Refrigeration and Air Conditioning (RAC)

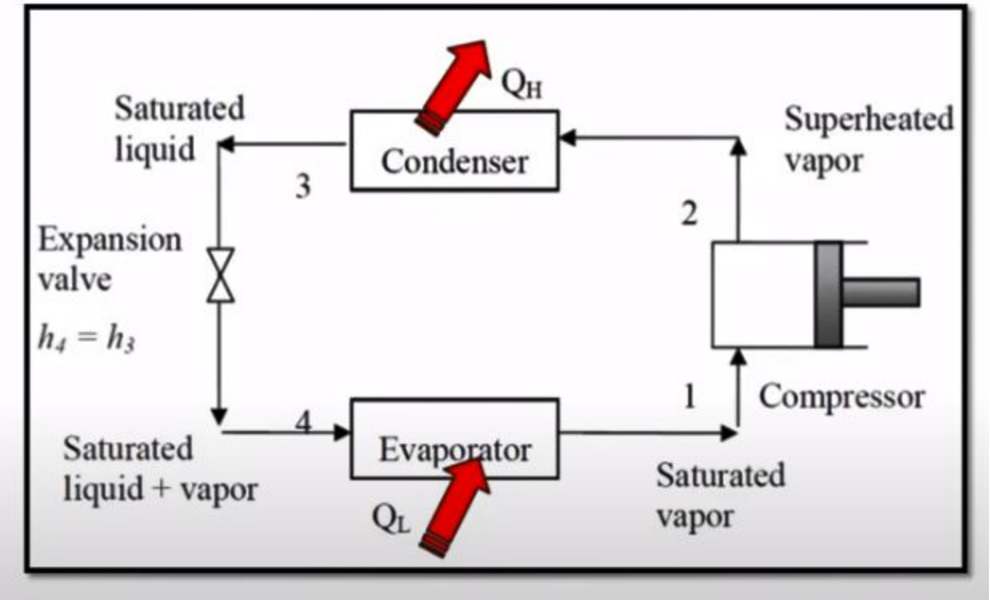
Refrigeration- It is a process of maintaining lower temperature compare to surrounding or It is a process of removing heat from a low temperature reservoir and transferring it to a high temperature reservoir.

- Refrigeration is a process, while Refrigerator is a device.

Air Conditioning- It is a process of removing heat and moisture from the interior of an occupied space, to improve the comfort of occupants.

Vapour Compression Refrigeration (VCR)

- It consist of mainly 4 components-
  - Compressor
  - Condensor
  - Expansion Valve
  - Evaporator
- In the all four component, the Refrigerant is flowing.
- Refrigerant - is a type of substance which is able to absorb the heat from the space to be cooled.
- From point 4, we have refrigerant in (liquid + vapour) state. When it passess through evaporator saturated vapor is created (after absorbing the heat).
- Now, at point 1, it is allowed to pass through the compressor. Compressor will increase the temperature and pressure of the refrigerant, which forms superheated vapor.

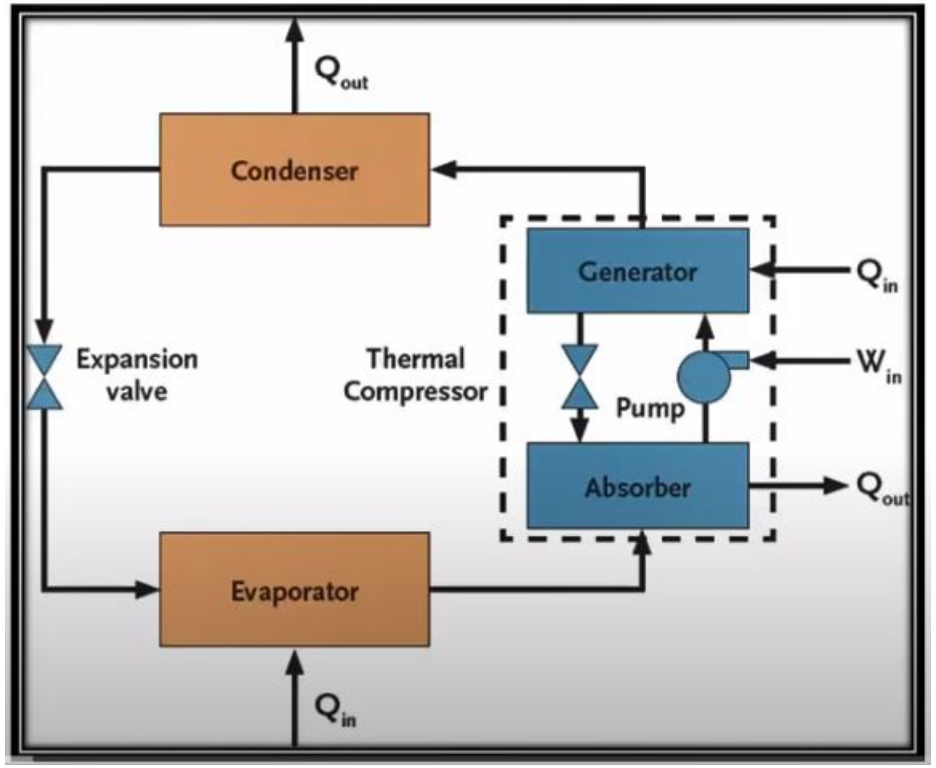


Saturated liquid - *The liquid which is about to vaporise.*
Example- hydrochlorofluorocarbons (used in most homes today) and Hydrofluorocarbons (used in cars), ammonia etc.

- Process 2 to 3 is condensation process. Here, refrigerant will reject heat to the surrounding. The state of refrigerant at point 3 is liquid state.
- This refrigerant is again allowed to flow through the expansion valve. Due to the expansion process, there is a reduction in temperature and pressure.

Vapour Absorption Refrigeration (VAR) System

- It is a type of system, in which instead of compressor we have 4 different elements- Generator, Absorber, Pump and Non-return Path.
- From the previous one, three components are common - Condenser, Expansion valve and Evaporator.
- At the inlet of evaporator, refrigerant - (Saturated liquid + vapor). As it passes through the evaporator it absorbs the heat from surrounding so that it can convert from (Saturated liquid + Vapor) to Saturated vapor.
- Now it is supplied to the Absorber which contains water. The vapor of refrigerand and water get mixed with each other after which converts it in pure liquid mixture of ammonia and water. The mixture of ammonia and water is pumped to the generator. In generator, heat (Q_{in}) is supplied which causes the evoporation of refrigerant (ammonia) and seperated from water. It happens due to the difference of their boiling point. The refrigerant will seperate first and leaving water.
- Superheated vapor will enter into the condensor while water flows back to the absorber via valve. The water releases Q_{out} heat in absorber and again mixed with new refrigerant.



In condensor, the refrigerand condenses by leaving the heat Q_{out} and process continues..

Difference Between VCR and VAR

SL.NO	PRINCIPLE	VAPOR COMPRESSION SYSTEM	VAPOR ABSORPTION SYSTEM
1	WORKING	Refrigerant vapor is compressed	Refrigerant vapor is absorbed & heated
2	TYPE OF ENERGY SUPLIED	Works on mechanical energy	Works on heat energy
3	COP	Higher	Lower
4	CAPACITY	can produce upto 1000 TOR	Can produce more than 1000 TOR
5	NOISE	More due to presence of compressor	Quiet in operation as there is no compressor
6	LEAKAGE	Due to high pressures, the chance of leakage of refrigerant is more	There is no leakage of the refrigerant
7	MAINTENANCE	High	Less
8	OPERATING COST	High, since electrical energy is used	Less because the thermal energy can be supplied from various sources

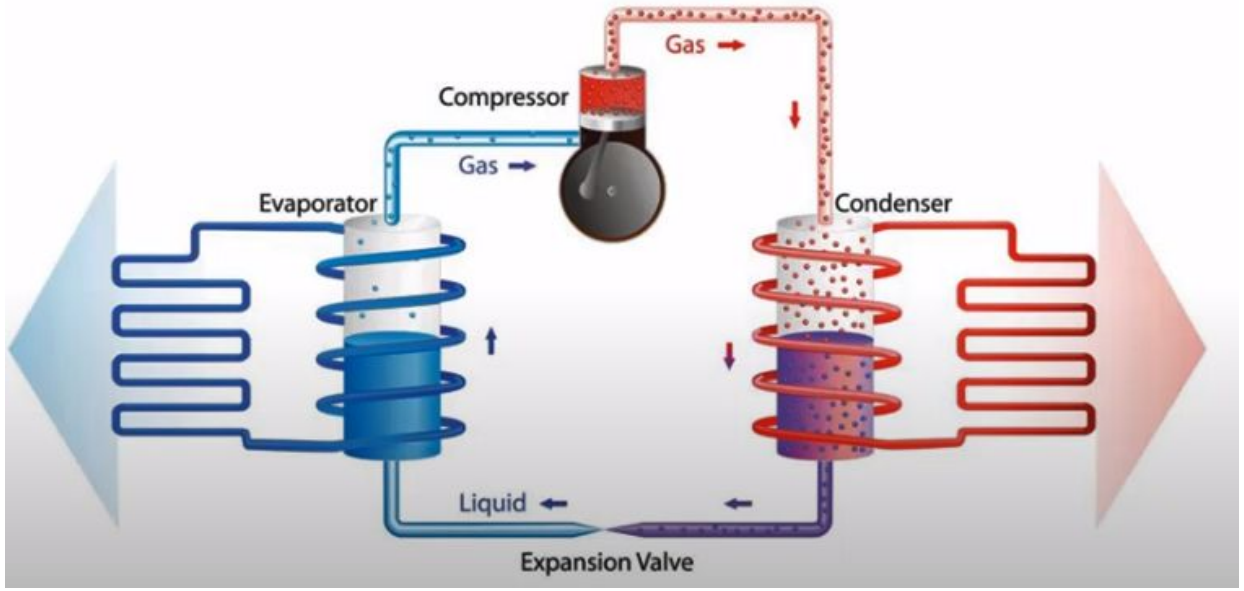
- TOR- ton of refrigeration,
- COP- Coefficient of Performance is a ratio of useful heating or cooling provided to work (energy) required.

Heat Pump

From construction point of view, heat pump and refrigerator are similar, but basic difference is -

In refrigerator, we use it for cooling application while in heat pump we use it for heating application.

The rejected heat from the condenser is utilized in heat pump for heating applications.



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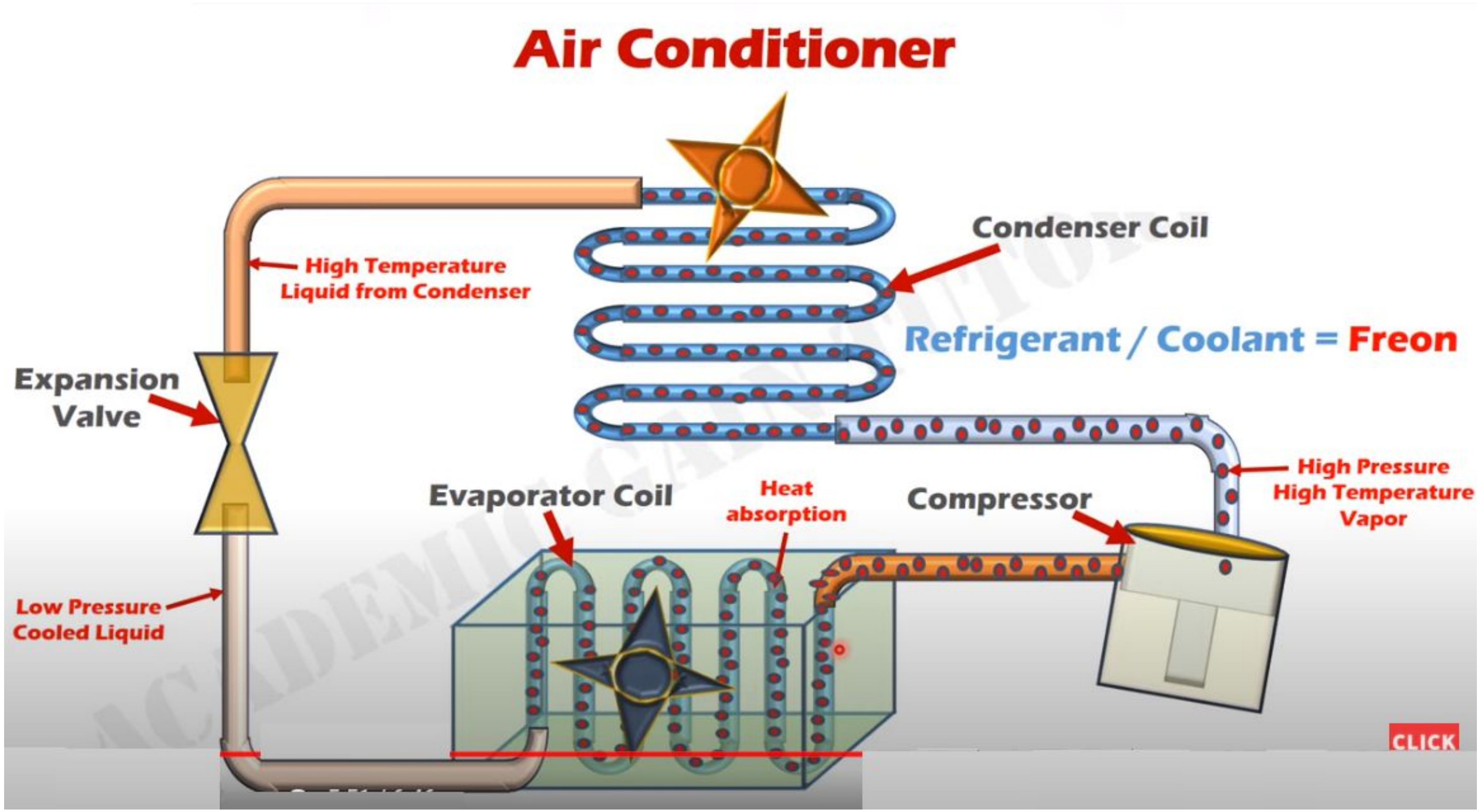

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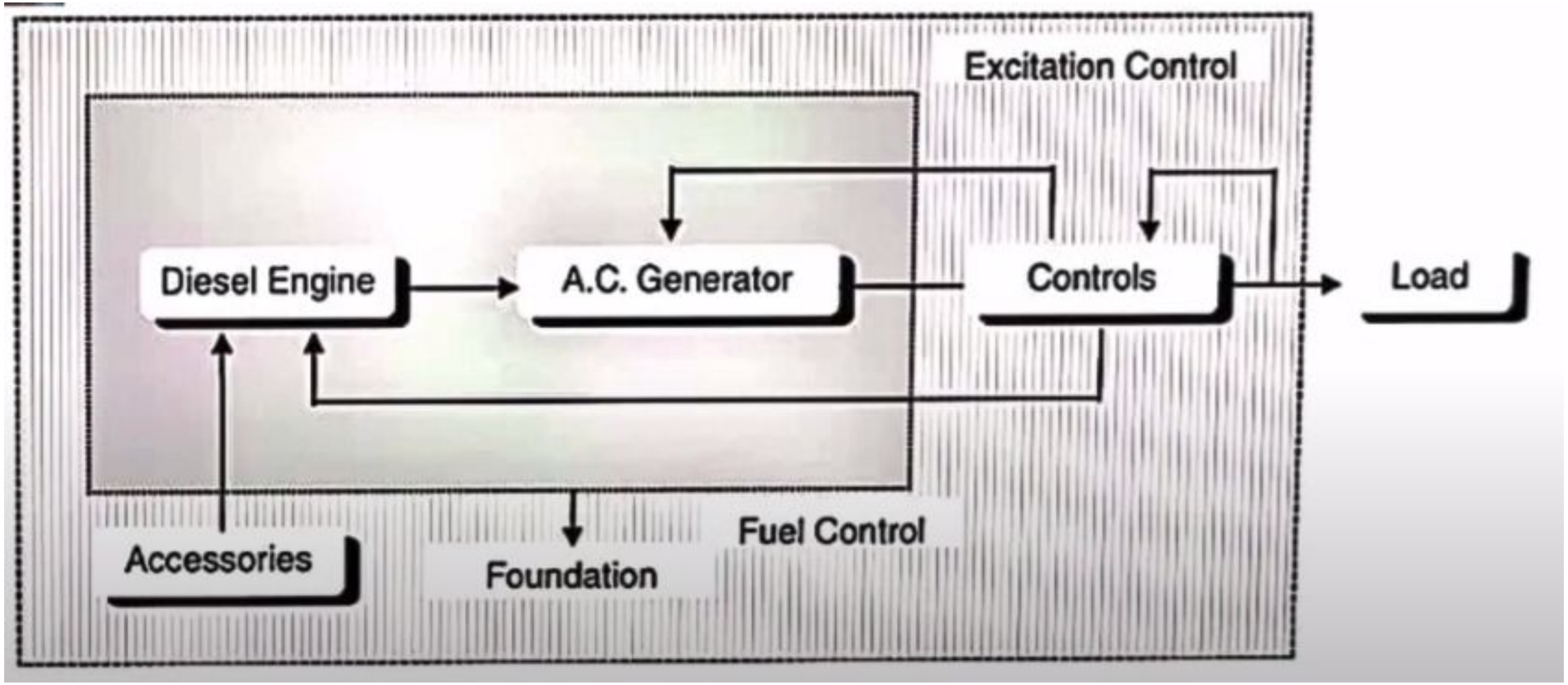
Factors affecting performance of Referigeration and Air Conditioning System-

- Maintenance of Heat Exchanger Surface
- System Design Features
- Capacity and control of Energy Efficiency
- Design of Process Heat Exchangers

DG Generator Sets (DG Sets)

- Consist of the following components-
   - Diesel Engine and its accessories
   - AC Generator
   - Control System and Switch Gear
   - The foundation and power house civil works
   - Connected loads

Block Diagram of DG Set



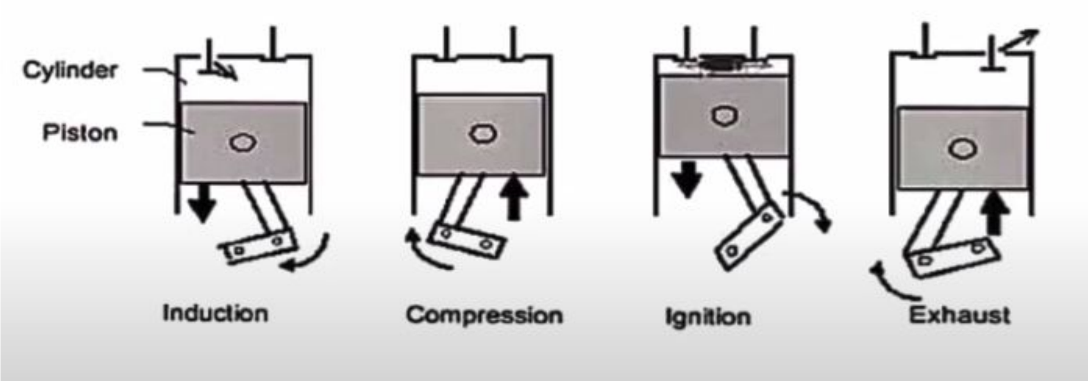
Continued...

- **<u>Diesel Engine-</u>** Diesel Engine is an prime mover which drives the alternator to produce electrical energy. In diesel engine, the air is drawn into the cylinder and compressed to high ratio (14:1 to 25:1).

- ❑ During the compression, the air heated the temperature of (700-900) °C.
- ❑ The fuel is injected at that time and it ignited spontaneously.

<u>Four Stroke Diesel Engine-</u>

1. Induction Stroke - Fresh air is drawn in.
2. Compression Stroke- Air is compressed.
3. Ignition and Power Stroke- Fuel is injected.
4. Exhaust Stroke- Gases gets exhausted



Diesel Genetrator Power Plants

- These power plants used in emergencies and used in small power systems. The main reason of using these power plants is its high efficiency of diesel engines compared with gas and steam turbines.

Advantages-

- Low installation cost
- High efficiency (43%-45%)
- Minimum cooling water required
- Short startup time.

Energy Efficiency Opportunities in DG Sets

- Ensure steady load conditions on DG Sets and provide and provide cold dustfree air to intake.
- Improve air filteration.
- Ensure fuel oil storage, handling as per recommendations of manufacturer.
- Consider fuel oil additives if they benefit oil properties.
- Caliberate fuel injection pump frequently.
- In case base load operation condition, considerwaste heat recovery system adoption.
- Consider parallel operation among DG Sets for improved loading and fuel economy.
- Carry out regular field trials to monitor the DG performance and maintenance planning as per the requirement.

https://www.youtube.com/watch?v=ib1OwRAGux4&ab_channel=e-BRILLIANTMINDS

Energy Efficiency in Building

- In current times the rapid infrastructure growth and looming energy crisis, there is a strong need to address and encorporate good practices for efficient energy and resource use while planning for building, be it for residential purpose or for commercial application.
- Some such good practices are presented in the Subsequent discussions:-
- **Green Buildings:**- *Feauture of Green Buildings are*-

 Use of energy efficient and Eco-FriendlyEquipment.

 Use of Recycled and Environment Friendly Building Materials

 Quality Indoor air for human safety and comfort

 Use of renewable energy

 Effective Controls and Building Management System

 Efficient use of water

 Use of Non-Toxic and Recycled Materials

 Effective use of existing landscapes

 Adoption of cost-effective and environement friendly technologies.

Benifits of Green Buildings

- (30-40)% reduction in operation cost.
- Green Corporate Image.
- Health and safety of the building occupant.
- Enhance Occupant Comfort.
- Improve productivity of occupant.

Typical Features of Green Buildings

- Reduction of Building footprints to minimise the impact on environment.
- Installation of high efficiency irrigation methods and selection of vegetation which have low water consumption.
- Harvesting of site energy.
- CFC free HVAC equipments.
- Energy efficient equipments for air conditioning and lighting system.
- Use of on-site renewable energy.
- Measurement & verification plan to ensure energy and water saving.
- Controls and building management system.

Continued...

- Segregation (isolation) collection and disposal of waste streams at source.
- Use of building materials having high recycled content.
- Use of rapidly renewable materials (the materials which could be replenished within a life cycle of 10 years).

Saving Opportunities in HVAC, Fans and Blowers

- **HVAC (Heating, Ventilation, and Air Conditioning)-**

1. BUILDING ORIENTATION/ ARCHITECTURAL FEATURES-

(i) Orientation
(ii) Double Glass
(iii) Insulation on roof
(iv) No Leakage (FromWindows/ Doors/ Ceiling)
(v) Long side should be having minimum heat gain.
(vi) Plant room and Air Handling Unit (AHU) locations should be such that ducting/ piping are minimum.
(vii) Fresh air intake should be sufficient to avoid "Sick Building Syndrome"
(viii) Sun shades over the glass area with proper inclination to avoid direct sunrays.
(viii) Partitions and closure of air grills of unutilized conditioned space.

