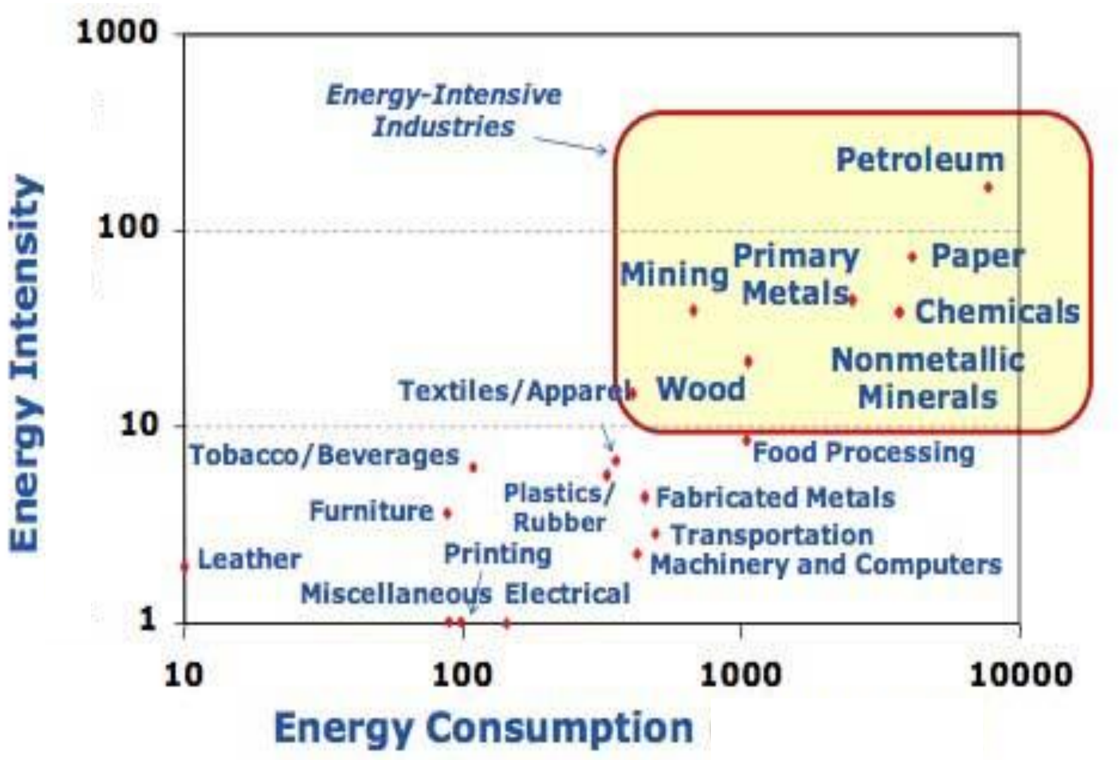
Energy Efficiency

1 Introduction

The petroleum industries are the largest worldwide energy consumer, hence any energy efficiency improvements here will have a significant impact on overall CO_2 and other emissions. This is shown in the data from the EIA Bureau of economic analysis.



The UK CO2 emissions targets (Climate targets Act) are as follows –

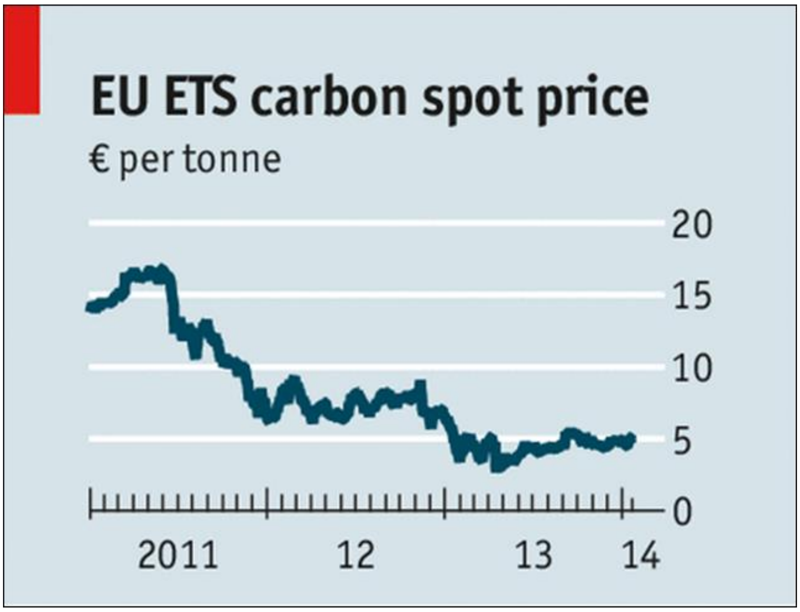
Budget	Carbon budget level	% reduction below base year
1st Carbon budget (2008-12)	3,018 MtCO2e	23%
2nd Carbon budget (2013-17)	2,782 MtCO2e	29%
3rd Carbon budget (2018-22)	2,544 MtCO2e	35% by 2020
4th Carbon budget (2023-27)	1,950 MtCO2e	50% by 2025

This module covers the common energy consumers, discusses the associated energy profiles and means for making equipment and unit operations more efficient.

2 EU Emissions Trading

Under the EU ETS, large emitters of carbon dioxide within the EU must monitor their CO_2 emissions, and annually report them, as they are obliged every year to return an amount of emission allowances to the government that is equivalent to their CO_2 emissions in that year. In order to neutralize annual irregularities in CO_2 emission levels that may occur due to extreme weather events (such as harsh winters or very hot summers), emission credits for any plant operator subject to the EU ETS are given out for a sequence of several years at once. Each such sequence of years is called a Trading Period. The 1st EU ETS Trading Period expired in December 2007; it had covered all EU ETS emissions since January 2005. With its termination, the 1st phase EU allowances became invalid. Since January 2008, the 2nd Trading lasted until December 2012. Currently, the installations receive trading credits from the NAPS (national allowance plans) which is part of each country's government. Besides receiving this initial allocation, an operator may purchase EU and international trading credits. If an installation has performed well at reducing its carbon emissions then it has the opportunity to sell its credits and make a profit. This allows the system to be more self contained and be part of the stock exchange without much government intervention.

The following shows the time movement of CO_2 spot prices. As can be seen these have been falling which is reducing the incentive to trade.



3 Pumps

The following statement has been issued by the EU;

'Pumps are the single largest user of electricity in Industry in the European Union, consuming 160 TWhpa of electricity, accounting for 79 Mton CO2'.

The US view follows;

"*..The U.S. Department of Energy also indicates that pumping systems account for nearly 20 percent of the world's electrical energy demand, and frequently they consume from 25 percent to 50 percent of the energy in industrial process plants. According to the U.S. Industrial Motor Systems Market Opportunities Assessment performed by the U.S. Department of Energy, pumps are the largest opportunity for energy efficiency improvements in industry..*"
"*A pump's efficiency can degrade* ***as much as 10% to 25%*** *before it is replaced, according to a study of industrial facilities commissioned by the U.S. Department of Energy (DOE), and efficiencies of 50% to 60% or lower are quite common. However, because these inefficiencies are not readily apparent, opportunities to save energy by repairing or replacing components and optimizing systems are often overlooked*."

Clearly energy efficiency improvements associated with pumps will have a significant impact on CO_2 footprint.

Pump Types

Like compressors pumps fall into similar categories;

- Centrifugal
- Rotary
- Reciprocating
- Hydraulic / Jet

Typical Upstream duties are as follows.

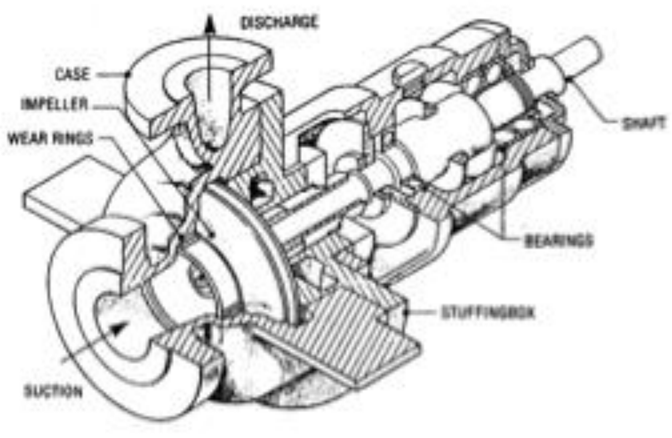
- Fire pumps
- Seawater lift
- Water injection
- Crude/condensate boosting

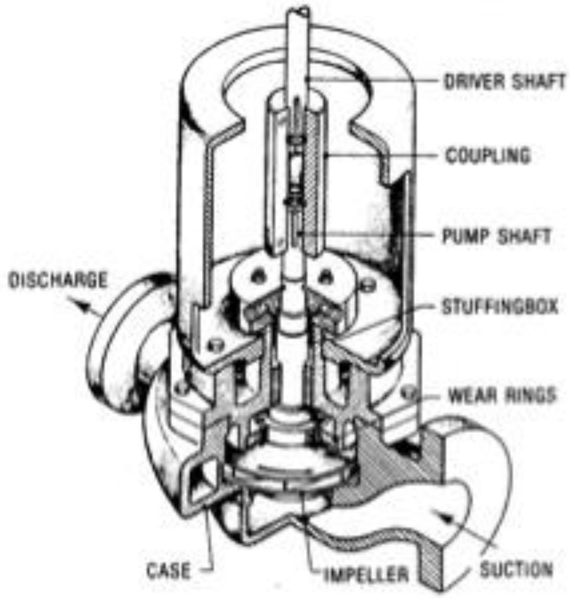
– Crude export
– Well fluids production
– General process services
– General utility services
– Cooling water
– Chemical injection

Again, like compressors the centrifugal configuration is very common.

4 Centrifugal Pump

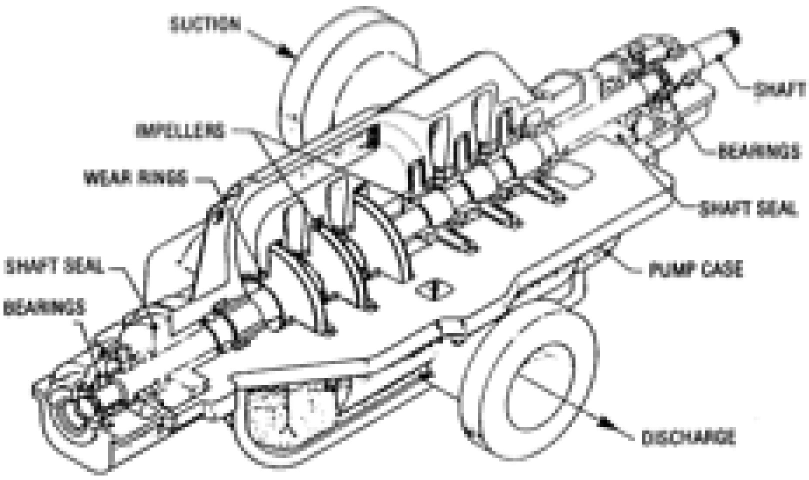
A centrifugal pump works on the principle of conversion of the kinetic energy of a flowing fluid (velocity pressure) into static pressure. This action is described by Bernoulli's principle. The rotation of the pump impeller accelerates the fluid as it passes from the impeller eye (centre) and outward through the impeller vanes to the periphery. As the fluid exits the impeller, a proportion of the fluid momentum is then converted to pressure. Typically the volute shape of the pump casing, or the diffuser vanes assists in the energy conversion. The energy conversion results in an increased pressure on the downstream side of the pump.





4.1 Water Injection and Oil Export Pumps

These pumps generally require high discharge pressure, typically 100 – 300 bar. Since a single impeller would be too large to achieve this level of discharge pressure, multiple impellers are required as shown.



Such pumps generally require many megawatts of power and, to supply an adequate net positive suction head (NPSH), a suction boost pump is often required.

4.2 Boost pumps

Booster pumps are required to provide sufficient NPSH for the main injection pumps. The inlet area is sized in to keep the inlet velocity low. High inlet velocities cause the inlet pressure to drop, hence potentially causing cavitation (see later). Boost pumps often operate at a different speed from the main injection pump; the booster typically operates at 1800 rpm, while the injection pump is typically geared at 5000 to 6000 rpm.

