(c - d\overline{V}) - c] \tag{7.2}$$

From Eqs (7.1) and (7.2).

$$\frac{dp}{c} = c d\rho \qquad \therefore c = \left[\frac{dp}{d\rho}\right]^{1/2}$$

Since the variations in pressure and temperature are negligibly small and the change of state is so fast as to be essentially adiabatic, and in the absence of any internal friction or viscosity, the process is reversible and isentropic. Hence, the sonic velocity is given, by

$$c = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} \tag{7.3}$$

No fluid is truly incompressible, although liquids show little change in density. The velocity of sound in common liquids is of the order of 1650 m/s.

7.2.2 Velocity of Sound in Ideal Gas

For an ideal gas, in an isentropic process

$$pv^{\gamma} = \text{constant} \qquad \text{or} \qquad \frac{p}{\rho^{\gamma}} = \text{constant}$$

By logarithmic differentiation (i.e., first taking logarithm and then differentia-ting),

$$\frac{dp}{p} - \gamma \frac{d\rho}{\rho} = 0$$

$$\frac{dp}{d\rho} = \gamma \frac{p}{\rho}$$

Since $\qquad c^2 = \dfrac{dp}{d\rho}$ and $p = \rho RT$ for an ideal gas,

$$c = \sqrt{(\gamma RT)} \tag{7.4}$$

where R is the characteristic gas constant, which is equal to the universal gas constant divided by the molecular weight of the gas.

The Mach number, M is defined as the ratio of the actual velocity V to the sonic velocity c

$$M = V/c$$

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When $M > 1$, the flow is supersonic, When $M < 1$, the flow is subsonic, and when $M = 1$, the flow is sonic.

7.2.3 Stagnation Properties

The isentropic *stagnation state* is defined as the state a fluid in motion would reach if it were brought to rest isentropically in steady flow, adiabatic, zero work output device. This is a reference state in compressible fluid flow and is commonly designated with the subscript zero. The stagnation enthalpy h_c (Fig. 7.3) is related to the enthalpy and velocity- of the moving fluid by

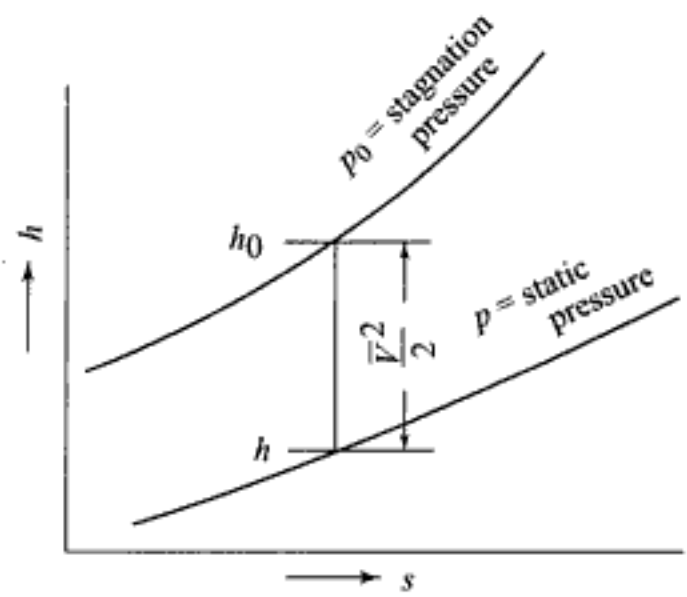

Fig. 7.3 *Stagnation state*

$$h_o = h + \frac{V^2}{2}$$

For an ideal gas, $c_p\,(T_o - T) = \dfrac{V^2}{2}$

The properties without subnscript denote static properties.

Since $$c_p = \frac{\gamma R}{\gamma - 1},\ T_o/T = 1 + \frac{V^2\,(\gamma - 1)}{2\gamma\, RT}$$

Using Eq. (7.4) and the Mach number,

$$\frac{T_o}{T} = 1 + \frac{\gamma - 1}{2} M^2 \tag{7.5}$$

The stagnation pressure p_o is related to the Mach number and static pressure in the case of an ideal gas by the following equation:

$$\frac{p_o}{p} = \left(\frac{T_o}{T}\right)^{\frac{\gamma}{\gamma-1}} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{\gamma}{\gamma-1}} \tag{7.6}$$

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7.2.4 Subsonic and Supersonic Nozzles

For isentropic flow through a nozzle,

$$h_o = h + \frac{\bar{V}^2}{2}$$

Since stangnation enthalpy does not change,

$$dh = -\bar{V}\ d\bar{V} \tag{7.7}$$

From the property relation

$$T\,ds = dh - v\,dp$$

For isentropic flow,

$$dh = \frac{dp}{\rho} \tag{7.8}$$

From Eqs (7.7) and (7.8)

$$dp = -\rho\bar{V}\ d\bar{V} \tag{7.9}$$

or

$$\frac{dp}{d\bar{V}} < 0 \tag{7.10}$$

Therefore, as pressure decreases for flow through a nozzle, the velocity increases, and *vice versa*. The continuity equation gives

$$\omega = \rho A V$$

By logarithmic differentiation,

$$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{d\bar{V}}{\bar{V}} = 0$$

or

$$\frac{dA}{A} = -\frac{d\bar{V}}{\bar{V}} - \frac{d\rho}{\rho}$$

Substituting from Eq. (7.9),

$$\frac{dA}{A} = \frac{dp}{\rho\bar{V}^2} - \frac{d\rho}{\rho} = \frac{dp}{\rho\bar{V}^2}\left[1 - \bar{V}^2\frac{d\rho}{dp}\right]$$

or

$$\frac{dA}{A} = \frac{dp}{\rho\bar{V}^2}\ (1 - M^2) \tag{7.11}$$

Also,

$$\frac{dA}{A} = (M^2 - 1)\ \frac{d\bar{V}}{\bar{V}} \tag{7.12}$$

When $M < 1$, i.e. the inlet velocity is subsonic, as flow area A decreases, the pressure decreases and the velocity increases. So, for subsonic flow, a convergent passage becomes a nozzle (Fig. 7.4a) and a divergent passage becomes a diffusor (Fig. 7.4b), When $M > 1$, i.e. when the inlet velocity is supersonic, as flow area A decreases, pressure increases and velocity decreases,

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and as flow area A increases; pressure decreases and velocity increases. So, for supersonic flow, a convergent passage is a diffusor (Fig. 7.4c) and a divergent passage is a nozzle (Fig. 7.4d).

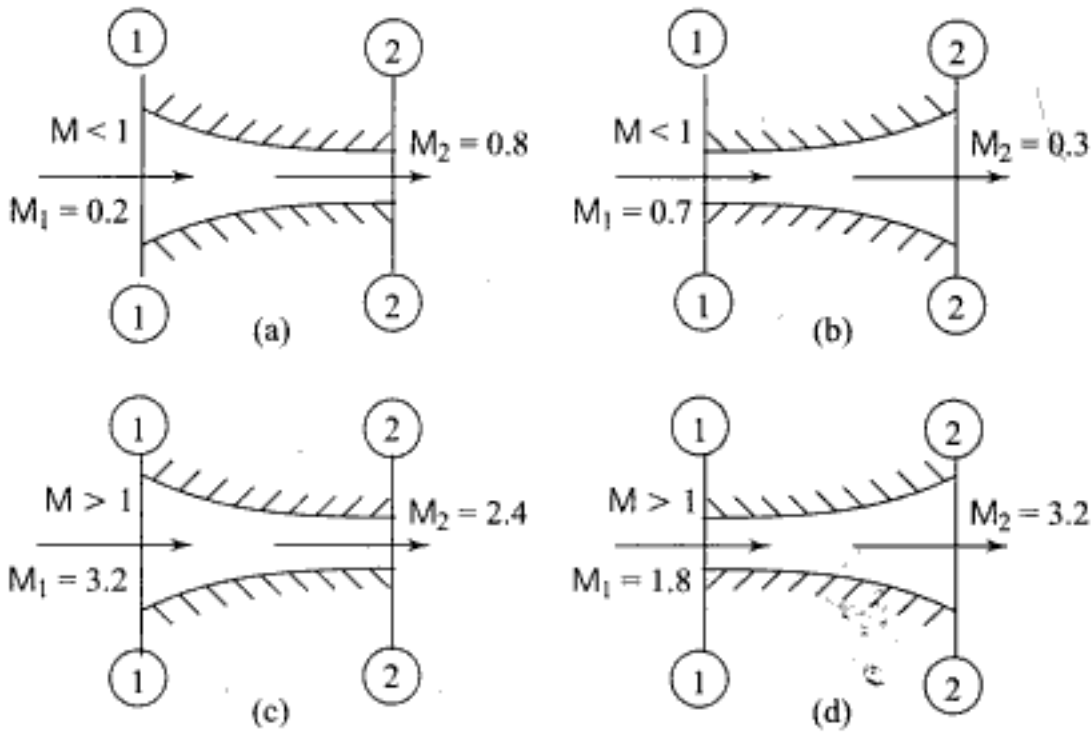

Fig. 7.4 *Effect of area change on subsonic and supersonic flow: (a) subsonic nozzle, (b) subsonic diffuser (c) supersonic diffuser, (d) supersoncic nozzle*

7.2.5 Critical Pressure ratio and Choked Flow

Let us consider the isentropic flow of an ideal gas through a convergent-divergent nozzle (Fig. 7.5).

The mass flow rate per unit area is given by

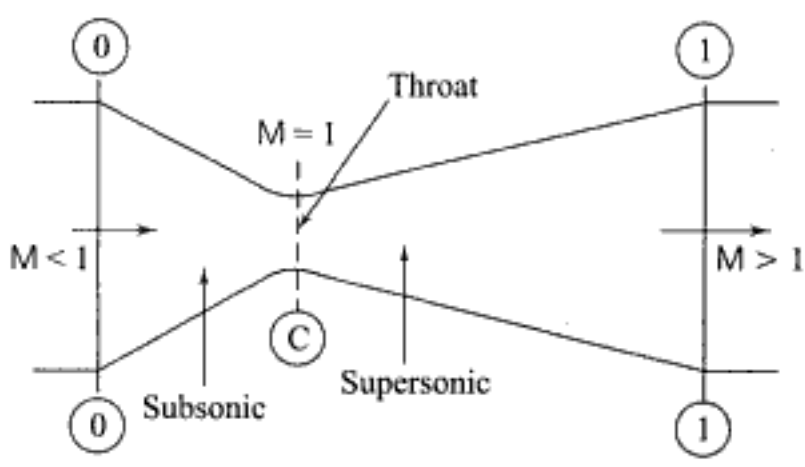

Fig 7.5 *Choked flow in a convergent-divergent nozzle*

$$\frac{\omega}{A} = \rho\bar{V} = \rho CM = \frac{p}{RT}\sqrt{(\gamma RT)}\cdot M = \sqrt{\left(\frac{\gamma}{R}\right)}\frac{p_0 M}{\sqrt{T_0}}\left(\frac{T_0}{T}\right)^{1/2}\frac{p}{p_0}$$

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$$= \sqrt{\left(\frac{\gamma}{R}\right)} \frac{p_0 M}{\sqrt{T_0}} \left(\frac{T_0}{T}\right)^{1/2} \frac{1}{\left(\frac{T_0}{T}\right)^{\gamma/(\gamma-1)}}$$

$$\frac{\omega}{A} = \sqrt{\left(\frac{\gamma}{R}\right)} \frac{p_0 M}{\sqrt{T_0}} \frac{1}{\left(1+\frac{\gamma-1}{2}M^2\right)^{(\gamma+1)/\{2(\gamma-1)\}}} \tag{7.13}$$

Since p_o, T_o, γ and R are constant, the discharge per unit area, ω/A, is a function of Mach number only. There is a particular value of M when ω/A is maximum. Differentiating Eq. (7.13) with respect to M and equating it to zero.

$$\frac{d(\omega/A)}{dM} = \sqrt{\left(\frac{\gamma}{R}\right)} \frac{p_0}{\sqrt{T_0}} \frac{1}{\left(1+\frac{\gamma-1}{2}M^2\right)^{(\gamma+1)/\{2(\gamma-1)\}}}$$

$$+ \sqrt{\left(\frac{\gamma}{R}\right)} \frac{p_0 M}{\sqrt{T_0}} \left[-\frac{\gamma+1}{2(\gamma-1)}\right] \left(1+\frac{\gamma-1}{2}M^2\right)^{-\frac{\gamma+1}{2(\gamma-1)}-1} \left(\frac{\gamma-1}{2}2M\right) = 0$$

or $$1 - \frac{M^2(\gamma+1)}{2\left(1+\frac{\gamma-1}{2}M^2\right)} = 0$$

or $$M^2(\gamma+1) = 2 + (\gamma-1)M^2$$

$$M^2 = 1 \quad \text{or} \quad M = 1$$

So, the discharge ω/A is maximum when $M = 1$.

Since $\bar{V} = cM = (\gamma RT)^{1/2}M$, by logarithmic differentiation,

$$\frac{d\bar{V}}{\bar{V}} = \frac{dM}{M} + \frac{1}{2}\frac{dT}{T} \tag{7.14}$$

Again, $$\frac{T}{T_o} = \left(1+\frac{\gamma-1}{2}M^2\right)^{-1}$$

By logarithmic differentiation,

$$\frac{dT}{T} = \frac{(\gamma-1)M^2}{1+\frac{\gamma-1}{2}M^2} \times \frac{dM}{M} \tag{7.15}$$

From Eqs (7.14) and (7.15),

$$\frac{d\bar{V}}{\bar{V}} = \frac{1}{1+\frac{\gamma-1}{2}M^2} \times \frac{dM}{M} \tag{7.16}$$

From Eqs (7.12) and (7.16),

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$$\frac{dA}{A} = \frac{(M^2 - 1)\,dM}{M\left(1 + \frac{\gamma - 1}{2} M^2\right)} \tag{7.17}$$

By substituting $M = 1$ in any of the Eqs (7. 11), (7.12) and (7.17), $dA = 0$, or A = constant, So $M = 1$ *occurs only at the throat and nowhere else, and this happens only when the discharge is the maximum.*

If the convergent-divergent duct (Fig. 7.5) acts as a nozzle, in the divergent part also, the pressure will fall continuously to yield a continuous rise in velocity. In other words, since $\bar{V} = \omega/\rho A$, for a certain mass flow rate ω with the decrease of pressure, density decreases at a rate faster than the rate at which area increases, as a result of which velocity continues to increase. This is true only for a compressible fluid. The velocity of the gas is subsonic before the throat, becomes sonic at the throat, and then supersonic till its exit, provided the exhaust pressure is low enough. The reverse situation prevails when the inlet velocity is supersonic. The whole duct then becomes a diffuser For an incompressible fluid, the convergent-divergent duct acts as a venturi, with the convergent part acting as the nozzle and the divergent part acting as the diffuser The maximum limiting speed of such a fluid is the sonic velocity.

When $M = 1$ occurs at the throat, the discharge is maximum and the nozzle is said to be *choked*. It is incapable of allowing more discharge even with further decrease in exhaust pressure (Fig. 7.6). This discharge is known as *critical discharge* and the properties at the throat under this condition are called *critical properties* designated by superscript asterisks (*).

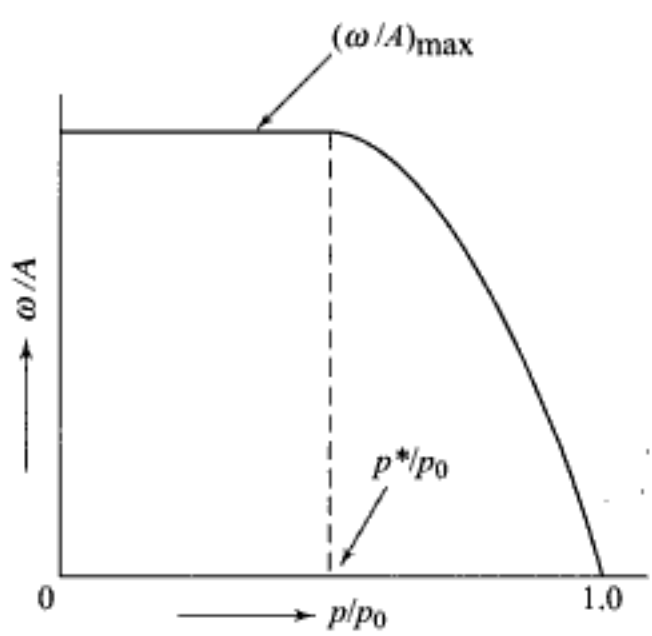

Fig. 7.6 *Choking of nozzle*

In a converging nozzle, the velocity at exit from the nozzle is sonic when the discharge through it is maximum, and the exit properties are critical properties. Substituting $M = 1$ in Eq. (7.5),

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$$\frac{T_0}{T^*} = 1 + \frac{\gamma - 1}{2} 1^2 = \frac{\gamma + 1}{2}$$

or

$$\frac{T^*}{T_0} = \frac{2}{\gamma + 1} \tag{7.18}$$

The ratio of the temperature at the throat where the velocity is sonic (T) to the inlet temperature (T_0) is called the critical temperature ratio.

The *critical pressure ratio*, p^*/p_0, defined as the ratio of the pressure at the throat to the inlet pressure, for choked flow when $M = 1$ occurs at the throat, is given by,

$$\frac{p^*}{p_0} = \left(\frac{2}{\gamma + 1}\right)^{\gamma/(\gamma - 1)} \tag{7.19}$$

Eqs (7.18) and (7.19) hold good only for an ideal gas. It is thus seen that the critical pressure ratio for an ideal gas depends only on the specific heat ratio γ. For air, $\gamma = 1.4$, and

$$\frac{p^*}{p_0} = \left(\frac{2}{1.4 + 1}\right)^{1.4/0.4} = 0.528$$

Hence, for air at 10 bar, a convergent nozzle requires a back pressure of 5.28 bar so that the flow is sonic at exit. For a correctly designed convergent-divergent nozzle with inlet pressure of 10 bar, the pressure at the throat is 5.28 bar for the maximum discharge. For helium, $\gamma = 1.67$ and

$$\frac{p^*}{p_0} = \left(\frac{2}{1.67 + 1}\right)^{1.67/0.67} = 0.487$$

Hence, for helium at 10 bar, a convergent nozzle requires a back pressure of 4.87 bar for sonic flow at exit, and the pressure at the throat of a convergent-divergent nozzle with inlet pressure of 10 bar is 4.87 bar.

The critical velocity at the throat of a convergent-divergent nozzle or at exit of convergent nozzle is given from Eq. (7.4),

$$\bar{V}^* = (\gamma R T^*)^{1/2} \tag{7.20}$$

For steam, Eqs (7.18), (7.19) and (7.20) are not strictly valid. However, a close approximation is obtained if it is assumed that steam expands isentropically in the nozzle following the law pv^k = constant, where k is the isentropic index and it is not a ratio of specific heats as is γ. For steam initially dry, saturated, $k = 1.135$; for steam initially superheated, $k = 1.3$.
Eq. (7.19) can then be written as

$$\frac{p^*}{p_0} = \left(\frac{2}{k + 1}\right)^{k/(k - 1)} \tag{7.21}$$

The proof of this equation in a steam nozzle is shown in the next Section. Therefore, when the steam entering a nozzle is dry, saturated,

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$$\frac{p^*}{p_0} = \left(\frac{2}{1.135+1}\right)^{1.135/0.135} = 0.577$$

and when the steam entering a nozzle is superheated,

$$\frac{p^*}{p_0} = \left(\frac{2}{1.3+1}\right)^{1.3/0.3} = 0.546$$

For wet steam, Zeuner's relation

$$k = 1.035 + 0.1\,x,$$

where x is the quality of steam, may be used. When $x = 1$, k becomes 1.135, as given earlier for dry, saturated steam.