Continued...

2. ESTABLISHING BASELINE PERFORMANCE INDICES:

- Usage time schedule – Working hours, holidays etc.
- tons / Sqr. Meter
- kW / ton
- kWh / day
- kWh / year

3. AUTOMATION AND BUILDING MANAGEMENT SYSTEM: Automation and building management systems are now increasingly used in the airconditioning systems for centralized monitoring and controlling the operations so as to ensure optimum operations of all the machines without any wastage of energy in overcooling or overheating of the areas.

4. VARIABLE VOLTAGE AND VARIABLE FREQUENCY DRIVES [VVVF] OR VARIABLE SPEED DRIVE:

In the VVVFD systems , the voltage and the frequency of electric supply to the induction motors for fans , pumps and compressors can be steplessly varied to control the speed of the motor in tune with the load requirements.

5.HEAT RECOVERY WHEEL AND DESICCANT (sustain a state of dryness) COOLING SYSTEM FOR FRESH AIR :

6. ROOF TOP CHILLERS: Roof top chillers are now increasingly used as they can be mounted on the

roof and the costly built up space inside the building can be saved.

- A chiller is a machine that removes heat from a liquid coolant via a vapor-compression, adsorption refrigeration, or absorption refrigeration cycles.

7. GEOTHERMAL SYSTEMS: Solar Energy Centre, Gulpahari Gurgaon constructed by TERI has used innovative scheme of providing airconditioning by harnessing in geo-thermal energy.

Continued...

• Fans and Blower-

Energy Saving Opportunities

Minimizing demand on the fan.

1. Minimising excess air level in combustion systems to reduce FD fan and ID fan load.
2. Minimising air in-leaks in hot flue gas path to reduce ID fan load, especially in case of kilns, boiler plants, furnaces, etc. Cold air in-leaks increase ID fan load tremendously, due to density increase of flue gases and in-fact choke up the capacity of fan, resulting as a bottleneck for boiler / furnace itself.
3. In-leaks / out-leaks in air conditioning systems also have a major impact on energy efficiency and fan power consumption and need to be minimized.

The findings of performance assessment trials will automatically indicate potential areas for improvement, which could be one or a more of the following:

1. Change of impeller by a high efficiency impeller along with cone.
2. Change of fan assembly as a whole, by a higher efficiency fan
3. Impeller de-rating (by a smaller dia impeller)
4. Change of metallic / Glass reinforced Plastic (GRP) impeller by the more energy efficient hollow FRP impeller with aerofoil design, in case of axial flow fans, where significant savings have been reported
5. Fan speed reduction by pulley dia modifications for derating
6. Option of two speed motors or variable speed drives for variable duty conditions
7. Option of energy efficient flat belts, or, cogged raw edged V belts, in place of conventional V belt systems, for reducing transmission losses.
8. Adopting inlet guide vanes in place of discharge damper control
9. Minimizing system resistance and pressure drops by improvements in duct system

- Industrial Process Fans are FD and ID.
- The main difference between a Forced Draft (FD) Fan and Induced Draft (ID) Fan:-
  - FD fan forces outside air into the heating system.
  - ID fan draws flue gases from the system out into the atmosphere.
  - So we can say ID Fans are more safe for boilers.

5. FANS AND BLOWERS

Syllabus
Fans and blowers: Types, Performance evaluation, Efficient system operation, Flow control strategies and energy conservation opportunities

5.1 Introduction

Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers, or other components in a fan system. The fan rotor receives energy from a rotating shaft and transmits it to the air.

Difference between Fans, Blowers and Compressors

Fans, blowers and compressors are differentiated by the method used to move the air, and by the system pressure they must operate against. As per American Society of Mechanical Engineers (ASME) the specific ratio - the ratio of the discharge pressure over the suction pressure – is used for defining the fans, blowers and compressors (see Table 5.1).

TABLE 5.1 DIFFERENCES BETWEEN FANS, BLOWER AND COMPRESSOR

Equipment	Specific Ratio	Pressure rise (mmWg)
Fans	Up to 1.11	1136
Blowers	1.11 to 1.20	1136 – 2066
Compressors	more than 1.20	–

5.2 Fan Types

Fan and blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Fan efficiencies differ from design to design and also by types. Typical ranges of fan efficiencies are given in Table 5.2.

Fans fall into two general categories: centrifugal flow and axial flow.

In centrifugal flow, airflow changes direction twice - once when entering and second when leaving (forward curved, backward curved or inclined, radial) (see Figure 5.1).

In axial flow, air enters and leaves the fan with no change in direction (propeller, tubeaxial, vaneaxial) (see Figure 5.2).

TABLE 5.2 FAN EFFICIENCIES

Type of fan	Peak Efficiency Range
Centrifugal Fans	
Airfoil, backward curved/inclined	79–83
Modified radial	72–79
Radial	69–75
Pressure blower	58–68
Forward curved	60–65
Axial fan	
Vane axial	78–85
Tube axial	67–72
Propeller	45–50

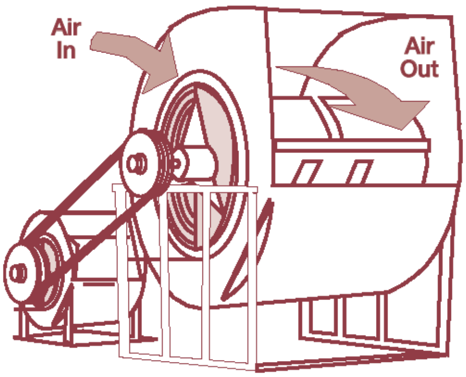


Figure 5.1 Centrifugal Fan

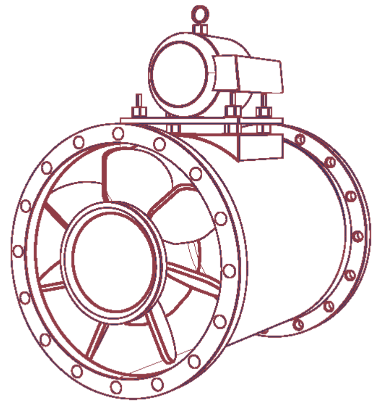


Figure 5.2 Axial Fan

Centrifugal Fan: Types

The major types of centrifugal fan are: *radial, forward curved and backward curved* (see Figure 5.3).

Radial fans are industrial workhorses because of their high static pressures (upto 1400 mm WC) and ability to handle heavily contaminated airstreams. Because of their simple design, radial fans are well suited for high temperatures and medium blade tip speeds.

Forward-curved fans are used in clean environments and operate at lower temperatures. They are well suited for low tip speed and high-airflow work - they are best suited for moving large volumes of air against relatively low pressures.

Backward-inclined fans are more efficient than forward-curved fans. Backward-inclined fans reach their peak power consumption and then power demand drops off well within their useable airflow range. Backward-inclined fans are known as "non-overloading" because changes in static pressure do not overload the motor.

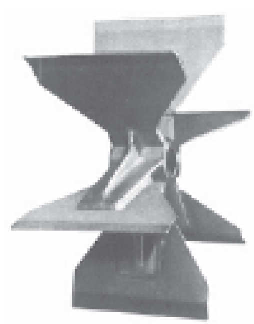
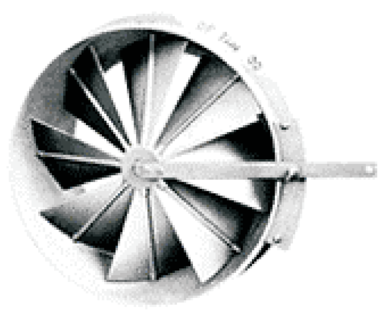
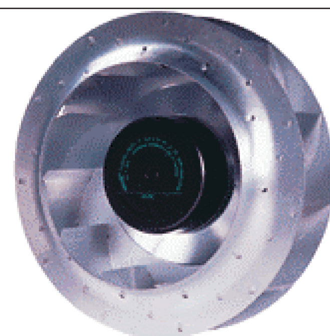
Paddle Blade (Radial blade)	Forward Curved (Multi-Vane)	Backward Curved
		

Figure 5.3 Types of Centrifugal Fans

Axial Flow Fan: Types

The major types of axial flow fans are: *tube axial, vane axial and propeller (see Figure 5.4.)*

Tubeaxial fans have a wheel inside a cylindrical housing, with close clearance between blade and housing to improve airflow efficiency. The wheel turn faster than propeller fans, enabling operation under high-pressures 250 – 400 mm WC. The efficiency is up to 65%.

Vaneaxial fans are similar to tubeaxials, but with addition of guide vanes that improve efficiency by directing and straightening the flow. As a result, they have a higher static pressure with less dependence on the duct static pressure. Such fans are used generally for pressures upto 500 mmWC. Vaneaxials are typically the most energy-efficient fans available and should be used whenever possible.

Propeller fans usually run at low speeds and moderate temperatures. They experience a large change in airflow with small changes in static pressure. They handle large volumes of air at low pressure or free delivery. Propeller fans are often used indoors as exhaust fans. Outdoor applications include air-cooled condensers and cooling towers. Efficiency is low – approximately 50% or less.

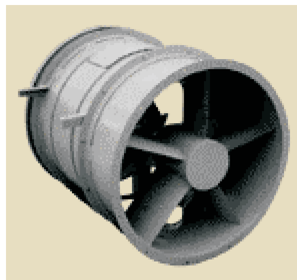
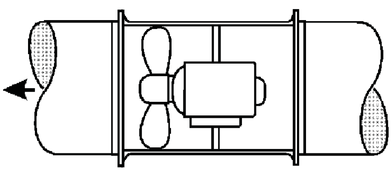
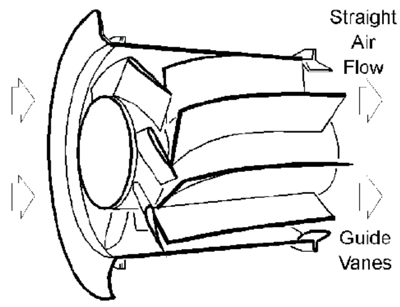
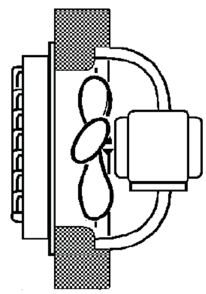
Tube Axial	Vane Axial	Propeller
		
		

Figure 5.4 Types of Axial Fans

The different types of fans, their characteristics and typical applications are given in Table 5.3.

Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 1.20 kg/cm^2. They are also used to produce negative pressures for industrial vacuum systems. Major types are: centrifugal blower and positive-displacement blower.

Centrifugal blowers look more like centrifugal pumps than fans. The impeller is typically gear-driven and rotates as fast as 15,000 rpm. In multi-stage blowers, air is accelerated as it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient.

Centrifugal blowers typically operate against pressures of 0.35 to 0.70 kg/cm^2, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure

TABLE 5.3 TYPES OF FANS, CHARACTERISTICS, AND TYPICAL APPLICATIONS

Centrifugal Fans			Axial-flow Fans		
Type	**Characteristics**	**Typical Applications**	**Type**	**Characteristics**	**Typical Applications**
Radial	High pressure, medium flow, efficiency close to tube-axial fans, power increases continuously	Various industrial applications, suitable for dust laden, moist air/gases	Propeller	Low pressure, high flow, low efficiency, peak efficiency close to point of free air delivery (zero static pressure)	Air-circulation, ventilation, exhaust
Forward-curved blades	Medium pressure, high flow, dip in pressure curve, efficiency higher than radial fans, power rises continuously	Low pressure HVAC, packaged units, suitable for clean and dust laden air / gases	Tube-axial	Medium pressure, high flow, higher efficiency than propeller type, dip in pressure-flow curve before peak pressure point.	HVAC, drying ovens, exhaust systems
Backward curved blades	High pressure, high flow, high efficiency, power reduces as flow increases beyond point of highest efficiency	HVAC, various industrial applications forced draft fans, etc.	Vane-axial	High pressure, medium flow, dip in pressure-flow curve, use of guide vanes improves efficiencyexhausts	High pressure applications including HVAC systems,
Airfoil type	Same as backward curved type, highest efficiency	Same as backward curved, but for clean air applications			

increases, which can be a disadvantage in material conveying systems that depend on a steady air volume. Because of this, they are most often used in applications that are not prone to clogging.

Positive-displacement blowers have rotors, which "trap" air and push it through housing. Positive-displacement blowers provide a constant volume of air even if the system pressure varies. They are especially suitable for applications prone to clogging, since they can produce enough pressure - typically up to 1.25 kg/cm^2 - to blow clogged materials free. They turn much slower than centrifugal blowers (e.g. 3,600 rpm), and are often belt driven to facilitate speed changes.

5.3 Fan Performance Evaluation and Efficient System Operation

System Characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the configuration of ducts, pickups, elbows and the pressure drops across equipment-for example backfilter or cyclone. *The system resistance varies with the square of the volume of air flowing through the system*. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.

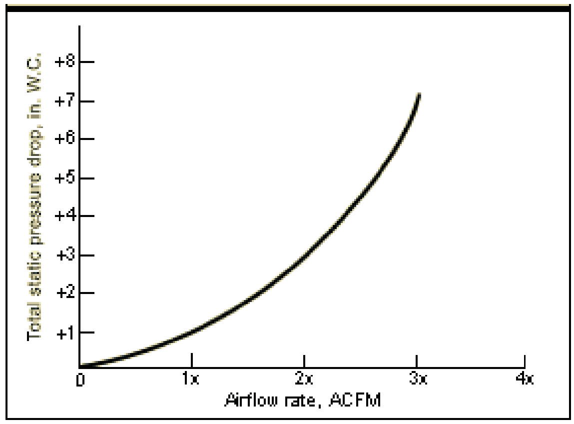


Figure 5.5 System Characteristics

Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics.

In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated. Typically a system resistance curve (see Figure 5.5) is generated with for various flow rates on the x-axis and the associated resistance on the y-axis.

Fan Characteristics

Fan characteristics can be represented in form of fan curve(s). The fan curve is a performance curve for the particular fan under a specific set of conditions. The fan curve is a graphical representation of a number of inter-related parameters. Typically a curve will be developed for a given set of conditions usually including: fan volume, system static pressure, fan speed, and brake horsepower required to drive the fan under the stated conditions. Some fan curves will also include an efficiency curve so that a system designer will know where on that curve the fan will be operating under the chosen conditions (see Figure 5.6). In the many curves shown in the Figure, the curve static pressure (SP) vs. flow is especially important.

The intersection of the system curve and the static pressure curve defines the operating point. When the system resistance changes, the operating point also changes. Once the operating point is fixed, the power required could be found by following a vertical line that passes through the operating point to an intersection with the power (BHP) curve. A horizontal line drawn through the intersection with the power curve will lead to the required power on the right vertical axis. In the depicted curves, the fan efficiency curve is also presented.

System Characteristics and Fan Curves

In any fan system, the resistance to air flow (pressure) increases when the flow of air is increased. As mentioned before, it varies as the square of the flow. The pressure required by a system over a range of flows can be determined and a "system performance curve" can be developed (shown as SC) (see Figure 5.7).

This system curve can then be plotted on the fan curve to show the fan's actual operating point at "A" where the two curves (N_1 and SC_1) intersect. This operating point is at air flow Q_1 delivered against pressure P_1.

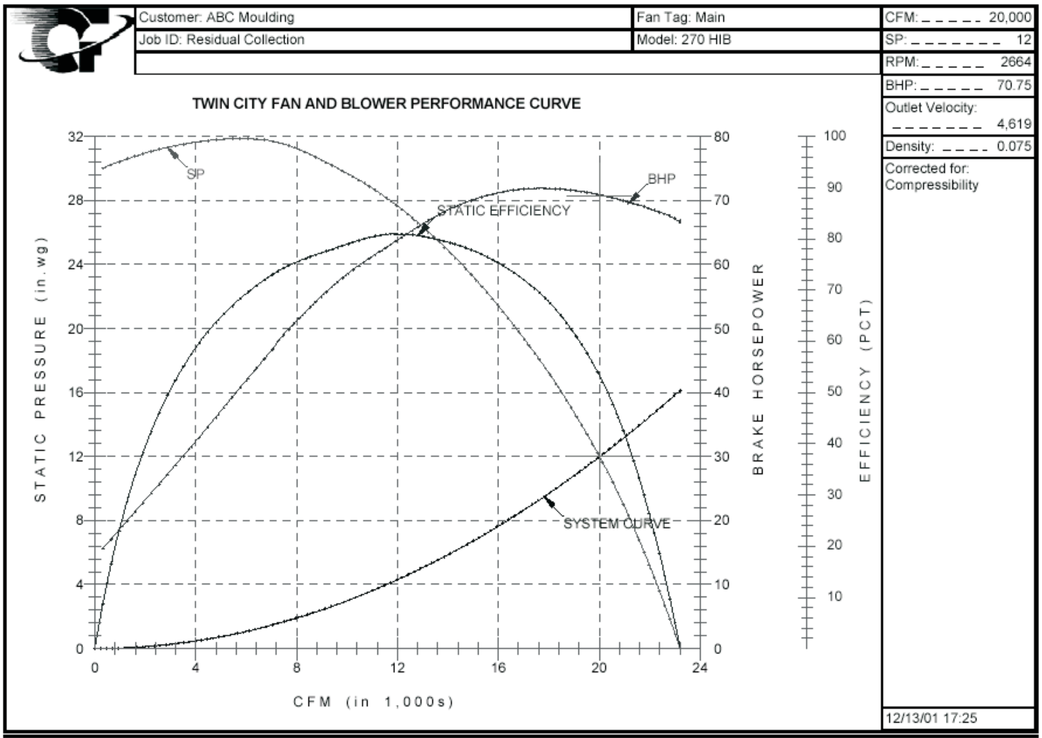


Figure 5.6 Fan Characteristics Curve by Manufacturer

A fan operates along a performance given by the manufacturer for a particular fan speed. (The fan performance chart shows performance curves for a series of fan speeds.) At fan speed N_1, the fan will operate along the N_1 performance curve as shown in Figure 5.7. The fan's actu-

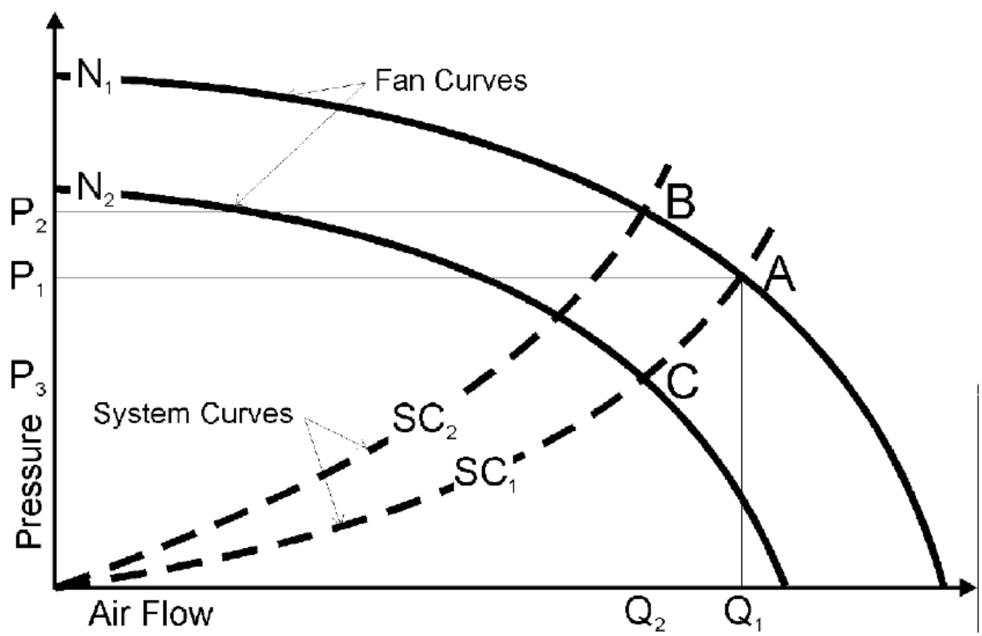


Figure 5.7 System Curve

al operating point on this curve will depend on the system resistance; fan's operating point at "A" is flow (Q_1) against pressure (P_1).

Two methods can be used to reduce air flow from Q_1 to Q_2:

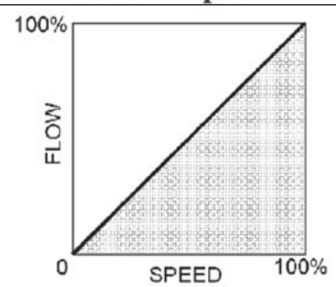
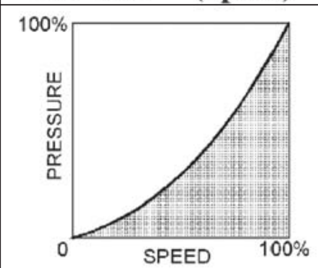
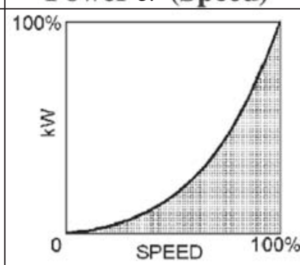
First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve (SC_2) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow Q_2 against higher pressure P_2.

Second method to reduce air flow is by reducing the speed from N_1 to N_2, keeping the damper fully open. The fan would operate at "C" to provide the same Q_2 air flow, but at a lower pressure P_3.

Thus, reducing the fan speed is a much more efficient method to decrease airflow since less power is required and less energy is consumed.

Fan Laws

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

<table>
<tr><th>Flow α Speed</th><th>Pressure α (Speed)^2</th><th>Power α (Speed)^3</th></tr>
<tr><td>
</td><td>
</td><td>
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<tr><td>$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$</td><td>$\frac{SP_1}{SP_2} = \left(\frac{N_1}{N_2}\right)^2$</td><td>$\frac{kW_1}{kW_2} = \left(\frac{N_1}{N_2}\right)^3$</td></tr>
<tr><td>Varying the RPM by 10% decreases or increases air delivery by 10%.</td><td>Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%</td><td>Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%</td></tr>
</table>

Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)

5.4 Fan Design and Selection Criteria

Precise determination of air-flow and required outlet pressure are most important in proper selection of fan type and size. The air-flow required depends on the process requirements; normally determined from heat transfer rates, or combustion air or flue gas quantity to be handled. System pressure requirement is usually more difficult to compute or predict. Detailed analysis should be carried out to determine pressure drop across the length, bends, contractions and expansions in the ducting system, pressure drop across filters, drop in branch lines, etc. These pressure drops should be added to any fixed pressure required by the process (in the case of ventilation fans there is no fixed pressure requirement). Frequently, a very conservative approach is adopted allocating large safety margins, resulting in over-sized fans which operate at flow rates much below their design values and, consequently, at very poor efficiency.

Once the system flow and pressure requirements are determined, the fan and impeller type are then selected. For best results, values should be obtained from the manufacturer for specific fans and impellers.