4.3 Seawater Lift Pumps

These centrifugal deliver water from around 30m depth to the platform decks for cooling and injection requirements. Two arrangements are common;

Lineshaft – motor is located on the platform deck with a long shaft to the submerged pump

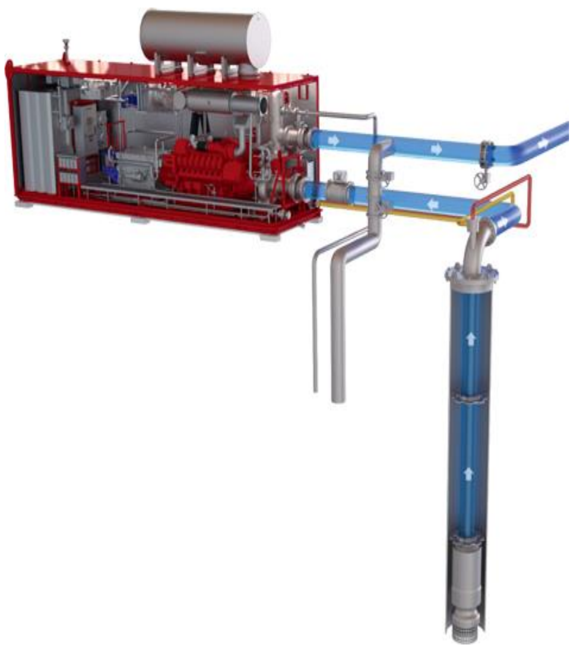
Submerged – the motor is located adjacent to the pump with a power cable.

These are shown.

1. Surface delivery
2. Vertical pipe
3. Impellers
4. Bearings
5. Cable connection
6. Motor
7. End fitting

4.4 Fire Water Pump

A fire pump is a totally independent, self contained system located in a blast and fire proof enclosure. The requirement on an installation is usually two independently powered 100% firewater pumps. A separate hydraulic system drives a submerged lift pump which supplies water to the booster pump

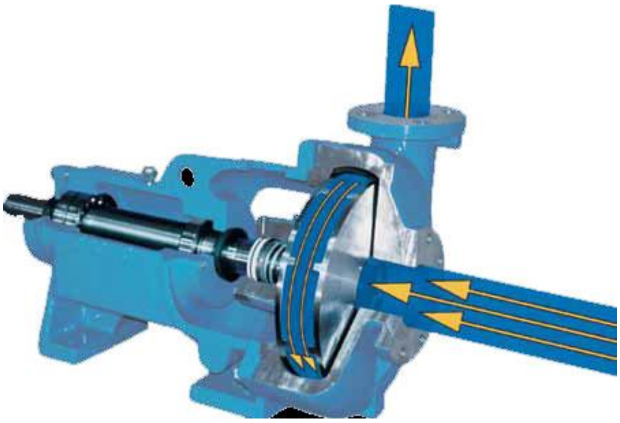


delivering the required discharge pressure into the fire main. Diesel driven pumps are the norm with typically a 24 hour diesel day tank.

4.5 Low Shear Pumps

Produced Water transfer pumps will have a tendency to shear the dispersed oil into smaller droplets thus making oil/water separation more difficult. To minimise the amount of pump shearing a Disc pump can be utilised.

Disc pumps operate on the principles of Boundary Layer and Viscous Drag. Under laminar flow conditions, streams of liquid travel at different velocities through a pipe, with the layer closest to the pipe being stationary – known as the Boundary Layer – and successive fluid layers flowing faster towards the centre of the pipe.

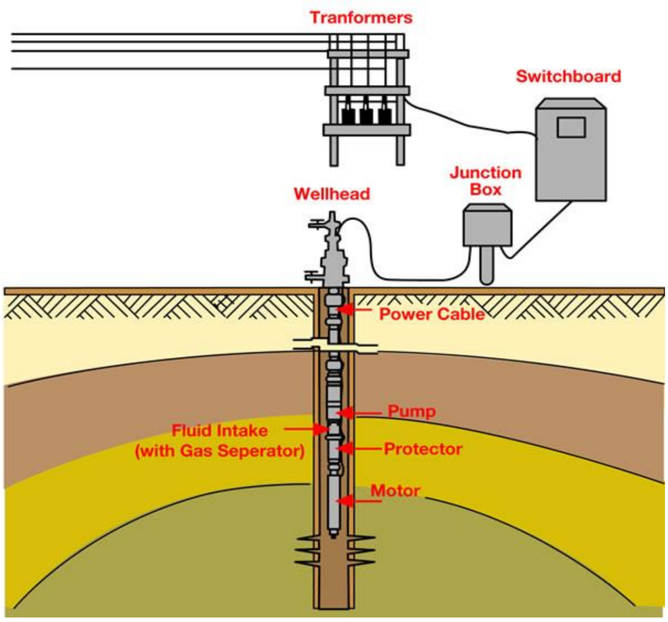


Similarly, when a fluid enters the disc pump, a boundary layer is formed on the surfaces of a series of parallel discs which form the pumping mechanism. As the discs rotate, energy is transferred to successive layers of molecules in the fluid between the discs via the Viscous Drag Principle, generating velocity and pressure gradients across the width of the disc pack. These pumps tend to be very inefficient

4.6 Electric Submersible Pump (ESP)

The system’s surface equipment includes transformers, a switchboard, junction box and surface power cables. Power passes through a cable running from the transformer to the switchboard and junction box, then to the wellhead

The ESP downhole assembly is located in the well at the bottom of the tubing. The motor, seal, intake and pump assembly, along with the power cable, goes in the well as the tubing is run. Below the pump is an intake that allows



fluid to enter the pump. Below the intake is a gas separator and a protector or seal, which equalizes internal and external pressures and protects the motor from well fluids. At the bottom is a motor that drives the pump. The assembly is positioned in the well above the perforations; this allows fluid entering the intake to flow past the motor and cool it.

4.7 Hydraulic Submersible Pump

The hydraulic submersible pump (HSP) system consists of a surface charge pump, power fluid control valve, HSP and a power fluid filter.

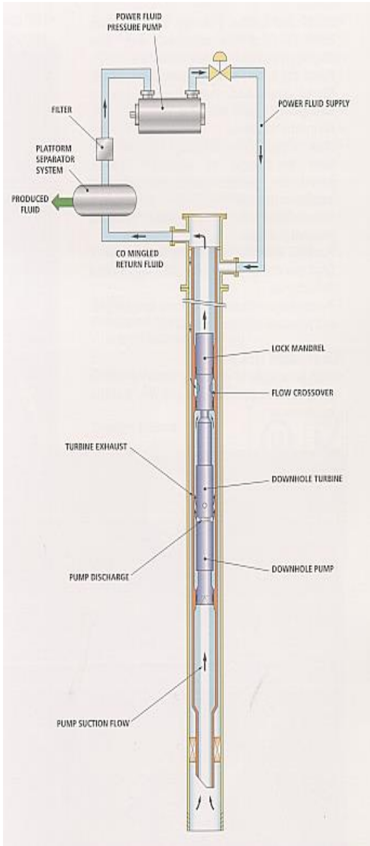
In operation, power fluid is pressure boosted by the surface charge pump and passed through the control valve before being injected down the well tubing and into the turbine.

The power fluid drives the turbine stages, causing the pump to rotate, before exhausting at the lower end of the turbine unit.

The pump suction flow enters at the bottom of the pump and is boosted in pressure through the various pump stages before discharging at the upper end of the pump unit.

The turbine power fluid can be produced water, aquifer water, or produced oil, depending on which is the most suitable for the application under consideration.

The ratio of power to produced fluid is in the region of 1 : 1, although this can be varied to suit specific operational flow and pressure requirements.

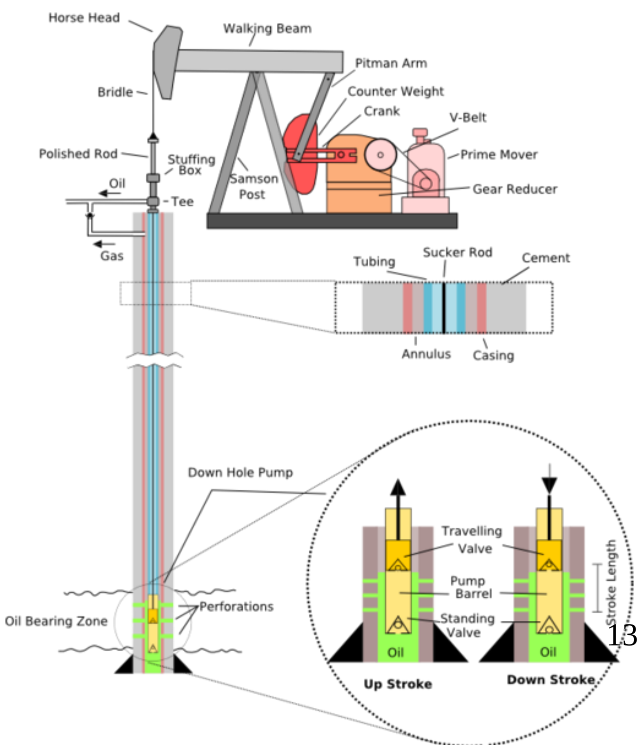


DOWNHOLE COMPLETION FOR WIRELINE CONVEYED HYDRAULIC DRIVE DOWNHOLE PUMP

5 Positive Displacement Pumps

5.1 Sucker Rod Beam Pump

This is the iconic image of oilfields – the nodding donkey. At the bottom of the tubing is the down-hole pump. This pump has two ball check valves: a stationary valve at the bottom called the standing valve, and a valve on the piston connected to the bottom of the sucker rods that



travels up and down as the rods reciprocate, known as the travelling valve. Reservoir fluid enters from the formation into the bottom of the borehole.

When the rods at the pump end are travelling up, the travelling valve is closed and the standing valve is open (due to the drop in pressure in the pump barrel). Consequently, the pump barrel fills with the fluid from the formation as the piston lifts the previous contents of the barrel upwards.

When the rods begin pushing down, the travelling valve opens and the standing valve closes (due to an increase in pressure in the pump barrel). The travelling valve drops through the fluid in the barrel (which had been sucked in during the upstroke). The piston then reaches the end of its stroke and begins its path upwards again, repeating the process.

5.2 Rotary Gear Pumps

A gear pump uses the meshing of gears to pump fluid by displacement. They are one of the most common types of pumps for hydraulic fluid power applications. Gear pumps are also widely used in chemical installations to pump fluid with high viscosities.

As the gears rotate they separate on the intake side of the pump, creating a void that is filled by fluid. The fluid is carried by the gears to the discharge side of the pump, where the meshing of the gears displaces the fluid.

The mechanical clearances are small, in the order of 10 µm. The tight clearances, along with the speed of rotation, effectively prevent the fluid from leaking backwards.

Applications are diesel oil transfer, lube oil distribution and viscous liquids.

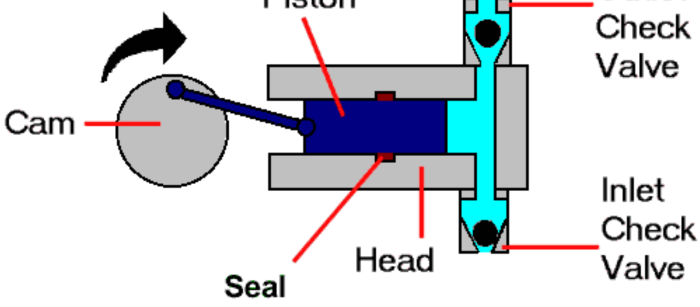
5.3 Screw/Progressive Cavity

A screw pump is a positive displacement pump that uses one or several screws to move fluids or solids along the screw(s) axis. In its simplest form (the Archimedes' screw pump), a single screw rotates in a cylindrical cavity, thereby moving the material along the screw's spindle. This ancient construction is still used in many low-tech applications, such as irrigation systems and in agricultural machinery for transporting grain and other solids. Development of the screw pump has led to a variety of multi-axis technologies where carefully crafted screws rotate in opposite directions or remains stationary within a cavity. The cavity can be profiled, thereby creating cavities where the pumped material is "trapped".

The progressive cavity pump consists of a helical rotor and a twin helix, twice the wavelength and double the diameter helical hole in a rubber stator. The rotor seals tightly against the rubber stator as it rotates, forming a set of fixed-size cavities in between. The cavities move when the rotor is rotated but their shape or volume does not change.