7.2.6 Flow in Steam Nozzle

Let us consider the flow of steam through a convergent-divergent nozzle (Fig. 7.7). If the flow is adiabatic, $Q = 0$, and since there is no work interaction involved, $W = 0$, and neglecting the potential energy change, the steady flow energy equation for the nozzle (Fig. 7.7 a) reduces to

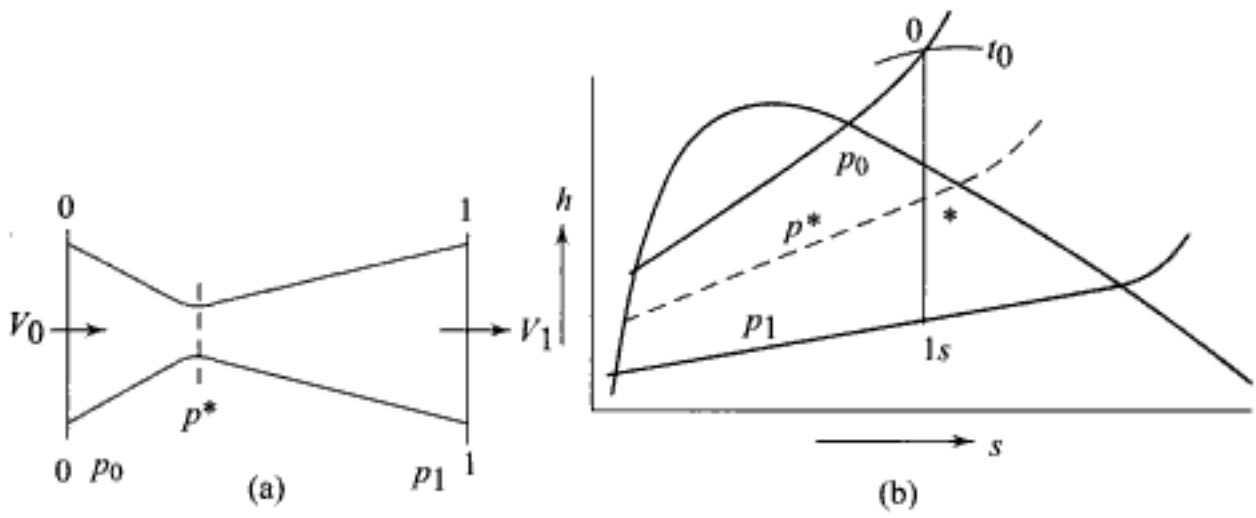

Fig. 7.7 *Expansion of steam in a convergent divergent nozzle*

$$h_0 + \frac{\bar{V}_0^2}{2} = h_1 + \frac{\bar{V}_1^2}{2} \tag{7.22}$$

If the velocity of approach, i.e. the inlet velocity $\bar{V}_0$ is small, the velocity at exit from the nozzle is given by,

$$\bar{V}_1 = [2(h_0 - h_1) \times 10^3]^{1/2} = 44.72(h_0 - h_1)^{1/2} \text{ m/s} \tag{7.23}$$

where $(h_0 - h_1)$ is the enthalpy drop across the nozzle in kJ/kg.
For isentropic flow in a nozzle.

$$Tds = dh - vdp = 0$$

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or,
$$\int_0^1 \mathrm{d}h = \int_0^1 v\,\mathrm{d}p$$

$$h_0 - h_1 = -\int_0^1 v\,\mathrm{d}p \tag{7.24}$$

From Eq. (7.22),

$$h_0 - h_1 = \frac{\bar{V}_1^2 - \bar{V}_0^2}{2} = -\int_0^1 v\,\mathrm{d}p$$

Assuming that the pressure and volume of steam during expansion obey the law pv^k = constant, where k is the isentropic index, and neglecting the inlet velocity V_0.

$$\frac{\bar{V}_1^2}{2} = -\int_0^1 \left(p_0 v_0^k\right)^{1/k} p^{-1/k}\mathrm{d}p = -\int_0^1 \left(p_1 v_1^k\right)^{1/k} p^{-1/k}\mathrm{d}p$$

$$= \frac{k}{k-1}(p_0 v_0 - p_1 v_1)$$

$$= \frac{k}{k-1} p_0 v_0 \left[1 - \left(\frac{p_1}{p_0}\right)^{(k-1)/k}\right]$$

or
$$\bar{V}_1 = \left[\frac{2k}{k-1} p_0 v_0 \left\{1 - \left(\frac{p_1}{p_0}\right)^{(k-1)/k}\right\}\right]^{1/2} \tag{7.25}$$

The rate of flow of steam is given by

$$\omega = \frac{A_1 \bar{V}_1}{v_1} = A_1 \left[\frac{2k}{k-1}\frac{p_0 v_0}{v_1^2}\left\{1 - \left(\frac{p_1}{p_0}\right)^{(k-1)/k}\right\}\right]^{1/2}$$

$$\frac{\omega}{A_1} = \left[\frac{2k}{k-1}\frac{p_0}{v_0}\left\{\left(\frac{p_1}{p_0}\right)^{2/k} - \left(\frac{p_1}{p_0}\right)^{(k+1)/k}\right\}\right]^{1/2} \tag{7.26}$$

By analogy, changing A_1 to A and p_1 to p,

$$\frac{\omega}{A} = \left[\frac{2k}{k-1}\frac{p_0}{v_0}\left\{\left(\frac{p}{p_0}\right)^{2/k} - \left(\frac{p}{p_0}\right)^{(k+1)/k}\right\}\right]^{1/2} \tag{7.27}$$

Equation (7.27) shows that for given values of p_0 and v_0, the mass flow rate of steam depends upon the pressure ratio p/p_0. There is a certain pressure ratio for which discharge (ω/A) is maximum. Therefore, by differentiating the term in square brackets and equating it to zero,

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$$\frac{d}{d(p/p_0)}\left[\left(\frac{p}{p_0}\right)^{2/k} - \left(\frac{p}{p_0}\right)^{(k+1)/k}\right] = 0$$

or
$$\frac{2}{k}\left(\frac{p}{p_0}\right)^{(2-k)/k} - \frac{k+1}{k}\left(\frac{p}{p_0}\right)^{1/k} = 0$$

$\therefore$
$$\frac{p^*}{p_0} = \left(\frac{2}{k+1}\right)^{k/(k-1)}$$

This is the same equation as given in Eq. (7.21). This is the critical pressure ratio.

If $p_{exit} < p^*$ a convergent-divergent nozzle is used, and if $p_{exit} > p^*$, a convergent nozzle is used.

The velocity of steam at the throat or the critical velocity is given by

$$V^* = 44.72\,(h_0 - h^*)^{1/2} \tag{7.28}$$

where $(h_0 - h^*)$ is the enthalpy drop of steam in the converging portion of the nozzle. The enthalpy drops $(h_0 - h_1)$ and $(h_0 - h^*)$ are read from the Mollier (or h–s) chart.

If the nozzle is only convergent, the minimum pressure to which steam can expand is the critical pressure when the discharge is maximum and the exit velocity of steam is the sonic velocity, which is given by Eq. (7.28).

The critical velocity can also be obtained from Eq. (7.25) on substituting the critical pressure ratio,

$$\bar{V}^* = \left[\frac{2k}{k-1}\,p_0 v_0\left\{1 - \frac{2}{k+1}\right\}\right]^{1/2}$$

$$= \left[\frac{2k}{k+1}\,p_0 v_0\right]^{1/2} \tag{7.29}$$

For superheated steam $k = 1.3$,

$$\bar{V}^* = 1.06\,(p_0 v_0)^{1/2} \tag{7.30}$$

For dry, saturated steam, $k = 1.135$

$$\bar{V}^* = 1.03\,(p_0 v_0)^{1/2} \tag{7.31}$$

where p_0 is in the N/m^2 and v is in m^3/kg. (7.31)

7.2.7 Nozzle Efficiency

Due to friction between the fluid and walls of the nozzle, the expansion process is irreversible, although still approximately adiabatic. In nozzle design, the usual practice is to base all calculations on isentropic flow and then to make an allowance for friction using a coefficient or efficiency.

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The nozzle efficiency, η_n, is defined as the ratio of the actual enthalpy drop to the isentropic enthalpy drop (Fig. 7.8),

$$\eta_n = \frac{h_0 - h_1}{h_0 - h_{1s}} \tag{7.32}$$

For an ideal gas,

$$\eta_n = \frac{c_p (T_0 - T_1)}{c_p (T_0 - T_{1s})} = \frac{T_0 - T_1}{T_0 - T_{1s}} \tag{7.33}$$

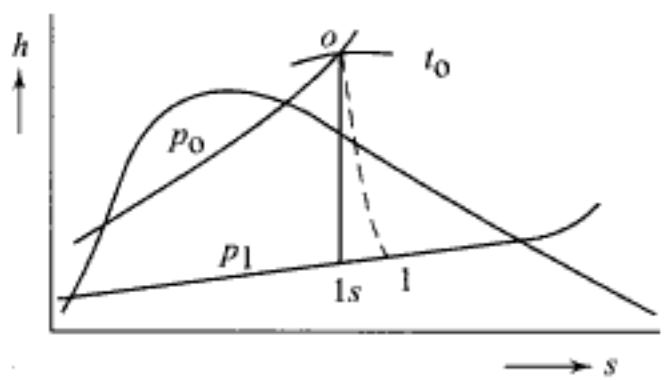

Fig. 7.8 *Nozzle efficiency*

If the actual velocity at exit from the nozzle is $\bar{V}_1$ and the velocity at exit when the flow is isentropic is $\bar{V}_{1s}$, then

$$\eta_n = \frac{\bar{V}_1^2 - \bar{V}_0^2}{\bar{V}_{1s}^2 - \bar{V}_0^2} \tag{7.34}$$

When the inlet velocity is very small, then

$$\eta_n = \frac{\bar{V}_1^2}{\bar{V}_{1s}^2} \tag{7.35}$$

Sometimes, a velocity coefficient ϕ is defined as the ratio of actual exit velocity to the exit velocity when the flow is isentropic between the same pressures,

$$\phi = \frac{\bar{V}_1}{\bar{V}_{1s}} \tag{7.36}$$

Therefore, $\phi = (\eta_n)^{1/2}$

Another coefficient which is frequently used is the ratio of the actual mass flow through the nozzle, ω, to the mass flow which would be passed if the flow were isentropic, ω_s. This is called the coefficient of discharge, C_d

$$C_d = \frac{\omega}{\omega_s} \tag{7.37}$$

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The convergent part of a nozzle is usually sharp, while the divergent part is gradual. Most of the friction loss occurs in the divergent portion of the nozzle. If the semi-divergence angle α is large (Fig. 7.9), there will be flow separation from the wall with the formation of eddies, which entails energy loss. If the angle α is small, the length of nozzle becomes large to provide the desired exit flow area, causing more energy loss due to friction. Usually, α varies from 5° to 8°.

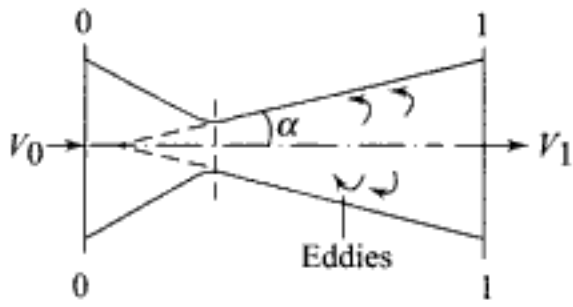

Fig. 7.9 *Semi-divergence angle a*

7.2.8 Nozzle Types

Probably the most common physical conception of the nozzle is the converging-diverging type, although its application in the turbine design is somewhat limited. More generally, the converging or subsonic nozzles are used. There are two types of nozzles:

1. Reamed or round nozzles
2. Foil nozzles

Reamed nozzles are used principally in the high pressure impulse stages of steam turbines. The principal advantages of the reamed nozzle are low cost, ease of manufacture, and adaptability to standardization. The principal disadvantages are lower efficiency, somewhat greater length, and an inability to utilize effectively for flow purposes the area of a given flow annulus. The angle of divergence in reamed nozzles, as mentioned earlier, is usually about 12° and rarely over 15°, in order to avoid flow separation with its attendant loss. The converging part is normally sharp and a good fillet or rounded entrance is provided.

The foil nozzle is formed by curved airfoil sections. The nozzle is short with well-rounded entrance edges and sharp exit edges affording a good issuing jet. These nozzles have high efficiency and are costlier. A foil nozzle block is made by welding the individual foils between sections of concentric rings. The block is then welded in place between the outer and inner shrouds. The foil nozzle finds wide application in large steam turbines.

7.2.9 Flow Area of Nozzle

If the condition of steam at inlet to the nozzle is known, the critical pressure p^* can be estimated by multiplying the inlet pressure p_0 by 0.546 or 0.577

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depending on whether the inlet steam is superheated or dry saturated, respectively. If the exit pressure is given and the nozzle efficiency is known, the exit velocity of steam, as well as the critical velocity of steam at the throat can be estimated from

$$\bar{V}_1 = 44.72\,[(h_0 - h_{1s})\eta_n]^{1/2}$$

and

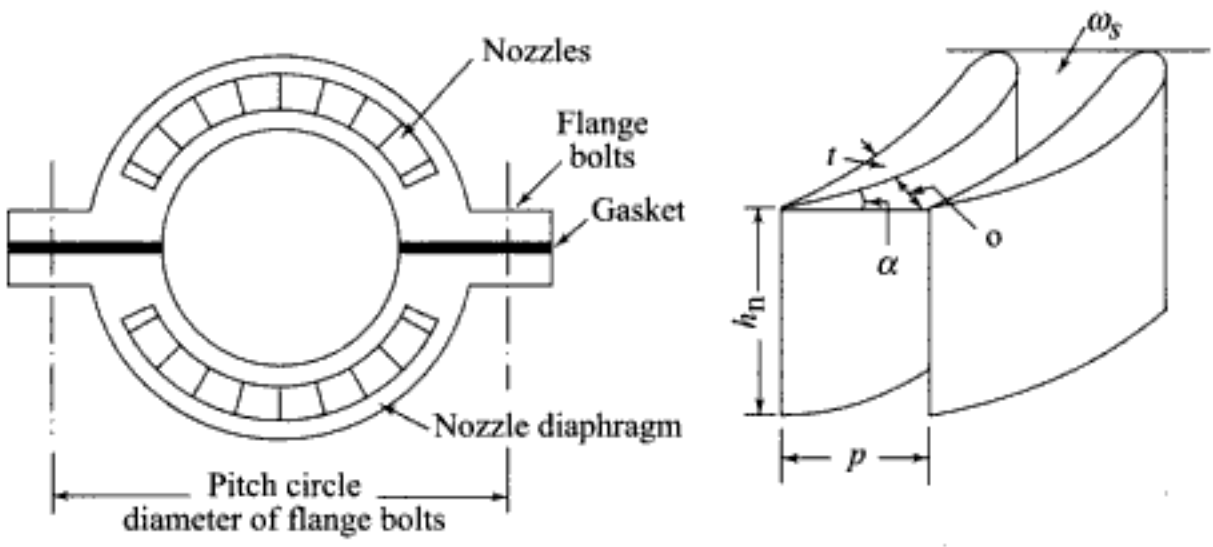
$$\bar{V}^* = 44.72\,[(h_0 - h_s^*)\eta_n]^{1/2}$$

where the enthalpy drops are read from the Mollier chart (kJ/kg). With the actual states of steam at exit and the throat being fixed on the Mollier diagram, the specific volumes can be obtained from it. Knowing the mass flow rate of steam (ω), the flow areas required at the nozzle exit, as well as at the throat can be estimated from

$$\omega = \frac{A^*\bar{V}^*}{v^*} = \frac{A_1\bar{V}_1}{v_1} \tag{7.38}$$

$$\therefore \qquad A^* = \frac{\omega v^*}{\bar{V}^*} \quad \text{and} \quad A_1 = \frac{\omega v_1}{\bar{V}_1}$$

If the nozzles are circular in cross-section, $A_1 = n\frac{\pi}{4} \times d^2_{exit}$, where n is the number of nozzles and d_{exit} is the exit diameter of the nozzles. The cross-section may be rectangular or square also. Foil type nozzles are mounted in a nozzle diaphragm which is horizontally split into two halves and joined by vertical flange bolts (Fig. 7.10a). The top halves of the diaphragms, which form the stator, can be lifted by cranes thus exposing the rotor for inspection and repair, if needed.



(a) Split casing (b) Foil type nozzles

Fig. 7.10 *Horizontally split diaphragm with nozzles*

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If D_m = mean diameter of the nozzle diaphragm,
h_n = height of nozzles,
p = pitch of nozzles on the mean diameter,
o = minimum opening for flow,
t = thickness of nozzles,
n = number of nozzles,

and α = nozzle angle, the angle subtended by the nozzle axis with the plane of rotation of the wheel (Fig. 7.10b), then the minimum area of nozzle open for steam flow,

$$A_n = o \cdot h_n \cdot n \tag{7.39}$$

Now, $o = p \sin\alpha - t$ and $n = \dfrac{\pi D_m}{p}$,

$$\therefore \quad A_n = (p \sin\alpha - t)\, h_n \cdot \frac{\pi D_m}{p} = \pi D_m h_n \sin\alpha \left(1 - \frac{t}{p \sin\alpha}\right)$$

$$= \pi D_m h_n \sin\alpha \times k_m \tag{7.40}$$

where k_m is the nozzle thickness factor.

7.2.10 Nozzles Operating in the Off-Design Condition

Let us consider a convergent nozzle as shown in Fig. 7.11, which also shows the pressure ratio p/p_0 varying along the length of the nozzle. The inlet condition of the fluid is the stagnation state at p_0, T_0, which is assumed to be constant. The pressure at the exit plane of the nozzle is denoted by p_E and the back pressure is p_B, which can be varied by a valve. As the back pressure p_B is varied, the mass flow rate ω and the exit plane pressure p_E/p_0 also vary, as shown in Fig. 7.12.

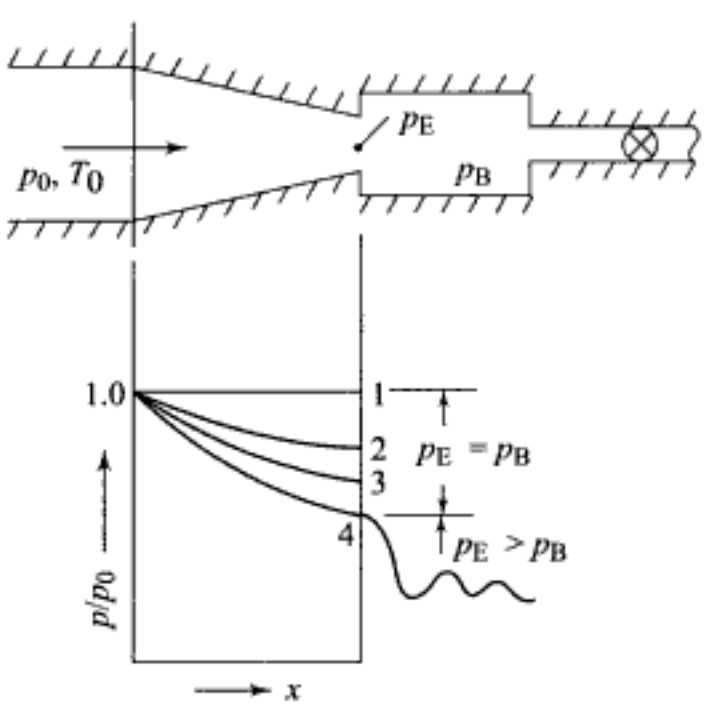

Fig. 7.11 *Pressure distribution in a convergent nozzle*

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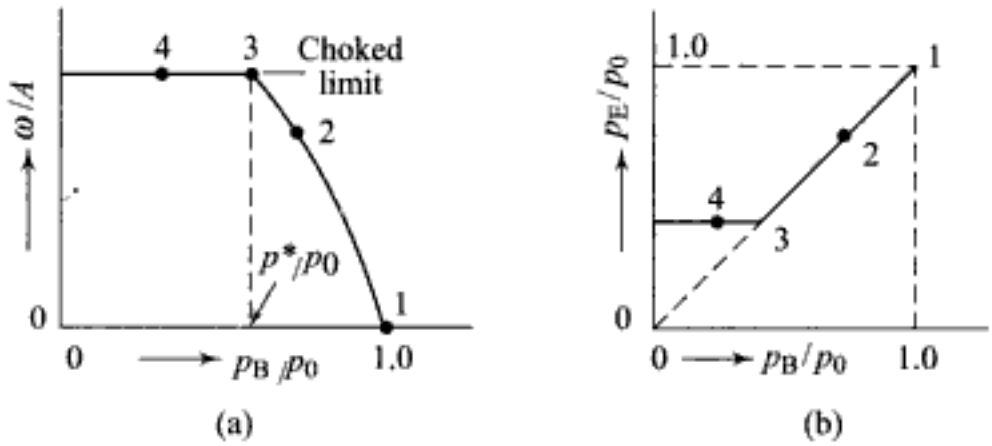

Fig. 7.12 *Mass flow rate and exit pressure as a function of back pressure in a convergent nozzle*

When $p_B/p_0 = 1$, there is no flow and $p_E/p_0 = 1$, as designated by point 1. If the back pressure is now decreased to 2, such that $p_B/p_0 > p^*/p_0$, $p_E = p_B$ and $M_E < 1$. Now, the back pressure is lowered to the critical pressure, point 3, when $M_E = 1$ and $p_E = p_B$. When p_B is decreased below the critical pressure, point 4, there is no increase in the mass flow rate, and $p_E = p^*$, $M_E = 1$. The drop in pressure from p_E to p_B occurs outside the nozzle exit. This is the choking limit which means that for given stagnation conditions the nozzle is passing the maximum possible mass flow.