The choice of fan type for a given application depends on the magnitudes of required flow and static pressure. For a given fan type, the selection of the appropriate impeller depends additionally on rotational speed. Speed of operation varies with the application. High speed small units are generally more economical because of their higher hydraulic efficiency and relatively low cost. However, at low pressure ratios, large, low-speed units are preferable.

Fan Performance and Efficiency

Typical static pressures and power requirements for different types of fans are given in the Figure 5.8.

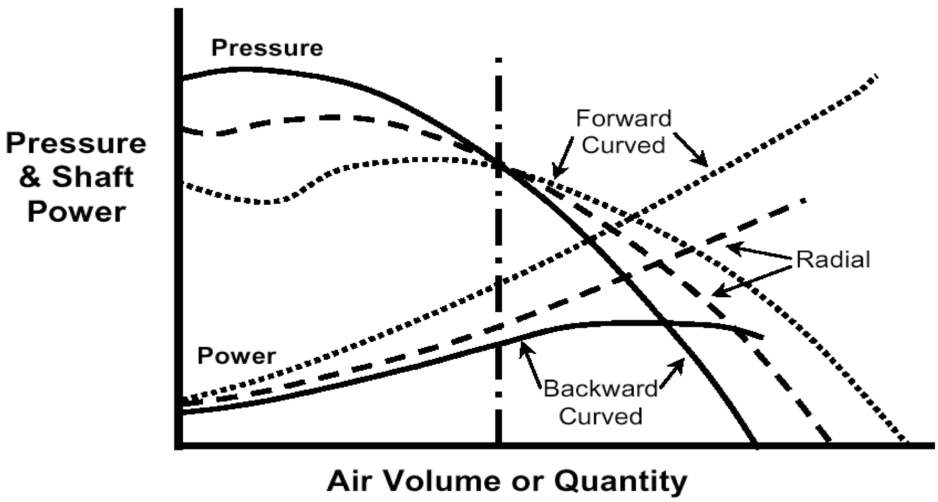

Figure 5.8 Fan Static Pressure and Power Requirements for Different Fans

Fan performance characteristics and efficiency differ based on fan and impeller type (See Figure 5.9).

In the case of centrifugal fans, the hub-to-tip ratios (ratio of inner-to-outer impeller diameter) the tip angles (angle at which forward or backward curved blades are curved at the blade tip - at the base the blades are always oriented in the direction of flow), and the blade width determine the pressure developed by the fan.

Forward curved fans have large hub-to-tip ratios compared to backward curved fans and produce lower pressure.

Radial fans can be made with different heel-to-tip ratios to produce different pressures.

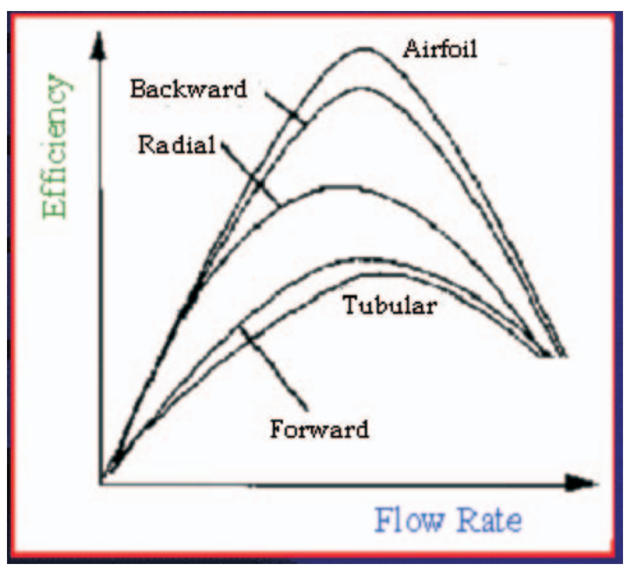

Figure 5.9 Fan Performance Characteristics Based on Fans/ Impellers

At both design and off-design points, backward-curved fans provide the most stable operation. Also, the power required by most backward –curved fans will decrease at flow higher than design values. A similar effect can be obtained by using inlet guide vanes instead of replacing the impeller with different tip angles. Radial fans are simple in construction and are preferable for high-pressure applications.

Forward curved fans, however, are less efficient than backward curved fans and power rises continuously with flow. Thus, they are generally more expensive to operate despite their lower first cost.

Among centrifugal fan designs, aerofoil designs provide the highest efficiency (upto 10% higher than backward curved blades), but their use is limited to clean, dust-free air.

Axial-flow fans produce lower pressure than centrifugal fans, and exhibit a dip in pressure before reaching the peak pressure point. Axial-flow fans equipped with adjustable / variable pitch blades are also available to meet varying flow requirements.

Propeller-type fans are capable of high-flow rates at low pressures. Tube-axial fans have medium pressure, high flow capability and are not equipped with guide vanes.

Vane-axial fans are equipped with inlet or outlet guide vanes, and are characterized by high pressure, medium flow-rate capabilities.

Performance is also dependant on the fan enclosure and duct design. Spiral housing designs with inducers, diffusers are more efficient as compared to square housings. Density of inlet air is another important consideration, since it affects both volume flow-rate and capacity of the fan to develop pressure. Inlet and outlet conditions (whirl and turbulence created by grills, dampers, etc.) can significantly alter fan performance curves from that provided by the manufacturer (which are developed under controlled conditions). Bends and elbows in the inlet or outlet ducting can change the velocity of air, thereby changing fan characteristics (the pressure drop in these elements is attributed to the system resistance). All these factors, termed as System Effect Factors, should, therefore, be carefully evaluated during fan selection since they would modify the fan performance curve.

Centrifugal fans are suitable for low to moderate flow at high pressures, while axial-flow fans are suitable for low to high flows at low pressures. Centrifugal fans are generally more expensive than axial fans. Fan prices vary widely based on the impeller type and the mounting (direct-or-belt-coupled, wall-or-duct-mounted). Among centrifugal fans, aerofoil and backward-curved blade designs tend to be somewhat more expensive than forward-curved blade designs and will typically provide more favourable economics on a lifecycle basis. Reliable cost comparisons are difficult since costs vary with a number of application-specific factors. A careful technical and economic evaluation of available options is important in identifying the fan that will minimize lifecycle costs in any specific application.

Safety margin

The choice of safety margin also affects the efficient operation of the fan. In all cases where the fan requirement is linked to the process/other equipment, the safety margin is to be decided, based on the discussions with the process equipment supplier. In general, the safety margin can be 5% over the maximum requirement on flow rate.

In the case of boilers, the induced draft (ID) fan can be designed with a safety margin of 20% on volume and 30% on head. The forced draft (FD) fans and primary air (PA) fans do not require any safety margins. However, safety margins of 10 % on volume and 20% on pressure are maintained for FD and PA fans.

Some pointers on fan specification

The right specification of the parameters of the fan at the initial stage, is pre-requisite for choosing the appropriate and energy efficient fan.

The user should specify following information to fan manufacturer to enable right selection:

Design operating point of the fan – volume and pressure

Normal operating point – volume and pressure

Maximum continuous rating

Low load operation - This is particularly essential for units, which in the initial few years may operate at lower capacities, with plans for upgradation at a later stage. The initial low load and the later higher load operational requirements need to be specified clearly, so that, the manufacturer can supply a fan which can meet both the requirements, with different sizes of impeller.

Ambient temperature – The ambient temperatures, both the minimum and maximum, are to be specified to the supplier. This affects the choice of the material of construction of the impeller.

The maximum temperature of the gas at the fan during upset conditions should be specified to the supplier. This will enable choice of the right material of the required creep strength.

Density of gas at different temperatures at fan outlet

Composition of the gas – This is very important for choosing the material of construction of the fan.

Dust concentration and nature of dust – The dust concentration and the nature of dust (e.g. bagasse – soft dust, coal – hard dust) should be clearly specified.

The proposed control mechanisms that are going to be used for controlling the fan.

The operating frequency varies from plant-to-plant, depending on the source of power supply. Since this has a direct effect on the speed of the fan, the frequency prevailing or being maintained in the plant also needs to be specified to the supplier.

Altitude of the plant

The choice of speed of the fan can be best left to fan manufacturer. This will enable him to design the fan of the highest possible efficiency. However, if the plant has some preferred speeds on account of any operational need, the same can be communicated to the fan supplier.

Installation of Fan

The installation of fan and mechanical maintenance of the fan also plays a critical role in the efficiency of the fan. The following clearances (typical values) should be maintained for the efficient operation of the impeller.

Impeller Inlet Seal Clearances

- Axial overlap –5 to 10 mm for 1 metre plus dia impeller
- Radial clearance –1 to 2 mm for 1 metre plus dia impeller
- Back plate clearance –20 to 30 mm for 1 metre plus dia impeller
- Labyrinth seal clearance –0.5 to 1.5 mm

The inlet damper positioning is also to be checked regularly so that the "full open" and "full close" conditions are satisfied. The fan user should get all the details of the mechanical clearances from the supplier at the time of installation. As these should be strictly adhered to, for efficient operation of the fan, and a checklist should be prepared on these clearances. A check on these clearances should be done after every maintenance, so that efficient operation of the fan is ensured on a continuous basis.

System Resistance Change

The system resistance has a major role in determining the performance and efficiency of a fan. The system resistance also changes depending on the process. For example, the formation of the coatings / erosion of the lining in the ducts, changes the system resistance marginally. In some cases, the change of equipment (e.g. Replacement of Multi-cyclones with ESP / Installation of low pressure drop cyclones in cement industry) duct modifications, drastically shift the operating point, resulting in lower efficiency. In such cases, to maintain the efficiency as before, the fan has to be changed.

Hence, the system resistance has to be periodically checked, more so when modifications are introduced and action taken accordingly, for efficient operation of the fan.

5.5 Flow Control Strategies

Typically, once a fan system is designed and installed, the fan operates at a constant speed. There may be occasions when a speed change is desirable, i.e., when adding a new run of duct that requires an increase in air flow (volume) through the fan. There are also instances when the fan is oversized and flow reductions are required.

Various ways to achieve change in flow are: pulley change, damper control, inlet guide vane control, variable speed drive and series and parallel operation of fans.

Pulley Change

When a fan volume change is required on a permanent basis, and the existing fan can handle the change in capacity, the volume change can be achieved with a speed change. The simplest way to change the speed is with a pulley change. For this, the fan must be driven by a motor through a v-belt system. The fan speed can be increased or decreased with a change in the drive pulley or the driven pulley or in some cases, both pulleys. As shown in the Figure 5.10, a higher sized fan operating with damper control was downsized by reducing the motor (drive) pulley size from 8" to 6". The power reduction was 15 kW.

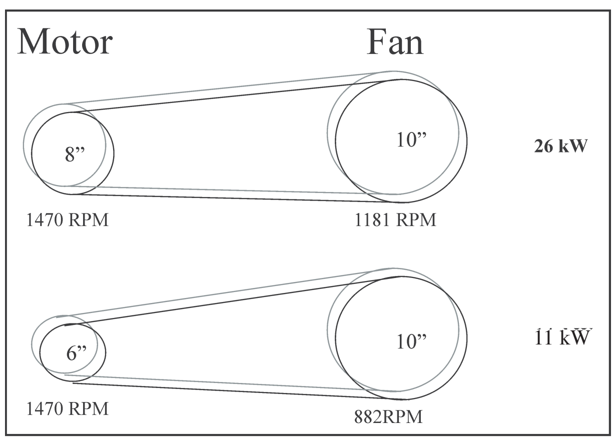

Figure 5.10 Pulley Change

Damper Controls

Figure 5.11 Damper change

Some fans are designed with damper controls (see Figure 5.11). Dampers can be located at inlet or outlet. Dampers provide a means of changing air volume by adding or removing system resistance. This resistance forces the fan to move up or down along its characteristic curve, generating more or less air without changing fan speed. However, dampers provide a limited amount of adjustment, and they are not particularly energy efficient.

Inlet Guide Vanes

Figure 5.12 Inlet Guide Vanes

Inlet guide vanes are another mechanism that can be used to meet variable air demand (see Figure 5.12). Guide vanes are curved sections that lay against the inlet of the fan when they are open. When they are closed, they extend out into the air stream. As they are closed, guide vanes pre-swirl the air entering the fan housing. This changes the angle at which the air is presented to the fan blades, which, in turn, changes the characteristics of the fan curve. Guide vanes are energy efficient for modest flow reductions – from 100 percent flow to about 80 percent. Below 80 percent flow, energy efficiency drops sharply.

Axial-flow fans can be equipped with variable pitch blades, which can be hydraulically or pneumatically controlled to change blade pitch, while the fan is at stationary. Variable-pitch blades modify the fan characteristics substantially and thereby provide dramatically higher energy efficiency than the other options discussed thus far.

Variable Speed Drives

Although, variable speed drives are expensive, they provide almost infinite variability in speed control. Variable speed operation involves reducing the speed of the fan to meet reduced flow requirements. Fan performance can be predicted at different speeds using the fan laws. Since power input to the fan changes as the cube of the flow, this will usually be the most efficient form of capacity control. However, variable speed control may not be economical for systems, which have infrequent flow variations. When considering variable speed drive, the efficiency of the control system (fluid coupling, eddy-current, VFD, etc.) should be accounted for, in the analysis of power consumption.

Series and Parallel Operation

Parallel operation of fans is another useful form of capacity control. Fans in parallel can be additionally equipped with dampers, variable inlet vanes, variable-pitch blades, or speed controls to provide a high degree of flexibility and reliability.

Combining fans in series or parallel can achieve the desired airflow without greatly increasing the system package size or fan diameter. Parallel operation is defined as having two or more fans blowing together side by side.

The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 5.13 shows, when a system curve is overlaid on the parallel performance curves, the higher the system resistance, the less increase in flow results with parallel fan operation. Thus, this type of application should only be used when the fans can operate in a low resistance almost in a free delivery condition.

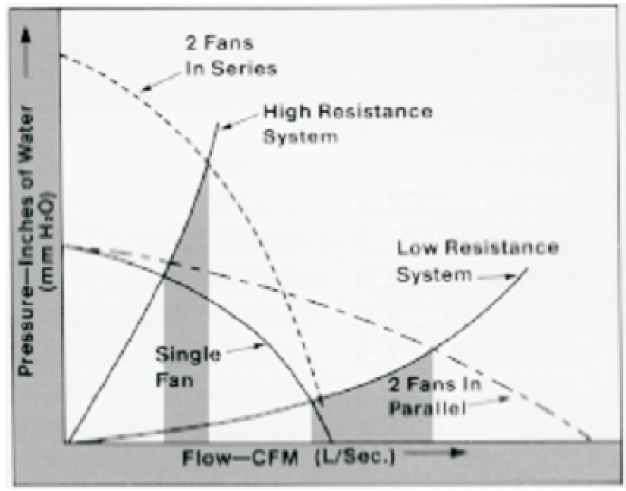

Figure 5.13 Series and Parallel Operation

Series operation can be defined as using multiple fans in a push-pull arrangement. By staging two fans in series, the static pressure capability at a given airflow can be increased, but again, not to double at every flow point, as the above Figure displays. In series operation, the best results are achieved in systems with high resistances.

In both series and parallel operation, particularly with multiple fans certain areas of the combined performance curve will be unstable and should be avoided. This instability is unpredictable and is a function of the fan and motor construction and the operating point.

Factors to be considered in the selection of flow control methods

Comparison of various volume control methods with respect to power consumption (%) required power is shown in Figure 5.14.

All methods of capacity control mentioned above have turn-down ratios (ratio of maximum–to–minimum flow rate) determined by the amount of leakage (slip) through the control elements. For example, even with dampers fully closed, the flow may not be zero due to leakage through the damper. In the case of variable-speed drives the turn-down ratio is limited by the control system. In many cases, the minimum possible flow will be determined by the characteristics of the fan itself. Stable operation of a fan requires that it operate in a region where the system curve has a positive slope and the fan curve has a negative slope.

The range of operation and the time duration at each operating point also serves as a guide to selection of the most suitable capacity control system. Outlet damper control due to its simplicity, ease of operation, and low investment cost, is the most prevalent form of capacity control. However, it is the most inefficient of all methods and is best suited for situations where only small, infrequent changes are required (for example, minor process variations due to seasonal changes. The economic advantage of one method over the other is determined by the time duration over which the fan operates at different operating points. The frequency of flow change is another important determinant. For systems requiring frequent flow control, damper adjustment may not be convenient. Indeed, in many plants, dampers are not easily accessible and are left at some intermediate position to avoid frequent control.

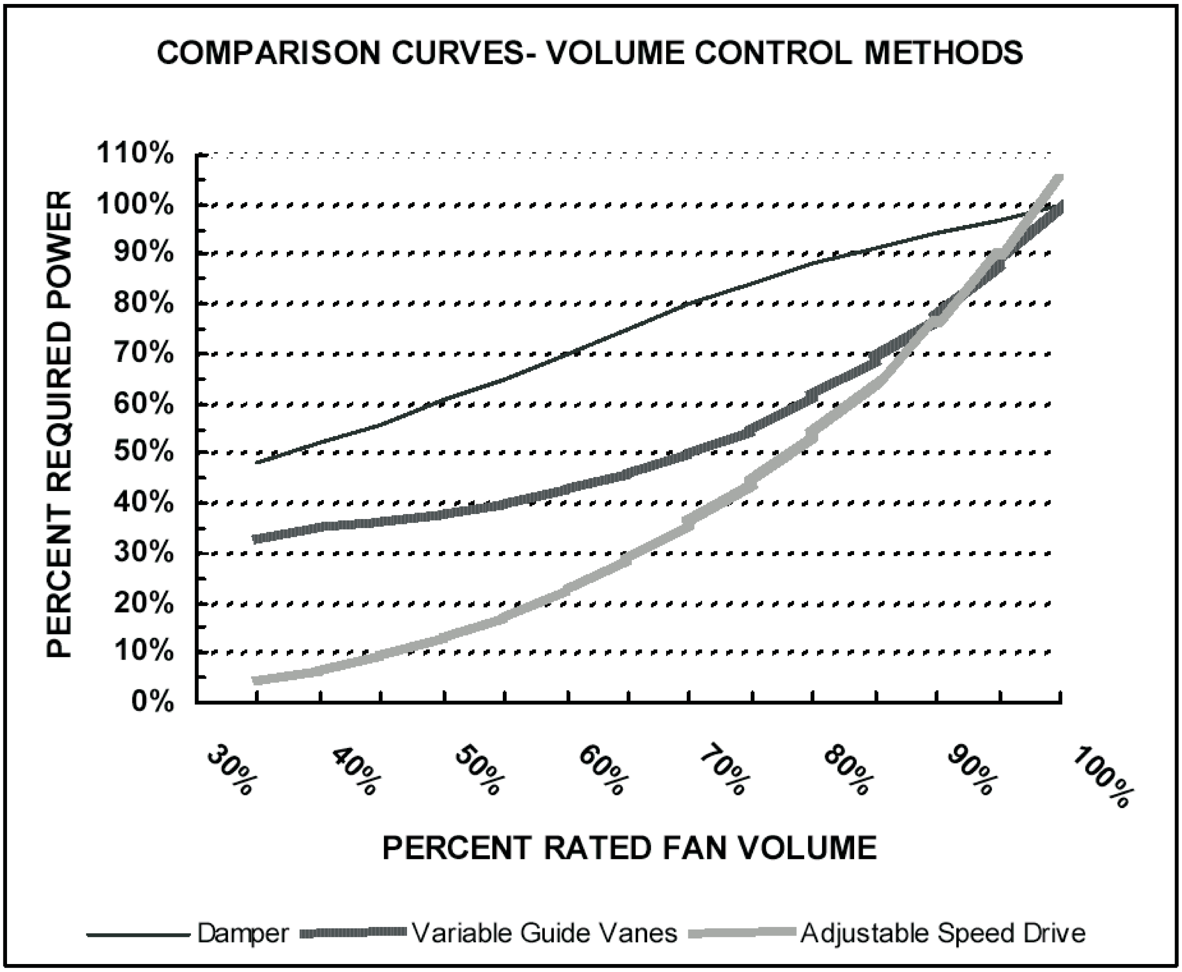

Figure 5.14 Comparison: Various Volume Control Methods

5.6 Fan Performance Assessment

The fans are tested for field performance by measurement of flow, head, temperature on the fan side and electrical motor kW input on the motor side.

Air flow measurement

Static pressure
Static pressure is the potential energy put into the system by the fan. It is given up to friction in the ducts and at the duct inlet as it is converted to velocity pressure. At the inlet to the duct, the static pressure produces an area of low pressure (see Figure 5.15).

Velocity pressure
Velocity pressure is the pressure along the line of the flow that results from the air flowing through the duct. The velocity pressure is used to calculate air velocity.

Total pressure
Total pressure is the sum of the static and velocity pressure. Velocity pressure and static pressure can change as the air flows though different size ducts, accelerating and decelerating the

velocity. The total pressure stays constant, changing only with friction losses. The illustration that follows shows how the total pressure changes in a system.

The fan flow is measured using pitot tube manometer combination, or a flow sensor (differential pressure instrument) or an accurate anemometer. Care needs to be taken regarding number of traverse points, straight length section (to avoid turbulent flow regimes of measurement) up stream and downstream of measurement location. The measurements can be on the suction or discharge side of the fan and preferably both where feasible.

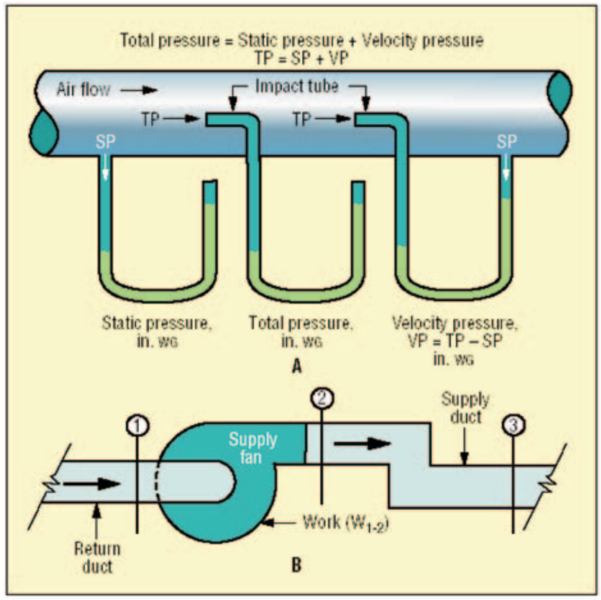

Figure 5.15 Static, Total and Velocity Pressure

Measurement by Pitot tube

The Figure 5.16 shows how velocity pressure is measured using a pitot tube and a manometer. Total pressure is measured using the inner tube of pitot tube and static pressure is measured using the outer tube of pitot tube. When the inner and outer tube ends are connected to a manometer, we get the velocity pressure. For measuring low velocities, it is preferable to use an inclined tube manometer instead of U tube manometer.