5.4 Reciprocating Pumps

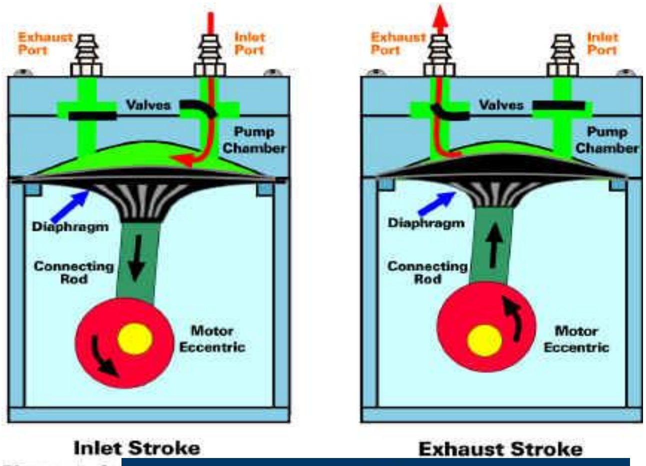
A pump consisting of a piston that moves back and forth or up and down in a cylinder. The cylinder is equipped with inlet (suction) and outlet (discharge) valves. On the intake stroke, the suction valves are opened, and fluid is drawn into the cylinder. On the discharge stroke, the suction valves close, the discharge valves open, and fluid is forced out of the cylinder. Drilling mud pumps are invariably reciprocating.



5.5 Diaphragm Pump

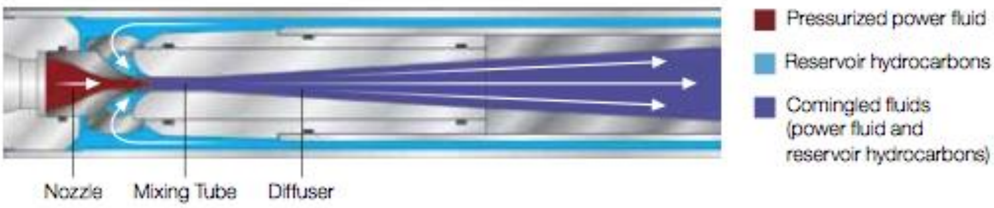
Diaphragm pumps are reciprocating, positive displacement type pumps, utilizing a valving system similar to a plunger pump. These pumps can deliver a small, precisely controlled amount of liquid at a moderate to very high discharge pressure.

Diaphragm pumps are commonly used as chemical injection pumps because of their controllable metering capability, the wide range of materials in which they can be fabricated, and their inherent leakproof design.



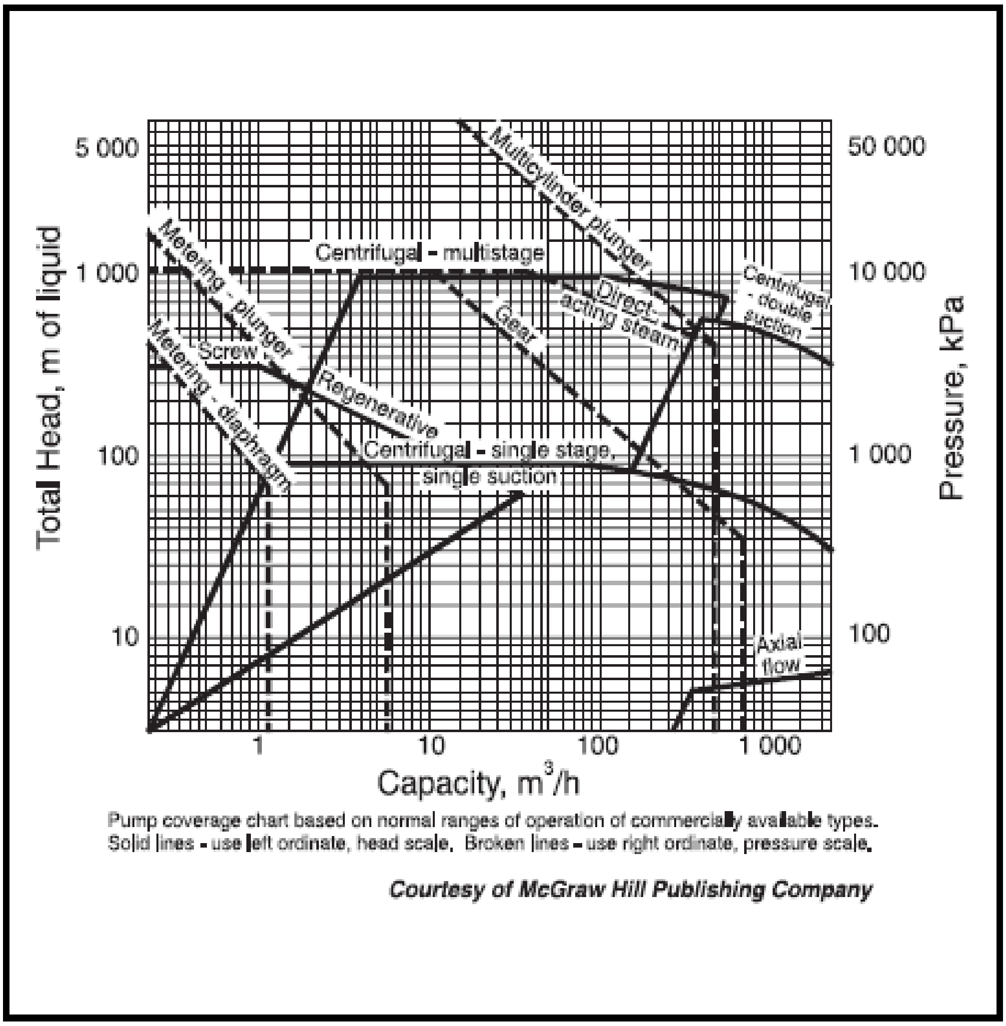
6 Jet Pumps - Eductors

Jet pumps operate on the Venturi Principle. A high pressure fluid is accelerated through a throat where the pressure drops. The reduced pressure allows a lower pressure system to be sucked in.



7 Preliminary Pump Selection

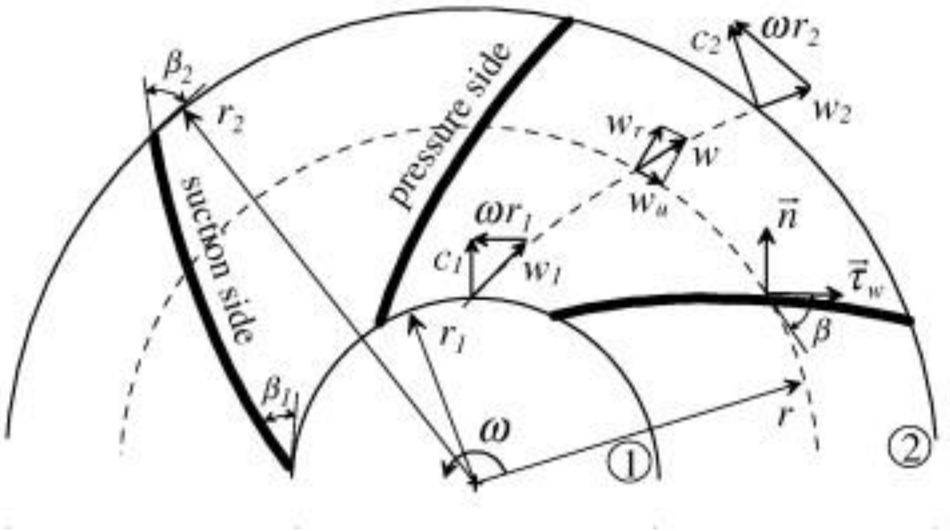
The following chart is useful for preliminary selection. Detailed selection will require an analysis of capital and operating cost, uptime, environment and safety.



8 Centrifugal Pump Features

These are the most commonly used industrial pumps and this section addresses pump efficiency.

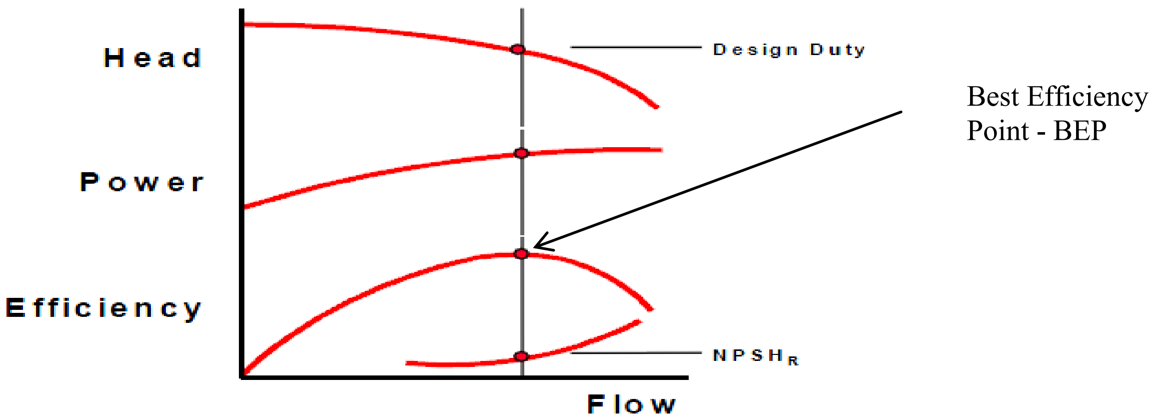
Recall that a centrifugal pump increases the absolute pressure of a fluid by adding kinetic (velocity) energy – ½ mv^2 and then converting that to pressure/head (potential energy) – m.g.h in the pump volute. Refer to the following figure - the fluid is drawn into the eye of the impeller (point 1) at a velocity v_1, approximately the volume flow divided by the cross sectional area of the impeller eye. The rotation of the impeller increases the velocity and pressure of the fluid . When the fluid reaches point 2 it is thrown from the impeller and then it is slowed down by the increasing area of the volute converting kinetic to potential energy.



The pump efficiency is affected by the angle and velocity at which the liquid is thrown from the impeller rim. Changing the flowrate alters the throw off vector and hence alters the efficiency of energy recovery to head/pressure. The pump designer provides an impeller design that gives maximum efficiency at the given design rate.

8.1 Centrifugal Pump Characteristics

Generic curves are illustrated in



The key pump components are illustrated

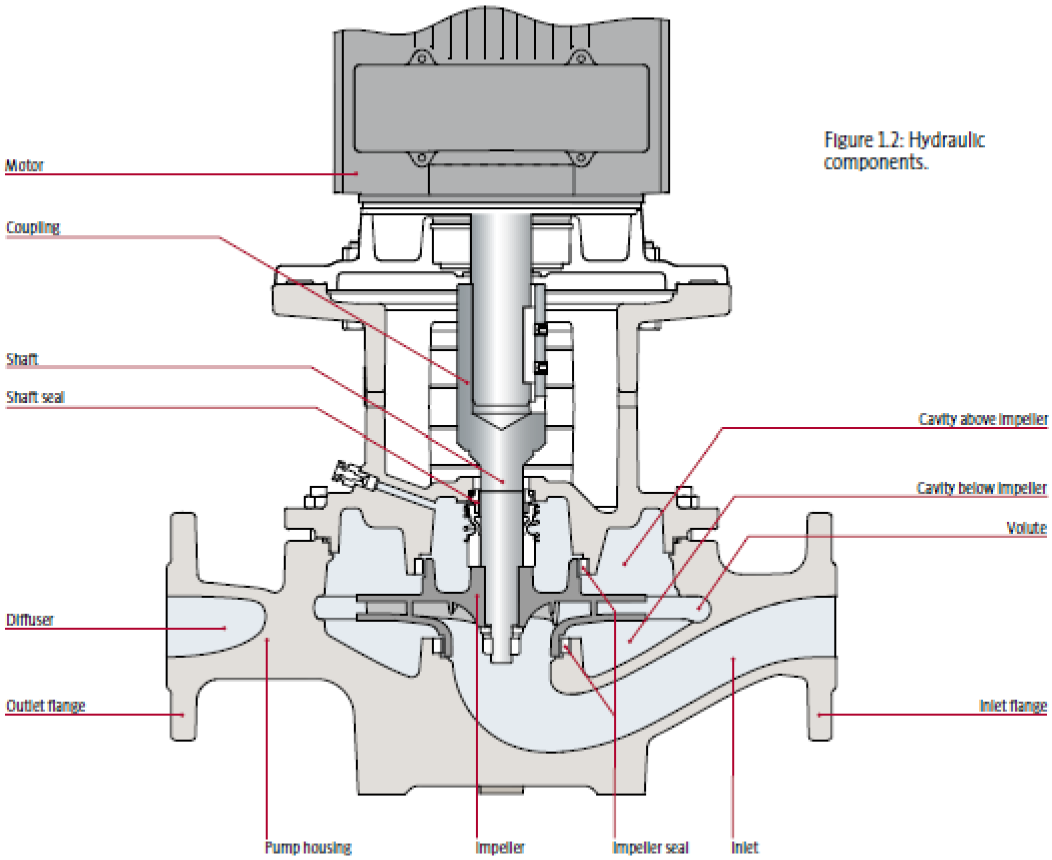
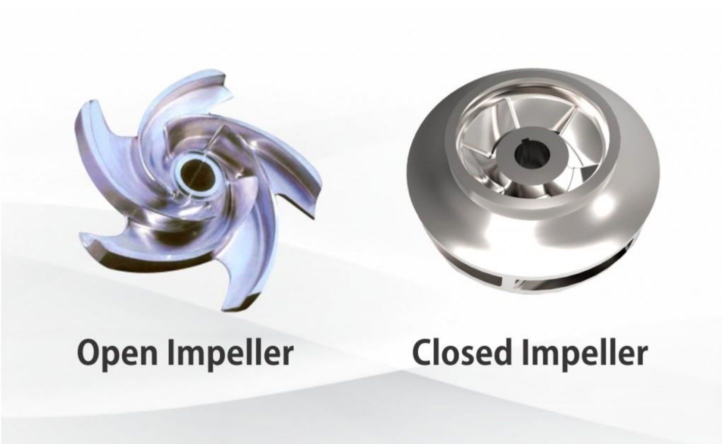


Figure 1.2: Hydraulic components.