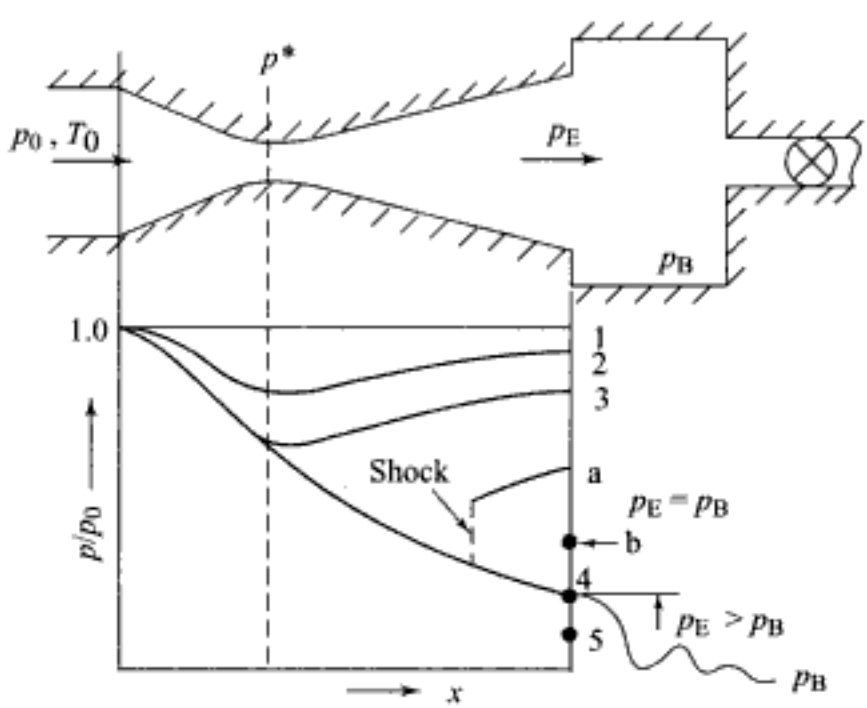

Fig. 7.13 *Pressure distribution in a convergent-divergent nozzle*

Let us next consider a convergent–divergent nozzle, as shown in Fig. 7.13. Point 1 designates the condition when $p_B = p_0$ and there is no flow. When p_B is lowered to the pressure denoted by 2, so that $p_B/p_0 < 1$, but $p_B/p_0 > p^*/p_0$, the velocity increases in the convergent section, but $M < 1$ occurs at the throat. The divergent section acts as the subsonic diffuser in which pressure increases and velocity decreases. Point 3 indicates the back pressure at which $M = 1$ occurs at the throat but the diverging section acts as a subsonic diffuser. Point 4

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indicates one other back pressure for which the flow is isentropic throughout and the diverging section acts as a supersonic nozzle with a continuous decrease in pressure and a continuous increase in velocity, and $p_{E_4} = p_{B_4}$. This condition of supersonic flow indicates the design pressure ratio of the nozzle. If the pressure ratio is lowered to 5, no further decrease in exit pressure occurs and the drop of pressure from p_E to p_B occurs outside the nozzle.

Between the back pressure between 3 and 4, flow is not isentropic in the diverging part, and it is accompanied by an irreversible phenomenon known as shocks. Shocks occur only when the flow is supersonic, and after the shock the flow becomes subsonic and the rest of the duct acts as a diffusor. Properties vary discontinuously across the shock. When the back pressure is as indicated by point *b*, the flow throughout the nozzle is isentropic, with pressure continuously decreasing the velocity increasing, but a shock appears just at the exist of the nozzle. When the back pressure is increased from *b* to *a*, the shock moves upstream as indicted. When the back pressure is further increased, the shock moves upstream and disappears at the nozzle throat where the back pressure corresponds to 3. Since the flow throughout is subsonic, no shock is possible.

7.2.11 Supersaturated Flow

When a superheated vapour expands slowly and isentropically, condensation within the vapour begins to form when the saturated vapour line is reached. As the expansion continues below this line into the wet region, condensation proceeds gradually with the progressive decrease of quality (x) and increase in the degree of wetness. Point *B* represents the point at which condensation within the vapour just begins (Fig. 7.14).

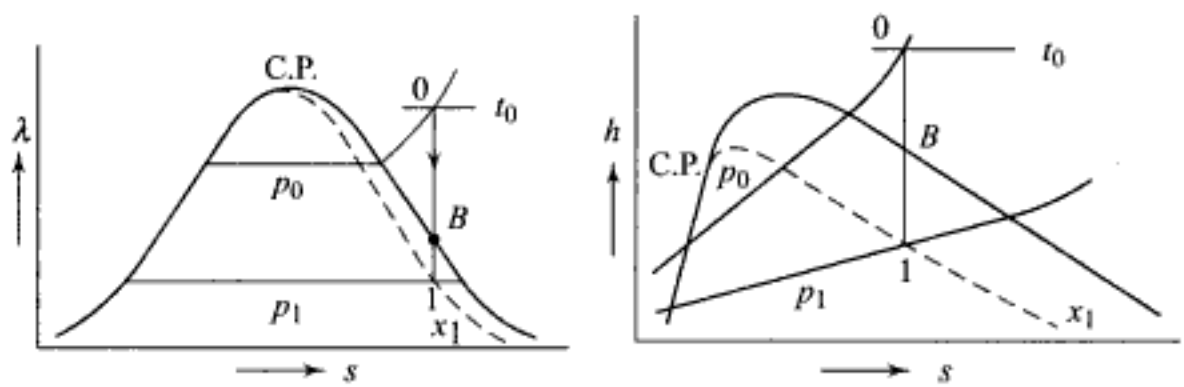

Fig. 7.14 *Slow and isentropic expansion of steam from the superheated into the two phase region*

However, when steam expands from superheated state to the two-phase wet region in a nozzle, the expansion occurs so rapidly that the vapour does not condense immediately as it crosses the dry saturated line, but somewhat later (at x = 0.96 to 0.97), when all the vapour suddenly condenses into liquid. Beyond the dry saturation line till the state when the vapour condenses, the flow is said to be *supersaturated* and the system is in *metastable equilibrium*, which means that it is stable to small disturbances but unstable to large disturbances. *Wilson line* (x = 0.96 to 0.97) is the locus of states below the dry saturation line where condensation within the vapour occurs at different pressures (Fig. 7.15).

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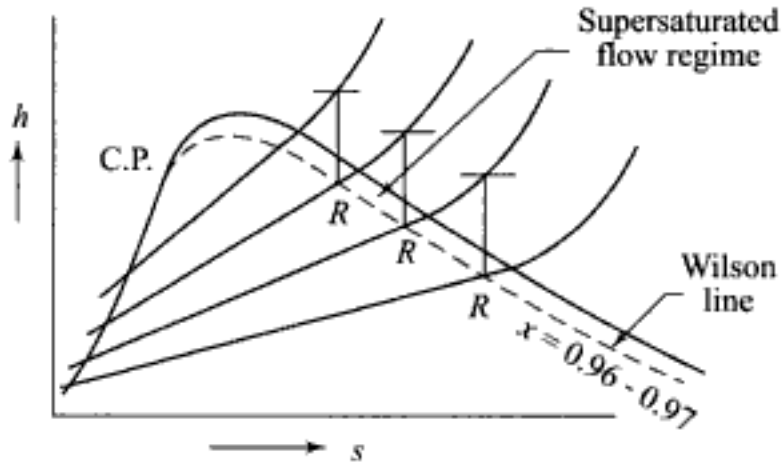

Fig. 7.15 *Wilson line*

In Fig. 7.16 (a and b), line 0–1 represents insentropic expansion (slow) with equilibrium at every stage of the expansion path. Line 0–*R* represents supersaturated expansion in which the vapour expands as if the saturated vapour line does not exist. It can be seen that the temperature of the supersaturated vapour at p_1 is t_R, which is less than the saturation temperature t_1 corresponding the pressure p_1. The vapour is said to be *supercooled*, where the *degree of supercooling* $= t_1 - t_R$. Sometimes a *degree of supersaturation* is defined as the ratio of the actual pressure p_1 to the saturation pressure at t_R (i.e. p_1/p_R)

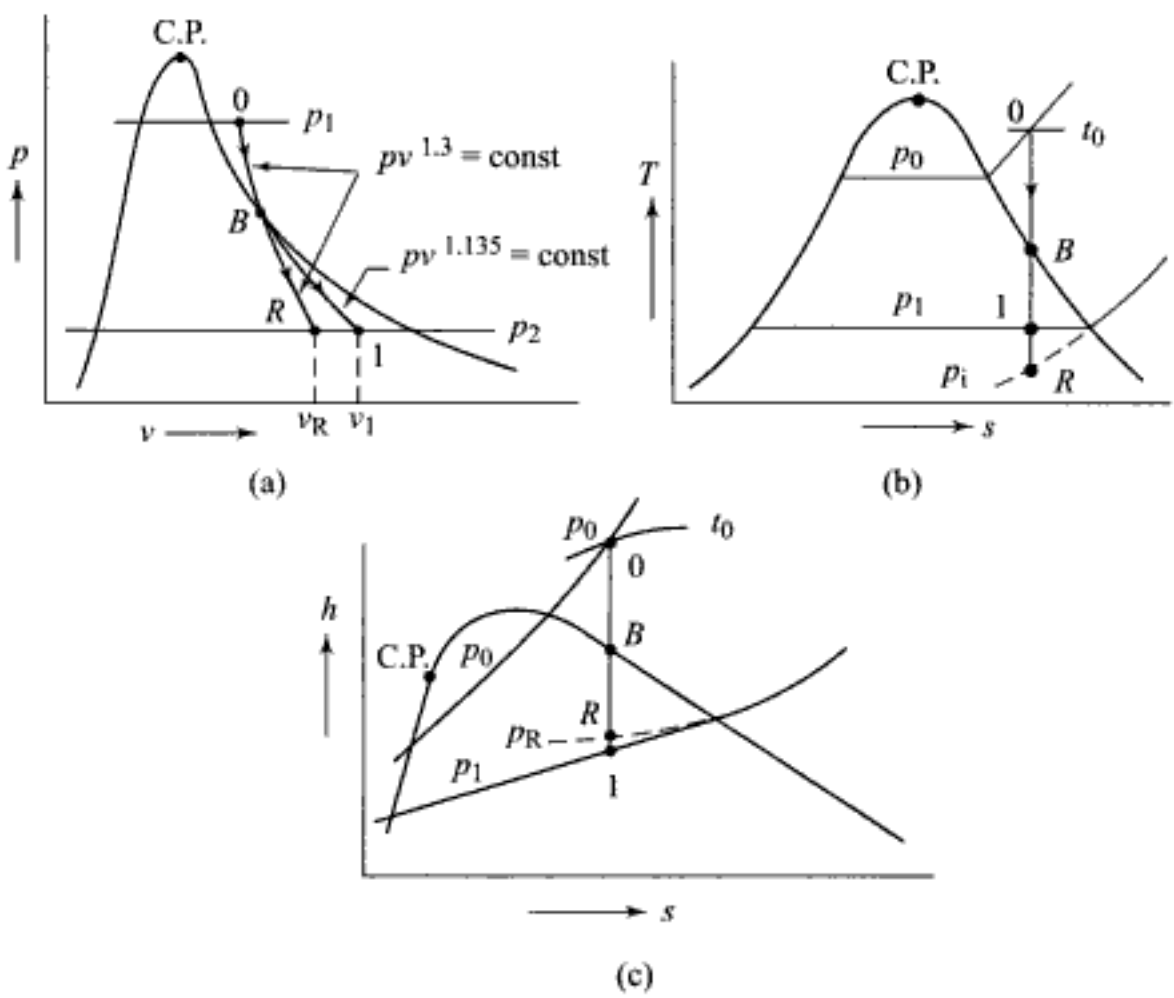

Fig. 7.16 *Supersaturated flow on (a) p–v, (b) T–s and (c) h–s diagrams*

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The exit velocity for insentropic flow is $\overline{V}_1 = 44.72\,(h_0 - h_1)^{1/2}$, whereas the exit velocity for supersaturated flow is $\overline{V}_{1sup} = 44.72\,(h_0 - h_R)^{1/2}$, which is less but not to a great extent since the difference in enthalpy drop is small.

If the expansion from 0 to B obeys the law $pv^{1.3}$ = constant, and that from B to 1 obeys the law $pv^{1.135}$ = constant, the isentropic and supersaturated expansion lines are shown on a p–v diagram in Fig. 7.16 (a). It is seen that the specific volume at exit with supersaturated flow, v_R, is considerably less than the specific volume at exit with isentropic flow, v_1. Now, the mass flow of steam through a given exit area A_1 is given by:

(a) For isentropic flow, $$\omega = \frac{A_1\overline{V}_1}{v_1}$$

(b) For supersaturated flow, $$\omega_{sup} = \frac{A_1\overline{V}_{1sup}}{v_R}$$

Since $\overline{V}_1 = \overline{V}_{1sup}$ and $v_1 > v_R$,

$\therefore$ $\omega < \omega_{sup}$

The mass flow with supersaturated flow is greater than the mass flow with isentropic flow. This fact, proved experimentally, led to the discovery of the phenomenon of supersaturation.

7.3 TURBINE BLADING

Depending upon the types of blades used and the method of energy transfer from the fluid to the rotor wheel, the turbines may be of two types:

(a) Impulse turbines
(b) Reaction turbines

7.3.1 Impulse Turbines

In impulse turbines, all pressure drops of steam occur in the nozzles and there is no pressure drop as steam flows through the passage between two blades. In Fig. 7.17(a), steam enters the nozzles at pressure p_0 with velocity $\overline{V}_0$, undergoes expansion to pressure p_1 with velocity increased to $\overline{V}_1$. High velocity jets of steam impinge upon the blades with velocity $\overline{V}_1$, get deflected by an angle, and come out at a lower velocity $\overline{V}_2$ impressing a torque on the blades. The pressure of steam p_1 remains essentially constant as steam flows through the blades.

From the principle of conservation of momentum,
Momentum of steam jets at inlet to the blades – momentum of jets at exit from the blades (both resolved in the direction of motion of the wheel) = momentum (angular) absorbed by the wheel in producing shaft work.

In Figs 7.17 (b) to 7.17 (d), it is seen that the wheel rotates only due to the impulsive effect of the jets (i.e., the difference of momenta of the jets deflected by the blades). The blades of such a wheel are called impulse blades.

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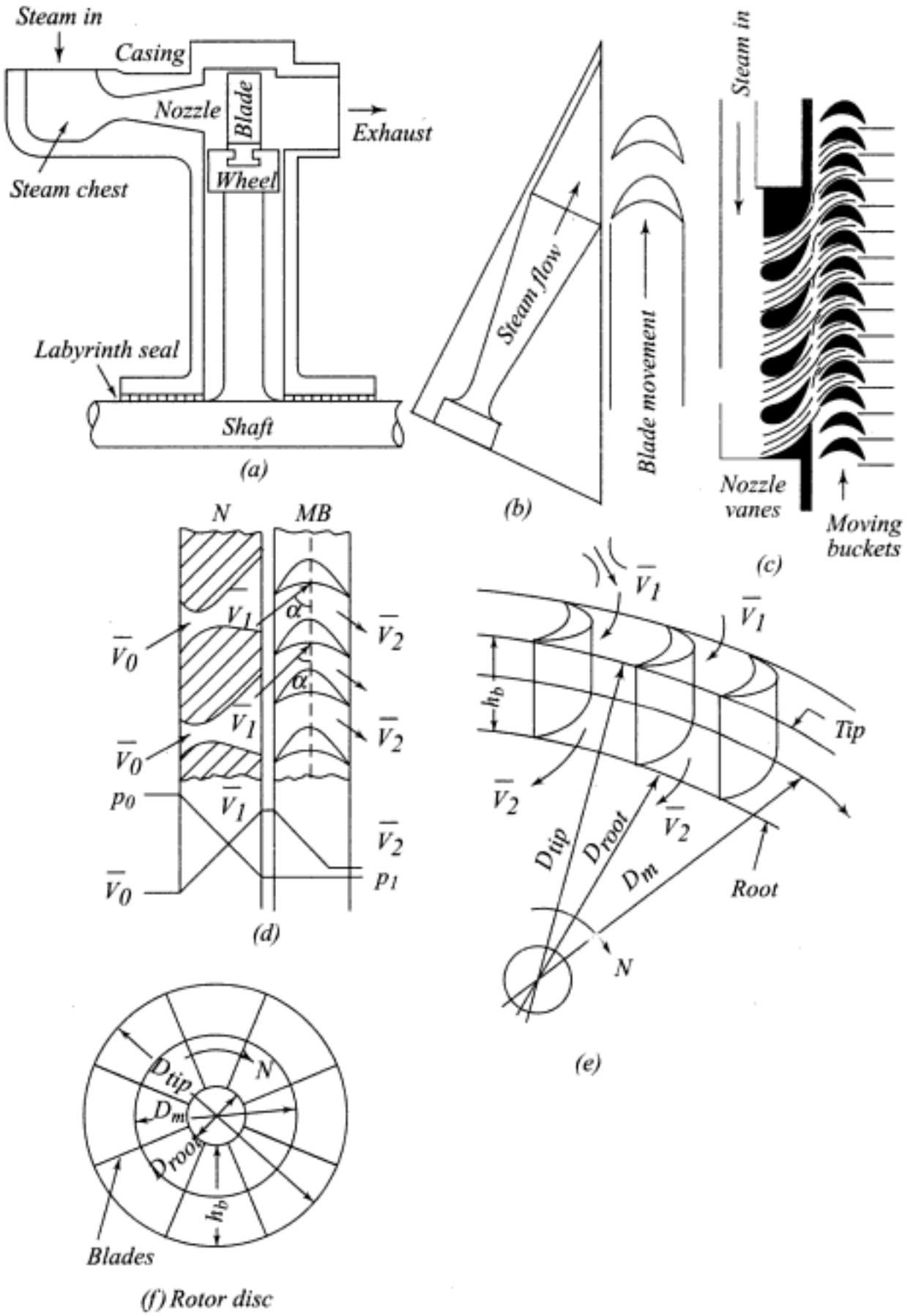

Fig. 7.17 *Flow through impulse blades*

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The mean peripheral velocity of the blades, also called the mean blade velocity, $\bar{V}_b$, is given by

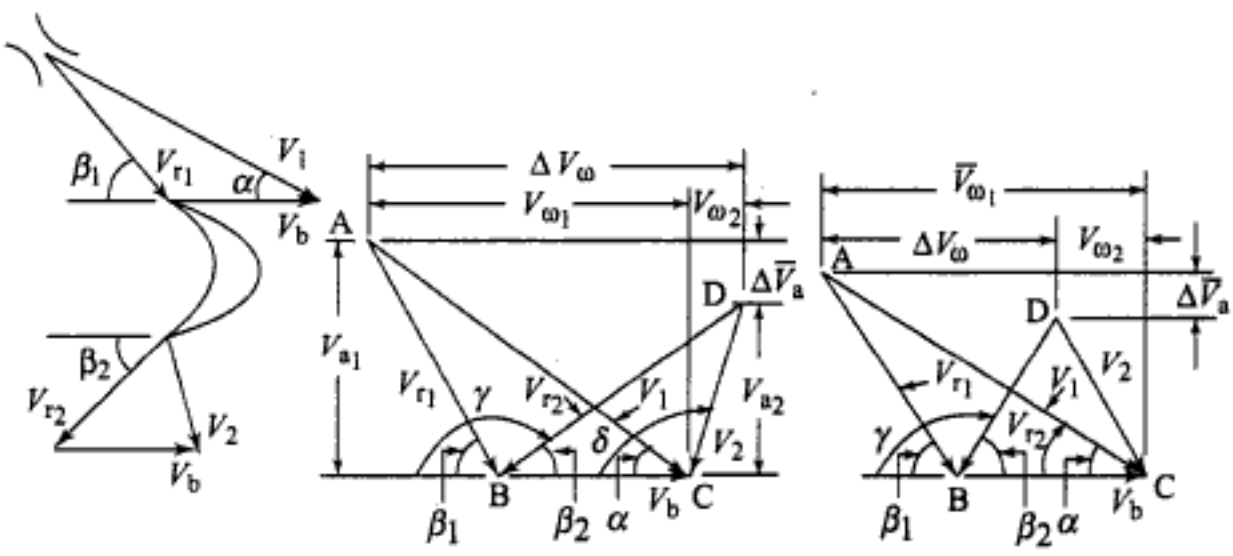
$$\bar{V}_b = \frac{\pi D_m N}{60}$$

where D_m is the mean diameter of the wheel and N is its rpm. The area of flow or blade annulus (Fig. 7.17(e)), A_b, is given to be

$$A_b = \frac{\pi}{4}\left(D^2{}_2 - D^2{}_1\right) = \pi \frac{D_2 + D_1}{2}\frac{D_2 - D_1}{2} = \pi D_m h_b \qquad (7.41)$$

where D_1 is the root diameter, D_2 is the tip diameter, and h_b is the height of the blades.