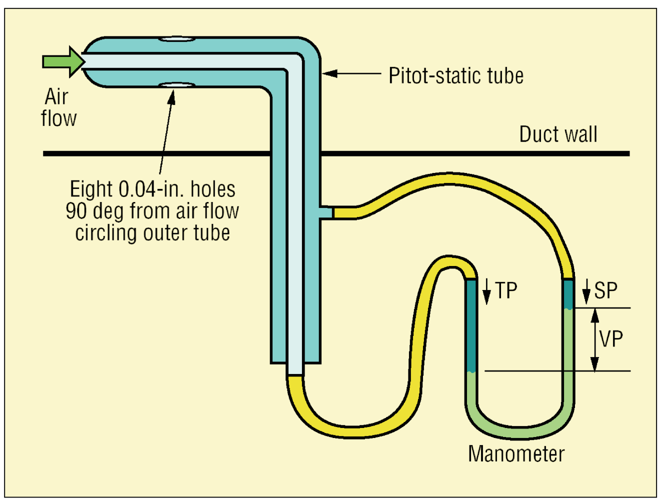

Figure 5.16 Velocity Measurement Using Pitot Tube

Measurements and Calculations

Velocity pressure/velocity calculation

When measuring velocity pressure the duct diameter (or the circumference from which to calculate the diameter) should be measured as well. This will allow us to calculate the velocity and the volume of air in the duct. In most cases, velocity must be measured at several places in the same system.

The velocity pressure varies across the duct. Friction slows the air near the duct walls, so the velocity is greater in the center of the duct. The velocity is affected by changes in the ducting configuration such as bends and curves. The best place to take measurements is in a section of duct that is straight for at least 3–5 diameters after any elbows, branch entries or duct size changes

To determine the average velocity, it is necessary to take a number of velocity pressure readings across the cross-section of the duct. The velocity should be calculated for each velocity pressure reading, and the average of the velocities should be used. Do not average the velocity pressure; average the velocities. For round ducts over 6 inches diameter, the following locations will give areas of equal concentric area (see Figure 5.17).

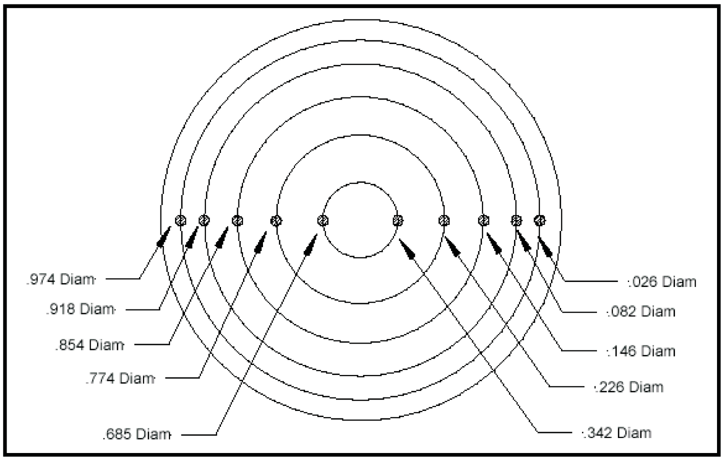

Figure 5.17 Traverse Points for Circular Duct

For best results, one set of readings should be taken in one direction and another set at a 90 ° angle to the first. For square ducts, the readings can be taken in 16 equally spaced areas. If it is impossible to traverse the duct, an approximate average velocity can be calculated by measuring the velocity pressure in the center of the duct and calculating the velocity. This value is reduced to an approximate average by multiplying by 0 .9.

Air density calculation

The first calculation is to determine the density of the air. To calculate the velocity and volume from the velocity pressure measurements it is necessary to know the density of the air. The density is dependent on altitude and temperature.

$$\text{Gas Density}(\gamma) = \frac{273 \quad x \ 1.293}{273 + t^{\circ}C}$$

t°C – temperature of gas/air at site condition

Velocity calculation

Once the air density and velocity pressure have been established, the velocity can be determined from the equation:

$$\text{Velocity v, m/s} = \frac{C_p \ x \ \sqrt{2 \times 9.81 \times \Delta p \times \gamma}}{\gamma}$$

C_p = Pitot tube constant, 0.85 (or) as given by the manufacturer

Δp = Average differential pressure measured by pitot tube by taking measurement at number of points over the entire cross section of the duct.

γ = Density of air or gas at test condition,

Volume calculation

The volume in a duct can be calculated for the velocity using the equation:

$$\text{Volumetric flow (Q), m}^3\text{/sec} = \text{Velocity,V(m / sec) x Area (m}^2\text{)}$$

Fan efficiency

Fan manufacturers generally use two ways to mention fan efficiency: mechanical efficiency (sometimes called the total efficiency) and static efficiency. Both measure how well the fan converts horsepower into flow and pressure.

The equation for determining mechanical efficiency is:

$$\text{Fan Mechanical Efficiency } \eta_{mechanical}\ \% = \frac{\text{Volume in } m^3 / \text{Sec} \times \Delta p \text{ (total pressure) in mmwc}}{102 \text{ x Power input to the fan shaft in (kW)}} \text{ x } 100$$

The static efficiency equation is the same except that the outlet velocity pressure is not added to the fan static pressure

$$\text{Fan Static Efficiency } \eta_{static}\ \% = \frac{\text{Volume in } m^3 / \text{Sec} \times \Delta p \text{ (static pressure) in mmwc}}{102 \text{ x Power input to the fan shaft in (kW)}} \text{ x } 100$$

Drive motor kW can be measured by a load analyzer. This kW multiplied by motor efficiency gives the shaft power to the fan.

5.7 Energy Saving Opportunities

Minimizing demand on the fan.

1. Minimising excess air level in combustion systems to reduce FD fan and ID fan load.
2. Minimising air in-leaks in hot flue gas path to reduce ID fan load, especially in case of kilns, boiler plants, furnaces, etc. Cold air in-leaks increase ID fan load tremendously, due to density increase of flue gases and in-fact choke up the capacity of fan, resulting as a bottleneck for boiler / furnace itself.
3. In-leaks / out-leaks in air conditioning systems also have a major impact on energy efficiency and fan power consumption and need to be minimized.

The findings of performance assessment trials will automatically indicate potential areas for improvement, which could be one or a more of the following:

1. Change of impeller by a high efficiency impeller along with cone.
2. Change of fan assembly as a whole, by a higher efficiency fan
3. Impeller de-rating (by a smaller dia impeller)
4. Change of metallic / Glass reinforced Plastic (GRP) impeller by the more energy efficient hollow FRP impeller with aerofoil design, in case of axial flow fans, where significant savings have been reported
5. Fan speed reduction by pulley dia modifications for derating
6. Option of two speed motors or variable speed drives for variable duty conditions
7. Option of energy efficient flat belts, or, cogged raw edged V belts, in place of conventional V belt systems, for reducing transmission losses.
8. Adopting inlet guide vanes in place of discharge damper control
9. Minimizing system resistance and pressure drops by improvements in duct system

Case Study – 1

VSD Applications

Cement plants use a large number of high capacity fans. By using liners on the impellers, which can be replaced when they are eroded by the abrasive particles in the dust-laden air, the plants have been able to switch from radial blades to forward-curved and backward-curved centrifugal fans. This has vastly improved system efficiency without requiring frequent impeller changes.

For example, a careful study of the clinker cooler fans at a cement plant showed that the flow was much higher than required and also the old straight blade impeller resulted in low system efficiency. It was decided to replace the impeller with a backward-curved blade and use liners to prevent erosion of the blade. This simple measure resulted in a 53 % reduction in power consumption, which amounted to annual savings of Rs. 2.1 million.

Another cement plant found that a large primary air fan which was belt driven through an arrangement of bearings was operating at system efficiency of 23 %. The fan was replaced with a direct coupled fan with a more efficient impeller. Power consumption reduced from 57 kW to 22 kW. Since cement plants use a large number of fans, it is generally possible to integrate the system such that air can be supplied from a common duct in many cases.

For example, a study indicated that one of the fans was operated with the damper open to only 5 %. By re-ducting to allow air to be supplied from another duct where flow was being throttled, it was possible to totally eliminate the use of a 55 kW fan.

The use of variable-speed drives for capacity control can result in significant power savings. A 25 ton-per-hour capacity boiler was equipped with both an induced-draft and forced-draft fan. Outlet dampers were used to control the airflow. After a study of the airflow pattern, it was decided to install a variable speed drive to control air flow. The average power consumption was reduced by nearly 41 kW resulting in annual savings of Rs. 0.33 million. The investment of Rs. 0.65 million for the variable-speed drive was paid back in under 2 years.

The type of variable-speed drive employed also significantly impacts power consumption. Thermal power stations install a hydraulic coupling to control the capacity of the induced-draft fan. It was decided to install a VFD on ID fans in a 200 MW thermal power plant. A comparison of the power consumption of the two fan systems indicated that for similar operating conditions of flow and plant power generation, the unit equipped with the VFD control unit consumed, on average, 4 million units / annum less than the unit equipped with the hydraulic coupling.

Case Study – 2

FRP Fans in Cooling Towers / Humidification Plants

The fans used for cooling tower applications are usually axial flow fans. Such fans are also commonly used in humidification plants. The conventional fans are made from aluminium / steel. These fans are being replaced in recent times by high efficiency FRP (fibre reinforced plastics) fans. The savings potential is shown below:

ILLUSTRATIVE DATA ON ENERGY SAVINGS WITH HIGH EFFICIENCY FRP BLADE AXIAL FLOW FANS					
(Source : PCRA Literature)					
Fan Data	**Type of Fan**	**Air flow cfm** (1 cfm = 1.7 m3/hr	**Static Pressure mmWC**	**Input Power kW**	**% of Power Saving**
Cooling Tower Fans					
24 ft.	Al	664480	3.65	68	Ref.
	FRP	740100	4.61	44	35.29
24 ft.	Al	817650	5.50	60	Ref.
	FRP	919400	6.00	46	23.33
16 ft.	Al	389200	--	45	Ref.
	FRP	391900	--	25	44.44
Humidification Fans					
5.25 ft.	Al	31382	--	7.9	
	FRP	31557	--	5.1	35.80
3.94 ft.	Al	38102	--	13.0	
	FRP	45935	--	13.0	17.05

QUESTIONS	
1.	Explain the difference between fans, blowers and compressors?
2.	Which fan you would chose for moving large flows against relatively low pressures
	a) Radial fan b) backward inclined fan c) forward curved fan d) axial fan
3.	If efficiency is the main consideration you would select
	a) Radial fan b) backward inclined fan c) forward curved fan d) axial fan
4.	For heavy dust conditions, which type of fan is ideally suited
	a) Radial fan b) backward inclined fan c) forward curved fan d) axial fan
5.	The system resistance refers to
	a) static pressure b) velocity pressure c) total pressure d) differential pressure
6.	System resistance varies as
	a) square of flow rate b) cube of flow rate c) directly proportional to square root of flow rate d) directly with flow rate
7.	The intersection of system curve with fan operating curve is called
	a) design point b) operating point c) selection point d) shut off point
8.	Varying the RPM of a fan by 10% varies the pressure by
	a) 19% b) 29% c) 10% d) does not vary
9.	Varying the RPM of a fan by 10% varies the flow by
	a) 10% b) 20% c) 30% d) does not vary
10.	Varying the RPM of a fan by 10% varies the power by
	a) 27% b) 37% c) 10% d) does not vary
11.	Explain the factors, which can change the system resistance?
12.	What are affinity laws as applicable to centrifugal fans?
13.	Explain the method of flow measurements using pitot tube?

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6. PUMPS AND PUMPING SYSTEM

Syllabus

Pumps and Pumping System: Types, Performance evaluation, Efficient system operation, Flow control strategies and energy conservation opportunities

6.1 Pump Types

Pumps come in a variety of sizes for a wide range of applications. They can be classified according to their basic operating principle as dynamic or displacement pumps. Dynamic pumps can be sub-classified as centrifugal and special effect pumps. Displacement pumps can be sub-classified as rotary or reciprocating pumps.

In principle, any liquid can be handled by any of the pump designs. Where different pump designs could be used, the centrifugal pump is generally the most economical followed by rotary and reciprocating pumps. Although, positive displacement pumps are generally more efficient than centrifugal pumps, the benefit of higher efficiency tends to be offset by increased maintenance costs.

Since, worldwide, centrifugal pumps account for the majority of electricity used by pumps, the focus of this chapter is on centrifugal pump.

Centrifugal Pumps

A centrifugal pump is of a very simple design. The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving part, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, polycarbonate, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and captures and directs the water off the impeller.

Water enters the center (eye) of the impeller and exits the impeller with the help of centrifugal force. As water leaves the eye of the impeller a low-pressure area is created, causing more water to flow into the eye. Atmospheric pressure and centrifugal force cause this to happen. Velocity is developed as the water flows through the impeller spinning at high speed. The water velocity is collected by the diffuser and converted to pressure by specially designed passageways that direct the flow to the discharge of the pump, or to the next impeller should the pump have a multi-stage configuration.

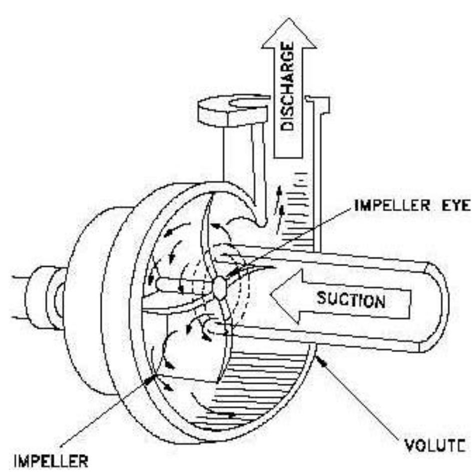
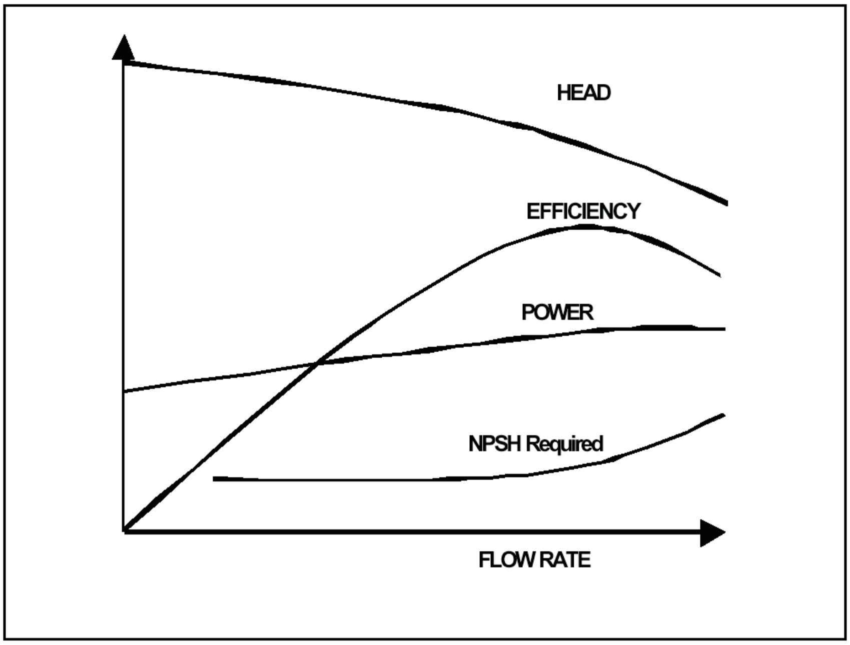


Figure 6.1 Centrifugal pump

The pressure (head) that a pump will develop is in direct relationship to the impeller diameter, the number of impellers, the size of impeller eye, and shaft speed. Capacity is determined by the exit width of the impeller. The head and capacity are the main factors, which affect the horsepower size of the motor to be used. The more the quantity of water to be pumped, the more energy is required.

A centrifugal pump is not positive acting; it will not pump the same volume always. The greater the depth of the water, the lesser is the flow from the pump. Also, when it pumps against increasing pressure, the less it will pump. For these reasons it is important to select a centrifugal pump that is designed to do a particular job.

Since the pump is a dynamic device, it is convenient to consider the pressure in terms of head i.e. meters of liquid column. The pump generates the same head of liquid whatever the density of the liquid being pumped. The actual contours of the hydraulic passages of the impeller and the casing are extremely important, in order to attain the highest efficiency possible. The standard convention for centrifugal pump is to draw the pump performance curves showing F[illegible]e horizontal axis and Head generated on the vertical axis. Efficiency, Power [illegible]PSH [illegible] (described later), are conventionally shown on the vertical axis, plotted [illegible]st Fl[illegible]strated in Figure 6.2.



System

Figure 6.2 Pump Performance Curve

Given the significant amount of electricity attributed to pumping systems, even small improvements in pumping efficiency could yield very significant savings of electricity. The pump is among the most inefficient of the components that comprise a pumping system, including the motor, transmission drive, piping and valves.

Hydraulic power, pump shaft power and electrical input power

Hydraulic power P_h = Q (m^3/s) x Total head, h_d - h_s (m) x ρ (kg/m^3) x g (m/s^2) / 1000

Where h_d – discharge head, h_s – suction head, ρ – density of the fluid, g – acceleration due to gravity

Pump shaft power P_s = Hydraulic power, P_h / pump efficiency, η_{Pump}

$$\textbf{Electrical input power} = \frac{\text{Pump shaft power } P_s}{\eta_{Motor}}$$

6.2 System Characteristics

In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination, e.g. filling a high level reservoir, or to circulate liquid around a system, e.g. as a means of heat transfer in heat exchanger.

A pressure is needed to make the liquid flow at the required rate and this must overcome head 'losses' in the system. Losses are of two types: static and friction head.

Static head is simply the difference in height of the supply and destination reservoirs, as in Figure 6.3. In this illustration, flow velocity in the pipe is assumed to be very small. Another example of a system with only static head is pumping into a pressurised vessel with short pipe runs. Static head is independent of flow and graphically would be shown as in Figure 6.4.

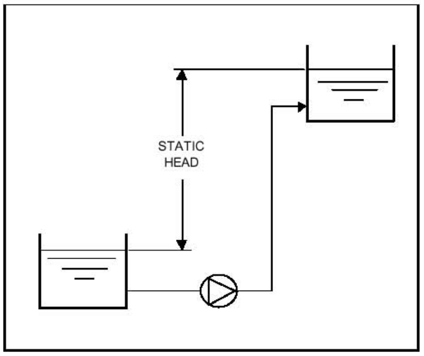


Figure 6.3 Static Head

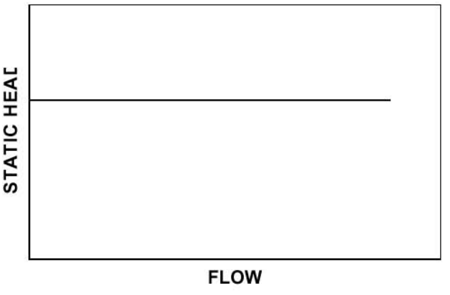


Figure 6.4 Static Head vs. Flow

Friction head (sometimes called dynamic head loss) is the friction loss, on the liquid being moved, in pipes, valves and equipment in the system. Friction tables are universally available for various pipe fittings and valves. These tables show friction loss per 100 feet (or metres) of a specific pipe size at various flow rates. In case of fittings, friction is stated as an equivalent length of pipe of the same size. The friction losses are proportional to the square of the flow rate. A closed loop circulating system without a surface open to atmospheric pressure, would exhibit only friction losses and would have a system friction head loss vs. flow curve as Figure 6.5.

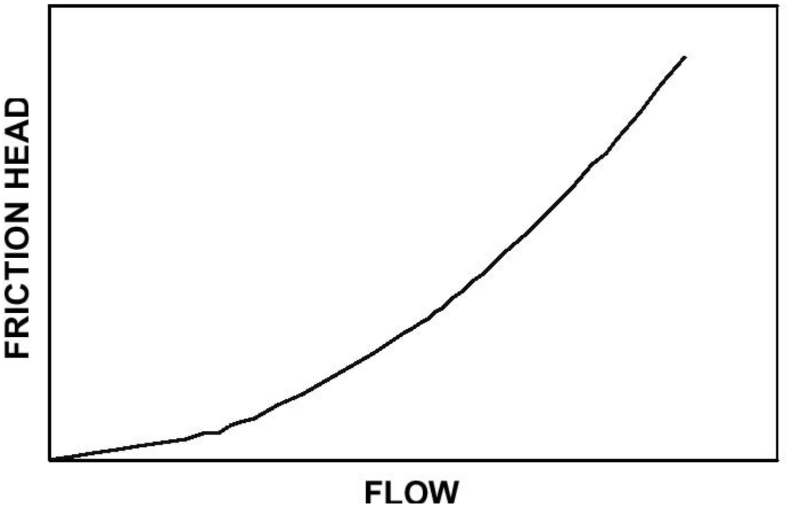


Figure 6.5 Friction Head vs. Flow

Most systems have a combination of static and friction head and the system curves for two cases are shown in Figures 6.6 and 6.7. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives which shall be discussed later.

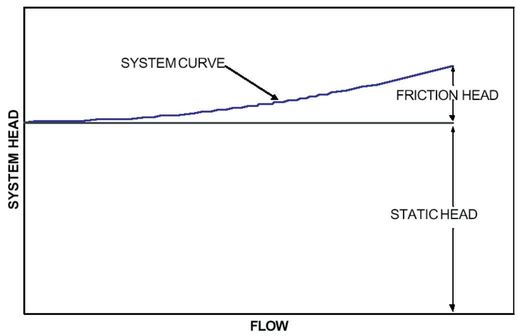


Figure 6.6 System with High Static Head

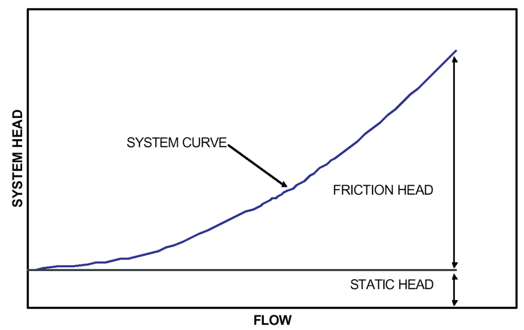


Figure 6.7 System with Low Static Head

Static head is a characteristic of the specific installation and reducing this head where this is possible, generally helps both the cost of the installation and the cost of pumping the liquid. Friction head losses must be minimised to reduce pumping cost, but after eliminating unnecessary pipe fittings and length, further reduction in friction head will require larger diameter pipe, which adds to installation cost.

6.3 Pump Curves

The performance of a pump can be expressed graphically as head against flow rate. The centrifugal pump has a curve where the head falls gradually with increasing flow. This is called the pump characteristic curve (Head - Flow curve) -see Figure 6.8.