8.2 Impellers

The open impeller is a series of vanes attached to a central hub for mounting on the shaft without any form of side wall or shroud. The semi-open impeller incorporates a single shroud at the back of the impeller. The closed impeller has a shroud on either side of the vanes. The impeller specific speed describes the shape of the impeller.

The shape of the head/ capacity curve is a function of specific speed, but the designer has some control of the head and capacity through the selection of the vane angle and the number of vanes.



8.3 Volutes - Diffusers

The volute casing collects the fluid from the impeller and leads into the outlet flange. The volute casing converts the dynamic pressure rise in the impeller to static pressure. The velocity is gradually reduced when the cross sectional area of the fluid flow is increased. This transformation is called velocity diffusion.

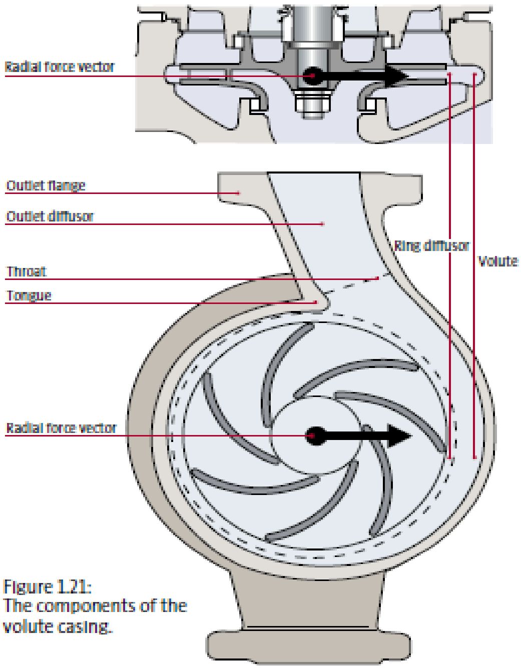


Figure 1.21:
The components of the volute casing.

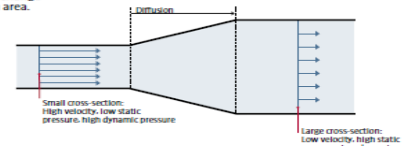


Figure 1.20: Change of fluid velocity in a pipe caused by change in the cross-section area.

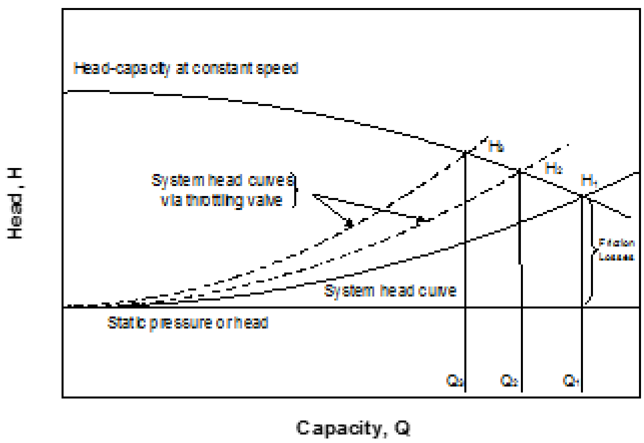
8.4 Pump Capacity Control

Key to evaluating system efficiency is linking the pump performance with the piping system. The pump provides pressure to overcome the flow resistances – elevation, friction, acceleration.

8.4.1 Control Valve Throttling

A control valve on the pump discharge provides a means for increasing or lowering the pump discharge pressure. The pump and piping system will reach a flow position where the pump discharge pressure is in balance with the flow resistances.

This is illustrated opposite.



8.4.2 Speed Control

The fan/affinity laws provide an insight into pump characteristics. The laws are as follows;

$$\frac{Q_1}{Q_2} = \left(\frac{N_1}{N_2}\right) \cdot \left(\frac{D_1}{D_2}\right)$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \cdot \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{W_1}{W_2} = \left(\frac{N_1}{N_2}\right)^3 \cdot \left(\frac{D_1}{D_2}\right)^3$$

Where;

Q – flowrate

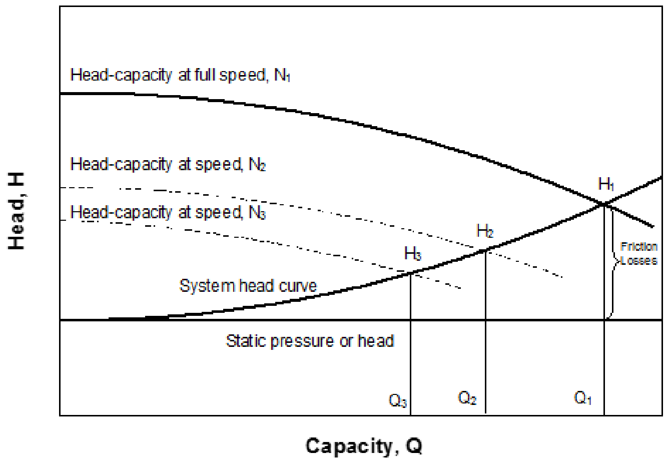
N – rotational speed

D – impeller diameter

H – pump head

W – power

The pump and system characteristics with speed control are shown.



Note that from an efficiency viewpoint, speed control is much more efficient than discharge throttling.

8.5 Net Positive Suction Head (NPSH)

NPSH (Net Positive Suction Head) can be considered as the amount of energy in the liquid at the pump datum. Sufficient NPSH must be provided to prevent formation of small gas bubbles (when the pressure is at or below the bubble point) - collapsing bubbles damaging the pump through the in rush of liquid to the void. This is termed cavitation.

The required NPSH is a pump characteristic provided by the supplier.

The required NPSH varies with pump design, pump size and operating conditions;

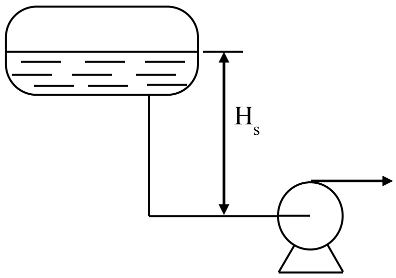
– rotation speed
– inlet area (eye)
– type and number of vanes

For reciprocating pumps:

– speed
– valve design

Available NPSH can be calculated as follows;

$$NPSH_{available} = \frac{10.19 \cdot \left(P_s - P_v - \Delta P_f\right)}{\gamma} + H_s - H_m - \frac{v^2}{2g} - H_{ac}$$



NPSH : Net Positive Suction Head (m)

P_s : Absolute pressure suction vessel (bara)

ΔP_f : Fitting and friction loss in suction line (bar)

H_s : Height between lowest drawdown line of suction vessel and centerline of pump suction (m)

H_{ac} : Acceleration head - reciprocating pumps (m)

v : Velocity at pump entry (m/s)

H_m : Extra available head above minimum – Design safety margin (m)

P_v : Vapour pressure of liquid at pump suction (bara)

g : Gravitational force (m/s^2)

γ : Liquid specific gravity (-)

Worked Example

Liquid propane (γ = 0.502), at its bubble point, is pumped from a pressure vessel (Operating Pressure 8.45 bara) by a centrifugal pump. The elevation of the liquid level in the suction vessel is 10 m above datum. The elevation of the pump suction nozzle is at 5 m above datum. The friction loss in the suction line is 0.1 bar.

What is the available NPSH ?

The $NPSH_{available}$ is calculated from:

$$NPSH_{available} = \frac{10.19 \cdot \left(P_s - P_v - \Delta P_f\right)}{\gamma} + H_s - H_m - \frac{v^2}{2g} - H_{ac}$$

The velocity term is assumed to be negligible

H_{ac} is only valid for reciprocating pumps (H_{ac} =0)

There is no allowance for extra available head (H_m = 0)

At bubble point, $P_s = P_v$

$$NPSH_{available} = \frac{10.19 \cdot \left(-\Delta P_f\right)}{\gamma} + H_s$$

$$NPSH_{available} = \frac{10.19 \cdot (-0.1)}{0.502} + (10 - 5) = 2.97\ m$$

8.6 Specific Speed

Specific speed is a function of the geometry (shape) of a pump impeller. Designers responsible for pump selection use specific speed information to :

- Select the shape of the pump curve – head/flow characteristic
- Determine the efficiency of the pump.
- Predict NPSH requirements.
- Select the lowest cost pump for their application.

Specific speed is defined as "the speed of an ideal pump geometrically similar to the actual pump, which when running at this speed will raise a unit of volume, in a unit of time through a unit of head".

The performance of a centrifugal pump is expressed in terms of pump speed, total head, and required flow. This information is available from the pump manufacturer's published curves.

Specific speed, N_s is determined from;

$$N_s = \frac{2.44 \cdot N \cdot Q^{0.5}}{H^{0.75}}$$

N_s pump speed (rpm)

q liquid flowrate (m^3/s)

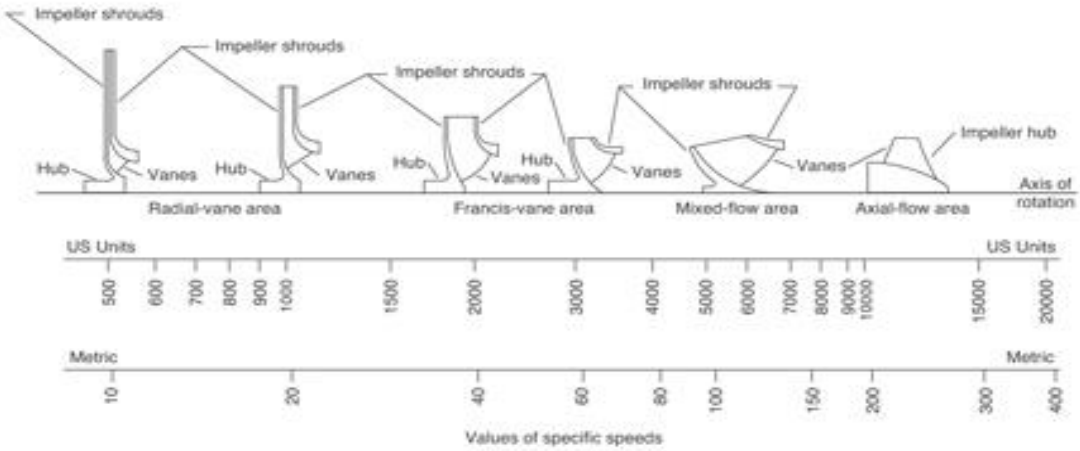
H pump head (m)

8.7 Impeller Types

Rotodynamic pump designs are generally described as any of three types: radial flow, mixed flow or axial flow. Radial flow impellers are designed so that the liquid exits perpendicular to the shaft centreline. They have lower specific speeds and most often are used for lower-flow, high-head applications.

As design flow increases, specific speed increases, and the impeller will become more axial in its configuration, with fluid flow in line with the shaft centreline. Fully axial impellers produce high flow rates with little head.

Between these two extremes, the liquid exit angle transitions from radial to axial. These transitional designs are referred to as mixed flow impellers.