1. Velocity diagrams, diagram work and diagram efficiency Steam coming out from the nozzles at absolute velocity $\bar{V}_1$ strikes the blades with relative velocity $\bar{V}_{r_1}$ while the blades rotate with mean peripheral velocity $\bar{V}_b$. Steam leaves the blades with relative velocity $\bar{V}_{r_2}$ while its absolute velocity is $\bar{V}_2$. As shown in Fig. 7.18(a), α is the nozzle angle subtended by the nozzle axis with the direction of rotation of the wheel, β_1 is the inlet blade angle and β_2 is the exit blade angle. The inlet and exit velocity triangles of Fig. 7.18(a) have been superposed on a common $\bar{V}_b$ in Fig. 7.18(b). If all the angles are measured clockwise, then γ is the exit blade angle ($\gamma = 180 - \beta_2$) and δ is the angle made by absolute exit velocity of steam leaving the blades with the plane of rotation of the wheel.



(a) (b) $\delta > 90°$ (c) $\delta < 90°$

Fig. 7.18 *Velocity diagrams for impulse blading*

It is the difference of the cosine components of the velocities of steam which drives the wheel and produces the torque. It is called the change in the velocity of whirl, $\Delta\bar{V}_\omega$, which is given by

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$\Delta \bar{V}_\omega$ = velocity of whirl at inlet-velocity of whirl at outlet

$$= \bar{V}_{\omega_1} - \bar{V}_{\omega_2} = \bar{V}_1 \cos\alpha - \bar{V}_2 \cos\delta \tag{7.42}$$

If $\delta > 90°$, $\bar{V}_{\omega_1}$ and $\bar{V}_{\omega_2}$ become additive in estimating $\Delta \bar{V}_\omega$ (Fig. 7.18(b)). If $\delta < 90°$, $\bar{V}_{\omega_2}$ is to be substracted from $\bar{V}_{\omega_1}$ to determine $\Delta \bar{V}_\omega$ (Fig. 7.18(c)).

From the inlet velocity triangle *ABC* (Fig. 7.18 (b)).

$$\bar{V}_1 \cos\alpha - \bar{V}_b = \bar{V}_{r_1} \cos\beta_1$$

$$\bar{V}_1 \sin\alpha = \bar{V}_{r_1} \sin\beta_1$$

Therefore, on division,

$$\tan\beta_1 = \frac{\bar{V}_1 \sin\alpha}{\bar{V}_1 \cos\alpha - \bar{V}_b} \tag{7.43}$$

From the exit velocity triangle *DBC* (Fig. 7.18 (b))

$$\bar{V}_2 \cos(180 - \delta) + \bar{V}_b = \bar{V}_{r_2} \cos(180 - \gamma)$$

$$\bar{V}_b - \bar{V}_2 \cos\delta = -\bar{V}_{r_2} \cos\gamma$$

$$\therefore \quad \bar{V}_2 \cos\delta = \bar{V}_b + \bar{V}_{r_2} \cos\gamma \tag{7.44}$$

The same result can also be obtained from Fig. (7.18 (c)).

$$\bar{V}_b - \bar{V}_2 \cos\delta = \bar{V}_{r_2} \cos(180 - \gamma) = -\bar{V}_{r_2} \cos\gamma$$

$$\bar{V}_2 \cos\delta = \bar{V}_b + \bar{V}_{r_2} \cos\gamma \tag{7.45}$$

The ratio of relative velocity at exit to that at inlet is called "the blade friction factor", k_b,

$$k_b = \bar{V}_{r_2} / \bar{V}_{r_1} \tag{7.46}$$

The energy loss due to friction in the blades is equal to

$$\frac{\bar{V}_{r_1}^2 - \bar{V}_{r_2}^2}{2}$$

Now, from Eqs (7.42) and (7.45)

$$\Delta \bar{V}_\omega = \bar{V}_1 \cos\alpha - \bar{V}_2 \cos\delta = \bar{V}_1 \cos\alpha - \bar{V}_b - k_b \bar{V}_{r_1} \cos\gamma$$

$$= \bar{V}_1 \cos\alpha - \bar{V}_b - k_b \frac{\bar{V}_1 \cos\alpha - \bar{V}_b}{\cos\beta_1} \cos\gamma$$

$$= \left(\bar{V}_1 \cos\alpha - \bar{V}_b\right)\left[1 - k_b \frac{\cos\gamma}{\cos\beta_1}\right] \tag{7.47}$$

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Since $\gamma = 180 - \beta_1$; $\cos\gamma = -\cos\beta_1$,

$$\Delta\bar{V}_\omega = (\bar{V}_1 \cos\alpha - \bar{V}_b)(1 + k_b)$$

Also,
$$\Delta V_\omega = V_{r_1}\cos\beta_1 + V_{r_2}\cos\beta_2 = V_{r_1}\cos\beta_1 + k_b V_{r_1}\cos\beta_1$$
$$= V_{r_1}\cos\beta_1(1 + k_b) = (V_1\cos\alpha - V_b)(1 + k_b) \quad (7.48)$$

Blades are said to be *symmetrical* if blade angles are equal, i.e. if $\beta_1 = \beta_2$. Impulse turbines mostly have symmetrical blades.

The tangential thrust impressed by the jets on the blades, P_t, is

$$P_t = \omega_s \cdot \Delta\bar{V}_\omega \quad (7.49)$$

where ω_s is the steam flow rate.

The axial thrust, i.e. the thrust produced by the difference in the axial components of the velocities (along the axis of the shaft),

$$P_a = \omega_s \cdot \Delta\bar{V}_a \quad (7.50)$$

where $\Delta\bar{V}_a = V_{r1}\sin\beta_1 - V_{r2}\sin\beta_2$ (7.51)

The rate at which work is done by the jets on the blades is called the *blading work or diagram work*, W_D, which is given by

$$W_D = P_t \times \bar{V}_b = \omega_s \cdot \Delta\bar{V}_\omega \bar{V}_b \quad (7.52)$$

The energy input to the blades is the kinetic energy of jets issuing out from the nozzles $= \omega_s \cdot \left(\bar{V}_1^2/2\right)$.

Therefore, the *blading or diagram efficiency*

$$\eta_D = \frac{\text{rate of work done on the blades}}{\text{rate of energy input to the blades}} = \frac{\omega_s \Delta\bar{V}_\omega \bar{V}_b}{\frac{1}{2}\omega_s \bar{V}_1^2}$$

or
$$\eta_D = \frac{2\Delta\bar{V}_\omega \bar{V}_b}{\bar{V}_1^2} \quad (7.53)$$

It is the fraction of kinetic energy of jets of steam issuing from nozzles which is converted to shaft work.

2. Optimum velocity ratio Substituting $\Delta\bar{V}_\omega$ from Eq. (7.48) in Eq. (7.53), the diagram efficiency is given by

$$\eta_D = \frac{2(\bar{V}_1\cos\alpha - \bar{V}_b)(1 + k_b)\bar{V}_b}{\bar{V}_1^2} = \frac{2\bar{V}_b^2\left(\frac{\bar{V}_1\cos\alpha}{\bar{V}_b} - 1\right)(1 + k_b)}{\bar{V}_1^2}$$

Defining the *velocity ratio* ρ as $\bar{V}_b/\bar{V}_1$, the ratio of mean blade velocity V_b to the jet velocity $\bar{V}_1$,

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$$\eta_D = 2\rho^2\left(\frac{\cos\alpha}{\rho} - 1\right)(1 + k_b) = 2(\rho\cos\alpha - \rho^2)(1 + k_b) \qquad (7.54)$$

There is a certain value of ρ which makes η_D the maximum. Differentiating η_D with respect to ρ and making it equal to zero.

$$\frac{d\eta_D}{d\rho} = 2(\cos\alpha - 2\rho)(1 + k_b) = 0$$

Therefore, the optimum velocity ratio for impulse blading is

$$\rho_{opt} = \frac{\cos\alpha}{2} \qquad (7.55)$$

Substituting this optimum value of ρ in Eq. (7.54), the maximum blading efficiency is

$$(\eta_D)_{max} = 2\left[\frac{\cos^2\alpha}{2} - \frac{\cos^2\alpha}{4}\right](1 + k_b) = \frac{1 + k_b}{2}\cos^2\alpha \qquad (7.56)$$

If the energy loss due to friction in blades is small, $k_b \simeq 1$, then

$$(\eta_D)_{max} = \cos^2\alpha \qquad (7.57)$$

where α is the nozzle angle. The lower is the nozzle angle, higher is the blading efficiency. However, too low a nozzle angle may cause energy loss at blade inlet. Therefore, the nozzle angle has to be maintained within a certain range, which varies from 16° to 22°.

3. Graphical method The diagram work and the diagram efficiency under a certain operating condition can also be estimated graphically by drawing to scale the inlet and exit velocity triangles. If the state of steam at inlet to the nozzles, the exit pressure and nozzle efficiency are known, then

$$\bar{V}_1 = 44.72\,[(h_0 - h_{1s})\eta_n]^{1/2}$$

can be estimated. Knowing D_m and N, $\bar{V}_b$ is found out from

$$\bar{V}_b = \frac{\pi D_m N}{60}$$

With α and k_b being given and with the blades being symmetrical ($\beta_1 = \beta_2$),a scale for the diagram is chosen, say, 1 cm = 50 m/s. A horizontal line AB is drawn, the length of which is proportional to $\bar{V}_b$ (Fig. 7.19). A straight line is drawn from B making an angle α and a length BC proportional to $\bar{V}_1$ is cut off. CA is joined. The angle $\angle CAD$ is measured. This is the inlet blade angle β_1. CA is measured, which represents $\bar{V}_{r_1}$. Again, a straight line AE is drawn from A making an angle β_1 (= β_2) and a length AE is cut off proportional to $\bar{V}_{r_2}$ (= $k_b V_{r_1}$). EB is joined. The velocity diagram is thus completed. From this diagram, $\Delta\bar{V}_\omega$, $\bar{V}_{a_1}$ and $\bar{V}_{a_2}$ are measured. Then the desired particulars,

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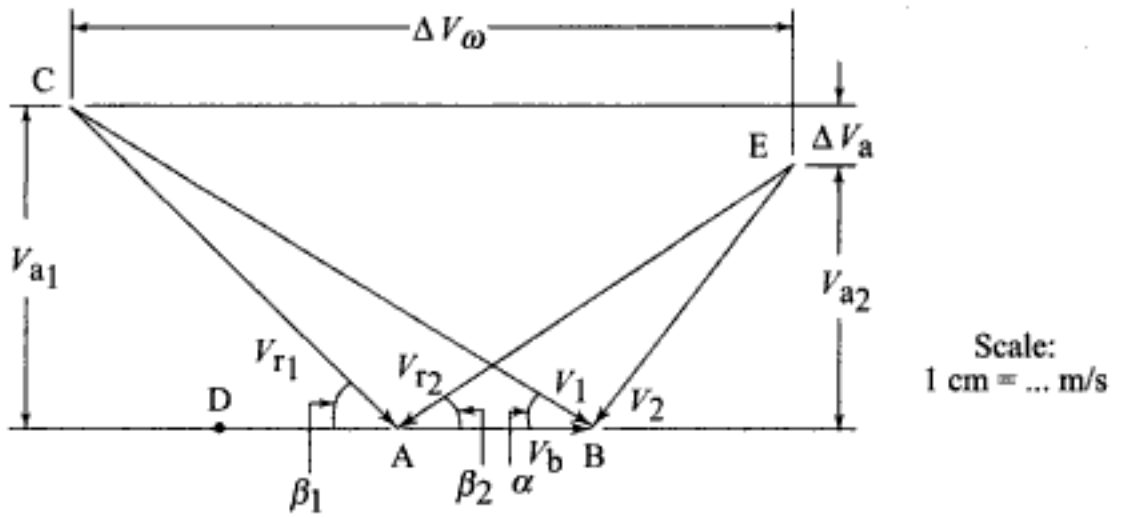

Fig. 7.19 *Construction of velocity diagram*

$P_t = \omega_s \Delta \bar{V}_\omega$; $P_a = \omega_s\ (\bar{V}_{a_1} - \bar{V}_{a_2})$; $W_D = P_t\ \bar{V}_b$; $\eta_D = (2\Delta\bar{V}_\omega \bar{V}_b)/\bar{V}_1^2$ are estimated.

4. Alternative way of drawing velocity diagrams The velocity diagrams with symmetrical impulse blading can also be drawn in a different way as shown in Fig. 7.20. The inlet velocity triangle *ABC* is drawn first as before. A length *CD* equal to ($k_b\ \bar{V}_{r_1}$) is cut off from *C* to represent $\bar{V}_{r_2}$ since $\beta_1 = \beta_2$. *ED* is drawn from *D*, the length being proportional to $\bar{V}_b$. *CE* is joined. The absolute exit velocity $\bar{V}_2$ is the vector sum of $\bar{V}_b$ and $\bar{V}_{r_2}$, as shown. Then, $\bar{V}_{\omega_1}, \bar{V}_{\omega_2}, \bar{V}_{a_1}$ and $\bar{V}_{a_2}$ are measured from the diagram to estimate the different output parameters.

If the blade friction factor k_b is unity, i.e. there is no energy loss due to friction as steam flows through the blades, and the discharge is axial, i.e. parallel to the axis of the shaft, for which $\delta = 90°$, the resultant velocity diagram becomes as shown in Fig. 7.21. The horizontal line *AB* ($= 2\bar{V}_b$) is drawn such that $AC = CB = \bar{V}_b$. From *A*, a perpendicular is drawn to *AB* and from *B*, a straight line is drawn making an angle α, which cuts the perpendicular at *D*. Then ΔDCA is the exit velocity triangle.

5. Compounding of steam turbines One row of nozzles followed by one row of blades is called a stage of a turbine. If steam is allowed to expand from boiler condition down to condenser vacuum in a single row of nozzles, due to a large enthalpy drop, the velocity at exit from the nozzles ($\bar{V}_1$) is very large (Fig. 7.22). Since $\bar{V}_b/\bar{V}_1 = \cos \alpha/2$, the blade velocity ($\bar{V}_b$) also becomes very large. Again $\bar{V}_b$ is equal to $\pi D_m N/60$. Therefore, if D_m is fixed at some economic value, *N* becomes very large. Thus single-stage impulse turbines, also called de Laval turbines, have inherently very high rotative speeds (*N*).

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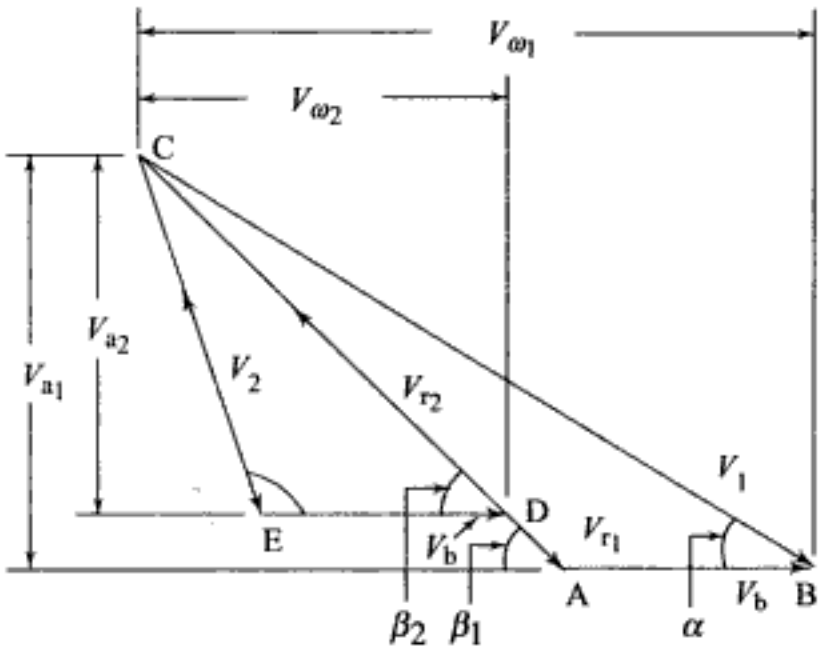

Fig. 7.20 *Velocity diagram with symmetrical impulse blading*

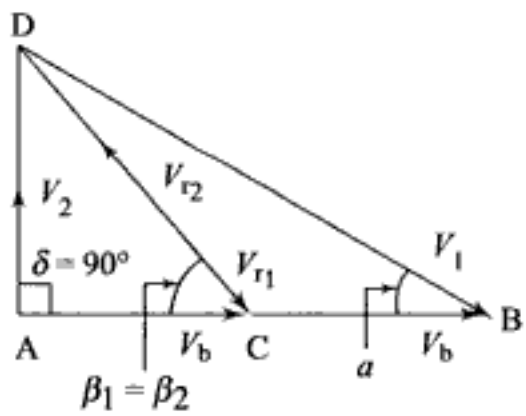

Fig. 7.21 *Velocity diagram for frictionless symmetrical impulse blading with axial discharge.*

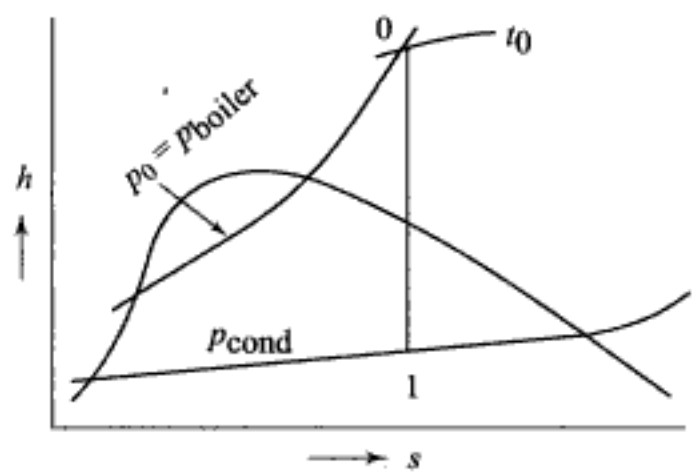

Fig. 7.22 *Single stage expansion of steam*

Such high rotational speeds cannot be properly utilized. It entails large friction losses. The centrifugal stresses also become very large. Alternatively, if

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N is fixed, the diameter of the wheel becomes too large. Moreover, with a single stage, the velocity of steam at exit is sufficiently high and there is a considerable loss of kinetic energy with exiting steam. To obviate these difficulties the turbines are compounded or staged, where steam instead of expanding in a single stage is made to expand in a number of stages, whereby the turbine speed is reduced while securing the same enthalpy drop of steam.

Basically, there are two ways of compounding steam turbines:

1. Pressure compounding or Rateau staging
2. Velocity compounding or Curtis staging

6. Pressure compounding or Rateau staging The pressure compounding or Rateau staging corresponds to putting a number of simple impulse stages in series (Fig. 7.23). The total enthalpy drop is divided equally among the stages (Fig. 7.24). The pressure drops only in the nozzles. There is no pressure drop (theoretically) while steam flows through the blades. The kinetic energy of steam increases in the nozzles at the expense of the pressure drop and and it is absorbed (partially) by the blades in each stage, in producing torque. In Fig. 7.24, the total isentropic enthalpy drop of steam $(h_0 - h_4)$ is divided equally among the four stages of the turbine. In Mollier diagram, the enthalpy drop $(h_0 - h_4)$ is measured, then the enthalpy drop, $h_0 - h_1 = h_1 - h_2 = h_2 - h_3 = h_3 - h_4 = (h_0 - h_4)/4$ are computed and inserted on the insentropic line. The interstage pressures noted from the diagram are p_1, p_2 and p_3, i.e. the pressure after first stage is p_1, the pressure after second stage is p_2 and so on.