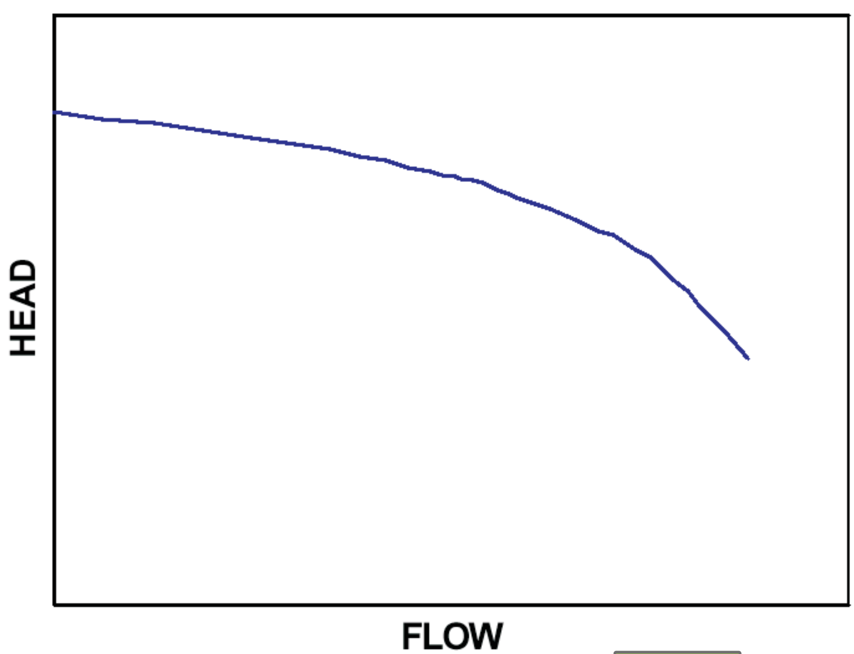


Figure 6.8 Head- Flow Curve

Pump operating point

When a pump is installed in a system the effect can be illustrated graphically by superimposing pump and system curves. The operating point will always be where the two curves intersect. Figure 6.9.

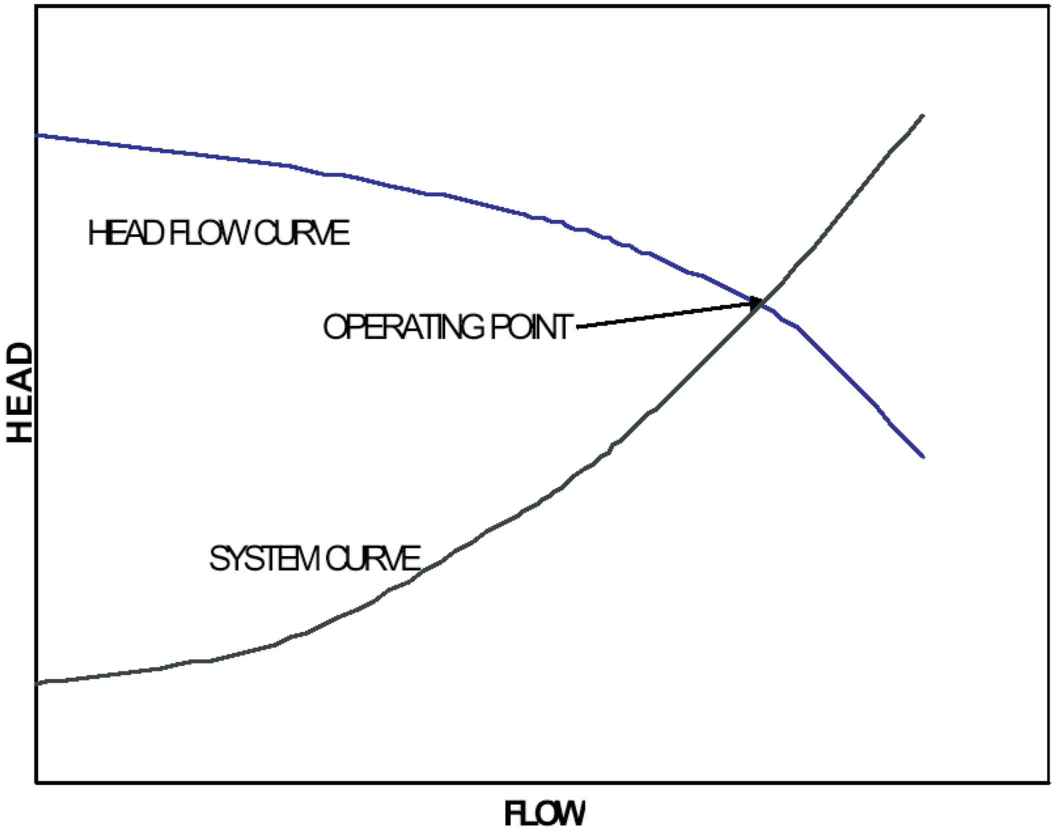


Figure 6.9 Pump Operating Point

If the actual system curve is different in reality to that calculated, the pump will operate at a flow and head different to that expected.

For a centrifugal pump, an increasing system resistance will reduce the flow, eventually to zero, but the maximum head is limited as shown. Even so, this condition is only acceptable for a short period without causing problems. An error in the system curve calculation is also likely to lead to a centrifugal pump selection, which is less than optimal for the actual system head losses. Adding safety margins to the calculated system curve to ensure that a sufficiently large pump is selected will generally result in installing an oversized pump, which will operate at an excessive flow rate or in a throttled condition, which increases energy usage and reduces pump life.

6.4 Factors Affecting Pump Performance

Matching Pump and System Head-flow Characteristics

Centrifugal pumps are characterized by the relationship between the flow rate (Q) they produce and the pressure (H) at which the flow is delivered. Pump efficiency varies with flow and pressure, and it is highest at one particular flow rate.

The Figure 6.10 below shows a typical vendor-supplied head-flow curve for a centrifugal pump. Pump head-flow curves are typically given for clear water. The choice of pump for a given application depends largely on how the pump head-flow characteristics match the requirement of the system downstream of the pump.

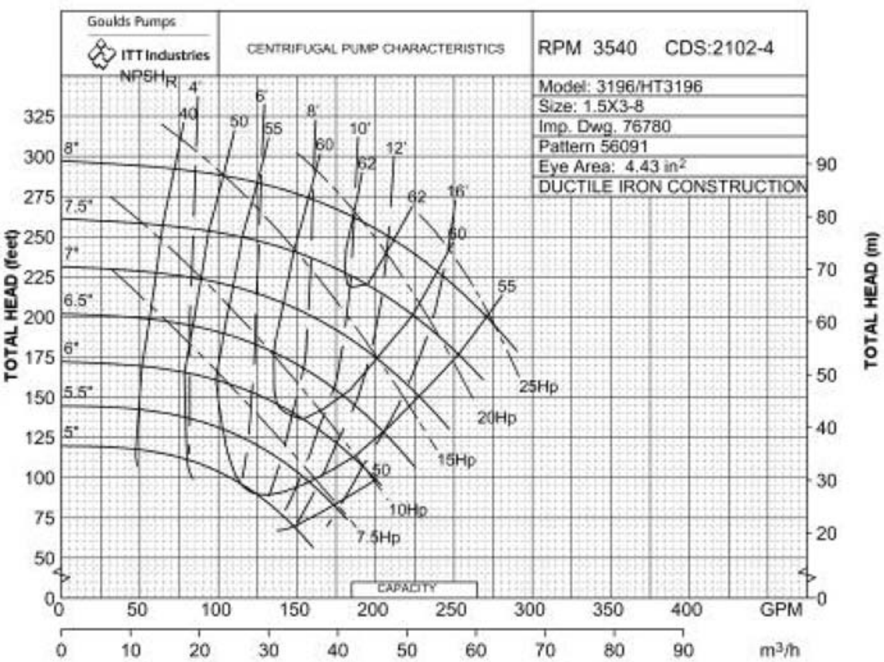


Figure 6.10 Typical Centrifugal Pump Performance Curve

Effect of over sizing the pump

As mentioned earlier, pressure losses to be overcome by the pumps are function of flow – the system characteristics – are also quantified in the form of *head-flow curves*. The system curve is basically a plot of system resistance i.e. head to be overcome by the pump versus various flow rates. The system curves change with the physical configuration of the system; for example, the system curves depends upon height or elevation, diameter and length of piping, number and type of fittings and pressure drops across various equipment - say a heat exchanger.

A pump is selected based on how well the pump curve and system head-flow curves match. The pump operating point is identified as the point, where the system curve crosses the pump curve when they are superimposed on each other.

The Figure 6.11 shows the effect on system curve with throttling.

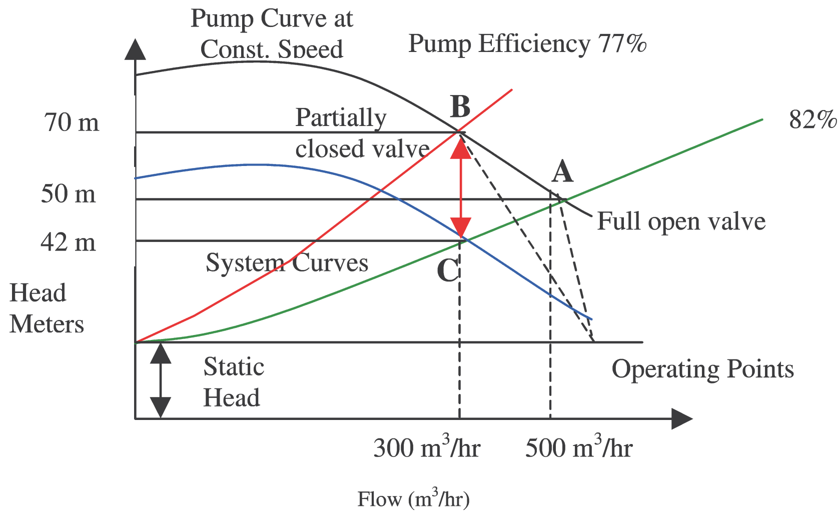


Figure 6.11 Effect on System Curve with Throttling

In the system under consideration, water has to be first lifted to a height – this represents the static head.

Then, we make a system curve, considering the friction and pressure drops in the system-this is shown as the green curve.

Suppose, we have estimated our operating conditions as 500 m^3/hr flow and 50 m head, we will chose a pump curve which intersects the system curve (Point A) at the pump's *best efficiency point* (BEP).

But, in actual operation, we find that 300 m^3/hr is sufficient. The reduction in flow rate has to be effected by a throttle valve. In other words, we are introducing an artificial resistance in the system.

Due to this additional resistance, the frictional part of the system curve increases and thus the new system curve will shift to the left -this is shown as the red curve.

So the pump has to overcome additional pressure in order to deliver the reduced flow. Now, the new system curve will intersect the pump curve at point B. The revised parameters are 300 m^3/hr at 70 m head. The red double arrow line shows the additional pressure drop due to throttling.

You may note that the best efficiency point has shifted from 82% to 77% efficiency.

So what we want is to actually operate at point C which is 300 m^3/hr on the original system curve. The head required at this point is only 42 meters.

What we now need is a new pump which will operate with its best efficiency point at C. But there are other simpler options rather than replacing the pump. The speed of the pump can be reduced or the existing impeller can be trimmed (or new lower size impeller). The blue pump curve represents either of these options.

Energy loss in throttling

Consider a case (see Figure [illegible] here we need to pump 68 m^3/hr of water at 47 m head. The pump characteristic curves [illegible] or a range of pumps are given in the Figure 6.12.

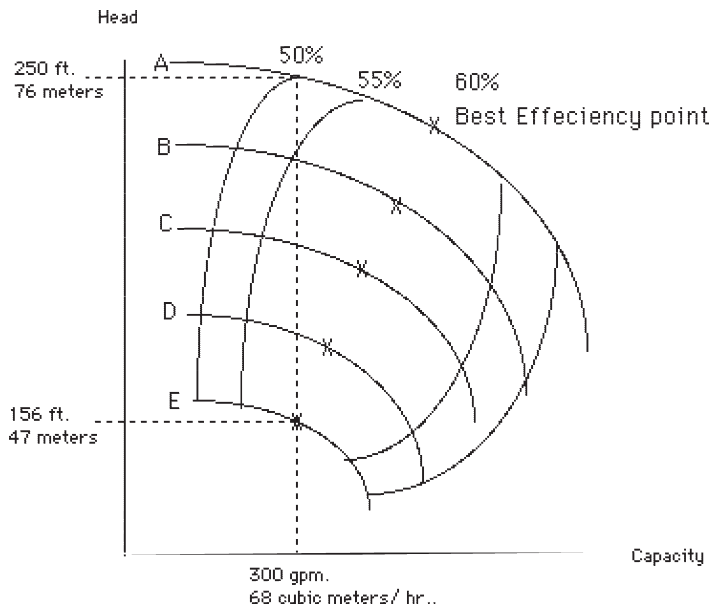


Figure 6.12 Pump Characteristic Curves

If we select E, then the pump efficiency is 60%

Hydraulic Power = Q (m^3/s) x Total head, h_d - h_s (m) x ρ (kg/m^3) x g (m/s^2) / 1000

$$= \frac{(68/3600) \times 47 \times 1000 \times 9.81}{1000}$$

= 8.7 kW

Shaft Power - 8.7 / 0.60 = 14.5 Kw
Motor Power - 14.5 / 0.9 = 16.1Kw (considering a motor efficiency of 90%)

If we select A, then the pump efficiency is 50% (drop from earlier 60%)

Obviously, this is an oversize pump. Hence, the pump has to be throttled to achieve the desired flow. Throttling increases the head to be overcome by the pump. In this case, head is 76 metres.

Hydraulic Power = Q (m^3/s) x Total head, h_d - h_s (m) x ρ (kg/m^3) x g (m /s^2) / 1000

$$= \frac{(68/3600) \times 76 \times 1000 \times 9.81}{1000}$$

= 14 kW

Shaft Power - 14 / 0.50 = 28 Kw
Motor Power - 28 / 0.9 = 31 Kw (considering a motor efficiency of 90%)

Hence, additional power drawn by A over E is 31 –16.1 = 14.9 kW.

Extra energy used = 8760 hrs/yr x 14.9 = 1,30,524 kwh/annum
= Rs. 5,22,096/annum

In this example, the extra cost of the electricity is more than the cost of purchasing a new pump.

6.5 Efficient Pumping System Operation

To understand a pumping system, one must realize that all of its components are interdependent. When examining or designing a pump system, the process demands must first be established and most energy efficiency solution introduced. For example, does the flow rate have to be regulated continuously or in steps? Can on-off batch pumping be used? What are the flow rates needed and how are they distributed in time?

The first step to achieve energy efficiency in pumping system is to target the end-use. A plant water balance would establish usage pattern and highlight areas where water consumption can be reduced or optimized. Good water conservation measures, alone, may eliminate the need for some pumps.

Once flow requirements are optimized, then the pumping system can be analysed for energy conservation opportunities. Basically this means matching the pump to requirements by adopting proper flow control strategies. Common symptoms that indicate opportunities for energy efficiency in pumps are given in the Table 6.1.

TABLE 6.1 SYMPTOMS THAT INDICATE POTENTIAL OPPORTUNITY FOR ENERGY SAVINGS

Symptom	Likely Reason	Best Solutions
Throttle valve-controlled systems	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower rpm
Bypass line (partially or completely) open	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower rpm
Multiple parallel pump system with the same number of pumps always operating	Pump use not monitored or controlled	Install controls
Constant pump operation in a batch environment	Wrong system design	On-off controls
High maintenance cost (seals, bearings)	Pump operated far away from BEP	Match pump capacity with system requirement

Effect of speed variation

As stated above, a centrifugal pump is a dynamic device with the head generated from a rotating impeller. There is therefore a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter and so varying the rotational speed has a direct effect on the performance of the pump. All the parameters shown in fig 6.2 will change if the speed is varied and it is important to have an appreciation of how these parameters vary in order to safely control a pump at different speeds. The equations relating rotodynamic pump performance parameters of flow, head and power absorbed, to speed are known as the *Affinity Laws*:

$$Q \propto N$$
$$H \propto N^2$$
$$P \propto N^3$$

Where:
Q = Flow rate
H = Head
P = Power absorbed
N = Rotating speed
Efficiency is essentially independent of speed

Flow: Flow is proportional to the speed

$\mathbf{Q_1 / Q_2 = N_1 / N_2}$
Example: $\mathbf{100 / Q_2 = 1750/3500}$
$\mathbf{Q_2 = 200\ m^3/hr}$

Head: Head is proportional to the square of speed

$H_1/H_2 = (N_1^2) / (N_2^2)$
Example: $100 / H_2 = 1750^2 / 3500^2$
$H_2 = 400$ m

Power(kW): Power is proportional to the cube of speed

$\mathbf{kW_1 / kW_2 = (N_1^3) / (N_2^3)}$
Example: $\mathbf{5/kW_2 = 1750^3 / 3500^3}$
$\mathbf{kW_2 = 40}$

As can be seen from the above laws, doubling the speed of the centrifugal pump will increase the power consumption by 8 times. Conversely a small reduction in speed will result in drastic reduction in power consumption. This forms the basis for energy conservation in centrifugal pumps with varying flow requirements. The implication of this can be better understood as shown in an example of a centrifugal pump in Figure 6.13 below.

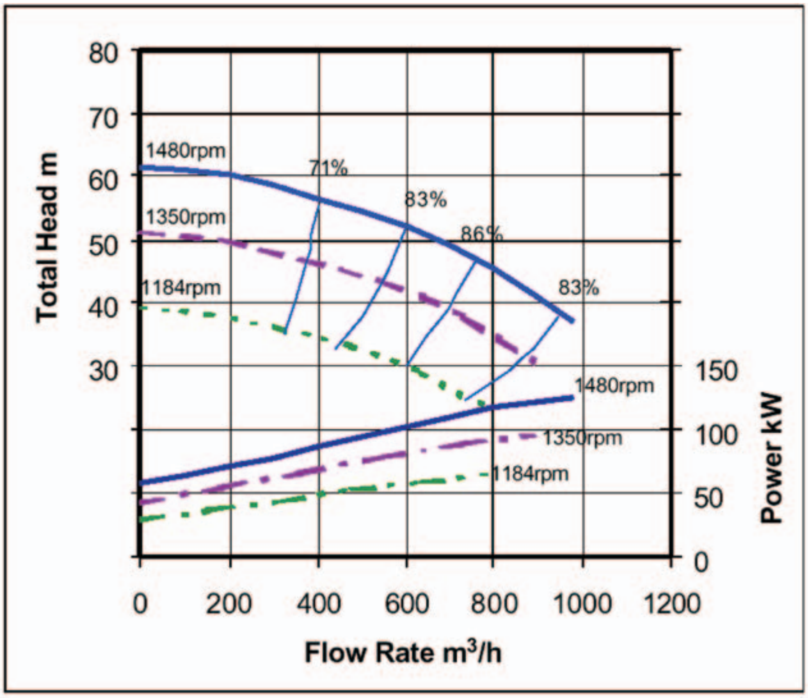


Figure 6.13 Example of Speed Variation Effecting Centrifugal Pump Performance

Points of equal efficiency on the curves for the 3 different speeds are joined to make the iso-efficiency lines, showing that efficiency remains constant over small changes of speed providing the pump continues to operate at the same position related to its best efficiency point (BEP).

The affinity laws give a good approximation of how pump performance curves change with speed but in order to obtain the actual performance of the pump in a system, the system curve also has to be taken into account.

Effects of impeller diameter change

Changing the impeller diameter gives a proportional change in peripheral velocity, so it follows that there are equations, similar to the affinity laws, for the variation of performance with impeller diameter D:

$$Q \propto D$$

$$H \propto D^2$$

$$P \propto D^3$$

Efficiency varies when the diameter is changed within a particular casing. Note the difference in iso-efficiency lines in Figure 6.14 compared with Figure 6.13. The relationships shown here apply to the case for changing only the diameter of an impeller within a fixed casing geometry, which is a common practice for making small permanent adjustments to the performance of a centrifugal pump. Diameter changes are generally limited to reducing the diameter to about 75% of the maximum, i.e. a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. However speed change can be used over a wider range without seriously reducing efficiency. For example reducing the speed by 50% typically results in a reduction of efficiency by 1 or 2 percentage points. The reason for the small loss of efficiency with the lower speed is that mechanical losses in seals and bearings, which generally represent <5% of total power, are proportional to speed, rather than speed cubed. It should be noted that if the change in diameter is more than about 5%, the accuracy of the squared and cubic relationships can fall off and for precise calculations, the pump manufacturer's performance curves should be referred to.

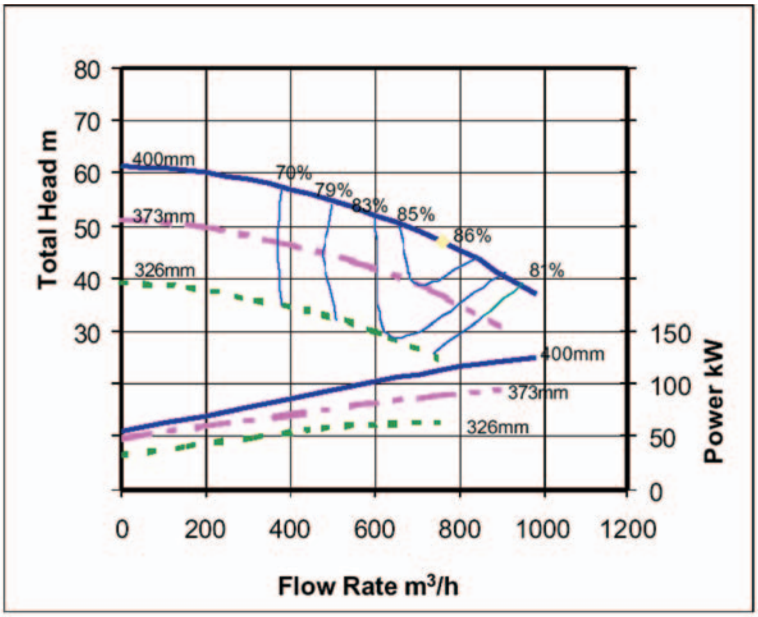


Figure 6.14 Example: Impeller Diameter Reduction on Centrifugal Pump Performance

The illustrated curves are typical of most centrifugal pump types. Certain high flow, low head pumps have performance curve shapes somewhat different and have a reduced operating region of flows. This requires additional care in matching the pump to the system, when changing speed and diameter.

Pump suction performance (NPSH)

Liquid entering the impeller eye turns and is split into separate streams by the leading edges of the impeller vanes, an action which locally drops the pressure below that in the inlet pipe to the pump.

If the incoming liquid is at a pressure with insufficient margin above its vapour pressure, then vapour cavities or bubbles appear along the impeller vanes just behind the inlet edges. This phenomenon is known as cavitation and has three undesirable effects:

1) The collapsing cavitation bubbles can erode the vane surface, especially when pumping water-based liquids.
2) Noise and vibration are increased, with possible shortened seal and bearing life.
3) The cavity areas will initially partially choke the impeller passages and reduce the pump performance. In extreme cases, total loss of pump developed head occurs.