8.8 Specific Speed

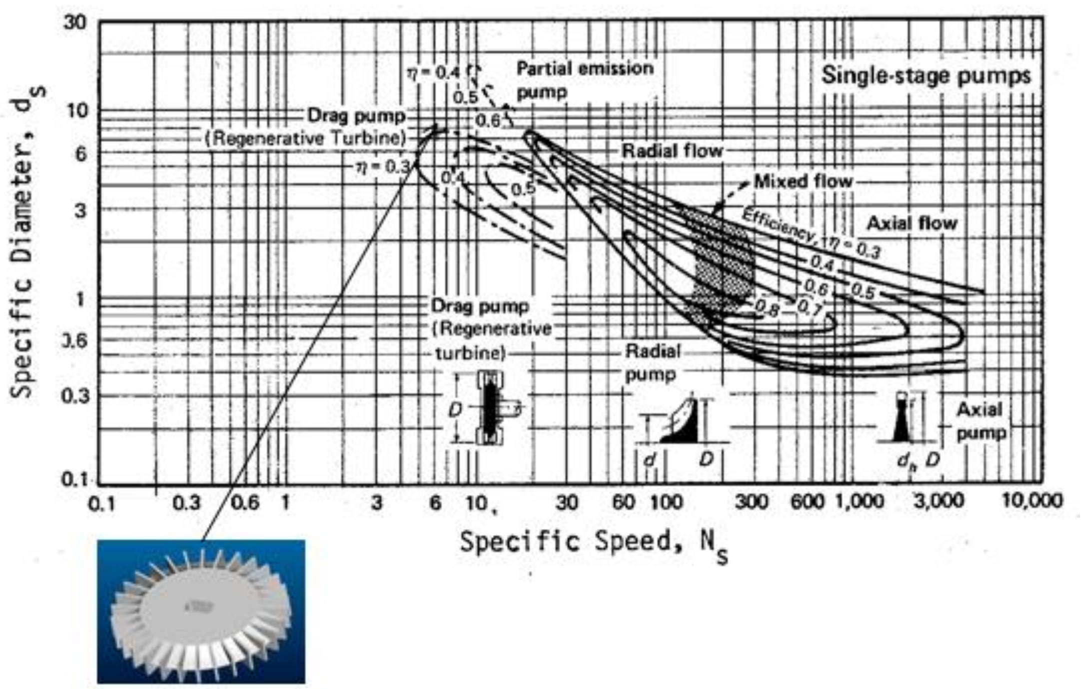
The specific diameter is a concept, which can be used with specific speed to make a general choice of pump type.

Specific diameter, d_s, is calculated from;

$$d_s = \frac{0.74 \cdot d \cdot H^{0.25}}{q^{0.5}}$$

d impeller diameter (m)

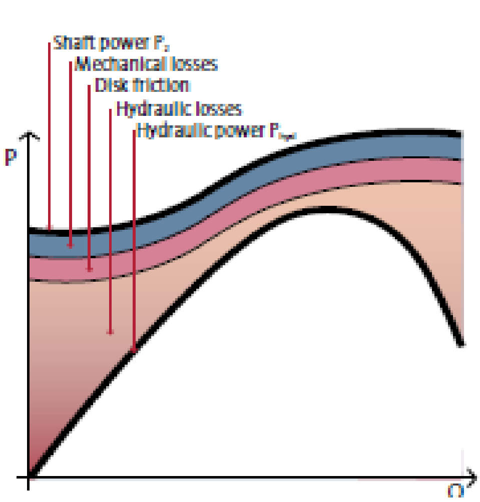
Specific speed and diameter are combined into the following chart.

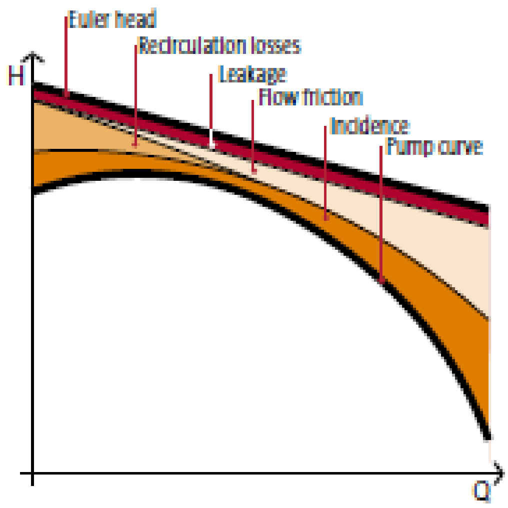


8.9 Centrifugal Pump Efficiency

Distinction is made between two primary types of losses: mechanical losses and hydraulic losses which can be divided into a number of subgroups. Table shows how the different types of loss affect flow (Q), head (H) and power consumption (P_2).

	Loss	Smaller flow (Q)	Lower head (H)	Higher power consumption (P_2)
Mechanical losses	Bearing			x
	Shaft seal			x
Hydraulic losses	Flow friction		x	
	Mixing		x	
	Recirculation		x	
	Incidence		x	
	Disk friction			x
	Leakage	x		





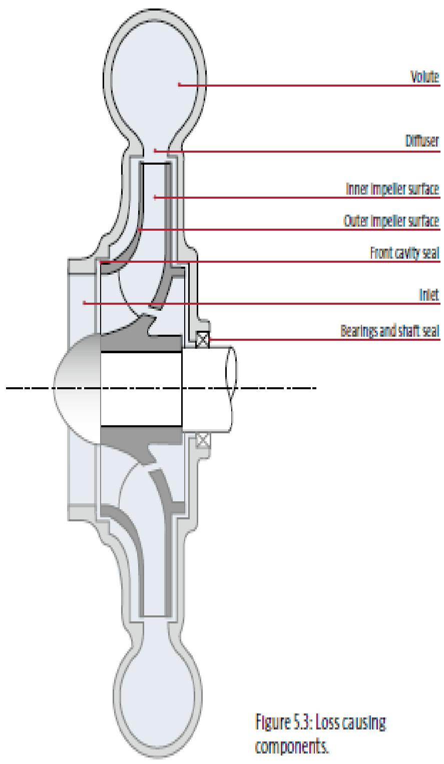


Figure 5.3: Loss causing components.

8.9.1 Mechanical losses

The pump coupling or drive consists of bearings, shaft seals, gear, depending on pump type. These components all cause mechanical friction loss.

Bearing and shaft seal losses - also called parasitic losses - are caused by friction. They are often modelled as a constant that is added to the power consumption. The size of the losses can, however, vary with pressure and rotational speed.

8.9.2 Flow Losses

Disk Friction

Disk friction is the increased power consumption which occurs on the shroud and hub of the impeller because it rotates in a fluid-filled pump casing. The fluid in the cavity between impeller and pump casing starts to rotate and creates a primary vortex. The rotation velocity equals the impeller's at the surface of the impeller, while it is zero at the surface of the pump casing. The average velocity of the primary vortex is therefore assumed to be equal to one half of the rotational velocity. The centrifugal force creates a secondary vortex movement because of the difference in rotation velocity between the fluid at the surfaces of the impeller and the fluid at the pump casing. The secondary vortex increases the disk friction because it transfers energy from the impeller surface to the surface of the pump casing. The size of the disk friction depends primarily on the speed, the impeller diameter as well as the dimensions of the pump housing in particular the distance between impeller and pump casing.

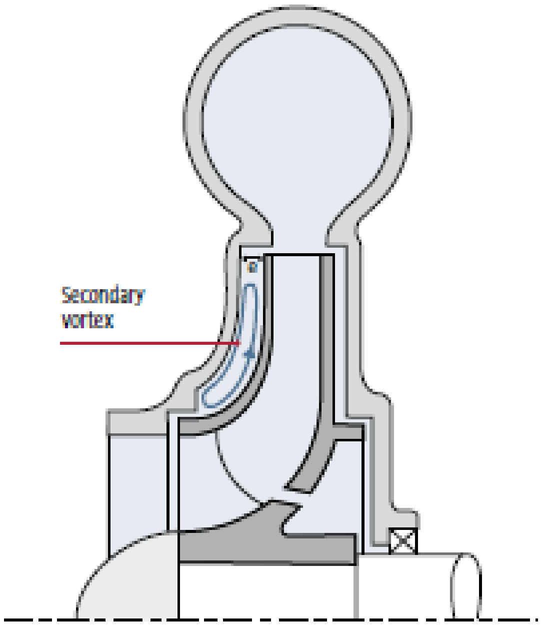


Figure 5.14: Disk friction on Impeller.

Furthermore, the impeller and pump housing surface roughness has a decisive importance for the size of the disk friction. The disk friction is also increased if there are rises or dents on the outer surface of the impeller.

Mixing Losses

Velocity energy is transformed to static pressure energy at cross-section expansions in the pump. The conversion is associated with a mixing loss. The reason is that velocity differences occur when the cross-section expands. The figure shows a diffuser with a sudden expansion because all fluid particles no longer move at the same speed, friction occurs between the molecules in the fluid which results in a discharge head loss.

Even though the velocity profile after the cross-section expansion gradually is evened out, a part of the velocity energy is turned into heat energy instead of static pressure energy. Mixing loss occurs at different places in the pump: At the outlet of the impeller where the fluid flows into the volute casing or return channel as well as in the diffuser.

When designing the hydraulic components, it is important to create small and smooth cross sections.

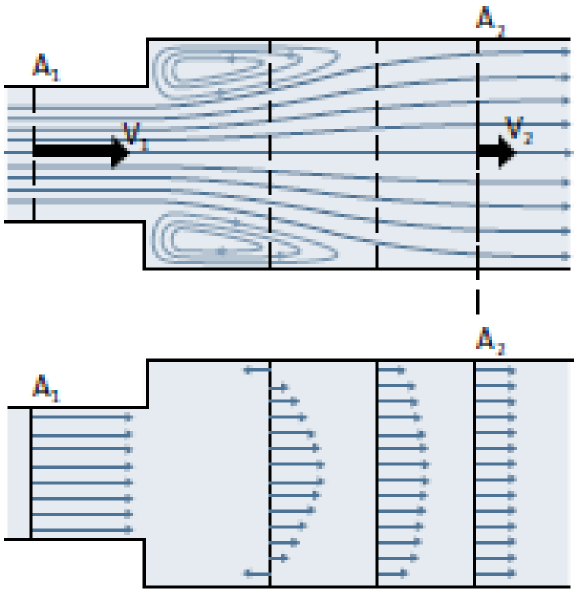
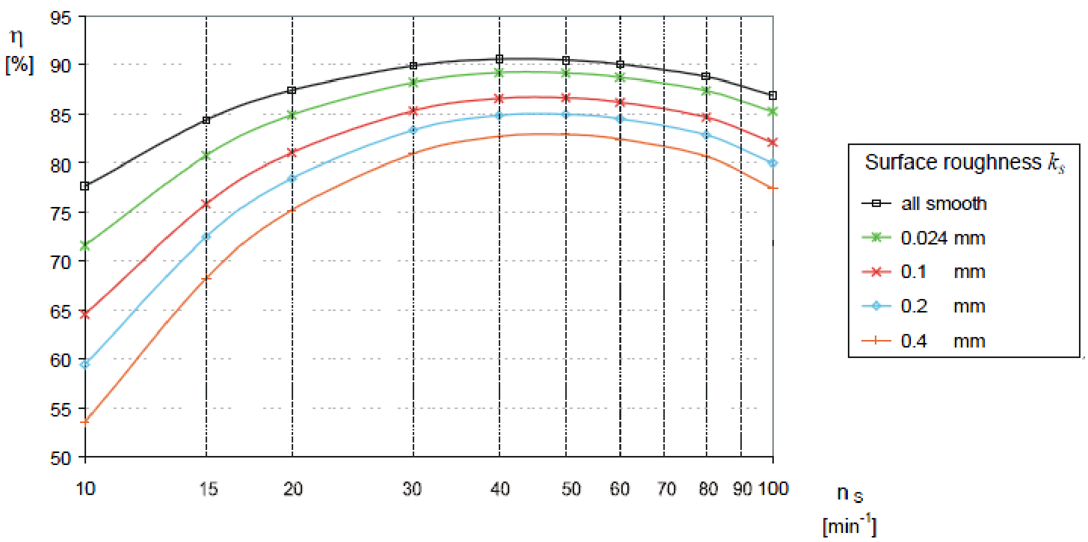


Figure 5.7: Mixing loss at cross-section expansion shown for a sudden expansion.

Surface Roughness

5.3 INFLUENCE OF DIFFERENT VALUES OF SURFACE ROUGHNESS



Increased roughness of the pump components will result in efficiency losses as indicated.