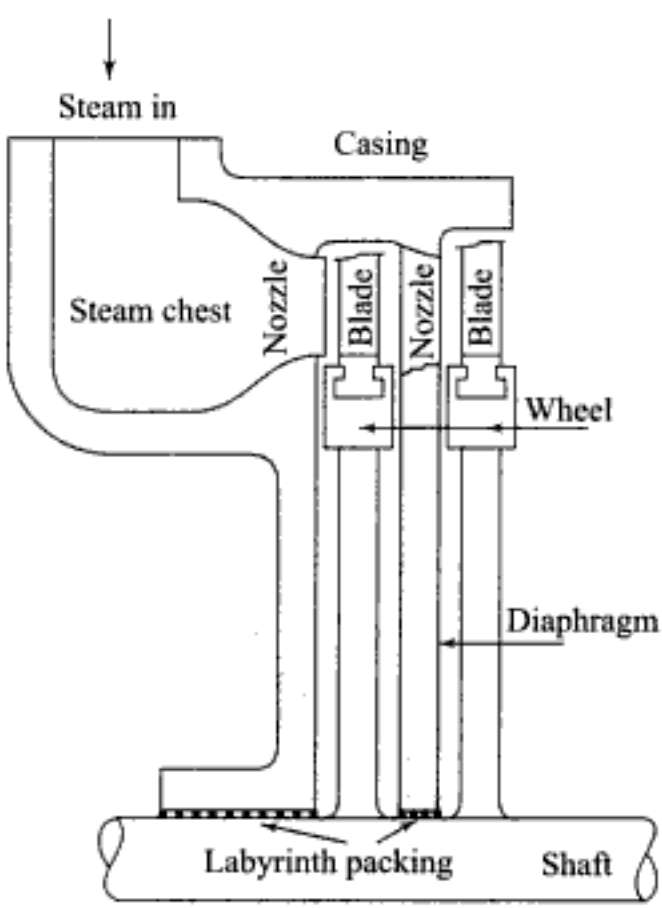

Fig. 7.23 (a) *Pressure-compounded impulse turbine*

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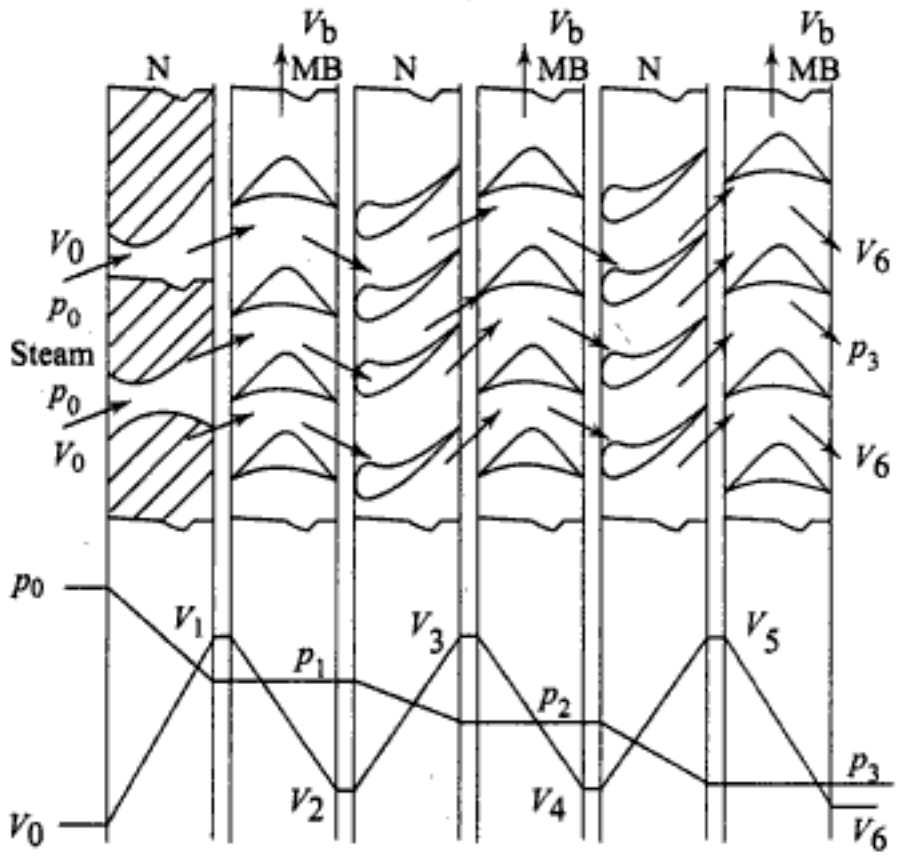

Fig. 7.23 (b) *Three pressure (or Rateau) stages in series*

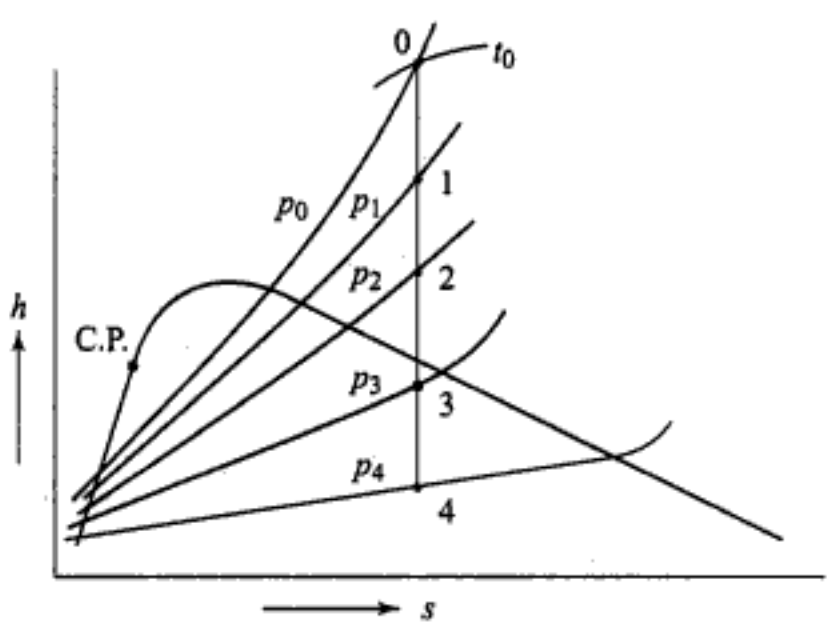

Fig. 7.24 *Enthalpy drop per stage in a 4-stage turbine*

The velocity of steam at exit from the first row of nozzles is

$$\bar{V}_1 = 44.72\,(h_0 - h_1)]^{1/2} = 44.72\,[(h_0 - h_4)/4]^{1/2}$$

or,

$$\bar{V}_1 = \frac{1}{2}\,(44.72)\,[(h_0 - h_4)]^{1/2} \tag{7.58}$$

If the KE of steam at inlet to each subsequent row of nozzles is small and neglected, then the velocity of steam leaving the nozzles in each row will be the same as given in Eq. (7.58). For a single-stage turbine the jet velocity would have been

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$$\bar{V}_1 = 44.72\,[h_0 - h_4]^{1/2} \tag{7.59}$$

Therefore, for a 4-stage turbine, the velocity of steam leaving the nozzles in each stage is half of that for a single-stage turbine. For a 9-stage turbine, it will be one-third.

For each impulse stage operating at its maximum blading efficiency, the blade velocity is given by

$$\bar{V}_b = \frac{\cos\alpha}{2}\bar{V}_1 \tag{7.60}$$

Therefore, along with $\bar{V}_1$, $\bar{V}_b$ also gets halved for a 4-stage turbine, and for a given D_m, N gets halved or for a given N, D_m gets halved. With more stages, N or D_m would further decrease.

If there are n stages in series in the turbine, the isentropic enthalpy drop per stage would be

$$(\Delta h)_{stage} = \frac{\Delta h_{total}}{n} \tag{7.61}$$

From Eq. (7.60), and for an impulse stage,

$$\bar{V}_1 = \frac{2}{\cos\alpha}\bar{V}_b = 44.72\,[\Delta h_{stage}]^{1/2}$$

$$\therefore \qquad (\Delta h)_{stage} = \left[\frac{2\bar{V}_b}{44.72\cos\alpha}\right]^2 = 4\left[\frac{\bar{V}_b}{44.72\cos\alpha}\right]^2 \tag{7.62}$$

From Eqs (7.61) and (7.62), the number of impulse stages required for a certain enthalpy drop $(\Delta h)_{total}$ can thus be estimated under ideal condition, i.e.,

$$n = \frac{(\Delta h_s)_{total}}{(\Delta h_s)_{stage}}$$

7. Velocity compounding or Curtis staging In velocity compounding or Curtis staging, all the pressure drop and hence, enthalpy drop of steam take place in a single row of nozzles and the resultant kinetic energy of steam is absorbed by the wheel in a number of rows of moving blades with guide blades in between two such rows. As shown in Fig. 7.25, steam expands in the single row of nozzles from p_0 to p_1 with velocity increasing from $\bar{V}_0$ to $\bar{V}_1$. The pressure p_1 remains essentially constant thereafter. The KE of steam jets $\left(\frac{1}{2}\omega_s\bar{V}_1^2\right)$ at nozzle exit is partially converted to shaft work in the first row of moving blades with velocity decreasing from $\bar{V}_1$ to $\bar{V}_2$. The exiting steam jets are then deflected by the stationary guide blades to the next row of moving blades where part of the remaining kinetic energy $\left(\frac{1}{2}\omega_s\bar{V}_2^2\right)$ is converted to shaft work. It is a two-row Curtis or velocity stage having two rows of moving blades with one row of guide blades in between. In a three row Curtis stage, the two-row stage is followed by a second row of guide blades and then a third row

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or moving blades, so that the energy conversion from fluid to rotor takes place in three stages. Steam with high KE exiting the nozzles works on the first row of moving blades, gets deflected by the first row of guide blades, works on the second row of moving blades, again gets deflected by the second row of guide blades and finally does work on the third row of moving blades.

The velocity diagrams for the first row of moving blades and the second row of moving blades are shown respectively in (a) and (b) of Fig. 7.26. The blade friction factor k_b may be assumed the same for both moving and guide blades.

Here,
$$\frac{\bar{V}_{r_2}}{\bar{V}_{r_1}} = \frac{\bar{V}_{r_3}}{\bar{V}_2} = \frac{\bar{V}_{r_4}}{\bar{V}_{r_3}} = k_b$$

α_1 = exit angle of the guide blades

β_1, β_2 = inlet and exit angles of the first row of moving blades

β_3, β_4 = inlet and exit angles of the second row of moving blades

$\Delta\bar{V}_{\omega_1}, \Delta\bar{V}_{\omega_2}$ = changes in the velocity of whirl in the first and second rows of moving blades

$\Delta V_{a1}, \Delta V_{a2}$ = changes in the axial components of velocity in the first and second rows of moving blades.

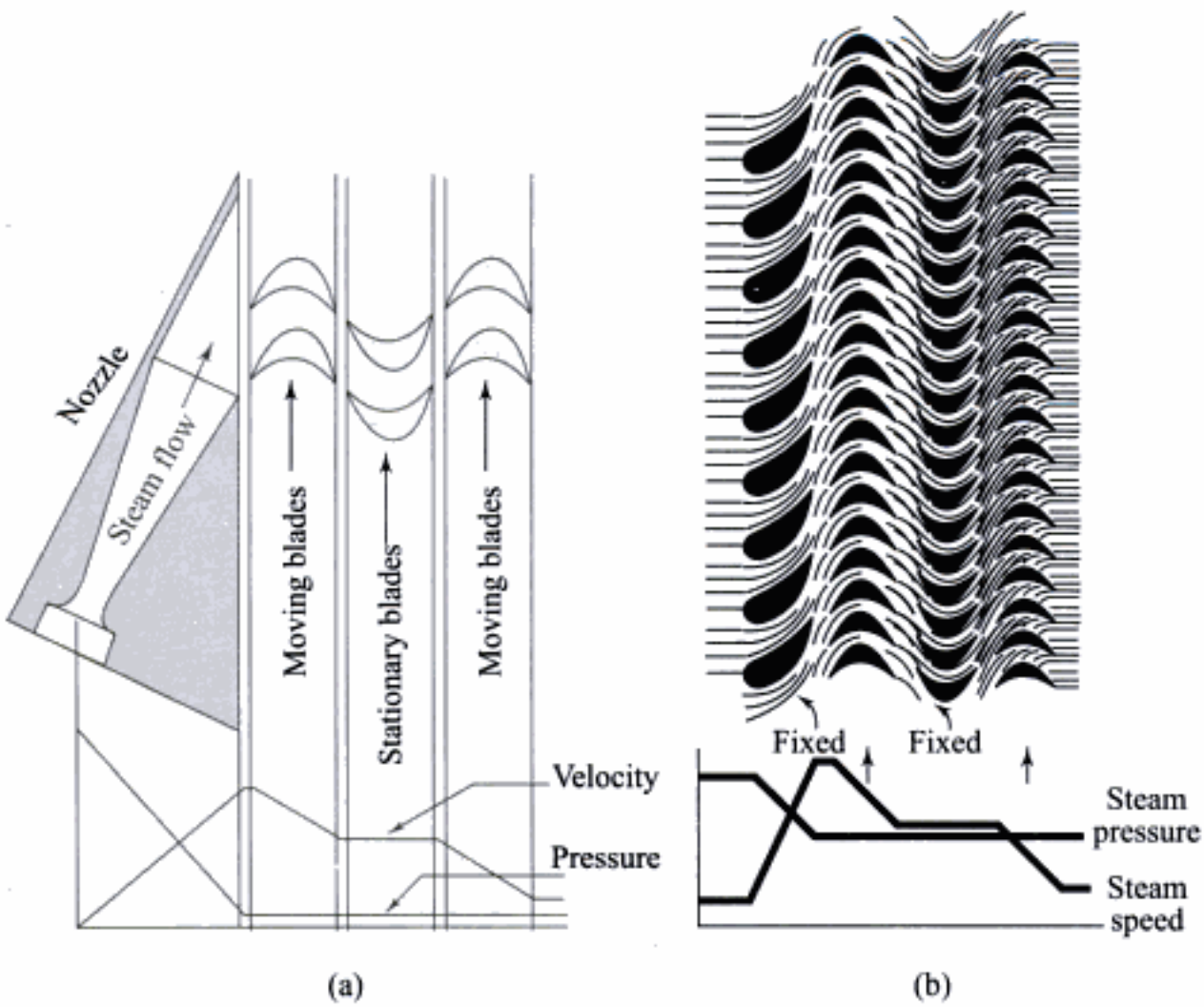

Fig. 7.25 (a) *A two-row Curtis stage, (b) Pressure and velocity variationin a two-row Curtis stage*

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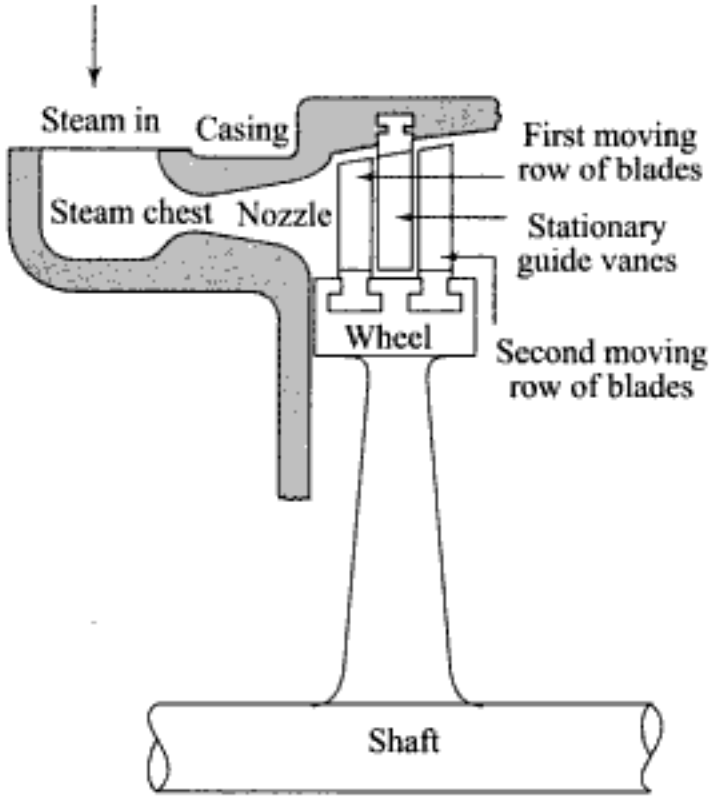

Fig. 7.25 *(c) Nozzles and blades in a two-row Curtis stage*

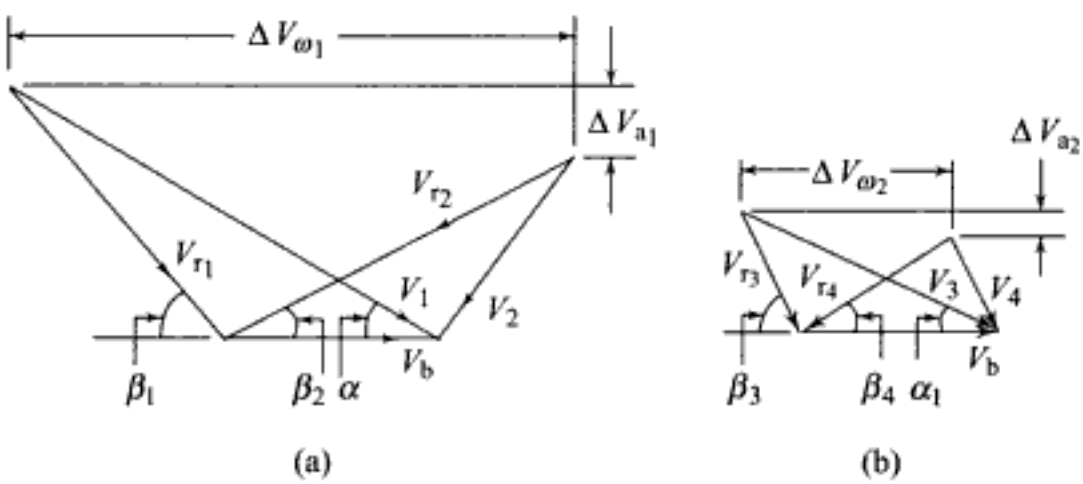

Fig. 7.26 *Velocity diagrams for a 2-row Curtis stage: (a) first row of moving blades, (b) second row of moving blades*

The tangential thrust,

$$P_t = \omega_s \Sigma \Delta \bar{V}_\omega = \omega_s \left(\Delta \bar{V}_{\omega_1} + \Delta \bar{V}_{\omega_2}\right)$$

The axial thrust,

$$P_a = \omega_s \Sigma \Delta \bar{V}_a = \omega_s \left(\Delta \bar{V}_{a_1} + \Delta \bar{V}_{a_2}\right)$$

Blading or diagram work, $W_D = P_t * \bar{V}_b$

Blading or diagram efficiency,

$$\eta_D = \frac{2 \Sigma \Delta \bar{V}_\omega \bar{V}_b}{\bar{V}_1^2}$$

These parameters can be estimated either graphically by drawing to scale the velocity diagrams or trigonometrically.

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For symmetrical blades ($\beta_1 = \beta_2$ and $\beta_3 = \beta_4$), the velocity triangles can also be drawn as shown in Fig. 7.27.