The value, by which the pressure in the pump suction exceeds the liquid vapour pressure, is expressed as a head of liquid and referred to as Net Positive Suction Head Available – (NPSHA). This is a characteristic of the system design. The value of NPSH needed at the pump suction to prevent the pump from cavitating is known as NPSH Required – (NPSHR). This is a characteristic of the pump design.

The three undesirable effects of cavitation described above begin at different values of NPSHA and generally there will be cavitation erosion before there is a noticeable loss of pump head. However for a consistent approach, manufacturers and industry standards, usually define the onset of cavitation as the value of NPSHR when there is a head drop of 3% compared with the head with cavitation free performance. At this point cavitation is present and prolonged operation at this point will usually lead to damage. It is usual therefore to apply a margin bywhich NPSHA should exceed NPSHR.

As would be expected, the NPSHR increases as the flow through the pump increases, see fig 6.2. In addition, as flow increases in the suction pipework, friction losses also increase, giving a lower NPSHA at the pump suction, both of which give a greater chance that cavitation will occur. NPSHR also varies approximately with the square of speed in the same way as pump head and conversion of NPSHR from one speed to another can be made using the following equations.

$$Q \propto N$$

$$NPSHR \propto N^2$$

It should be noted however that at very low speeds there is a minimum NPSHR plateau, NPSHR does not tend to zero at zero speed It is therefore essential to carefully consider NPSH in variable speed pumping.

6.6 Flow Control Strategies

Pump control by varying speed

To understand how speed variation changes the duty point, the pump and system curves are over-laid. Two systems are considered, one with only friction loss and another where static head is high in relation to friction head. It will be seen that the benefits are different. In Figure 6.15,

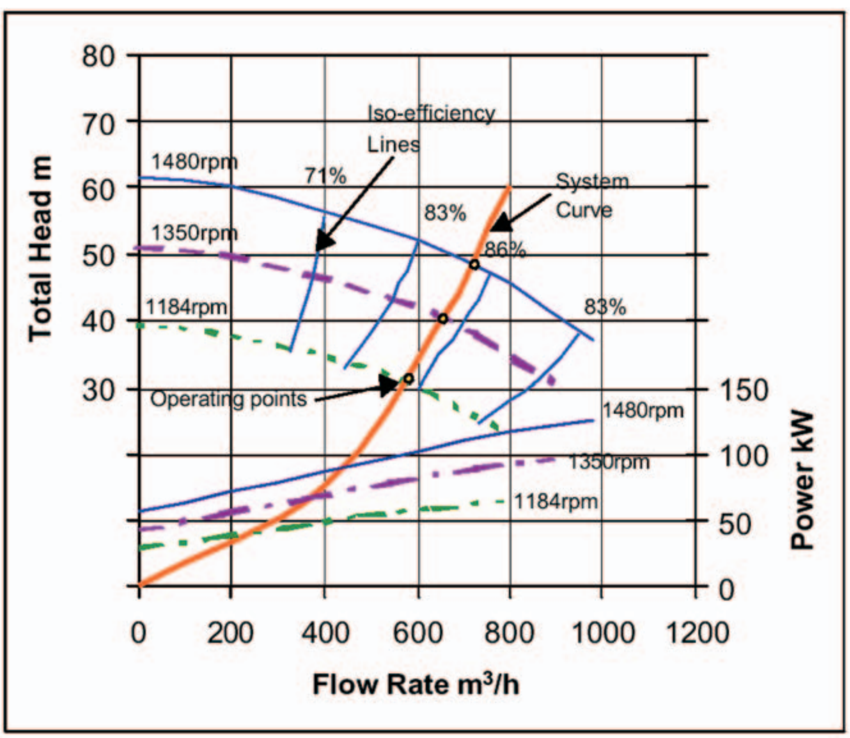


Figure 6.15 Example of the Effect of Pump Speed Change in a System With Only Friction Loss

reducing speed in the friction loss system moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump, relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region. The affinity laws are obeyed which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method for systems with friction loss.

In a system where static head is high, as illustrated in Figure 6.16, the operating point for the pump moves relative to the lines of constant pump efficiency when the speed is changed. The reduction in flow is no longer proportional to speed. A small turn down in speed could give a big reduction in flow rate and pump efficiency, which could result in the pump operating in a region where it could be damaged if it ran for an extended period of time even at the lower speed. At the lowest speed illustrated, (1184 rpm), the pump does not generate sufficient head to pump any liquid into the system, i.e. pump efficiency and flow rate are zero and with energy still being input to the liquid, the pump becomes a water heater and damaging temperatures can quickly be reached.

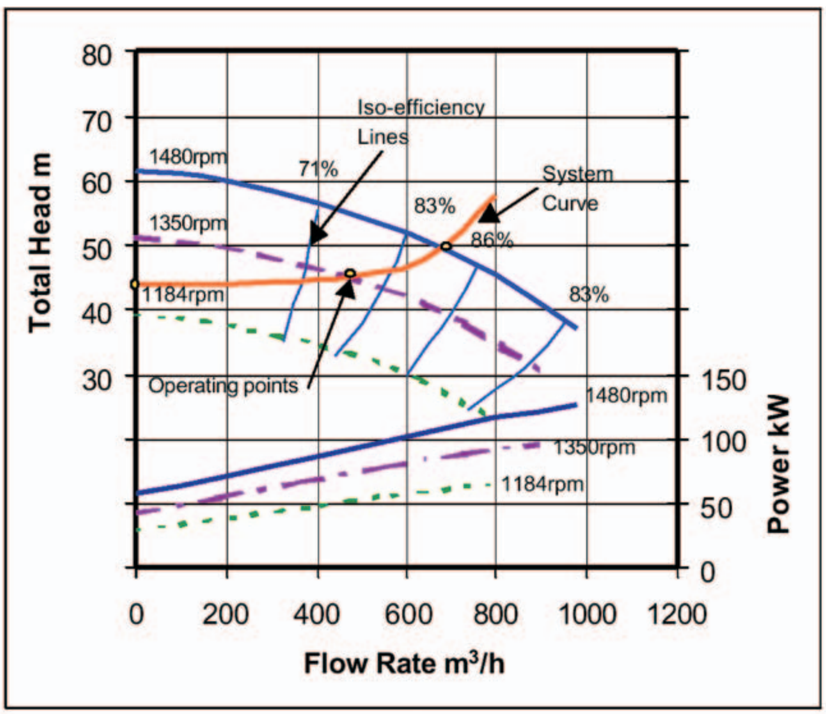


Figure 6.16 Example for the Effect of Pump Speed Change with a System with High Static Head.

The drop in pump efficiency during speed reduction in a system with static head, reduces the economic benefits of variable speed control. There may still be overall benefits but economics should be examined on a case-by-case basis. Usually it is advantageous to select the pump such that the system curve intersects the full speed pump curve to the right of best efficiency, in order that the efficiency will first increase as the speed is reduced and then decrease. This can extend the useful range of variable speed operation in a system with static head. The pump manufacturer should be consulted on the safe operating range of the pump.

It is relevant to note that flow control by speed regulation is always more efficient than by control valve. In addition to energy savings there could be other benefits of lower speed. The hydraulic forces on the impeller, created by the pressure profile inside the pump casing, reduce approximately with the square of speed. These forces are carried by the pump bearings and so reducing speed increases bearing life. It can be shown that for a centrifugal pump, bearing life is inversely proportional to the 7^{th} power of speed. In addition, vibration and noise are reduced and seal life is increased providing the duty point remains within the allowable operating range.

The corollary to this is that small increases in the speed of a pump significantly increase power absorbed, shaft stress and bearing loads. It should be remembered that the pump and motor must be sized for the maximum speed at which the pump set will operate. At higher speed the noise and vibration from both pump and motor will increase, although for small increases the change will be small. If the liquid contains abrasive particles, increasing speed will give a corresponding increase in surface wear in the pump and pipework.

The effect on the mechanical seal of the change in seal chamber pressure, should be reviewed with the pump or seal manufacturer, if the speed increase is large. Conventional mechanical seals operate satisfactorily at very low speeds and generally there is no requirement for a minimum speed to be specified, however due to their method of operation, gas seals require a minimum peripheral speed of 5 m/s.

Pumps in parallel switched to meet demand

Another energy efficient method of flow control, particularly for systems where static head is a high proportion of the total, is to install two or more pumps to operate in parallel. Variation of flow rate is achieved by switching on and off additional pumps to meet demand. The combined pump curve is obtained by adding the flow rates at a specific head. The head/flow rate curves for two and three pumps are shown in Figure 6.17.

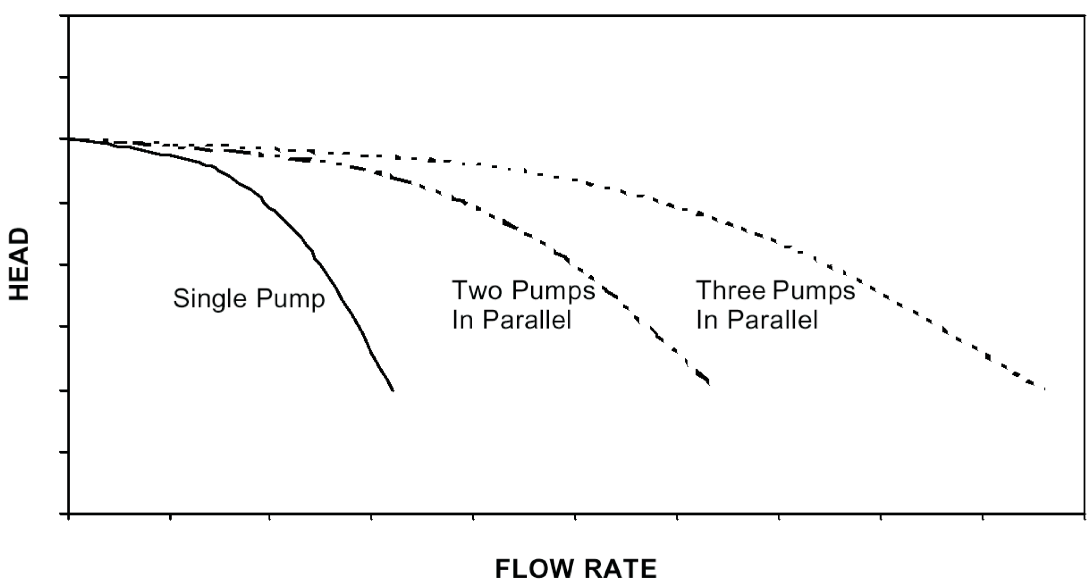


Figure 6.17 Typical Head-Flow Curves for Pumps in Parallel

The system curve is usually not affected by the number of pumps that are running. For a system with a combination of static and friction head loss, it can be seen, in Figure 6.18, that the operating point of the pumps on their performance curves moves to a higher head and hence lower flow rate per pump, as more pumps are started. It is also apparent that the flow rate with two pumps running is not double that of a single pump. If the system head were only static, then flow rate would be proportional to the number of pumps operating.

It is possible to run pumps of different sizes in parallel provided their closed valve heads are similar. By arranging different combinations of pumps running together, a larger number of different flow rates can be provided into the system.

Care must be taken when running pumps in parallel to ensure that the operating point of the pump is controlled within the region deemed as acceptable by the manufacturer. It can be seen from Figure 6.18 that if 1 or 2 pumps were stopped then the remaining pump(s) would operate well out along the curve where NPSH is higher and vibration level increased, giving an increased risk of operating problems.

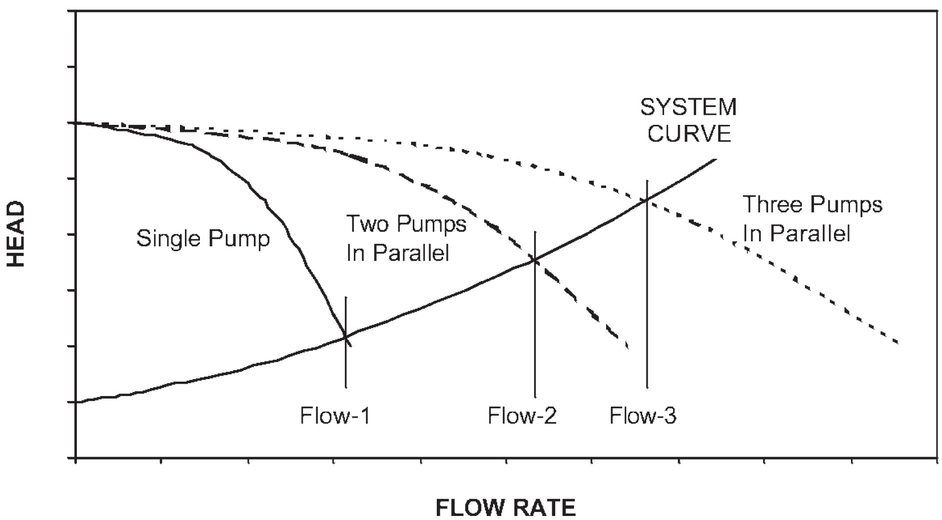


Figure 6.18 Typical Head-Flow Curves for Pumps in Parallel, With System Curve Illustrated.

Stop/start control

In this control method, the flow is controlled by switching pumps on or off. It is necessary to have a storage capacity in the system e.g. a wet well, an elevated tank or an accumulator type pressure vessel. The storage can provide a steady flow to the system with an intermittent operating pump. When the pump runs, it does so at the chosen (presumably optimum) duty point and when it is off, there is no energy consumption. If intermittent flow, stop/start operation and the storage facility are acceptable, this is an effective approach to minimise energy consumption.

The stop/start operation causes additional loads on the power transmission components and increased heating in the motor. The frequency of the stop/start cycle should be within the motor design criteria and checked with the pump manufacturer.

It may also be used to benefit from "off peak" energy tariffs by arranging the run times during the low tariff periods.

To minimise energy consumption with stop start control it is better to pump at as low flow rate as the process permits. This minimises friction losses in the pipe and an appropriately small pump can be installed. For example, pumping at half the flow rate for twice as long can reduce energy consumption to a quarter.

Flow control valve

With this control method, the pump runs continuously and a valve in the pump discharge line is opened or closed to adjust the flow to the required value.

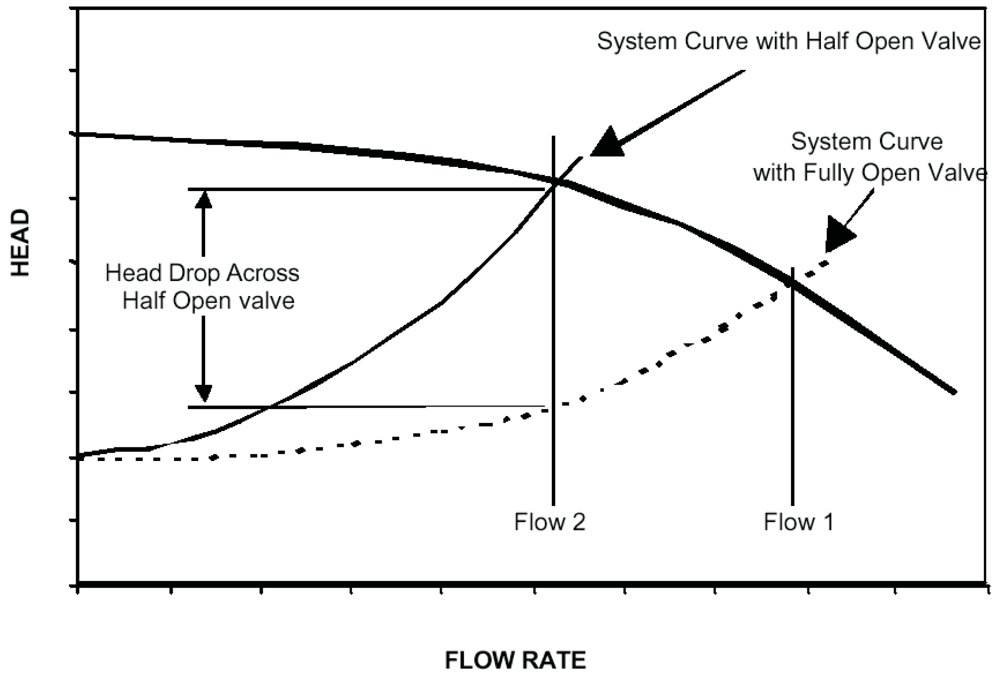


Figure 6.19 Control of Pump Flow by Changing System Resistance Using a Valve.

To understand how the flow rate is controlled, see Figure 6.19. With the valve fully open, the pump operates at "Flow 1". When the valve is partially closed it introduces an additional friction loss in the system, which is proportional to flow squared. The new system curve cuts the pump curve at "Flow 2", which is the new operating point. The head difference between the two curves is the pressure drop across the valve.

It is usual practice with valve control to have the valve 10% shut even at maximum flow. Energy is therefore wasted overcoming the resistance through the valve at all flow conditions. There is some reduction in pump power absorbed at the lower flow rate (see Figure 6.19), but the flow multiplied by the head drop across the valve, is wasted energy. It should also be noted that, while the pump will accommodate changes in its operating point as far as it is able within its performance range, it can be forced to operate high on the curve, where its efficiency is low, and its reliability is affected.

Maintenance cost of control valves can be high, particularly on corrosive and solids-containing liquids. Therefore, the lifetime cost could be unnecessarily high.

By-pass control

With this control approach, the pump runs continuously at the maximum process demand duty, with a permanent by-pass line attached to the outlet. When a lower flow is required the surplus liquid is bypassed and returned to the supply source.

An alternative configuration may have a tank supplying a varying process demand, which is kept full by a fixed duty pump running at the peak flow rate. Most of the time the tank over-flows and recycles back to the pump suction. This is even less energy efficient than a control valve because there is no reduction in power consumption with reduced process demand.

The small by-pass line sometimes installed to prevent a pump running at zero flow is not a means of flow control, but required for the safe operation of the pump.

Fixed Flow reduction

Impeller trimming

Impeller trimming refers to the process of machining the diameter of an impeller to reduce the energy added to the system fluid.

Impeller trimming offers a useful correction to pumps that, through overly conservative design practices or changes in system loads are oversized for their application.

Trimming an impeller provides a level of correction below buying a smaller impeller from the pump manufacturer. But in many cases, the next smaller size impeller is too small for the pump load. Also, smaller impellers may not be available for the pump size in question and impeller trimming is the only practical alternative short of replacing the entire pump/motor assembly. (see Figures 6.20 & 6.21 for before and after impeller trimming).

Impeller trimming reduces tip speed, which in turn directly lowers the amount of energy imparted to the system fluid and lowers both the flow and pressure generated by the pump.

The Affinity Laws, which describe centrifugal pump performance, provide a theoretical relationship between impeller size and pump output (assuming constant pump speed):

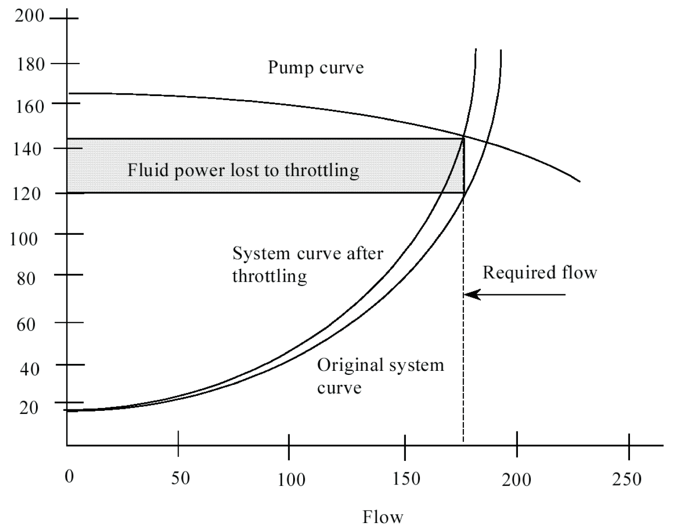


Figure 6.20 Before Impeller trimming

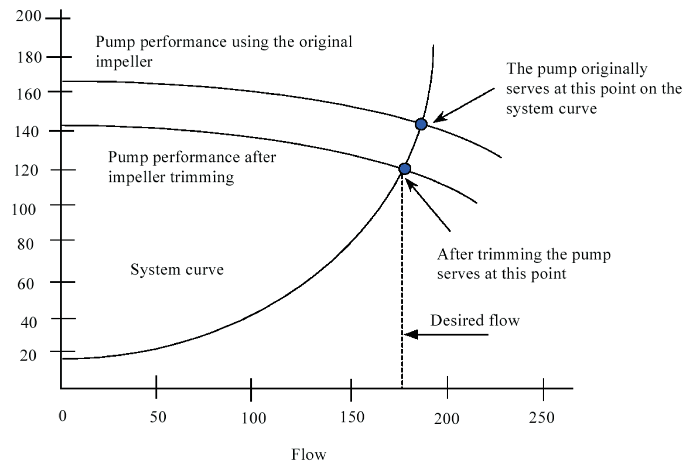


Figure 6.21 After Impeller Trimming

Where:

Q = flow

H = head

BHP = brake horsepower of the pump motor

Subscript 1 = original pump,

Subscript 2 = pump after impeller trimming

D = Diameter

$$Q_2 = \frac{D_2}{D_1} Q_1$$

$$H_2 = \left[\frac{D_2}{D_1}\right]^2 H_1$$

$$BHP_2 = \left[\frac{D_2}{D_1}\right]^3 BHP_1$$

Trimming an impeller changes its operating efficiency, and the non-linearities of the Affinity Laws with respect to impeller machining complicate the prediction of pump performance. Consequently, impeller diameters are rarely reduced below 70 percent of their original size.

Meeting variable flow reduction

Variable Speed Drives (VSDs)

In contrast, pump speed adjustments provide the most efficient means of controlling pump flow. By reducing pump speed, less energy is imparted to the fluid and less energy needs to be throttled or bypassed. There are two primary methods of reducing pump speed: multiple-speed pump motors and variable speed drives (VSDs).

Although both directly control pump output, multiple-speed motors and VSDs serve entirely separate applications. Multiple-speed motors contain a different set of windings for each motor speed; consequently, they are more expensive and less efficient than single speed motors. Multiple speed motors also lack subtle speed changing capabilities within discrete speeds.

VSDs allow pump speed adjustments over a continuous range, avoiding the need to jump from speed to speed as with multiple-speed pumps. VSDs control pump speeds using several different types of mechanical and electrical systems. Mechanical VSDs include hydraulic clutches, fluid couplings, and adjustable belts and pulleys. Electrical VSDs include eddy current clutches, wound-rotor motor controllers, and variable frequency drives (VFDs). VFDs adjust the electrical frequency of the power supplied to a motor to change the motor's rotational speed. VFDs are by far the most popular type of VSD.