Leakage

Leakage loss occurs because of smaller circulation through gaps between the rotating and fixed parts of the pump. Leakage loss results in a loss in efficiency because the flow in the impeller is increased compared to the flow through the entire pump. Leakage occurs many different places in the pump and depends on the pump type. The pressure differences in the pump which drives the leakage .

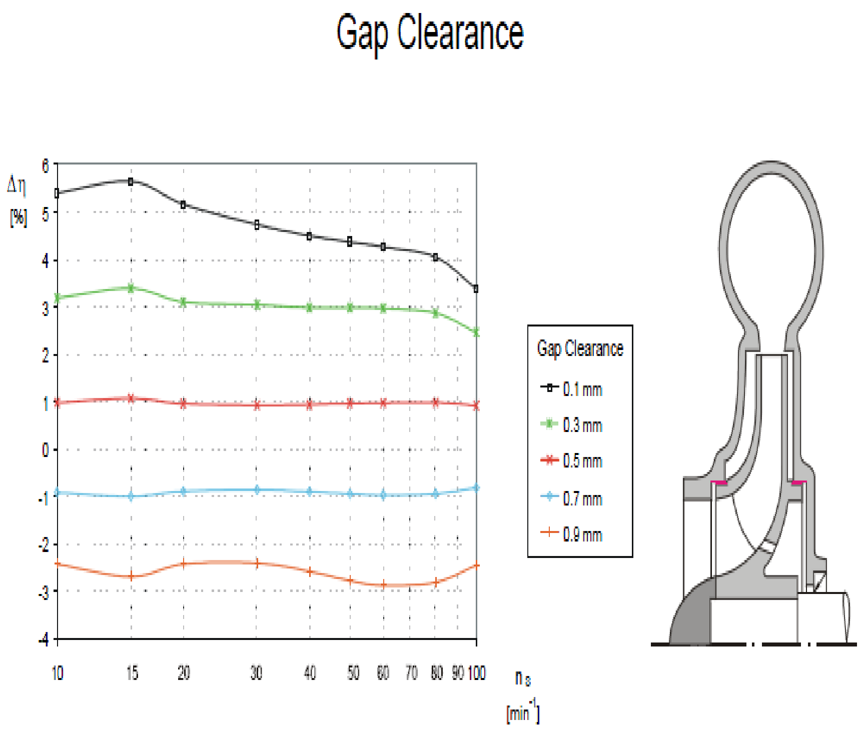


Fig. 5.6: The influence of secondary flow through the sealing gaps

The leakage between the impeller and the casing at impeller eye and through axial relief are typically of the same size.

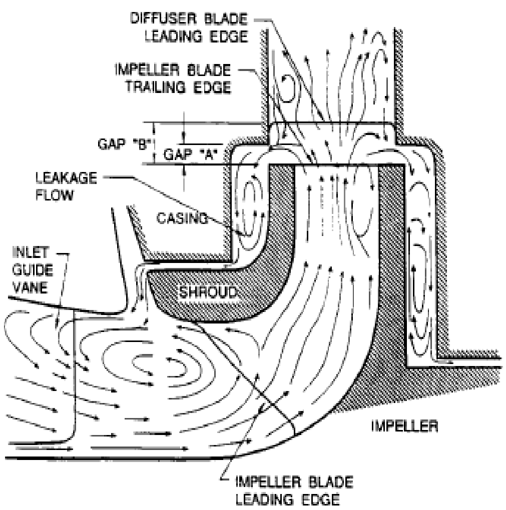
To minimise the leakage flow, it is important to make the gaps as small as possible. When the pressure difference across the gap is large, it is in particular important that the gaps are small.

Recirculation Losses

Recirculation zones in the hydraulic components typically occur at part load when the flow is below the design flow. The opposite figure shows an example of recirculation in the impeller. The recirculation zones reduce the effective cross-section area which the flow experiences.

High velocity gradients occurs in the flow between the main flow which has high velocity and the eddies which have a velocity close to zero. The result is a considerable mixing loss.

Recirculation zones can occur in inlet, impeller, return channel or volute casing.



The extent of the zones depends on geometry and operating point.

When designing hydraulic components, it is important to minimise the size of the recirculation zones in the primary operating points.

Incidence Losses

Incidence loss occurs when there is a difference between the flow angle and blade angle at the impeller or guide vane leading edges. This is typically the case at part load or when prerotation exists.

A recirculation zone occurs on one side of the blade when there is difference between the flow angle and the blade angle. The recirculation zone causes a flow contraction after the blade leading edge. The flow must once again decelerate after the contraction to fill the entire blade channel and mixing loss occurs.

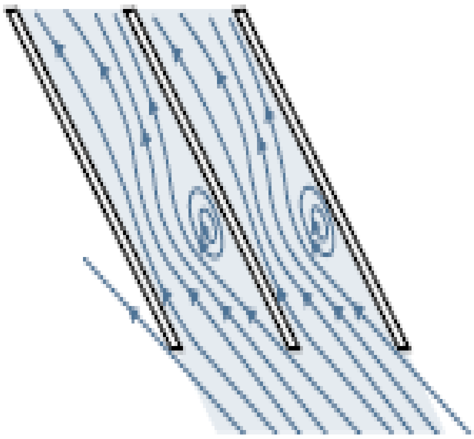
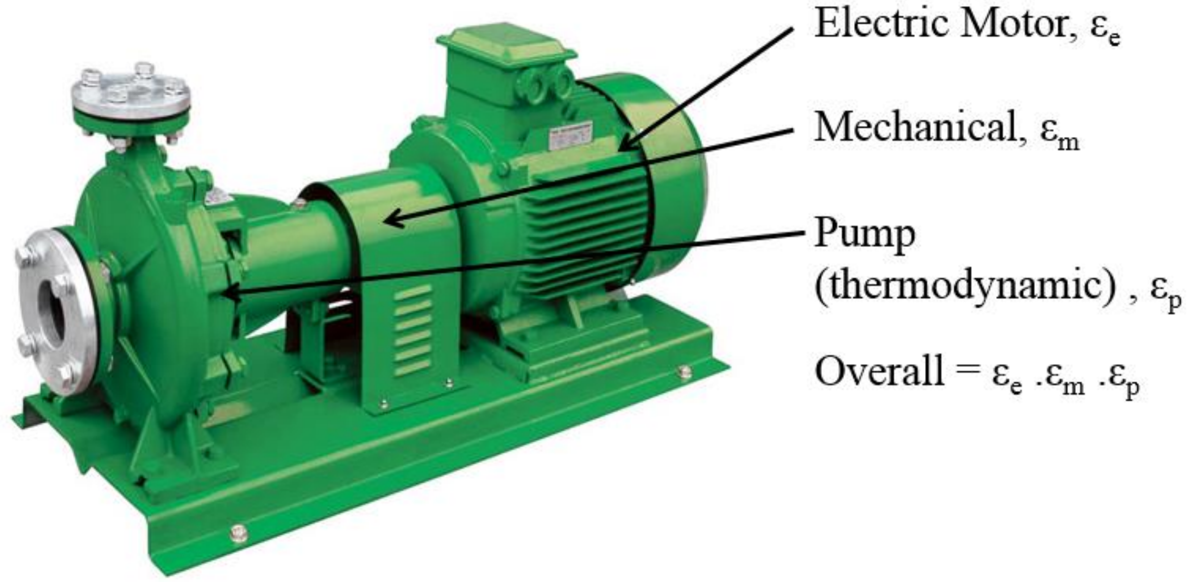


Figure 5.11: Incidence loss at inlet to impeller or guide vanes.

At off-design flow, incidence losses also occur at the volute tongue. The designer must therefore make sure that flow angles and blade angles match each other so the incidence loss is minimised. Rounding blade edges and volute casing tongue can reduce the incidence loss. The magnitude of the incidence loss depends on the difference between relative velocities before and after the blade leading edge.

8.10 Pump Overall Efficiency

In addition to Mechanical and Pump (Flow) losses the driver will also contribute to the overall efficiency.



Pump thermodynamic power is calculated as follows;

$$\int_{P_1}^{P_2} VdP = \Delta H = W_{theoretical} = W_{reversible}$$

For an incompressible flow this becomes;

$$W_{actual} = \frac{q \cdot (P_2 - P_1)}{E}$$

Where;

W : Power (kW)

q : Flowrate (m^3/s)

E : Overall thermodynamic efficiency (-)

P : Pressure (kPa)

8.11 Pump Configuration

Pumps should be designed to cater for peak and turndown rates. Dependent upon the application a range of pump configurations can be specified e.g. 1x100%, 3x50%, 4x33%, 2x67%. The configuration selected again depends on cost, uptime, safety and environment. As flows vary, the pump arrangement that gives best efficiency will alter.

Large pumps do not operate efficiently under turndown conditions hence a spill back (recycle) is often deployed. Hydraulic instability occurs at low flows, causing cavitation, surging, and excessive vibration in the pump. Operation of centrifugal pumps below their minimum flow requirements is one of the main cause of premature pump failure.
Note again that significant efficiency gains can be achieved by the application of variable speed drives. Variable speed drives are complex and expensive and this is a barrier to their use.

8.12 Impeller Replacement

If flow conditions change it may be beneficial to replace the impeller(s) to a design more suited to the new conditions. Most pump casings can accommodate a range of impeller sizes and an impeller change likely to be much more cost effective than a pump replacement.

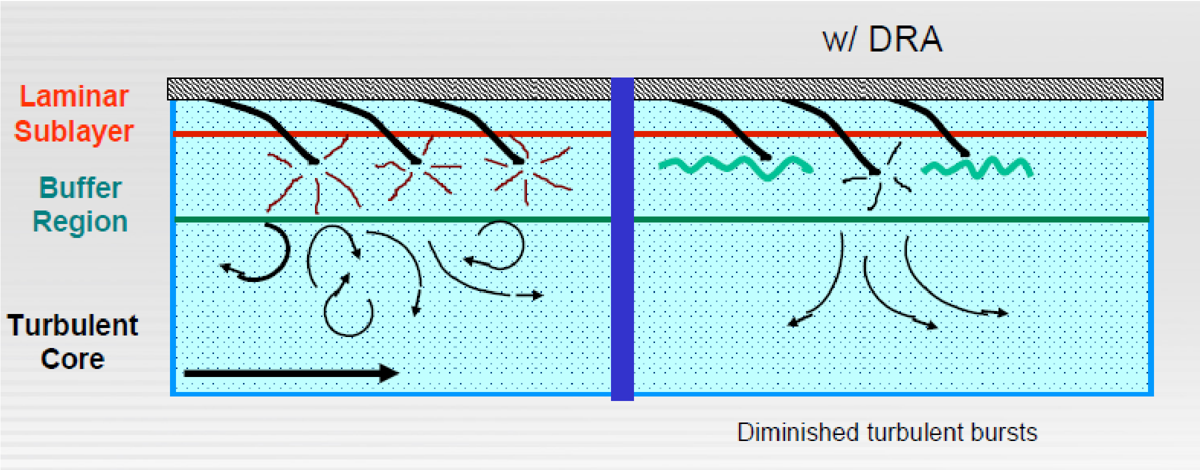
9 Pipeline Losses

Very significant energy losses are incurred in overcoming friction losses in pipelines. Many megawatts of lost work are encountered here. Hence means for reducing friction losses will reduce energy consumption. The most obvious way of reducing pressure drop is to install a larger diameter pipeline but this will clearly be more costly.

9.1 Drag Reduction Agents (DRA)

An option used for pressure loss reduction used in many pipelines is the use of Drag Reducing Agents.
Frictional pressure drop, or drag, is a result of the resistance encountered by flowing fluid coming into contact with a solid surface, such as a pipe wall. There are generally two types of flow; laminar and turbulent. The friction pressures observed in laminar flow cannot be changed unless the physical properties of the fluid are changed. DRAs do not change fluid properties and hence they are effective only in turbulent flow.
In a turbulent flow regime, the fluid molecules move in a random manner, causing much of the energy applied to them to be wasted as eddy currents and other indiscriminate motion. DRAs work by an interaction of the polymer molecules with the turbulence of the flowing fluid.
In order to understand how drag reducers decrease the turbulence, it is necessary to describe the structure of turbulent flow in a pipeline. The image below shows a typical turbulent flow in a pipeline that has three parts to the flow. In the very centre of the pipe is a turbulent core. It is the largest region and includes most of the fluid in the pipeline. This is the zone of the eddy currents and random motions of turbulent flow. Nearest to the pipeline wall is the

laminar sub layer. In this zone, the fluid moves laterally in sheets. Between the laminar layer and the turbulent core lies the buffer zone. This is illustrated in the following

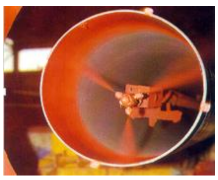


Drag Reduction occurs due to suppression of the energy dissipation by turbulent eddy currents near the pipe wall during turbulent flow.
There still is much to be learned about polymeric drag reduction, as there still is much to be learned about the complex phenomenon of turbulence. Recent research into this area tells us that the buffer zone is very important because this is where turbulence is formed first. A portion of the laminar sub layer, called a "streak", will occasionally move to the buffer region. There, the streak begins to vortex and oscillate, moving faster as it gets closer to the turbulent core. Finally, the streak becomes unstable and breaks up as it throws fluid into the core of the flow. This ejection of fluid into the turbulent core is called a turbulent burst. This bursting motion and growth of the bursts in the turbulence core results in wasted energy.
Drag reducing polymers interfere with the bursting process and reduce the turbulence in the core. The polymers absorb the energy in the streak, like a shock absorber, thereby reducing subsequent turbulent bursts. As such, drag reducing polymers are most active in the buffer zone.