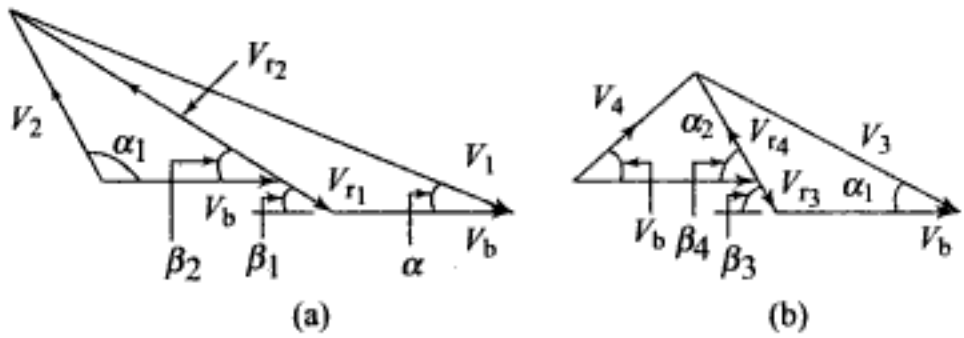

Fig. 7.27 *Two-row Curtis stage with symmetrical blades: (a) first row of moving blades, (b) second row of moving blades*

8. Effectiveness of moving rows in a Curtis stage Let us consider a two-row Curtis stage. Let us assume for simplicity symmetrical blading ($\beta_1 = \beta_2$, $\beta_3 = \beta_4$), the blade friction factor as unity and axial discharge ($\delta = 90°$). The velocity diagrams of the first and second rows of moving blades can be combined into one diagram (Fig. 7.28). The horizontal line $AB (= 4\bar{V}_b)$ is drawn such that $AC = CD = DE = EB$ $(= \bar{V}_b)$. At A, a perpendicular is drawn, and from B a line is drawn making an angle α with AB, which cuts the perpendicular at point F. FC, FD and FE are joined. The directions of velocities are shown in the diagram.

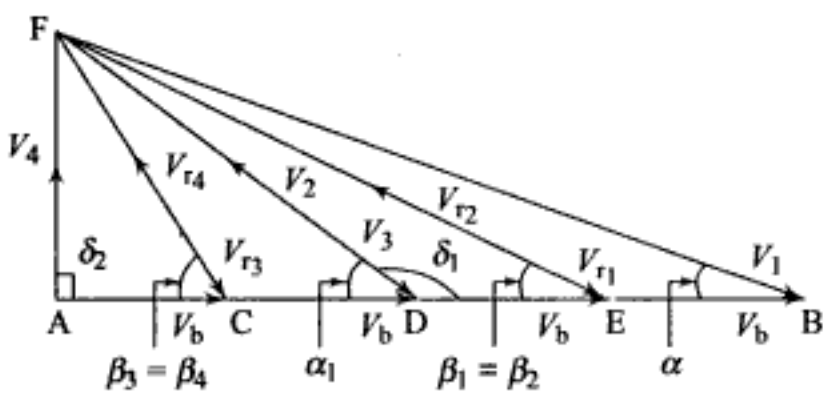

Fig. 7.28 *Velocity diagram of a two-row Curtis stage having frictionless symmetrical blading with axial discharge of steam*

Since, $\delta_1 > 90°$, $\Delta\bar{V}_{\omega_1} = \bar{V}_1 \cos\alpha + \bar{V}_2 \cos\alpha_1$

$$= 4\bar{V}_b + 2\bar{V}_b = 6\bar{V}_b$$

Now, $\Delta\bar{V}_{\omega_2} = \bar{V}_3 \cos\alpha_1 - \bar{V}_4 \cos\delta = 2\bar{V}_b$

Work done by the steam jets on the first row of moving blades,

$$W_{D_1} = \omega_s \, \Delta\bar{V}_{\omega_1} \bar{V}_b = \omega_s 6 \, \bar{V}_b . \, \bar{V}_b = 6\omega_s \, \bar{V}_b^2$$

Work done by the steam jets on the second row of moving blades,

$$W_{D_2} = \omega_s \Delta \, \bar{V}_{\omega_2} \, \bar{V}_b = \omega_s 2 \, \bar{V}_b \bar{V}_b = 2\omega_s \, \bar{V}_b^2$$

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Thus

$$W_{D_1} : W_{D_2} = 6 : 2 = 3 : 1 \tag{7.63}$$

It means that three-fourth of the total work is done by the steam jets on the first row of moving blades and one-fourth of the total work is done on the second row of moving blades.

In a 3-row Curtis stage, it can similarly be shown that

$$W_{D_1} : W_{D_2} : W_{D_3} = 5 : 3 : 1 \tag{7.64}$$

Only one-ninth of the total work in the stage is done in the 3rd row of moving blades. Therefore, as the number of rows of moving blades in a Curtis stage increases, the effectiveness of the later rows decreases. In conventional designs only two rows of moving blades are used in a Curtis stage. It is not justifiable to use more than two rows.

9. Optimum velocity ratio for a Curtis stage Let us again consider a two-row Curtis stage having symmetrical blading but with discharge not being axial. The velocity diagram is shown in Fig. 7.29.

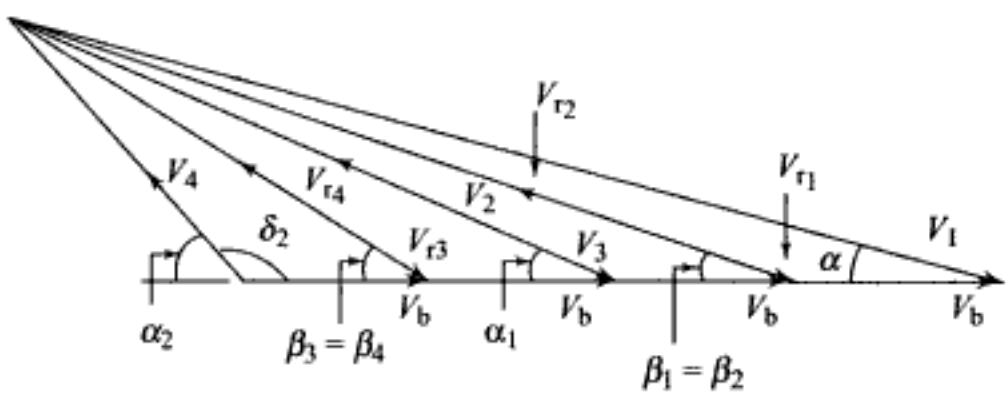

Fig. 7.29 *Velocity diagram of a two-row Curtis stage with frictionless symmetrical blading and non-axial discharge*

$$\Delta \bar{V}_{\omega_1} = \bar{V}_1 \cos\alpha + \bar{V}_2 \cos\alpha_1 = \bar{V}_1 \cos\alpha + \bar{V}_1 \cos\alpha - 2\bar{V}_b$$

$$= 2(\bar{V}_1 \cos\alpha - \bar{V}_b)$$

$$\Delta \bar{V}_{\omega_2} = \bar{V}_3 \cos\alpha_1 + \bar{V}_4 \cos\alpha_2$$

$$= \bar{V}_1 \cos\alpha - 2\bar{V}_b + \bar{V}_1 \cos\alpha - 4\bar{V}_b$$

$$= 2(\bar{V}_1 \cos\alpha - 3\bar{V}_b)$$

$$\therefore \quad \Sigma\Delta \bar{V}_\omega = \Delta\bar{V}_{\omega_1} + \Delta\bar{V}_{\omega_2} = 4(\bar{V}_1 \cos\alpha - 2\bar{V}_b)$$

The rate of energy transfer from fluid to rotor,

$$W_D = \omega_s \, \Delta\bar{V}_\omega \cdot \bar{V}_b$$

and the diagram efficiency (fraction of fluid energy converted to shaft work),

$$\eta_D = \frac{2\Sigma\Delta\bar{V}_\omega \bar{V}_b}{\bar{V}_1^2} = \frac{2 \times 4\left(\bar{V}_1 \cos\alpha - 2\bar{V}_b\right)\bar{V}_b}{\bar{V}_1^2} = \frac{8\bar{V}_b^2\left(\dfrac{\bar{V}_1 \cos\alpha}{\bar{V}_b} - 2\right)}{\bar{V}_1^2}$$

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$$= 8\rho^2\left(\frac{\cos\alpha}{\rho} - 2\right) = 8(\rho\cos\alpha - 2\rho^2)$$

$$\frac{d\eta_D}{d\rho} = 8(\cos\alpha - 4\rho) = 0$$

$$\therefore \qquad \rho_{opt} = \frac{\cos\alpha}{4} \tag{7.65}$$

For a three-row Curtis stage, it can similarly be shown that

$$\rho_{opt} = \frac{\cos\alpha}{6}$$

Therefore, for a Curtis stage having Z-rows of moving blades

$$\rho_{opt} = \frac{\cos\alpha}{2Z} \tag{7.66}$$

If $Z = 1$, i.e. for a simple impulse stage, $\rho_{opt} = \frac{\cos\alpha}{2}$ as obtained earlier.

For a two-row Curtis stage, $Z = 2$,

$$\rho_{opt} = \frac{\bar{V}_b}{\bar{V}_1} = \frac{\cos\alpha}{4} \qquad \therefore \bar{V}_b = \frac{1}{2}\left(\frac{\cos\alpha}{2}\bar{V}_1\right)$$

Therefore, in a 2-row Curtis stage, the blade velocity is half of the value for a single-stage turbine. For a certain D_m, N gets halved. Substituting in Eq. (7.64).

$$(\eta_D)_{max} = 8\left(\frac{\cos^2\alpha}{4} - \frac{\cos^2\alpha}{8}\right) = \cos^2\alpha \tag{7.67}$$

This holds good for the conditions as specified before, i.e. there is no friction and blades are symmetrical, and the discharge is axial. The same efficiency (i.e. $\cos^2\alpha$) was also obtained for symmetrical and frictionless simple impulse blading (Eq. (7.57)). However, had friction been taken into account the blading efficiency of the Curtis stage would have been considerably lower than the efficiency of the simple impulse blading.

$$\bar{V}_1 = \frac{4\bar{V}_b}{\cos\alpha} = 44.72\,[(\Delta h_s)_{Curtis})]^{1/2}$$

$$\therefore \qquad (\Delta h_s)_{Curtis} = 16\left[\frac{\bar{V}_b}{44.72\cos\alpha}\right]^2 \tag{7.68}$$

This is the isentropic enthalpy drop in the 2-row Curtis stage.

7.3.2 Reaction Turbines

In these turbines, pressure drop occurs both in the nozzles or the fixed row of blades, as well as in the moving row of blades (Fig. 7.30), since the moving

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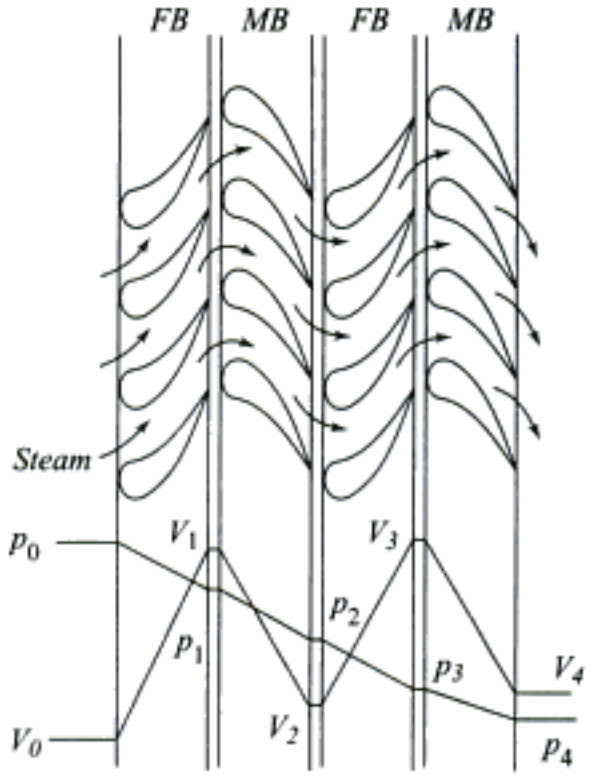
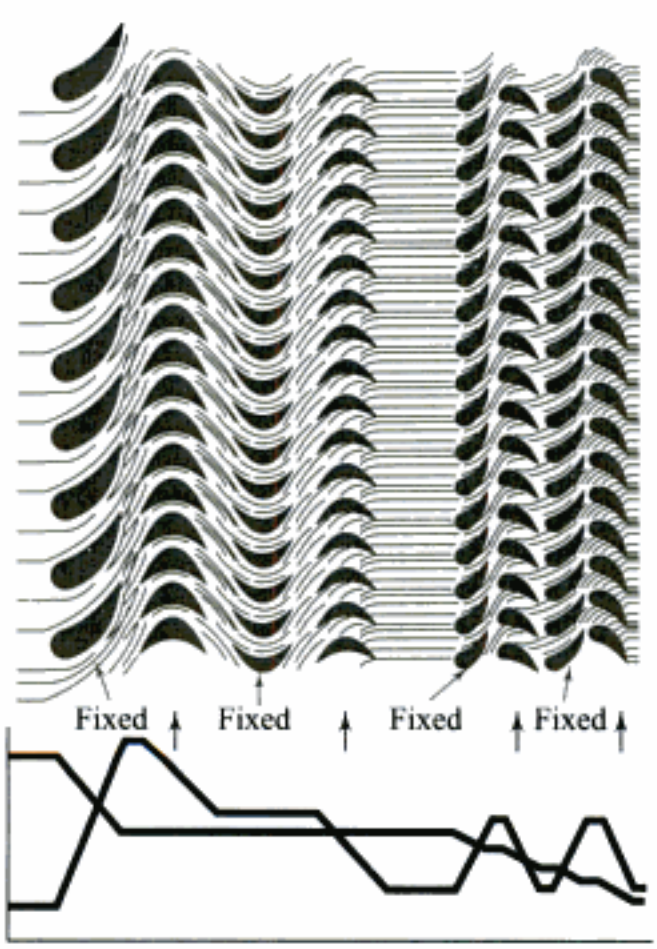


(a)



(b)

Fig. 7.30 *Variation of pressure and velocity in (a) two reaction stages,(b) in a two-row Curtis stage followed by two reaction stages*

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blade channels are also of the nozzle shape. Due to the expansion of steam while flowing through the blades, there is an increase in kinetic energy, which gives rise to reaction in the opposite direction (by Newton's third law of motion). Blades rotate due to both the impulse effect of the jets (due to change in their momentum) and the reaction force of the exiting jets impressed on the blades in the opposite direction. Such turbines are called impulse-reaction turbines, or to distinguish them from impulse turbines, simply reaction turbines. The degree of reaction (R) of these turbines is defined as

$$R = \frac{\Delta h_{mb}}{\Delta h_{fb} + \Delta h_{mb}} \tag{7.69}$$

where the subscripts "mb" and "fb" represent moving blades and fixed blades, respectively.

If $(\Delta h)_{mb} = 0$, $R = 0$, which is the case of pure impulse turbines where there is no enthalpy drop of steam in the moving blades, and all the enthalpy drop of the stage take place only in nozzles.

If $\Delta h_{fb} = 0$, $R = 1$, which is the case of a pure reaction (R = 100%) turbine, e.g. Hero's turbine.

If equal enthalpy drops occur in the fixed and moving blades, i.e. if $\Delta h_{fb} = \Delta h_{mb} = (\Delta h_{stage})/2$, $R = 1/2$ or 50%. Sometimes, 50% reaction turbines are also called *Parsons turbines*.

The velocity diagrams for the moving blades of a 50% reaction turbine are shown in Fig. 7.31. With a simple impulse type the value of V_{r_2} ($= k_b V_{r_1}$) would be given by BE, but in the reaction turbine this velocity is increased to BC by further expansion of the steam in the nozzle-shape blade channels.

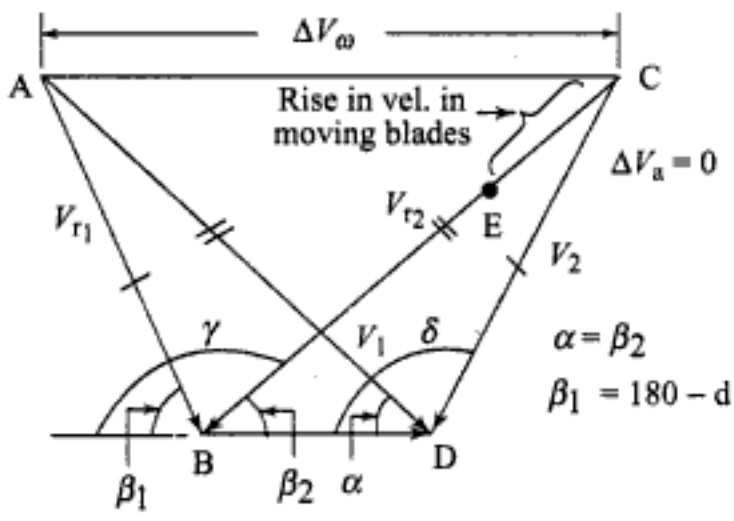

Fig. 7.31 *Velocity diagram of a 50% reaction turbine*

For manufacturing advantage, both fixed blades and moving blades are made similar in shape so that they can be extruded from the same set of dies.

Since $\Delta h_{fb} = \Delta h_{mb}$, $V_1 = V_{r2}$

Again, for similar geometry, $\alpha = \beta_2 = 180° - \gamma$

The triangle ABD and DBC are similar.

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$\therefore \qquad V_{r1} = V_2 \quad$ and $\quad \beta_1 = 180° - \delta$

Since $\beta_1 \neq \beta_2$, the blades are unsymmetrical. Again, since $\Delta V_a = 0$, there is no axial thrust imposed on the blades due to change in axial velocity in a 50% reaction turbine. However, there will be considerable axial thrust produced due to the pressure difference across the blades in each rotor disc since there is pressure drop of steam across the moving blades.

$$\Delta V_\omega = V_1 \cos\alpha - V_2 \cos\delta = V_1 \cos\alpha + V_{r2} \cos\beta_2 - V_b \quad (\because \delta > 90°)$$

$$= 2V_1 \cos\alpha - V_b$$

Also, $$\Delta V_\omega = V_{r_1} \cos\beta_1 + V_{r2} \cos\beta_2 = V_1 \cos\alpha - V_b + V_1 \cos\alpha$$

$$= 2\,V_1 \cos\alpha - V_b \tag{7.70}$$

The diagram work per kg of steam,

$$W_D = \Delta V_\omega\, V_b = (2V_1 \cos\alpha - V_b)V_b \tag{7.71}$$

Energy input to blades per kg of steam

$$= \frac{V_1^2}{2} + \frac{V_{r_2}^2 - V_{r_1}^2}{2} = \frac{V_1^2}{2} + \frac{V_1^2}{2} - \frac{V_{r_1}^2}{2} = V^2{}_1 - \frac{V_{r_1}^2}{2}$$

Now, $$V^2{}_{r_1} = V_1^2 + V_b^2 - 2V_1\, V_b \cos\alpha$$

$\therefore \qquad$ Energy input $= V_1^2 - \dfrac{V_1^2 + V_b^2 - 2V_1 V_b \cos\alpha}{2}$

$$= \frac{V_1^2 - V_b^2 + 2V_1 V_b \cos\alpha}{2} \tag{7.72}$$

Diagram efficiency of the blades,

$$\eta_D = \eta_{b1} = \frac{2(2V_1 \cos\alpha - V_b)V_b}{V_1^2 - V_b^2 + 2V_1 V_b \cos\alpha} = \frac{2V_b^2\left[2\dfrac{V_1 \cos\alpha}{V_b} - 1\right]}{V_1^2\,[1 - (V_b^2/V_1^2) + 2(V_b/V_1)\cos\alpha]}$$

Putting $V_b/V_1 = \rho$, the velocity ratio,

$$\eta_D = \frac{2\rho^2\left[2\dfrac{\cos\alpha}{\rho} - 1\right]}{1 - \rho^2 + 2\rho\cos\alpha} = \frac{2(2\rho\cos\alpha - \rho^2)}{1 - \rho^2 + 2\rho\cos\alpha} \tag{7.73}$$

There is a particular value of ρ for which η_D is a maximum. Differentiating η_D with respect to ρ and equating it zero,

$$\frac{d\eta_D}{d\rho} = \frac{(1 - \rho^2 + 2\rho\cos\alpha)\,2(\cos\alpha - 2\rho) - 2\rho\,(2\cos\alpha - \rho)(-2\rho + 2\cos\alpha)}{(1 - \rho^2 + 2\rho\cos\alpha)^2} = 0$$

$$4(1 - \rho^2 + 2\rho\cos\alpha)(\cos\alpha - \rho) - 4\rho\,(2\cos\alpha - \rho)(\cos\rho - \rho) = 0$$

$$4(\cos\alpha - \rho)(1 - \rho^2 + 2\rho\cos\alpha - 2\rho\cos\alpha + \rho^2) = 0$$

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$$\therefore \qquad \rho_{opt} = \cos\alpha \tag{7.74}$$

or $$V_b = V_1 \cos\alpha$$

Substituting the value of ρ_{opt} in Eq. (7.73),

$$(\eta_D)_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha} \tag{7.75}$$

From Eqs (7.71) and (7.74), the specific blading work corresponding to maximum blading efficiency.

$$W_D = (2V_b - V_b)V_b = V_b^2 \tag{7.76}$$

The velocity diagrams for a 50% reaction turbine operating with maximum blading efficiency are shown in Fig. 7.32.