However, pump speed adjustment is not appropriate for all systems. In applications with high static head, slowing a pump risks inducing vibrations and creating performance problems that are similar to those found when a pump operates against its shutoff head. For systems in which the static head repre-

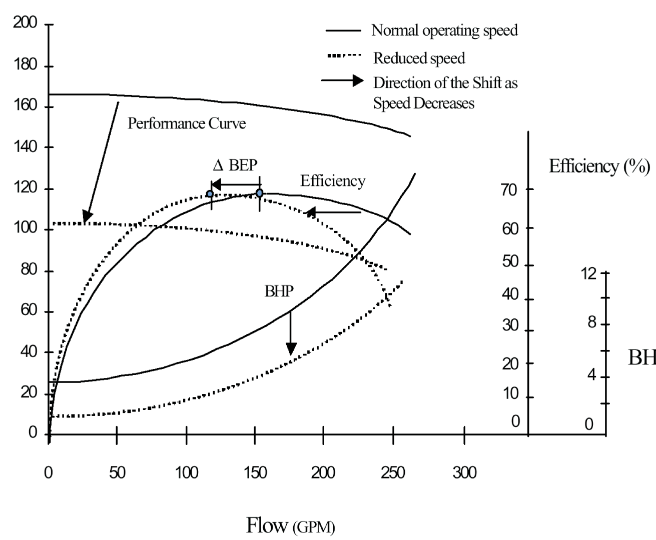


Figure 6.22 Effect of VFD

sents a large portion of the total head, caution should be used in deciding whether to use VFDs. Operators should review the performance of VFDs in similar applications and consult VFD manufacturers to avoid the damage that can result when a pump operates too slowly against high static head.

For many systems, VFDs offer a means to improve pump operating efficiency despite changes in operating conditions. The effect of slowing pump speed on pump operation is illustrated by the three curves in Figure 6.22. When a VFD slows a pump, its head/flow and brake horsepower (BHP) curves drop down and to the left and its efficiency curve shifts to the left. This efficiency response provides an essential cost advantage; by keeping the operating efficiency as high as possible across variations in the system's flow demand, the energy and maintenance costs of the pump can be significantly reduced.

VFDs may offer operating cost reductions by allowing higher pump operating efficiency, but the principal savings derive from the reduction in frictional or bypass flow losses. Using a system perspective to identify areas in which fluid energy is dissipated in non-useful work often reveals opportunities for operating cost reductions.

For example, in many systems, increasing flow through bypass lines does not noticeably impact the backpressure on a pump. Consequently, in these applications pump efficiency does not necessarily decline during periods of low flow demand. By analyzing the entire system, however, the energy lost in pushing fluid through bypass lines and across throttle valves can be identified.

Another system benefit of VFDs is a soft start capability. During startup, most motors experience in-rush currents that are 5 – 6 times higher than normal operating currents. This high current fades when the motor spins up to normal speed. VFDs allow the motor to be started with a lower startup current (usually only about 1.5 times the normal operating current). This reduces wear on the motor and its controller.

6.7 Energy Conservation Opportunities in Pumping Systems

- Ensure adequate NPSH at site of installation
- Ensure availability of basic instruments at pu[illegible] pressure gauges, flow meters.
- Operate pumps near best efficiency point.
- Modify pumping system and pumps losses to minimize throttling.
- Adapt to wide load variation with variable speed drives or sequenced control of multiple units.
- Stop running multiple pumps - add an auto-start for an on-line spare or add a booster pump in the problem area.
- Use booster pumps for small loads requiring higher pressures.
- Increase fluid temperature differentials to reduce pumping rates in case of heat exchangers.
- Repair seals and packing to minimize water loss by dripping.
- Balance the system to minimize flows and reduce pump power requirements.
- Avoid pumping head with a free-fall return (gravity); Use siphon effect to advantage:
- Conduct water balance to minimise water consumption
- Avoid cooling water re-circulation in DG sets, air compressors, refrigeration systems, cooling towers feed water pumps, condenser pumps and process pumps.

- In multiple pump operations, carefully combine the operation of pumps to avoid throttling
- Provide booster pump for few areas of higher head
- Replace old pumps by energy efficient pumps
- In the case of over designed pump, provide variable speed drive, or downsize / replace impeller or replace with correct sized pump for efficient operation.
- Optimise number of stages in multi-stage pump in case of head margins
- Reduce system resistance by pressure drop assessment and pipe size optimisation

QUESTIONS	
1.	What is NPSH of a pump and effects of inadequate NPSH?
2.	State the affinity laws as applicable to centrifugal pumps?
3.	Explain what do you understand by static head and friction head?
4.	What are the various methods of pump capacity control normally adopted?
5.	Briefly explain with a diagram the energy loss due to throttling in a centrifugal pump.
6.	Briefly explain with a sketch the concept of pump head flow characteristics and system resistance.
7.	What are the effects of over sizing a pump?
8.	If the speed of the pump is doubled, power goes up by a) 2 times b) 6 times c) 8 times d) 4 times
9.	How does the pump performance vary with impeller diameter?
10.	State the relationship between liquid kW, flow and pressure in a pumping application.
11.	Draw a pump curve for parallel operation of pumps (2 nos).
12.	Draw a pump curve for series operation of pumps (2 nos).
13.	List down few energy conservation opportunities in pumping system.

REFERENCES

1. British Pump Manufacturers' Association
2. BEE (EMC) Inputs
3. PCRA Literature

7. COOLING TOWER

Syllabus
Cooling Tower: Types and performance evaluation, Efficient system operation, Flow control strategies and energy saving opportunities, Assessment of cooling towers

7.1 Introduction

Cooling towers are a very important part of many chemical plants. The primary task of a cooling tower is to reject heat into the atmosphere. They represent a relatively inexpensive and dependable means of removing low-grade heat from cooling water. The make-up water source is used to replenish water lost to evaporation. Hot water from heat exchangers is sent to the cooling tower. The water exits the cooling tower and is sent back to the exchangers or to other units for further cooling. Typical closed loop cooling tower system is shown in Figure 7.1.

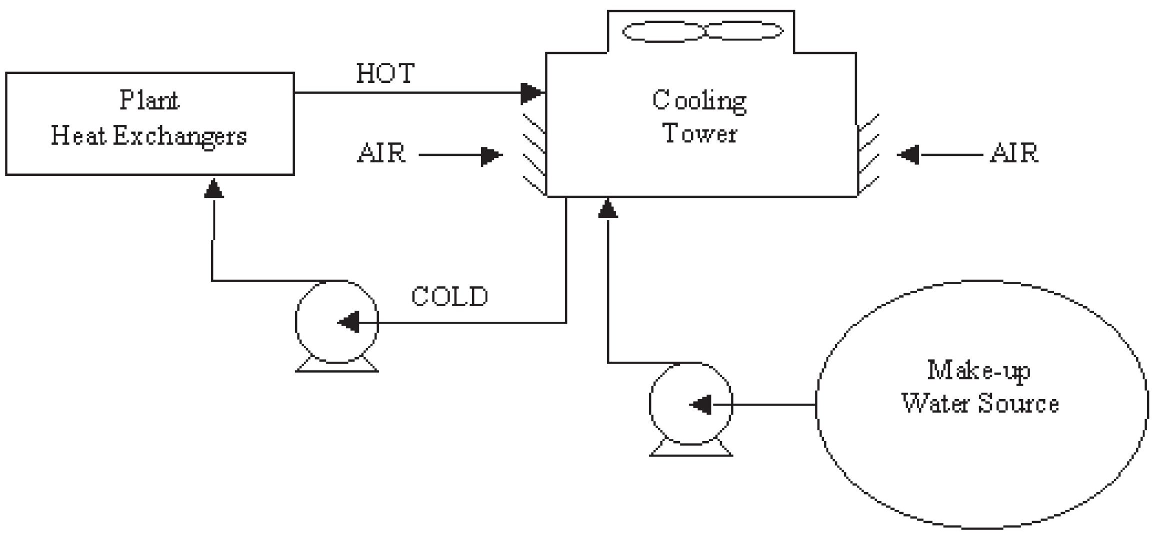


Figure 7.1 Cooling Water System

Cooling Tower Types

Cooling towers fall into two main categories: Natural draft and Mechanical draft.

Natural draft towers use very large concrete chimneys to introduce air through the media. Due to the large size of these towers, they are generally used for water flow rates above 45,000 m^3/hr. These types of towers are used only by utility power stations.

Mechanical draft towers utilize large fans to force or suck air through circulated water. The water falls downward over fill surfaces, which help increase the contact time between the water and the air - this helps maximise heat transfer between the two. Cooling rates of Mechanical draft towers depend upon their fan diameter and speed of operation. Since, the mechanical draft cooling towers are much more widely used, the focus is on them in this chapter.

Mechanical draft towers

Mechanical draft towers are available in the following airflow arrangements:

1. Counter flows induced draft.
2. Counter flow forced draft.
3. Cross flow induced draft.

In the counter flow induced draft design, hot water enters at the top, while the air is introduced at the bottom and exits at the top. Both forced and induced draft fans are used.

In cross flow induced draft towers, the water enters at the top and passes over the fill. The air, however, is introduced at the side either on one side (single-flow tower) or opposite sides (double-flow tower). An induced draft fan draws the air across the wetted fill and expels it through the top of the structure.

The Figure 7.2 illustrates various cooling tower types. Mechanical draft towers are available in a large range of capacities. Normal capacities range from approximately 10 tons, 2.5 m^3/hr flow to several thousand tons and m^3/hr. Towers can be either factory built or field erected - for example concrete towers are only field erected.

Many towers are constructed so that they can be grouped together to achieve the desired capacity. Thus, many cooling towers are assemblies of two or more individual cooling towers or "cells." The number of cells they have, e.g., an eight-cell tower, often refers to such towers. Multiple-cell towers can be lineal, square, or round depending upon the shape of the individual cells and whether the air inlets are located on the sides or bottoms of the cells.

Components of Cooling Tower

The basic components of an evaporative tower are: Frame and casing, fill, cold water basin, drift eliminators, air inlet, louvers, nozzles and fans.

Frame and casing: Most towers have structural frames that support the exterior enclosures (casings), motors, fans, and other components. With some smaller designs, such as some glass fiber units, the casing may essentially be the frame.

Fill: Most towers employ fills (made of plastic or wood) to facilitate heat transfer by maximising water and air contact. Fill can either be splash or film type.

With splash fill, water falls over successive layers of horizontal splash bars, continuously breaking into smaller droplets, while also wetting the fill surface. Plastic splash fill promotes better heat transfer than the wood splash fill.

Film fill consists of thin, closely spaced plastic surfaces over which the water spreads, forming a thin film in contact with the air. These surfaces may be flat, corrugated, honeycombed, or other patterns. The film type of fill is the more efficient and provides same heat transfer in a smaller volume than the splash fill.

Cold water basin: The cold water basin, located at or near the bottom of the tower, receives the cooled water that flows down through the tower and fill. The basin usually has a sump or low point for the cold water discharge connection. In many tower designs, the cold water basin is beneath the entire fill.

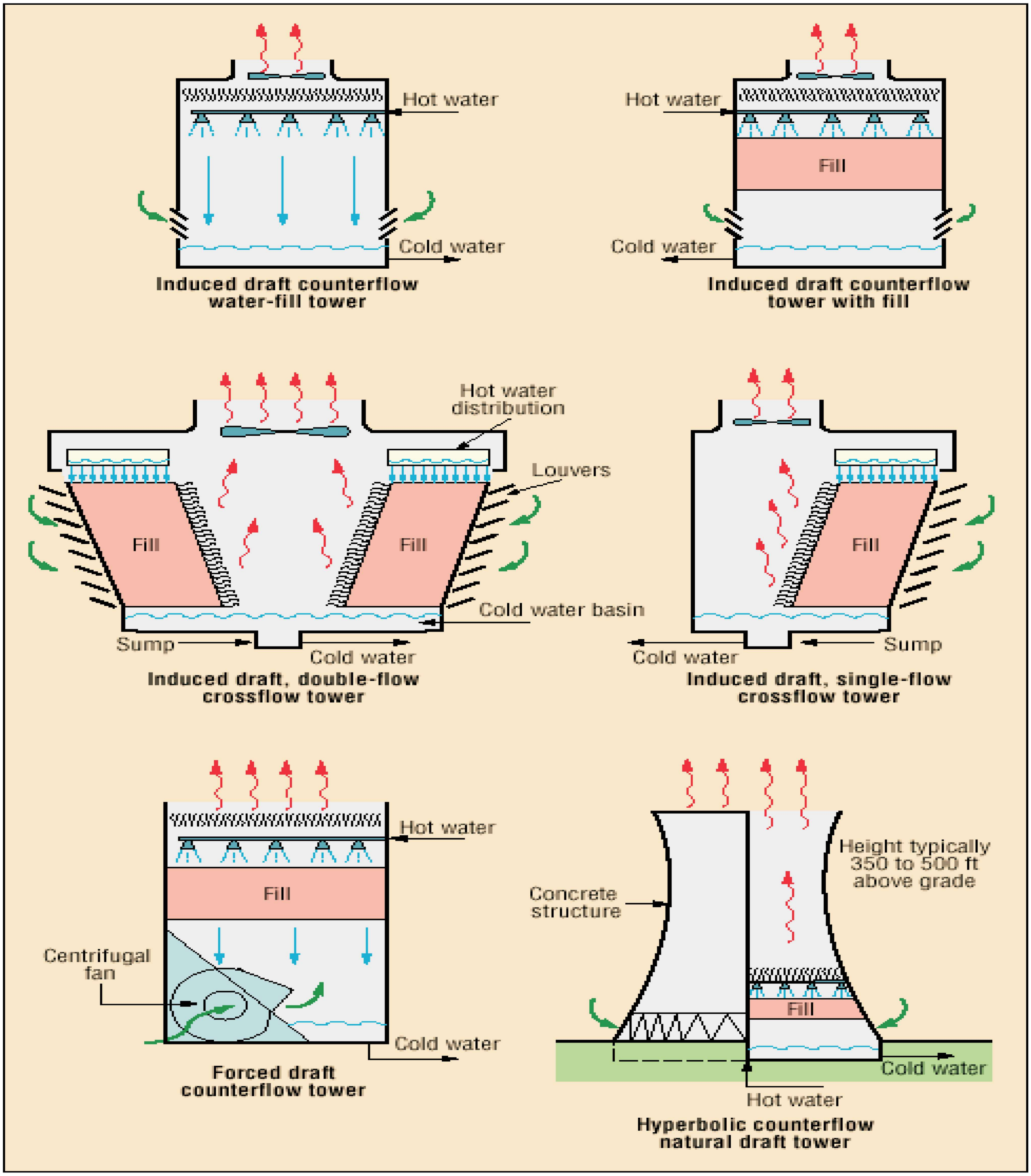


Figure 7.2 Cooling Tower Types

In some forced draft counter flow design, however, the water at the bottom of the fill is channeled to a perimeter trough that functions as the cold water basin. Propeller fans are mounted beneath the fill to blow the air up through the tower. With this design, the tower is mounted on legs, providing easy access to the fans and their motors.

Drift eliminators: These capture water droplets entrapped in the air stream that otherwise would be lost to the atmosphere.

Air inlet: This is the point of entry for the air entering a tower. The inlet may take up an entire side of a tower–cross flow design– or be located low on the side or the bottom of counter flow designs.

Louvers: Generally, cross-flow towers have inlet louvers. The purpose of louvers is to equalize air flow into the fill and retain the water within the tower. Many counter flow tower designs do not require louvers.

Nozzles: These provide the water sprays to wet the fill. Uniform water distribution at the top of the fill is essential to achieve proper wetting of the entire fill surface. Nozzles can either be fixed in place and have either round or square spray patterns or can be part of a rotating assembly as found in some circular cross-section towers.

Fans: Both axial (propeller type) and centrifugal fans are used in towers. Generally, propeller fans are used in induced draft towers and both propeller and centrifugal fans are found in forced draft towers. Depending upon their size, propeller fans can either be fixed or variable pitch.

A fan having non-automatic adjustable pitch blades permits the same fan to be used over a wide range of kW with the fan adjusted to deliver the desired air flow at the lowest power consumption.

Automatic variable pitch blades can vary air flow in response to changing load conditions.

Tower Materials

In the early days of cooling tower manufacture, towers were constructed primarily of wood. Wooden components included the frame, casing, louvers, fill, and often the cold water basin. If the basin was not of wood, it likely was of concrete.

Today, tower manufacturers fabricate towers and tower components from a variety of materials. Often several materials are used to enhance corrosion resistance, reduce maintenance, and promote reliability and long service life. Galvanized steel, various grades of stainless steel, glass fiber, and concrete are widely used in tower construction as well as aluminum and various types of plastics for some components.

Wood towers are still available, but they have glass fiber rather than wood panels (casing) over the wood framework. The inlet air louvers may be glass fiber, the fill may be plastic, and the cold water basin may be steel.

Larger towers sometimes are made of concrete. Many towers–casings and basins–are constructed of galvanized steel or, where a corrosive atmosphere is a problem, stainless steel. Sometimes a galvanized tower has a stainless steel basin. Glass fiber is also widely used for cooling tower casings and basins, giving long life and protection from the harmful effects of many chemicals.

Plastics are widely used for fill, including PVC, polypropylene, and other polymers. Treated wood splash fill is still specified for wood towers, but plastic splash fill is also widely used when water conditions mandate the use of splash fill. Film fill, because it offers greater heat transfer efficiency, is the fill of choice for applications where the circulating water is generally free of debris that could plug the fill passageways.

Plastics also find wide use as nozzle materials. Many nozzles are being made of PVC, ABS, polypropylene, and glass-filled nylon. Aluminum, glass fiber, and hot-dipped galvanized steel are commonly used fan materials. Centrifugal fans are often fabricated from galvanized steel. Propeller fans are fabricated from galvanized, aluminum, or moulded glass fiber reinforced plastic.

7.2 Cooling Tower Performance

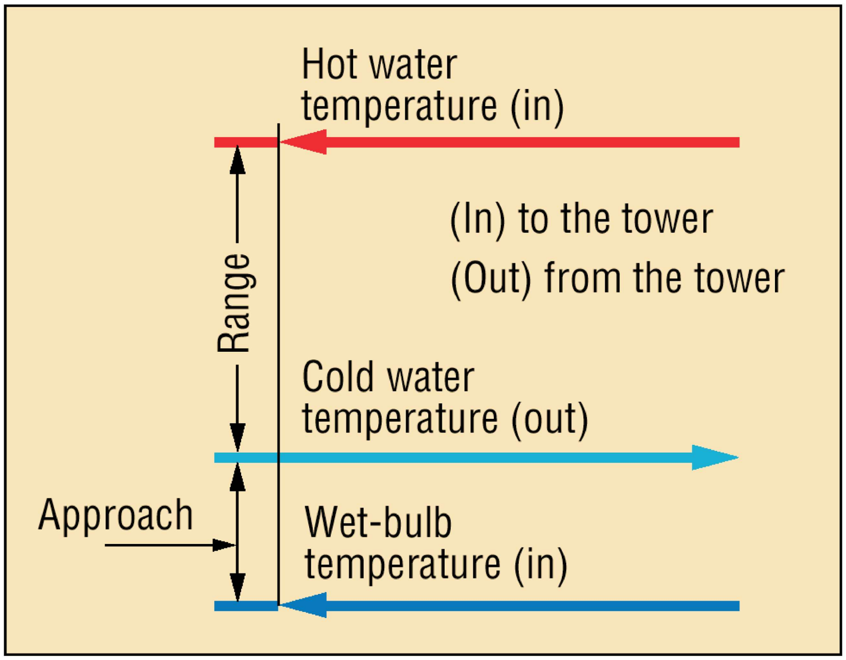


Figure 7.3 Range and Approach

The important parameters, from the point of determining the performance of cooling towers, are:

i) "Range" is the difference between the cooling tower water inlet and outlet temperature. (See Figure 7.3).

ii) "Approach" is the difference between the cooling tower outlet cold water temperature and ambient wet bulb temperature. Although, both range and approach should be monitored, the 'Approach' is a better indicator of cooling tower performance. (see Figure 7.3).

iii) Cooling tower effectiveness (in percentage) is the ratio of range, to the ideal range, i.e., difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = Range / (Range + Approach).

iv) Cooling capacity is the heat rejected in kCal/hr or TR, given as product of mass flow rate of water, specific heat and temperature difference.

v) Evaporation loss is the water quantity evaporated for cooling duty and, theoretically, for every 10,00,000 kCal heat rejected, evaporation quantity works out to 1.8 m^3. An empirical relation used often is:

$$*\text{Evaporation Loss } (m^3/hr) = 0.00085 \times 1.8 \times \text{circulation rate } (m^3/hr) \times (T_1\text{-}T_2)$$

T_1-T_2 = Temp. difference between inlet and outlet water.

*Source: Perry’s Chemical Engineers Handbook (Page: 12-17)

vi) Cycles of concentration (C.O.C) is the ratio of dissolved solids in circulating water to the dissolved solids in make up water.

vii) Blow down losses depend upon cycles of concentration and the evaporation losses and is given by relation:

$$\text{Blow Down} = \text{Evaporation Loss} / (\text{C.O.C.} - 1)$$

viii) Liquid/Gas (L/G) ratio, of a cooling tower is the ratio between the water and the air mass flow rates. Against design values, seasonal variations require adjustment and tuning of water and air flow rates to get the best cooling tower effectiveness through measures like water box loading changes, blade angle adjustments.

Thermodynamics also dictate that the heat removed from the water must be equal to the heat absorbed by the surrounding air:

$$L(T_1 - T_2) = G(h_2 - h_1)$$

$$\frac{L}{G} = \frac{h_2 - h_1}{T_1 - T_2}$$

where:

L/G = liquid to gas mass flow ratio (kg/kg)
T_1 = hot water temperature (°C)
T_2 = cold water temperature (°C)
h_2 = enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (same units as above)
h_1 = enthalpy of air-water vapor mixture at inlet wet-bulb temperature (same units as above)

Factors Affecting Cooling Tower Performance

Capacity

Heat dissipation (in kCal/hour) and circulated flow rate (m^3/hr) are not sufficient to understand cooling tower performance. Other factors, which we will see, must be stated along with flow rate m^3/hr. For example, a cooling tower sized to cool 4540 m^3/hr through a 13.9°C range might be larger than a cooling tower to cool 4540 m^3/hr through 19.5°C range.

Range

Range is determined not by the cooling tower, but by the process it is serving. The range at the exchanger is determined entirely by the heat load and the water circulation rate through the exchanger and on to the cooling water.

Range °C = Heat Load in kcals/hour / Water Circulation Rate in LPH

Thus, Range is a function of the heat load and the flow circulated through the system.

Cooling towers are usually specified to cool a certain flow rate from one temperature to another temperature at a certain wet bulb temperature. For example, the cooling tower might be specified to cool 4540 m^3/hr from 48.9°C to 32.2°C at 26.7°C wet bulb temperature.