9.2 Low Friction Coatings

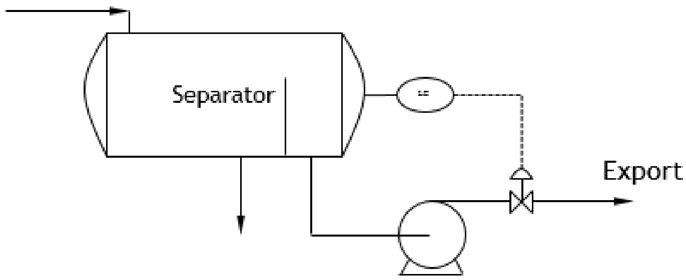
Friction losses are reduced by a smooth layer coating on the pipe. This can produce 10% savings to pump and compressor power requirements as illustrated.

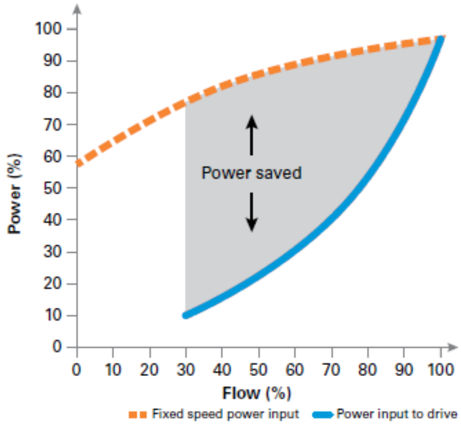
Reynolds Number	Friction factor normal pipe	Friction factor internally coated pipe	Percentage reduction in pressure drop
200,000	0.017	0.0156	8.2
300,000	0.0161	0.0144	10.6



10 Variable Speed Drives

As stated previously variable speed drives can deliver significant energy savings. For example for an oil export pump, if export flow rate declines, a fixed speed pump will use significantly more power than a variable speed as illustrated. However, a variable speed drive will cost more than a fixed speed.

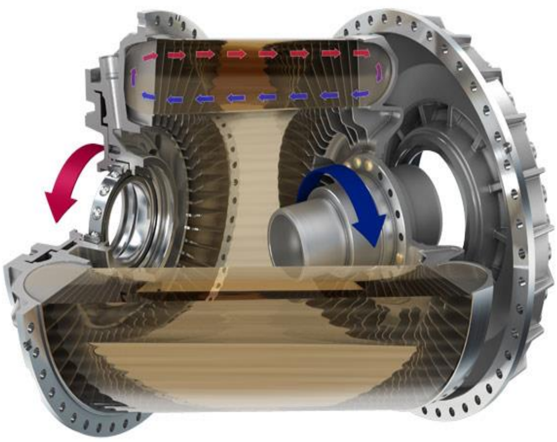




10.1 Fluid Coupling

Voith constant-fill fluid couplings (Turbo Couplings) and fill-controlled fluid couplings (Turbo Couplings) are the invention of Professor Hermann Föttinger (1877-1945).

The principle of hydrodynamic power transmission is based on the interaction of a pump and a turbine. In a Voith turbo coupling, this principle is realised by two bladed wheels. Together with a surrounding shell, these bladed wheels form the working chamber in which the operating fluid circulates.

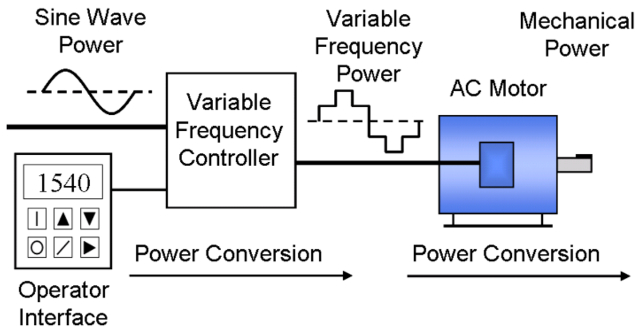


The speed of an equipment item driven by a fixed-speed electric motor can be varied with a fluid coupling. This is essentially a pump discharging to a power-recovery turbine, both in the same casing. The pump is connected to the driver shaft and the turbine to the driven shaft. The turbine speed is varied by varying the amount of fluid in the casing. Increasing the fluid increases the circulation between the pump and turbine, thereby increasing the speed of the turbine.

10.2 Variable Electrical Frequency

A **variable-frequency drive (VFD)** is a system for controlling the rotational speed of an alternating current (AC) electric motor by controlling the frequency of the electrical power supplied to the motor. A variable frequency drive is a specific type of adjustable-speed drive. The motor used in a VFD system is usually a three-phase induction motor . Some types of single-phase motors can be used, but three-phase motors are usually preferred. Various types of synchronous motors offer advantages in some situations, but induction motors are suitable for most purposes and are generally the most economical choice. Motors that are designed for fixed-speed operation are often used.

Variable frequency drive controllers are solid state electronic power conversion devices. The usual design first converts AC input power to DC intermediate power using a rectifier or converter bridge. The rectifier is usually a three-phase, full-wave-diode bridge. The DC intermediate power is then converted to quasi-sinusoidal AC power using an inverter switching circuit. The inverter circuit is probably the most important section of the VFD, changing DC energy into three channels of AC energy that can be used by an AC motor. These units provide improved power factor, less harmonic distortion, and low sensitivity to the incoming phase sequencing than older phase controlled converter VFD's.



A standard a-c motor operating at 50 hertz will operate at a constant speed, depending upon the number of magnetic poles it has in accordance with the formula:

RPM = 120 x frequency/number of poles

If the input frequency can be varied in accordance with the speed requirements, then a wide range of speeds can be obtained. For example, with a frequency range from 40 to 120 hertz, a 4-pole motor has a speed range from 1200 through 3600 rpm.

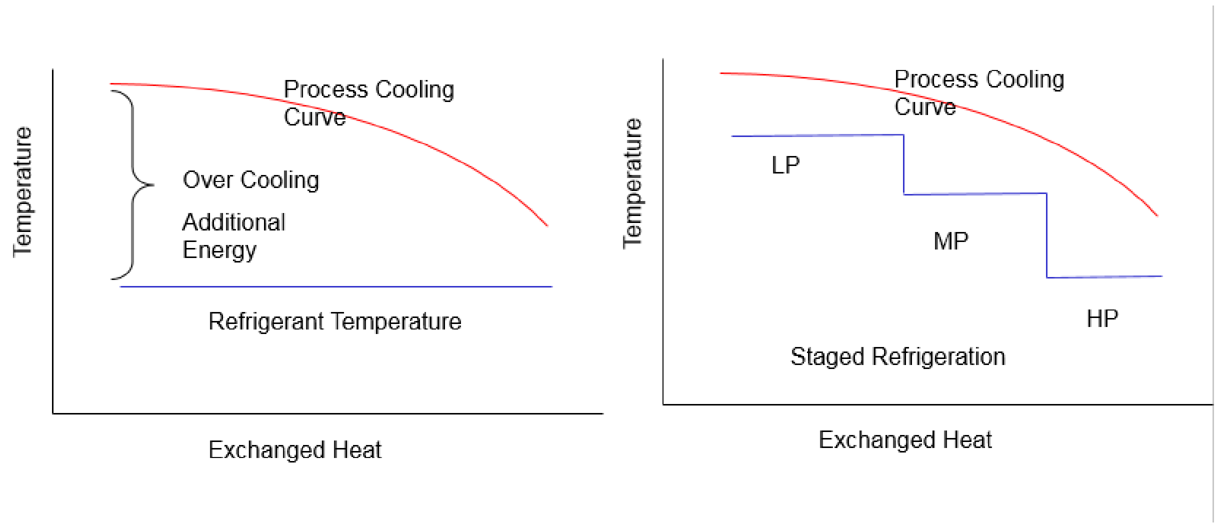
10.3 Turbines

Gas and power turbines (e.g. steam) can also be deployed to provide a variable speed driver. Gas turbines are discussed in detail later.

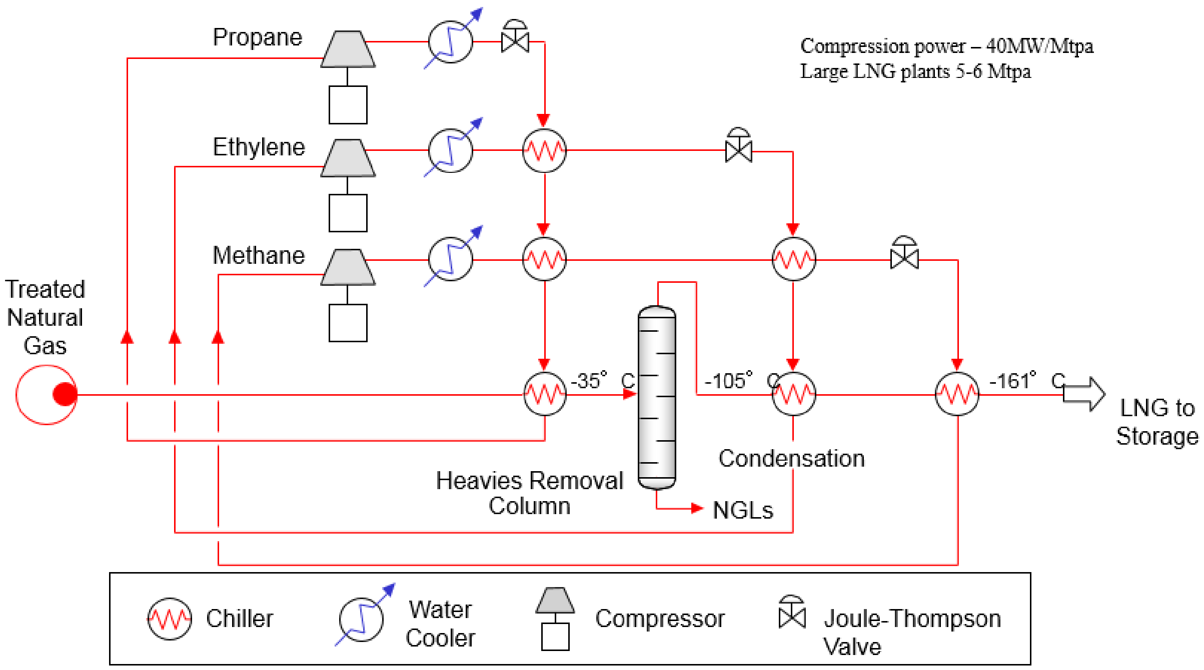
11 Refrigeration Systems

As presented in the Gas Treatment module refrigeration systems can be high energy consumers. Indeed for large LNG plants refrigeration compression power requirements can exceed 100 megawatts.

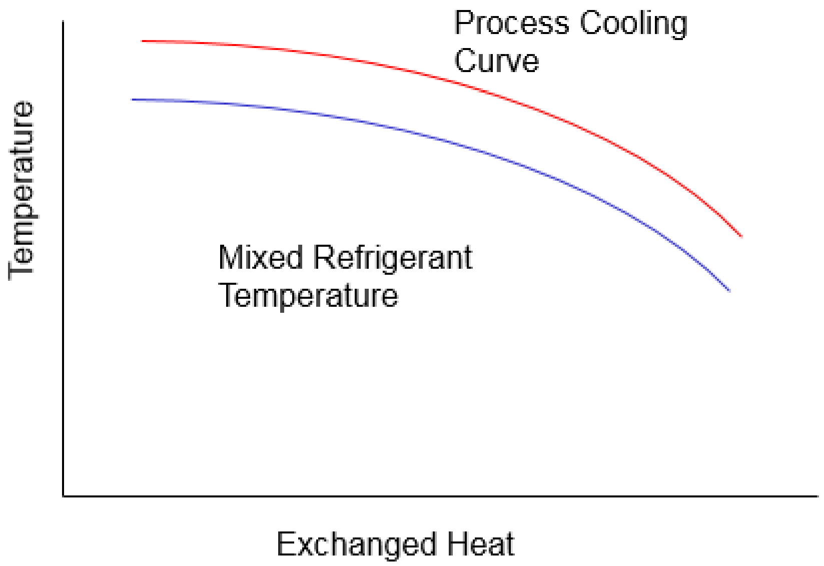
Staged refrigeration is a means of reducing power requirements by avoiding inefficient over cooling. This is illustrated as follows.