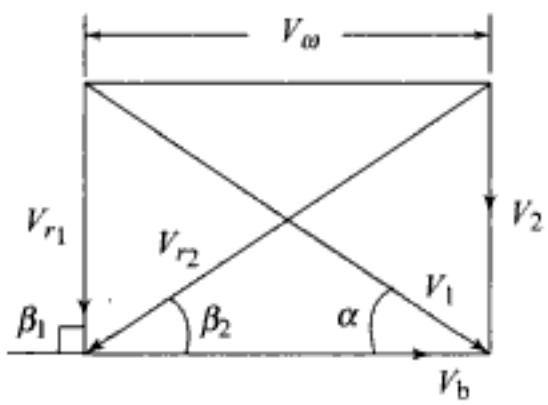

Fig. 7.32 *Velocity diagram of a 50% reaction turbine operating with maximum blading efficiency*

$$\Delta V_\omega = V_1 \cos\alpha = V_b$$

$$\therefore \qquad W_D = \Delta V_\omega V_b = V_b^2$$

It is no longer convenient to talk of nozzles and blades, since in the reaction turbine both fixed and moving blades act as nozzles. It is usual to refer to the two sets of blades as the stator or fixed blades and the rotor or moving blades.

Since the isentropic enthalpy drop of the stage is equally divided between the fixed blades and moving blades (Fig. 7.33),

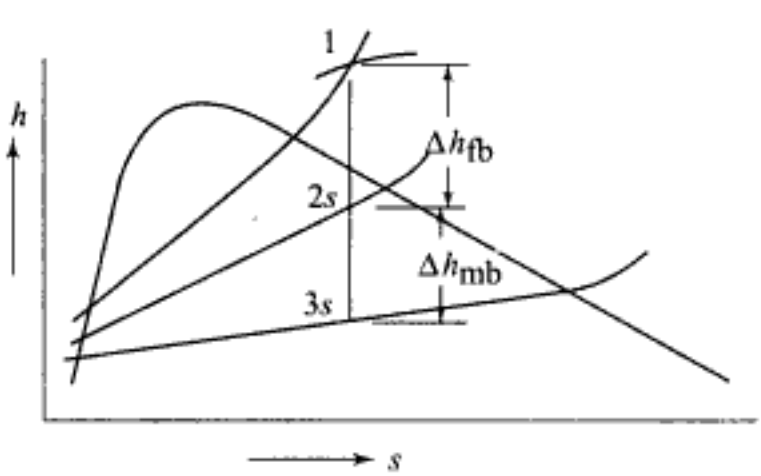

Fig. 7.33 *Enthalpy drop in a 50% reaction stage*

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$$(\Delta h_s)_{fb} = h_1 - h_{2s} = \frac{V_1^2 - V_2^2}{2}$$

$$(\Delta h_s)_{mb} = h_{2s} - h_{3s} = \frac{V_{r_2}^2 - V_{r_1}^2}{2} = \frac{V_1^2 - V_2^2}{2}$$

$$\therefore \quad (\Delta h_s)_{fb} = (\Delta h_s)_{mb}$$

$$(\Delta h_s)_{stage} = h_1 - h_{3s} = 2(h_{2s} - h_{3s}) = 2(\Delta h_s)_{mb}$$

$$\therefore \quad R = \frac{h_{2s} - h_{3s}}{h_1 - h_{3s}} = \frac{\Delta h_{mb}}{2\,(\Delta h_{mb})} = \frac{1}{2}$$

1. Carry-over efficiency For symmetrical staging (R = 50%), the isentropic enthalpy drop across the stage is evenly distributed between the stationary and moving rows of blades (Fig. 7.33) so that

$$(\Delta h_s)_{fb} = (\Delta h_s)_{mb} = \frac{(\Delta h_s)_{stage}}{2} = (\Delta h_s)_{row}$$

Assuming the same nozzle efficiency for both fixed and moving rows, the kinetic energy of fluid gained per row, fixed as well as moving, is

$$\eta_n \cdot \frac{(\Delta h_s)_{stage}}{2}$$

Now, the kinetic energy associated with the leaving velocity of the preceding stage is available to do work in the following stage. Of course, some loss is involved in the journey from one stage to the next and is taken into consideration by what is known as *carry-over efficiency*. Hence, the kinetic energy leaving one stage and available to the next is given by $\eta_{CO}\, V_2^2/2$, where η_{CO} is the carry-over efficiency.

The energy balance for the fixed row of blades gives

$$(V_1^2/2) + h_1 = \eta_{CO}\,\frac{V_2^2}{2} + h_2$$

$$\frac{V_1^2 - \eta_{CO} V_2^2}{2} = h_2 - h_1 = \eta_n\,\frac{(\Delta h_s)_{stage}}{2}$$

$$\therefore \quad (\Delta h_s)_{stage} = \frac{V_1^2 - \eta_{CO} V_2^2}{\eta_n} \tag{7.77}$$

This is the energy available for conversion in one stage. The energy output or diagram work per kg of steam is given by Eq. (7.71). Therefore, the combined nozzle and blade efficiency η_{nb} is given by

$$\eta_{nb} = \frac{(2V_1 \cos\alpha - V_b)\,V_b}{(V_1^2 - \eta_{CO} V_2^2)/\eta_n}$$

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or
$$\eta_{nb} = \eta_n \frac{V_b^2}{V_1^2} \frac{\left(\frac{2V_1 \cos\alpha}{V_b} - 1\right)}{1 - \eta_{CO}\left(\frac{V_2^2}{V_1^2}\right)} \tag{7.78}$$

Again,
$$V_2^2 = V_{r2}^2 + V_b^2 - 2V_{r2} V_b \cos\alpha$$

$\therefore$
$$\frac{V_2^2}{V_1^2} = 1 + \rho^2 - 2\rho\cos\alpha, \text{ where } \rho = V_b/V_1$$

Substituting in Eq. (7.78),

$$\eta_{nb} = \eta_n \frac{\rho^2\left(\frac{2\cos\alpha}{\rho} - 1\right)}{1 - \eta_{CO}(1 + \rho^2 - 2\rho\cos\alpha)}$$

or
$$\eta_{nb} = \eta_n \frac{2\rho\cos\alpha - \rho^2}{1 - \eta_{CO}(1 + \rho^2 - 2\rho\cos\alpha)} \tag{7.79}$$

When $\eta_{CO} = 1$, $\eta_{nb} = \eta_n$, i.e. the losses are confined to the nozzle friction only.

When $\eta_{CO} = 0$, there is no carry-over of kinetic energy to the next stage (velocity of approach is zero), and

$$\eta_{nb} = \eta_n (2\rho\cos\alpha - \rho^2) \tag{7.80}$$

which is the efficiency of a single row of blades.
If $\eta_n = 1$,

$$\eta_{nb} = \eta_{b1} = \eta_D = 2\rho\cos\alpha - \rho^2 \tag{7.81}$$

For maximum blading efficiency,

$$\frac{d\eta_D}{d\rho} = 2\cos\alpha - 2\rho = 0$$

$$\rho_{opt} = \cos\alpha$$

which is the same as Eq. (7.74).
Substituting in Eq. (7.81)

$$(\eta_{b1})_{max} = \cos^2\alpha \tag{7.82}$$

which is somewhat less than that obtained from Eq. (7.75).

A comparison of Eqs (7.57), (7.67) and (7.82) reveals that the optimum efficiencies for simple impulse, Curtis and reaction blading are all equal. However, when friction is taken into account, the reaction stage is found to be the most efficient, followed by Rateau and Curtis staging in that order. The friction losses are less significant in the reaction stage since the flow velocities are lower.

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2. Comparison of enthalpy drops in various stages Assuming the blades operating at maximum efficiency, for a symmetric (50% R) reaction stage,

$$\rho_{opt} = \cos\alpha = V_b/V_1$$

$\therefore$ V_1 = velocity of steam exiting the nozzles or fixed blades

$$= V_b/\cos\alpha = 44.72\left[\frac{(\Delta h_s)_{stage}}{2}\right]^2$$

$$\therefore \qquad (\Delta h_s)_{stage} = 2\left[\frac{V_b}{44.72\cos\alpha}\right]^2 \tag{7.83}$$

This is the isentropic enthalpy drop in a 50% reaction stage. If we compare this with the isentropic enthalpy drop for a simple impulse stage, Eq. (7.59) and that for a 2-row Curtis stage, Eq. (7.68),

$$(\Delta h_s)_{impulse\ stage} = 4\left[\frac{V_b}{44.72\cos\alpha}\right]^2$$

and $$(\Delta h_s)_{2\text{-row Curtis stage}} = 16\left[\frac{V_b}{44.72\cos\alpha}\right]^2$$

we find for the same values of $\bar{V}_b$ and α,

$$(\Delta h_s)_{50\%\ reaction\ stage} : (\Delta h_s)_{simple\ impulse\ stage} : (\Delta h_s)_{2\text{-row curtis stage}} = 1 : 2 : 8 \tag{7.84}$$

Therefore, a 2-row Curtis stage is equivalent (ideally) to four simple impulse stages and eight 50% reaction stages.

Since the number of stages required for a certain total isentropic enthalpy drop is

$$n = \frac{(\Delta h_s)_{total}}{(\Delta h_s)_{stage}}$$

the lower is the enthalpy drop per stage, the higher is the number of stages required for the same output.

For V_b = 320 m/s and α = 16°,

$$(\Delta h_s)_{impulse\ stage} = 221.64 \text{ kJ/kg}$$
$$(\Delta h_s)_{50\%\ reaction\ stage} = 110.82 \text{ kJ/kg}$$
$$(\Delta h_s)_{2\text{-row Curtis stage}} = 886.56 \text{ kJ/kg}$$

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Thus, for the same output of a turbine the number of reaction (50%) stages required will be about twice the number of impulse or pressure (Rateau) stages and about 8 times the number of 2-row Curtis stages. So, reaction turbines are costlier than impulse turbines for the same output.

Again, for the same mean blade speed V_b,

$$V_1 = V_b/\cos\alpha \text{ for a 50\% reaction stage}$$
$$V_1 = 2V_b/\cos\alpha \text{ for a simple impulse stage, and}$$
$$V_1 = 4V_b/\cos\alpha \text{ for a 2-row Curtis stage.}$$

The energy loss due to friction is proportional to the square of the velocity of a fluid (steam). Since the fluid velocity is the highest for a 2-row Curtis stage and lowest in the 50% reaction stage, so the energy loss due to friction in the reaction stage is the least while that in the Curtis stage is the highest. The energy loss in the impulse stage will be in between these two values. Therefore, the efficiency of the reaction stage will be the highest and that of the Curtis stage will be the lowest, while that of impulse stage will lie in between.

$$\eta_{\text{reaction (50\%) stage}} > \eta_{\text{simple impulse stage}} > \eta_{\text{2-row Curtis}}$$

Figure 7.34 gives a comparison of diagram efficiencies with respect to the velocity ratio. Reaction turbines are more efficient but are more costly.

In modern turbines, a 2-row Curtis stage is generally used as the first stage. It is followed by a series of reaction stages. Sometimes impulse stages are also used after the Curtis stage. The use of the initial Curtis stage reduces the length of the rotor, i.e. the number of stages required (since a 2-row Curtis stage can [illegible]

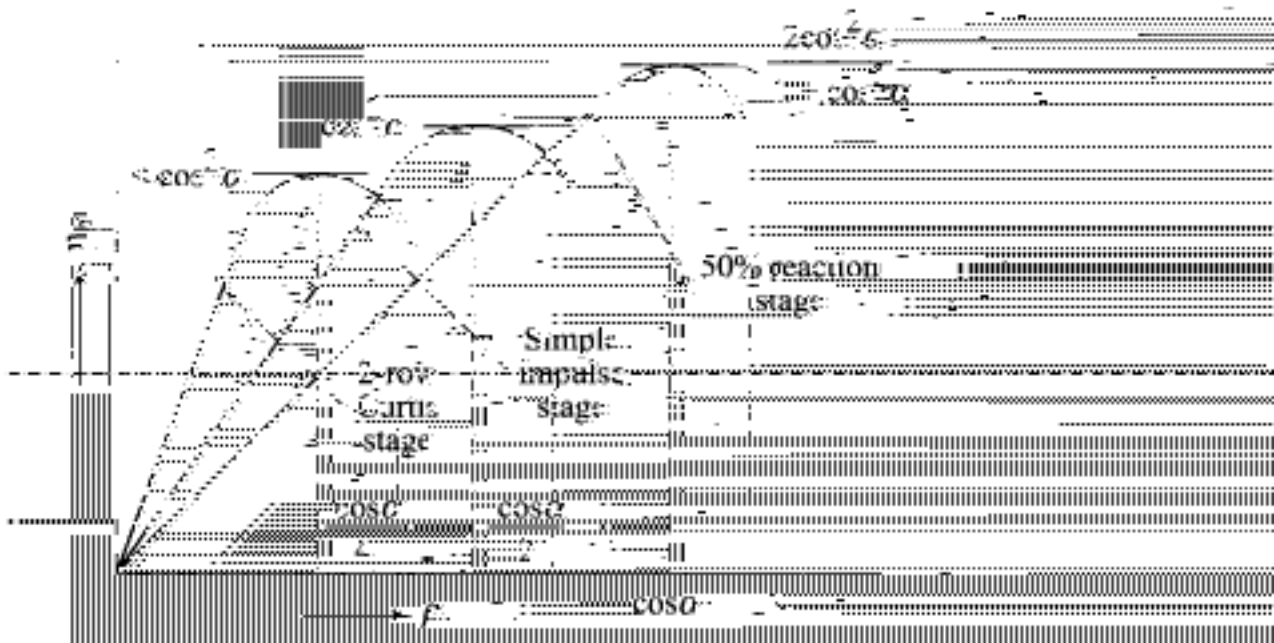

Fig. 7.34 Variation of diagram efficiency with speed ratio

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are simplified and the overall length of the rotor is reduced. Thus, it is a good compromise to use one 2-row Curtis stage initially, sacrificing some loss in efficiency. A large drop in enthalpy occurs in this Curtis stage. The remaining enthalpy drop occurs in the subsequent stages, the number of which can be estimated from

$$n = \frac{(\Delta h_s)_{\text{total}} - (\Delta h_s)_{\text{2-row Curtis stage}}}{(\Delta h_s)_{\text{stage}}} \tag{7.85}$$

where Δh_s represents the isentropic enthalpy drop.
The arrangement of stages for a 15-stage turbine is given below:

$$\frac{\text{2-row Curtis stage}}{\text{N MB GB MB}} \; \frac{1}{\text{FB MB}} \; \frac{2}{\text{FB MB}} \; \frac{3}{\text{FB MB}} \; \frac{4}{\text{FB MB}} \cdots \frac{13}{\text{FB MB}} \; \frac{14}{\text{FB MB}},$$

where N : nozzle; MB moving blades, GB guide blades, and FB fixed blades.

These stages are distributed in a number of cylinders or casings, as discussed later. Simple layouts of an impulse and a reaction turbine are shown in Fig. 7.35. Figure 7.36 shows typical single cylinder designs of impulse and reaction turbines. In practice, the number of impulse and reaction turbines in operation are approximately equal, the choice being dependent on the preference of the turbine manufacturer. In some designs the stage may have some degree of reaction, 10% or less to produce a small pressure drop across the bucket. This small drop can be very useful in keeping the buckets running full of steam, with the shaft thrust positive in one direction.

7.3.3 Variation of Blade Velocity Along Blade Height

In the low pressure region, the blade heights are quite large. The velocity at the blade root $[(\pi D_{\text{root}} N)/60]$ will be much smaller than that at the mid-point, and still smaller than that at the tip $[(\pi D_{\text{tip}} N)/60]$. Hence, for a good efficiency, the blade angles should vary with the diameter (Fig. 7.37). For this reason, *twisted (or warped) blades are used in the later stages of the turbine.* The vertical line in Fig. 7.37 represents the radius of the wheel. The vertical distance between the tip and the root radii is the blade height. Steam enters and leaves the blades with velocities V_1 and V_2, respectively. Since V_b is proportional to the radius, angle β_1 will vary accordingly. It is seen that the vector diagram at the root looks like one for an impulse blade, while the diagram at the tip looks like one for a reaction blade. The blade work, $V_b \Delta V_\omega$, is constant.

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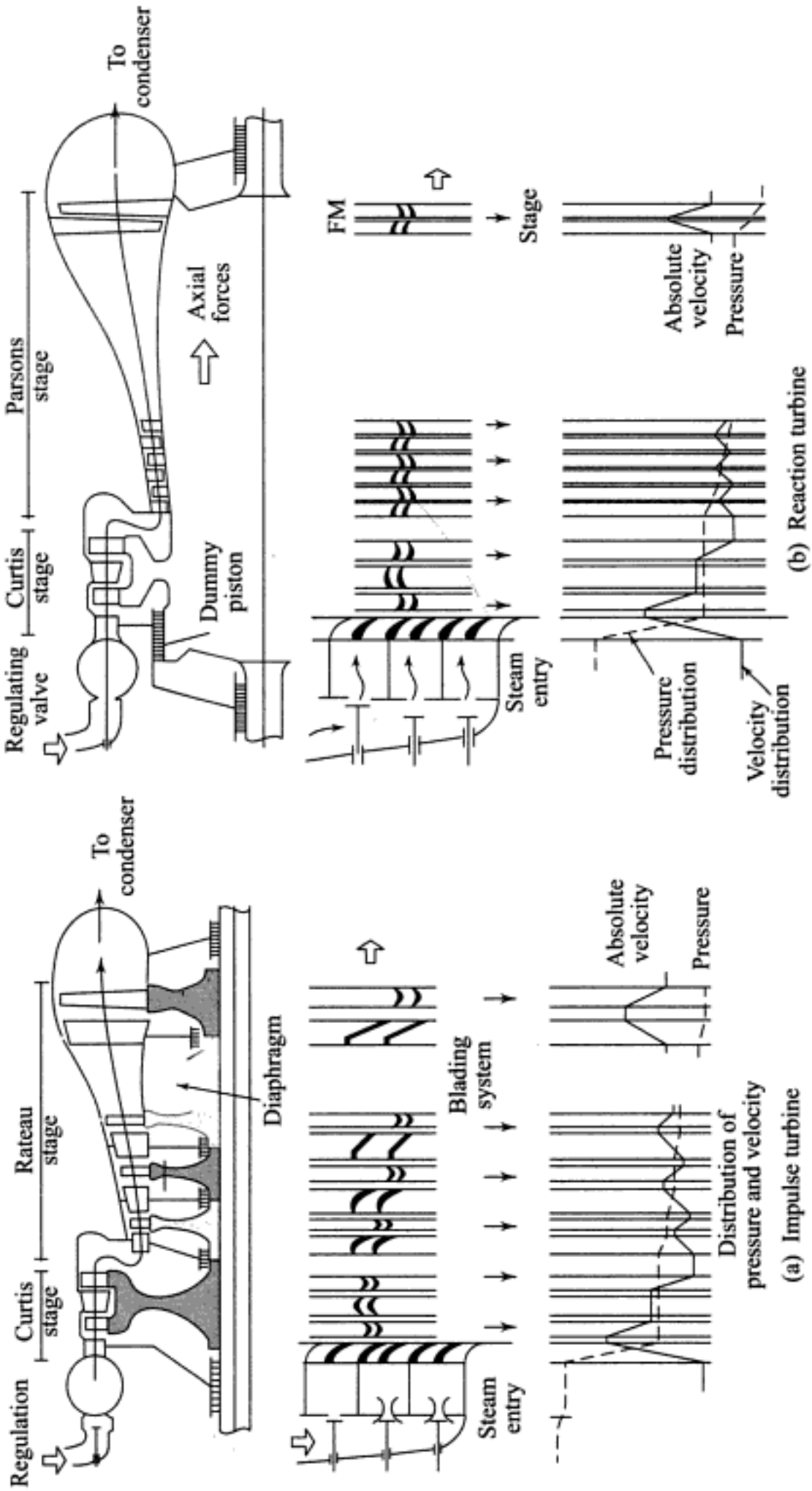