Cold Water Temperature 32.2°C – Wet Bulb Temperature (26.7°C) = Approach (5.5°C)

As a generalization, the closer the approach to the wet bulb, the more expensive the cooling tower due to increased size. Usually a 2.8°C approach to the design wet bulb is the coldest water temperature that cooling tower manufacturers will guarantee. If flow rate, range, approach and wet bulb had to be ranked in the order of their importance in sizing a tower, approach would be first with flow rate closely following the range and wet bulb would be of lesser importance.

Factors that affect cooling tower size

Cooling tower size is affected by the heat load, range, approach, and WBT. When three of these four quantities are held constant, tower size varies in the following manner:

- Directly with the heat load
- Inversely with the range
- Inversely with the approach
- Inversely with the entering WBT

Heat Load

The heat load imposed on a cooling tower is determined by the process being served. The degree of cooling required is controlled by the desired operating temperature level of the process. In most cases, a low operating temperature is desirable to increase process efficiency or to improve the quality or quantity of the product. In some applications (e.g. internal combustion engines), however, high operating temperatures are desirable. The size and cost of the cooling tower is proportional to the heat load. If heat load calculations are low undersized equipment will be purchased. If the calculated load is high, oversize and more costly, equipment will result.

Process heat loads may vary considerably depending upon the process involved. Determination of accurate process heat loads can become very complex but proper consideration can produce satisfactory results. On the other hand, air conditioning and refrigeration heat loads can be determined with greater accuracy.

Information is available for the heat rejection requirements of various types of power equipment. A sample list is as follows:

*	Air Compressor		
	- Single-stage	-	129 kCal/kW/hr
	- Single-stage with after cooler	-	862 kCal/kW/hr
	- Two-stage with intercooler	-	518 kCal/kW/hr
	- Two-stage with intercooler and after cooler	-	862 kCal/kW/hr
*	Refrigeration, Compression	-	63 kCal/min/TR
*	Refrigeration, Absorption	-	127 kCal/min/TR
*	Steam Turbine Condenser	-	555 kCal/kg of steam
*	Diesel Engine, Four-Cycle, Supercharged	-	880 kCal/kW/hr
*	Natural Gas Engine, Four-cycle (18 kg/cm^2 compression)	-	1523 kCal/kW/hr

Wet Bulb Temperature

Wet bulb temperature is an important factor in performance of evaporative water cooling equipment. It is a controlling factor from the aspect of minimum cold water temperature to which water can be cooled by the evaporative method. Thus, the wet bulb temperature of the air entering the cooling tower determines operating temperature levels throughout the plant, process, or system. Theoretically, a cooling tower will cool water to the entering wet bulb temperature, when operating without a heat load. However, a thermal potential is required to reject heat, so it is not possible to cool water to the entering air wet bulb temperature, when a heat load is applied. The approach obtained is a function of thermal conditions and tower capability.

Initial selection of towers with respect to design wet bulb temperature must be made on the basis of conditions existing at the tower site. The temperature selected is generally close to the average maximum wet bulb for the summer months. An important aspect of wet bulb selection is, whether it is specified as ambient or inlet. The ambient wet bulb is the temperature, which exists generally in the cooling tower area, whereas inlet wet bulb is the wet bulb temperature of the air entering the tower. The later can be, and often is, affected by discharge vapours being recirculated into the tower. Recirculation raises the effective wet bulb temperature of the air entering the tower with corresponding increase in the cold water temperature. Since there is no initial knowledge or control over the recirculation factor, the ambient wet bulb should be specified. The cooling tower supplier is required to furnish a tower of sufficient capability to absorb the effects of the increased wet bulb temperature peculiar to his own equipment.

It is very important to have the cold water temperature low enough to exchange heat or to condense vapours at the optimum temperature level. By evaluating the cost and size of heat exchangers versus the cost and size of the cooling tower, the quantity and temperature of the cooling tower water can be selected to get the maximum economy for the particular process.

The Table 7.1 illustrates the effect of approach on the size and cost of a cooling tower. The towers included were sized to cool 4540 m^3/hr through a 16.67°C range at a 26.7°C design wet bulb. The overall width of all towers is 21.65 meters; the overall height, 15.25 meters, and the pump head, 10.6 m approximately.

TABLE 7.1 APPROACH VS. COOLING TOWER SIZE (4540 m^3/hr; 16.67°C Range 26.7°C Wet Bulb; 10.7 m Pump Head)

Approach °C	2.77	3.33	3.88	4.44	5.0	5.55
Hot Water °C	46.11	46.66	47.22	47.77	48.3	48.88
Cold Water °C	29.44	30	30.55	31.11	31.66	32.22
No. of Cells	4	4	3	3	3	3
Length of Cells Mts.	10.98	8.54	10.98	9.76	8.54	8.54
Overall Length Mts.	43.9	34.15	32.93	29.27	25.61	25.61
No. of Fans	4	4	3	3	3	3
Fan Diameter Mts.	7.32	7.32	7.32	7.32	7.32	6.71
Total Fan kW	270	255	240	202.5	183.8	183.8

Approach and Flow

Suppose a cooling tower is installed that is 21.65 m wide × 36.9 m long × 15.24m high, has three 7.32 m diameter fans and each powered by 25 kW motors. The cooling tower cools from 3632 m^3/hr water from 46.1°C to 29.4°C at 26.7°C WBT dissipating 60.69 million kCal/hr. The Table 7.2 shows what would happen with additional flow but with the range remaining constant at 16.67°C. The heat dissipated varies from 60.69 million kCal/hr to 271.3 million kCal/hr.

TABLE 7.2 FLOW VS. APPROACH FOR A GIVEN TOWER (Tower is 21.65 m × 36.9 M; Three 7.32 M Fans; Three 25 kW Motors; 16.7°C Range with 26.7°C Wet Bulb)

Flow m^3/hr	Approach °C	Cold Water °C	Hot Water °C	Million kCal/hr
3632	2.78	29.40	46.11	60.691
4086	3.33	29.95	46.67	68.318
4563	3.89	30.51	47.22	76.25
5039	4.45	31.07	47.78	84.05
5516	5.00	31.62	48.33	92.17
6060.9	5.56	32.18	48.89	101.28
7150.5	6.67	33.29	50.00	119.48
8736	8.33	35.00	51.67	145.63
11590	11.1	37.80	54.45	191.64
13620	13.9	40.56	57.22	226.91
16276	16.7	43.33	60.00	271.32

For meeting the increased heat load, few modifications would be needed to increase the water flow through the tower. However, at higher capacities, the approach would increase.

Range, Flow and Heat Load

Range is a direct function of the quantity of water circulated and the heat load. Increasing the range as a result of added heat load does require an increase in the tower size. If the cold water temperature is not changed and the range is increased with higher hot water temperature, the driving force between the wet bulb temperature of the air entering the tower and the hot water temperature is increased, the higher level heat is economical to dissipate.

If the hot water temperature is left constant and the range is increased by specifying a lower cold water temperature, the tower size would have to be increased considerably. Not only would the range be increased, but the lower cold water temperature would lower the approach. The resulting change in both range and approach would require a much larger cooling tower.

Approach & Wet Bulb Temperature

The design wet bulb temperature is determined by the geographical location. Usually the design wet bulb temperature selected is not exceeded over 5 percent of the time in that area. Wet bulb temperature is a factor in cooling tower selection; the higher the wet bulb temperature, the smaller the tower required to give a specified approach to the wet bulb at a constant range and flow rate.

A 4540 m^3/hr cooling tower selected for a 16.67°C range and a 4.45°C approach to 21.11°C wet bulb would be larger than a 4540 m^3/hr tower selected for a 16.67°C range and a 4.45°C approach to a 26.67°C wet bulb. Air at the higher wet bulb temperature is capable of picking up more heat. Assume that the wet bulb temperature of the air is increased by approximately 11.1°C. As air removes heat from the water in the tower, each kg of air entering the tower at 21.1°C wet bulb would contain 18.86 kCals and if it were to leave the tower at 33.2°C wet bulb it would contain 24.17 kCal per kg of air. In the second case, each kg of air entering the tower at 26.67°C wet bulb would contain 24.17 kCals and were to leave at 37.8°C wet bulb it would contain 39.67 kCal per kg of air. In going from 21.1°C to 32.2°C, 12.1 kCal per kg of air is picked up, while 15.5 kCal/kg of air is picked up in going from 26.67°C to 37.8°C.

Fill Media Effects

In a cooling tower, hot water is distributed above fill media which flows down and is cooled due to evaporation with the intermixing air. Air draft is achieved with use of fans. Thus some power is consumed in pumping the water to a height above the fill and also by fans creating the draft.

An energy efficient or low power consuming cooling tower is to have efficient designs of fill media with appropriate water distribution, drift eliminator, fan, gearbox and motor. Power savings in a cooling tower, with use of efficient fill design, is directly reflected as savings in fan power consumption and pumping head requirement.

Function of Fill media in a Cooling Tower

Heat exchange between air and water is influenced by surface area of heat exchange, time of heat exchange (interaction) and turbulence in water effecting thoroughness of intermixing. Fill media in a cooling tower is responsible to achieve all of above.

Splash and Film Fill Media: As the name indicates, splash fill media generates the required heat exchange area by splashing action of water over fill media and hence breaking into smaller water droplets. Thus, surface of heat exchange is the surface area of the water droplets, which is in contact with air.

Film Fill and its Advantages

In a film fill, water forms a thin film on either side of fill sheets. Thus area of heat exchange is the surface area of the fill sheets, which is in contact with air.

TABLE 7.3 TYPICAL COMPARISONS BETWEEN VARIOUS FILL MEDIA

	Splash Fill	Film Fill	Low Clog Film Fill
Possible L/G Ratio	1.1 – 1.5	1.5 – 2.0	1.4 – 1.8
Effective Heat Exchange Area	30 – 45 m^2/m^3	150 m^2/m^3	85 – 100 m^2/m^3
Fill Height Required	5 – 10 m	1.2 – 1.5 m	1.5 – 1.8 m
Pumping Head Requirement	9 – 12 m	5 – 8 m	6 – 9 m
Quantity of Air Required	High	Much low	Low

Typical comparison between various fill media is shown in Table 7.3.

Due to fewer requirements of air and pumping head, there is a tremendous saving in power with the invention of film fill.

Recently, low-clog film fills with higher flute sizes have been developed to handle high turbid waters. For sea water, low clog film fills are considered as the best choice in terms of power saving and performance compared to conventional splash type fills.

Choosing a Cooling Tower

The counter-flow and cross flows are two basic designs of cooling towers based on the fundamentals of heat exchange. It is well known that counter flow heat exchange is more effective as compared to cross flow or parallel flow heat exchange.

Cross-flow cooling towers are provided with splash fill of concrete, wood or perforated PVC. Counter-flow cooling towers are provided with both film fill and splash fill.

Typical comparison of Cross flow Spash Fill, Counter Flow Tower with Film Fill and Splash fill is shown in Table 7.4. The power consumption is least in Counter Flow Film Fill fol-

TABLE 7.4 TYPICAL COMPARISON OF CROSS FLOW SPLASH FILL, COUNTER FLOW TOWER WITH FILM FILL AND SPLASH FILL

Number of Towers	:	2
Water Flow	:	16000 m^3/hr.
Hot Water Temperature	:	41.5°C
Cold Water Temperature	:	32.5°C
Design Wet Bulb Temperature	:	27.6°C

	Counter Flow Film Fill	Counter Flow Splash Fill	Cross-Flow Splash Fill
Fill Height, Meter	1.5	5.2	11.0
Plant Area per Cell	14.4 × 14.4	14.4 × 14.4	12.64 × 5.49
Number of Cells per Tower	6	6	5
Power at Motor Terminal/Tower, kW	253	310	330
Static Pumping Head, Meter	7.2	10.9	12.05

lowed by Counter Flow Splash Fill and Cross-Flow Splash Fill.

7.3 Efficient System Operation

Cooling Water Treatment

Cooling water treatment is mandatory for any cooling tower whether with splash fill or with film type fill for controlling suspended solids, algae growth, etc.

With increasing costs of water, efforts to increase Cycles of Concentration (COC), by Cooling Water Treatment would help to reduce make up water requirements significantly. In large industries, power plants, COC improvement is often considered as a key area for water conservation.

Drift Loss in the Cooling Towers

It is very difficult to ignore drift problem in cooling towers. Now-a-days most of the end user specification calls for 0.02% drift loss.

With technological development and processing of PVC, manufacturers have brought large change in the drift eliminator shapes and the possibility of making efficient designs of drift eliminators that enable end user to specify the drift loss requirement to as low as 0.003 – 0.001%.

Cooling Tower Fans

The purpose of a cooling tower fan is to move a specified quantity of air through the system, overcoming the system resistance which is defined as the pressure loss. The product of air flow and the pressure loss is air power developed/work done by the fan; this may be also termed as fan output and input kW depends on fan efficiency.

The fan efficiency in turn is greatly dependent on the profile of the blade. An aerodynamic profile with optimum twist, taper and higher coefficient of lift to coefficient of drop ratio can provide the fan total efficiency as high as 85–92 %. However, this efficiency is drastically affected by the factors such as tip clearance, obstacles to airflow and inlet shape, etc.

As the metallic fans are manufactured by adopting either extrusion or casting process it is always difficult to generate the ideal aerodynamic profiles. The FRP blades are normally hand moulded which facilitates the generation of optimum aerodynamic profile to meet specific duty condition more efficiently. Cases reported where replacement of metallic or Glass fibre reinforced plastic fan blades have been replaced by efficient hollow FRP blades, with resultant fan energy savings of the order of 20–30% and with simple pay back period of 6 to 7 months.

Also, due to lightweight, FRP fans need low starting torque resulting in use of lower HP motors. The lightweight of the fans also increases the life of the gear box, motor and bearing is and allows for easy handling and maintenance.

Performance Assessment of Cooling Towers

In operational performance assessment, the typical measurements and observations involved are:

- Cooling tower design data and curves to be referred to as the basis.
- Intake air WBT and DBT at each cell at ground level using a whirling pyschrometer.
- Exhaust air WBT and DBT at each cell using a whirling psychrometer.
- CW inlet temperature at risers or top of tower, using accurate mercury in glass or a digital thermometer.
- CW outlet temperature at full bottom, using accurate mercury in glass or a digital thermometer.
- Process data on heat exchangers, loads on line or power plant control room readings, as relevant.
- CW flow measurements, either direct or inferred from pump motor kW and pump head and flow characteristics.
- CT fan motor amps, volts, kW and blade angle settings
- TDS of cooling water.
- Rated cycles of concentration at the site conditions.
- Observations on nozzle flows, drift eliminators, condition of fills, splash bars, etc.

The findings of one typical trial pertaining to the Cooling Towers of a Thermal Power Plant 3 x 200 MW is given below:

Observations

*	Unit Load 1 & 3 of the Station	=	398 MW
*	Mains Frequency	=	49.3
*	Inlet Cooling Water Temperature °C	=	44 (Rated 43°C)
*	Outlet Cooling Water Temperature °C	=	37.6 (Rated 33°C)
*	Air Wet Bulb Temperature near Cell °C	=	29.3 (Rated 27.5°C)
*	Air Dry Bulb Temperature near Cell °C	=	40.8°C
*	Number of CT Cells on line with water flow	=	45 (Total 48)
*	Total Measured Cooling Water Flow m^3/hr	=	70426.76
*	Measured CT Fan Flow m^3/hr	=	989544

Analysis

*	CT Water Flow/Cell, m^3/hr	=	1565 m^3/hr (1565000 kg/hr) (Rated 1875 m^3/hr)
*	CT Fan Air Flow, m^3/hr (Avg.)	=	989544 m^3/hr (Rated 997200 m^3/hr)
*	CT Fan Air Flow kg/hr (Avg.) @ Density of 1.08 kg/m^3	=	1068708 kg/hr
*	L/G Ratio of C.T. kg/kg	=	1.46 (Rated 1.74 kg/kg)
*	CT Range	=	(44 – 37.6) = 6.4°C
*	CT Approach	=	(37.6 – 29.3) = 8.3°C
*	% CT Effectiveness	=	$\frac{\text{Range}}{(\text{Range + Approach})}$ x100

	=	$\frac{6.4}{(6.4 + 8.3)}$ x100
	=	43.53
* Rated % CT Effectiveness	=	100 * (43 – 33) / (43 – 27.5)
	=	64.5%
* Cooling Duty Handled/Cell in kCal (i.e., Flow * Temperature Difference in kCal/hr)	=	1565 * 6.4 * 10^3
	=	10016 * 10^3 kCal/hr (Rated 18750 * 10^3 kCal/hr)
* Evaporation Losses in m^3/hr	=	0.00085 x 1.8 x circulation rate (m^3/hr) x (T_1-T_2)
	=	0.00085 x 1.8 x 1565 x (44-37.6)
	=	15.32 m^3/hr per cell
* Percentage Evaporation Loss	=	[15.32/1565]*100
	=	0.97%
* Blow down requirement for site COC of 2.7	=	Evaporation losses/COC-1
	=	15.32/(2.7-1) per cell i.e., 9.01 m^3/hr
* Make up water requirement/cell in m^3/hr	=	Evaporation Loss + Blow down Loss
	=	15.32 + 9.01
	=	24.33

Comments

- Cooling water flow per cell is much lower, almost by 16.5%, need to investigate CW pump and system performance for improvements. Increasing CW flow through cell was identified as a key result area for improving performance of cooling towers.
- Flow stratification in 3 cooling tower cells identified.
- Algae growth identified in 6 cooling tower cells.
- Cooling tower fans are of GRP type drawing 36.2 kW average. Replacement by efficient hollow FRP fan blades is recommended.

7.4 Flow Control Strategies

Control of tower air flow can be done by varying methods: starting and stopping (On-off) of fans, use of two- or three-speed fan motors, use of automatically adjustable pitch fans, use of variable speed fans.

On-off fan operation of single speed fans provides the least effective control. Two-speed fans provide better control with further improvement shown with three speed fans. Automatic adjustable pitch fans and variable-speed fans can provide even closer control of tower cold-water temperature. In multi-cell towers, fans in adjacent cells may be running at different speeds or some may be on and others off depending upon the tower load and required water temperature. Depending upon the method of air volume control selected, control strategies can be determined to minimise fan energy while achieving the desired control of the Cold water temperature.

7.5 Energy Saving Opportunities in Cooling Towers

- Follow manufacturer's recommended clearances around cooling towers and relocate or modify structures that interfere with the air intake or exhaust.
- Optimise cooling tower fan blade angle on a seasonal and/or load basis.
- Correct excessive and/or uneven fan blade tip clearance and poor fan balance.
- On old counter-flow cooling towers, replace old spray type nozzles with new square spray ABS practically non-clogging nozzles.
- Replace splash bars with self-extinguishing PVC cellular film fill.
- Install new nozzles to obtain a more uniform water pattern
- Periodically clean plugged cooling tower distribution nozzles.
- Balance flow to cooling tower hot water basins.
- Cover hot water basins to minimise algae growth that contributes to fouling.
- Optimise blow down flow rate, as per COC limit.
- Replace slat type drift eliminators with low pressure drop, self extinguishing, PVC cellular units.
- Restrict flows through large loads to design values.
- Segregate high heat loads like furnaces, air compressors, DG sets, and isolate cooling towers for sensitive applications like A/C plants, condensers of captive power plant etc. A 1°C cooling water temperature increase may increase A/C compressor kW by 2.7%. A 1°C drop in cooling water temperature can give a heat rate saving of 5 kCal/kWh in a thermal power plant.
- Monitor L/G ratio, CW flow rates w.r.t. design as well as seasonal variations. It would help to increase water load during summer and times when approach is high and increase air flow during monsoon times and when approach is narrow.
- Monitor approach, effectiveness and cooling capacity for continuous optimisation efforts, as per seasonal variations as well as load side variations.
- Consider COC improvement measures for water savings.
- Consider energy efficient FRP blade adoption for fan energy savings.
- Consider possible improvements on CW pumps w.r.t. efficiency improvement.
- Control cooling tower fans based on leaving water temperatures especially in case of small units.
- Optimise process CW flow requirements, to save on pumping energy, cooling load, evaporation losses (directly proportional to circulation rate) and blow down losses.

Some typical problems and their trouble shooting for cooling towers are given in Table 7.5.

TABLE 7.5 TYPICAL PROBLEMS AND TROUBLE SHOOTING FOR COOLING TOWERS

Problem / Difficulty	Possible Causes	Remedies/Rectifying Action
Excessive absorbed current / electrical load	1. Voltage Reduction	Check the voltage
	2a. Incorrect angle of axial fan blades	Adjust the blade angle
	2b. Loose belts on centrifugal fans (or speed reducers)	Check belt tightness
	3. Overloading owing to excessive air flow-fill has minimum water loading per m^2 of tower section	Regulate the water flow by means of the valve
	4. Low ambient air temperature	The motor is cooled proportionately and hence delivers more than name plate power
Drift/carry-over of water outside the unit	1. Uneven operation of spray nozzles	Adjust the nozzle orientation and eliminate any dirt
	2. Blockage of the fill pack	Eliminate any dirt in the top of the fill
	3. Defective or displaced droplet eliminators	Replace or realign the eliminators
	4. Excessive circulating water flow (possibly owing to too high pumping head)	Adjust the water flow-rate by means of the regulating valves. Check for absence of damage to the fill
Loss of water from basins/pans	1. Float-valve not at correct level	Adjust the make-up valve
	2. Lack of equalising connections	Equalise the basins of towers operating in parallel
Lack of cooling and hence increase in temperatures owing to increased temperature range	1. Water flow below the design valve	Regulated the flow by means of the valves
	2. Irregular airflow or lack of air	Check the direction of rotation of the fans and/or belt tension (broken belt possible)
	3a. Recycling of humid discharge air	Check the air descent velocity
	3b. Intake of hot air from other sources	Install deflectors
	4a. Blocked spray nozzles (or even blocked spray tubes)	Clean the nozzles and/or the tubes
	4b. Scaling of joints	Wash or replace the item
	5. Scaling of the fill pack	Clean or replace the material (washing with inhibited aqueous sulphuric acid is possible but long, complex and expensive)

QUESTIONS

1.	What do you understand by the following terms in respect of cooling towers? a) Approach, b) Cooling Duty c) Range d) Cooling Tower Effectiveness
2.	Explain with a sketch the different types of cooling towers.
3.	What do you mean by the term of Cycles of Concentration and how it is related to cooling tower blow down?
4.	Explain the term L/G ratio?
5.	CT Observations at an industrial site were * CW Flow : 5000 m^3/hr * CW in Temperature : 42°C * CW Out Temperature : 36°C * Wet Bulb Temperature : 29°C What is the Effectiveness of the cooling tower?
6.	What is the function of fill media in a cooling tower?
7.	List the factors affecting cooling tower performance.
8.	List the energy conservation opportunities in a cooling tower system.
9.	Explain the difference between evaporation loss and drift loss?
10.	What is the Blow-down Loss, if the Cycles of Concentration (COC) is 3.0?

REFERENCES

1. ASHRAE Handbook
2. NPC Case Studies