Fig. 7.35 *Layouts of impulse and reaction steam turbines*

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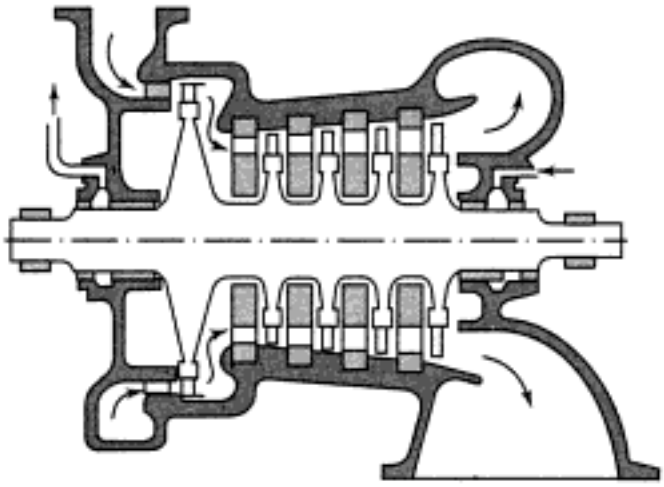
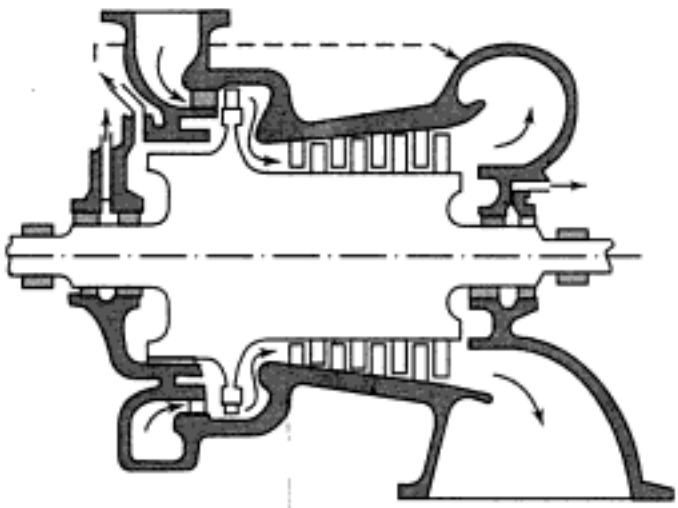


(a) Impulse turbine



(b) Reaction turbine

Fig. 7.36 *Typical designs of impulse and reaction steam turbines*

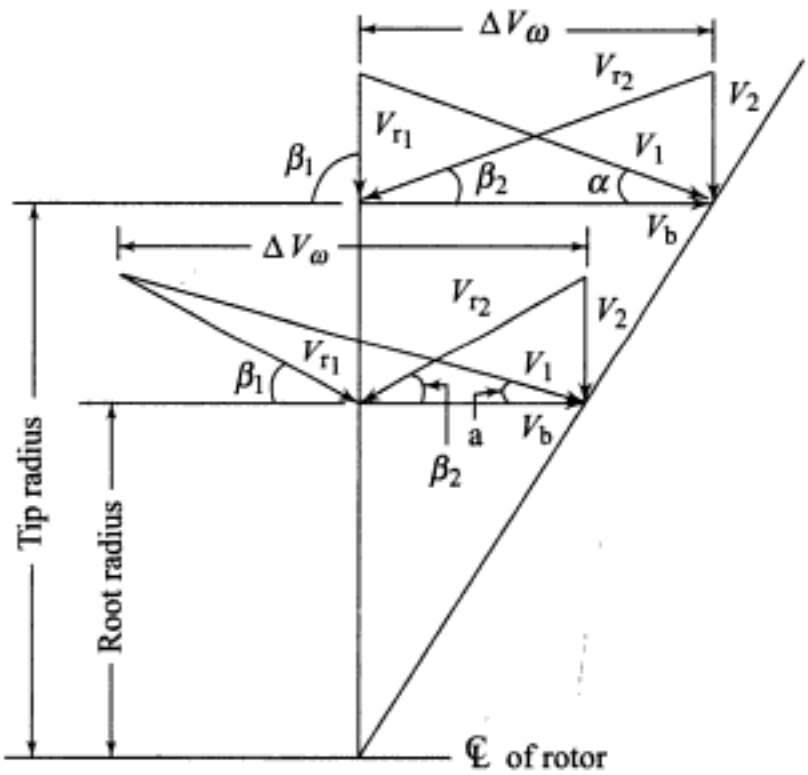


Fig. 7.37 *Variation in low-pressure blade angles as influenced by vector diagrams for root and tip velocity*

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7.3.4 Nozzle and Blade Heights

Nozzle or blade height depends on the total annular area required to pass the desired flow of fluid. From Fig. 7.38, it can be seen that the area available for flow at exit of one nozzle passage is approximately.

$$A = Oh_n$$

where O = width of flow passage at exit at mean nozzle height, and h_n = nozzle height.

It was already discussed in section, 7.2.9 how to estimate the nozzle annular area and its height. For coherence, it is being done here too.

From Fig. 7.38(c),

$$p \sin \alpha = 0 + t$$

where p = pitch of nozzles at mean nozzle height, and t = nozzle wall thickness, as shown.

$$\therefore \qquad A = (p \sin \alpha - t)h_n$$

Assuming full peripheral admission, i.e. the nozzle diaphragm is completely occupied by nozzles, the number of nozzles, z, is given by

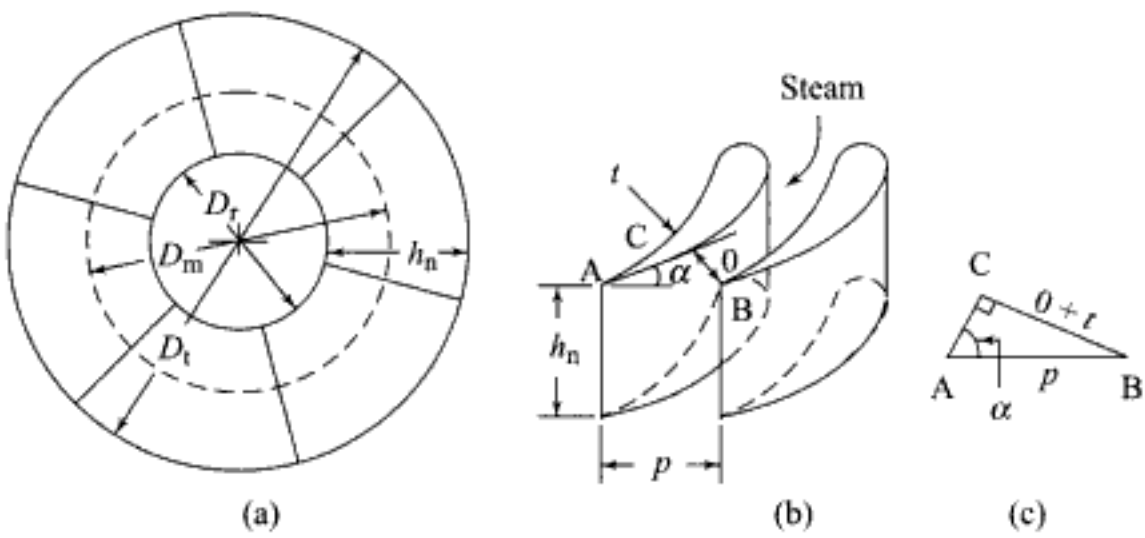

Fig. 7.38 *Flow through a nozzle passage*

$$z = (\pi D_m)/p$$

Total nozzle area is

$$A_n = p \sin \alpha \left(1 - \frac{t}{p \sin \alpha}\right) h_n \frac{\pi D_m}{p}$$

or

$$A_n = pD_m h_n k_{tn} \sin a \qquad (7.86)$$

where k_{tn} is the nozzle thickness factor.

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Similarly, it can be shown that the area available for flow at the entrance to the blades is

$$A_b = \pi D_m h_b k_{tb} \sin \beta_1 \tag{7.87}$$

where k_{tb} is the edge thickness factor for blades.
From the continuity equation, it follows that

$$\omega_s = \frac{A_n V_1}{v_1} = \frac{A_b V_{r_1}}{v_1}$$

or $$\pi D_m h_n k_{tn} \sin \alpha V_1 = \pi D_m h_b k_{tb} \sin \beta_1 V_{r1} \tag{7.88}$$

For all practical purposes, it can be assumed that $k_{tn} = k_{tb}$, so that

$$h_b/h_n = (V_1 \sin \alpha)/(V_{r_1} \sin \beta_1) = 1 \tag{7.89}$$

Although Eq. (7.89) indicates that the blade height at entrance is equal to the nozzle height at exit, it is customary to increase the blade entrance height slightly. This is done in order to avoid spilling of the fluid issuing from the nozzles. This increase in the blade entrance height is called "over lap" and is given equally at root and tip of the blade. This h_b exceeds h_n by 1.6 mm in high pressure stages and by about 20 mm in low pressure stages of large turbines, i.e. the over lap varies from 1.6 mm to 20 mm.

1. First stage nozzles and blades From the continuity equation, the nozzle area required for a given flow rate ω_s is given by

$$A_n = \frac{\omega_s V_1}{v_1}$$

At high pressure of steam, V_1 is small and $V_1 = V_b/\cos \alpha$ for 50% reaction blading, and $V_1 = 2V_b/\cos \alpha$ for impulse blading if the blades are assumed to operate at the maximum efficiency. Thus, at the initial stage of turbine, the flow area (A_n) required is small. Now,

$$A_n = \pi D_m h_n \sin \alpha k_{tn}$$

where D_m is estimated from the given value of $V_b = (\pi D_m N)/60$. Since A_n is small and for given values of α and k_{tn}, the estimated value of h_n is often found to be too small to manufacture. If the nozzles are conical, the diameter of each nozzle also is found to be very small. It has been experimentally found that small diameter nozzles are less efficient than nozzles of larger diameter.

Furthermore, the blades following the nozzles are thus very short, which is difficult to fabricate. Such short blades are also less efficient. It is advisable to use a minimum height of 20 mm for the initial blades. Therefore, the nozzle height in the first stage cannot be much less than 20 mm, or

$$(h_b)_{min} \cong (h^*_n)_{min} \cong 20 \text{ mm}$$

The required nozzle height h_n for a given flow is much less than this. Thus,

$$A_n = x\pi D_m (h_n)_{min} \sin \alpha k_{tn} \tag{7.90}$$

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where x is the fraction of the total arc of nozzles which is open for steam flow. This is called *partial admission of steam*. Some nozzles in the arc are blocked by steel plates (Fig. 7.39). Thus, by increasing the estimated h_n to its minimum practicable value (~ 20 mm) and simultaneously blocking part of the periphery of the nozzle diaphragm, the required flow area (A_n) for a certain mass flow of steam at the given condition is obtained. For full admission, the entire arc or periphery of the nozzle diaphragm (πD_m) is open for steam flow. For partial admission, the arc or periphery open for steam flow is ($x\pi D_m$), where x is less than unity.

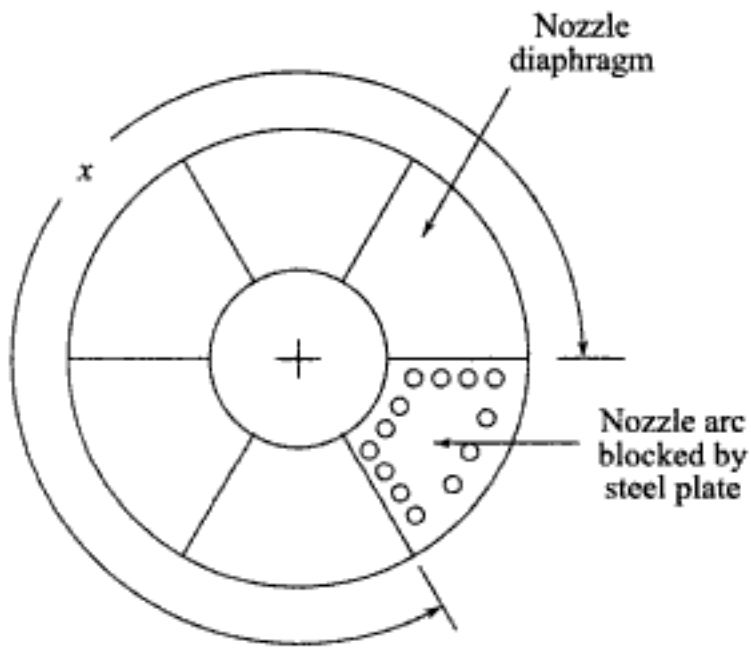

Fig. 7.39 *Partial admission of steam in the initial stage*

With partial admission, a given blade passage will not receive flow from the nozzles at all times. Thus exposed to alternate flows of high velocity steam, blades are subjected to vibrations which may prove dangerous in long blades.

2. Last stage blade height The last stage is very important in steam turbine design. As the pressure of steam decreases during expansion, the specific volume increases. The volume flow of steam increases necessitating an increased flow area. Consequently, h_b and D_m increase with decrease of pressure. At the last stage, the pressure is the minimum, and so h_b and D_m should have maximum values.

Blades are held at one end with the rotor while the other end is free. So they act as cantilevers with distributed load of steam on them. They are subjected to bending stresses. Since they are rotating at a high rpm, so they are subjected to centrifugal stresses also. As the blade height increases, both bending and centrifugal stresses increase. Due to these stresses, both blade height and blade

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diameter get restricted. The maximum blade velocity is also limited depending on the material of the blades, which is about 350–400 m/s.

$$(V_b)_{max} = 350 \text{ m/s (say)} = \frac{\pi (D_m)_{max} N}{60}$$

The rpm of the rotor is fixed from the generator side. For a 2-pole, 50 Hz alternator,

$$N = \frac{120 f}{p} = \frac{120 \times 50}{2} = 3000 \text{ rpm}$$

$$\therefore \qquad (D_m)_{max} = \frac{350 \times 60}{\pi \times 3000} = 2.23 \text{ m}$$

For straight blades, the maximum blade height is about 20% of the mean blade ring diameter, or

$$(h_b/D_m)_{max} = 0.20$$

Where it is necessary to exceed this ratio because of flow requirements, the blades may be tapered or twisted (Fig. 7.40), thereby reducing both bending and centrifugal stresses. With these modifications, the blade height may approach about 30% of the mean blade ring diameter. Therefore, for twisted or tapered blades,

$$(h_b/D_m)_{max} = 0.3$$

$$\therefore \qquad (h_b)_{max} = 2.23 \times 0.3 = 0.67 \text{ m}$$

The flow area (A_b) could also be increased by increasing the blade angles, which would, in turn, increase the nozzle angle α. This would, however, decrease the blade efficiency. By reducing the rpm, both h_b and D_m can be increased. But, this will also increase the weight and bulk of the rotor along with the cost. So, this is not justified.

When the blade height becomes a significant part of the total stage diameter, the ratio of steam to blade speed changes over the height of the blade. Figure 7.41 shows velocity diagrams at the root and tip of the twisted blade that receives a steam jet moving in a vortex flow ($V_1 r_1 = V_2 r_2$). The blade root has been designed for impulse flow, when the tip is exposed to a pure reaction force.

3. Parallel exhausts: number of last stages The maximum flow area is provided in the last stage where the volume flow is maximum.

$$(A_b)_{max} = \pi (D_m h_b)_{max} \sin \beta_1 \, k_{tb}$$

The maximum volume flow that the last stage can accommodate

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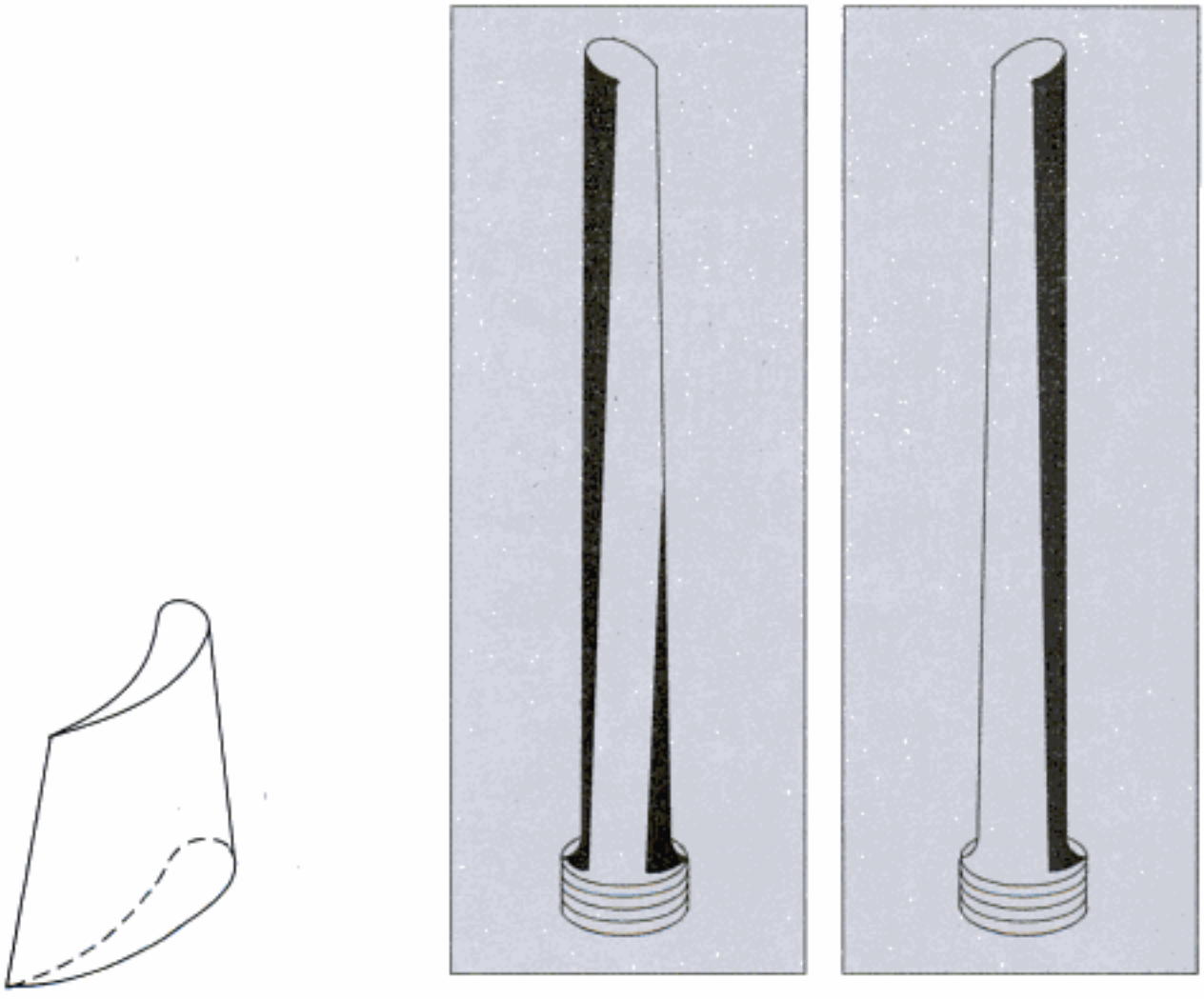

(a) Tapered blade (b) Twisted blades

Fig. 7.40 *Tapered and twisted blades*

$$= \pi (D_m h_b)_{max} \sin \beta_1 \, k_{tb} \, V_{r1}$$

$$= \pi (D_m h_b)_{max} \, k_{tb} \, V_1 \sin \alpha \, [\text{since } V_{r1} \sin \beta_1 = V_1 \sin \alpha] \qquad (7.91)$$

If impulse blading is used,

$$\frac{V_b}{V_1} = \frac{\cos \alpha}{2} \qquad \therefore V_1 = 700/\cos \alpha$$

From Eq. (7.91), for $\alpha = 20°$ and $k_{tb} = 0.9$ (assumed),

$$\text{Maximum volume flow} = \pi \times 2.23 \times 0.67 \times \frac{700}{\cos 20°} \sin 20° \times 0.9 = 1075 \text{ m}^3/\text{s}$$

The maximum mass flow of steam that the last stage can accommodate (Fig. 7.42).

$$(\omega_s)_{max} = \frac{1075}{v_2} \text{ kg/s}$$

For the turbine exhaust condition at 0.075 bar, 0.88 quality,

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