Staged refrigeration can be seen in many LNG plants using propane, ethylene and methane loops in cascade as shown.



An ideal refrigerant would show the following trend. Mixed refrigerants have been developed to provide this characteristic. The refrigerant consists of a blend of nitrogen and hydrocarbons from methane through pentane.



12 Oil Plant Energy Focus

An oil separation and key energy points are shown in the next page

Points to note are;

Control Valves and Chokes – a valve is designed to use pressure for control. Loss of pressure is a potential waste of energy.

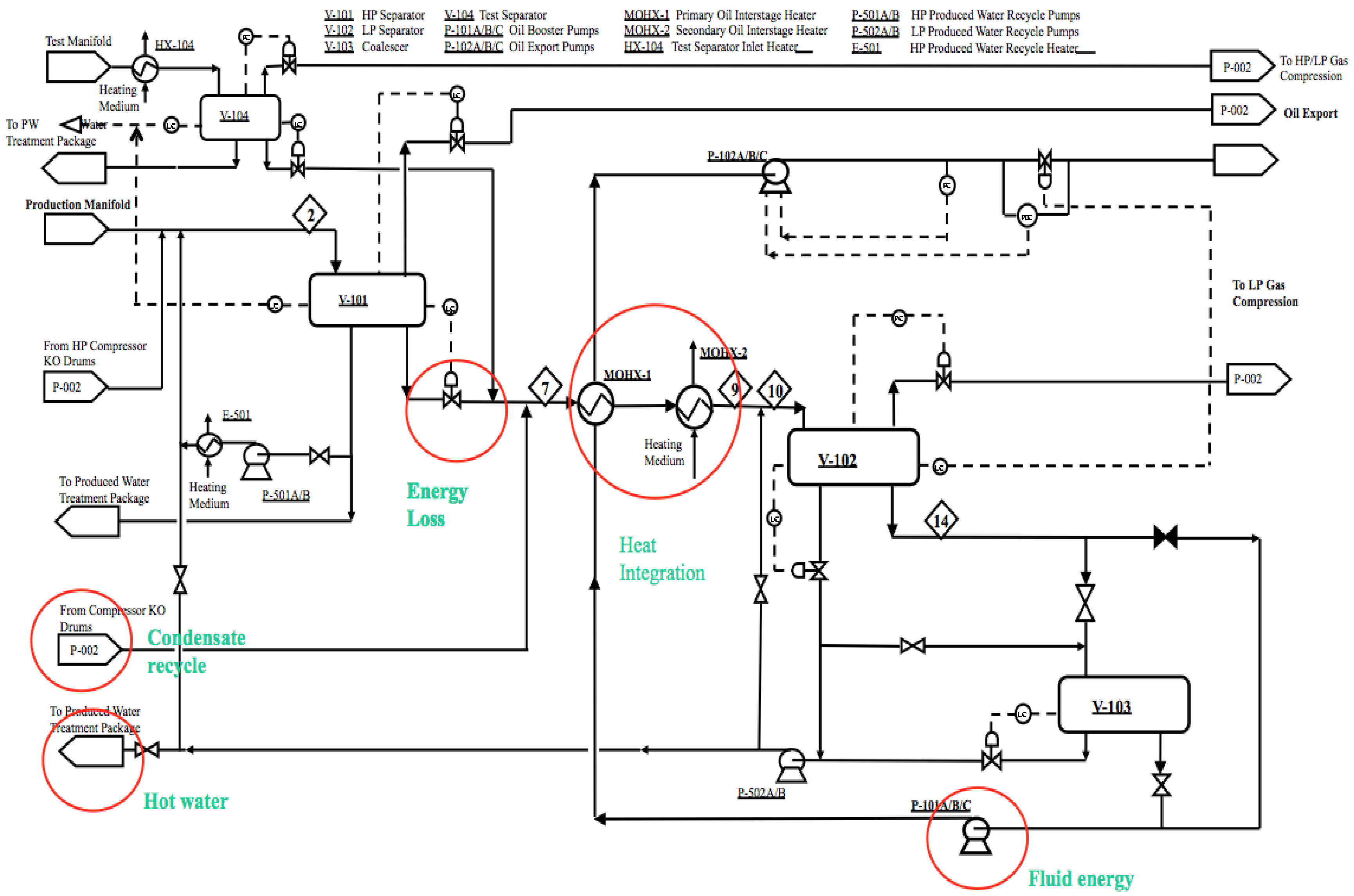
Pumps will most likely be required, hence pump efficiencies will be important.

If heat is required to aid emulsion treatment are there opportunities for heat integration.

If condensate is being recycled from the compression system this will reflash and could set up power intensive recycle loops.

The plant may be disposing of a considerable amount of hot water. This might be source of both pressure and thermal energy.

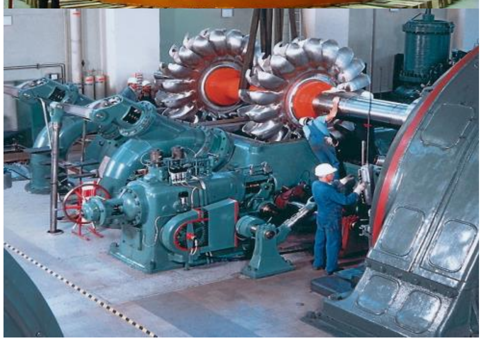
Power may be recovered from produced water using a recovery turbine.


V-101 HP Separator
V-102 LP Separator
V-103 Coalescer
V-104 Test Separator
P-101A/B/C Oil Booster Pumps
P-102A/B/C Oil Export Pumps
MOHX-1 Primary Oil Interstage Heater
MOHX-2 Secondary Oil Interstage Heater
HX-104 Test Separator Inlet Heater
P-501A/B HP Produced Water Recycle Pumps
P-502A/B LP Produced Water Recycle Pumps
E-501 HP Produced Water Recycle Heater
Test Manifold
HX-104
Heating Medium
To PW Treatment Package
Water
V-104
Production Manifold
V-101
From HP Compressor KO Drums
P-002
E-501
P-501A/B
To Produced Water Treatment Package
Energy Loss
MOHX-1
MOHX-2
Heat Integration
V-102
V-103
P-102A/B/C
P-101A/B/C
P-502A/B
From Compressor KO Drums
Condensate recycle
Hot water
Fluid energy
To HP/LP Gas Compression
Oil Export
To LP Gas Compression
2
7
9
10
14

13 Power Recovery Turbines

Energy recovery could take place where a pressure drop is experienced by using some form of recovery turbine. Two major types of centrifugal liquid power recovery turbines (HPRT) are used. For gases a recovery turbine has been discussed in Gas Treatment within the Turbo-Expansion process.

1. Reaction—Single or multistage Francis-type rotor with fixed or variable guide vanes.
2. Impulse—Pelton Wheel, usually specified for relatively high differential pressures.



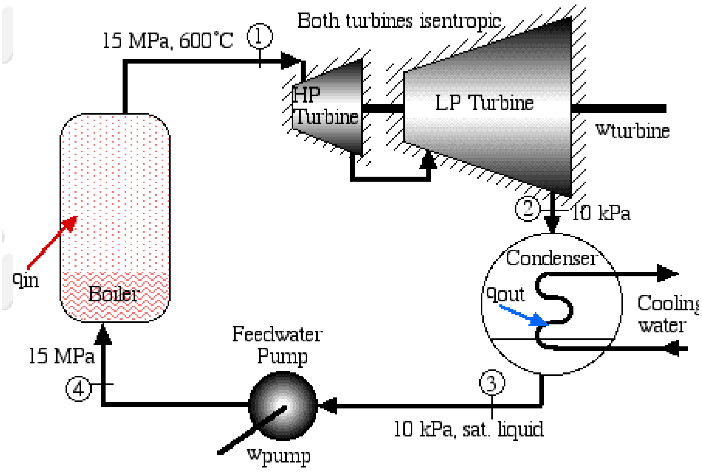
HPRTs with Francis-type rotors are similar to centrifugal pumps. In fact, a good centrifugal pump can be expected to operate with high efficiency as an HPRT when the direction of flow is reversed.

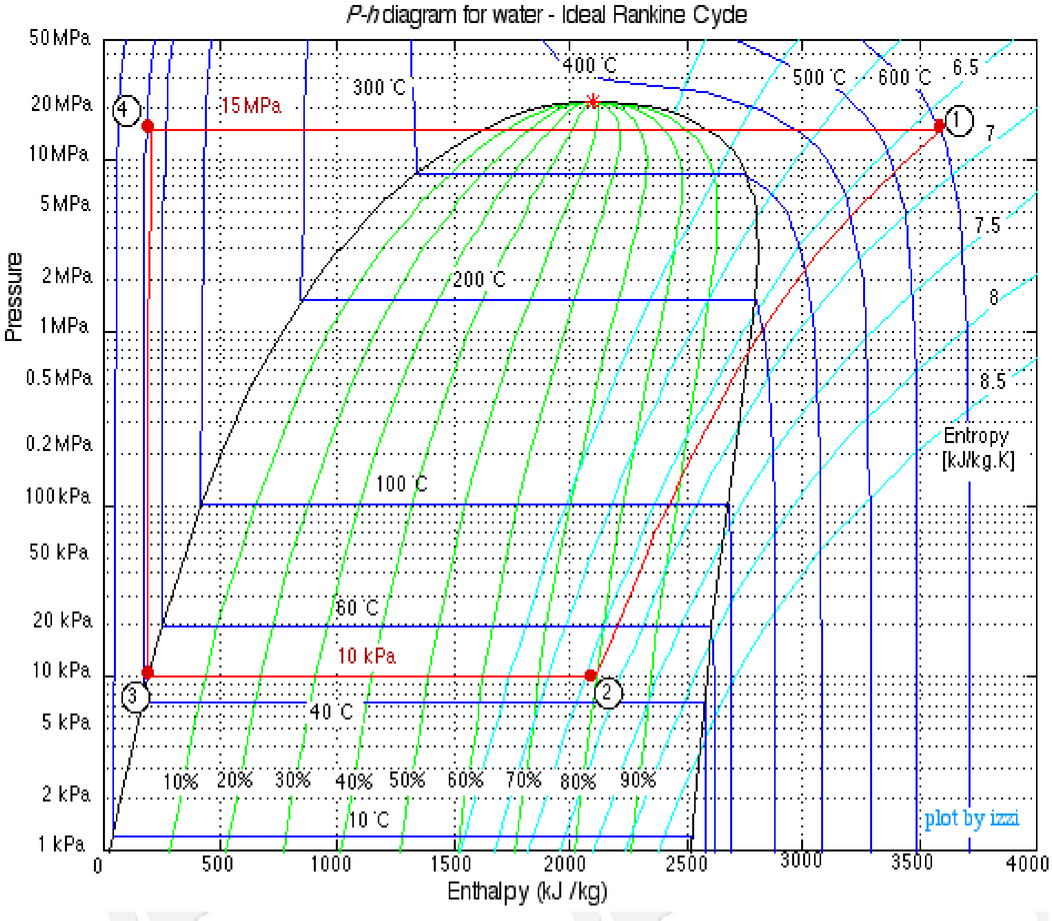
The Pelton Wheel or impulse runner type HPRT is used in high head applications. The impulse type turbine has a nozzle which directs the high pressure fluid against bowl-shaped buckets on the impulse wheel. This type of turbines' performance is dependent upon back pressure, while the reaction type is less dependent upon back pressure.

14 Energy Recovery from Low Grade Heat

14.1 Rankine Cycle

An option for recovering energy from low grade heat – such as produced water – is the application of the Rankine Cycle. Here heat is used to vapourise a liquid. The vapour is used to drive a turbine to extract work. The vapour is cooled to its liquid state then pumped to the vapouriser and the cycle continues. This shown in the following figure and in the associated Mollier chart.

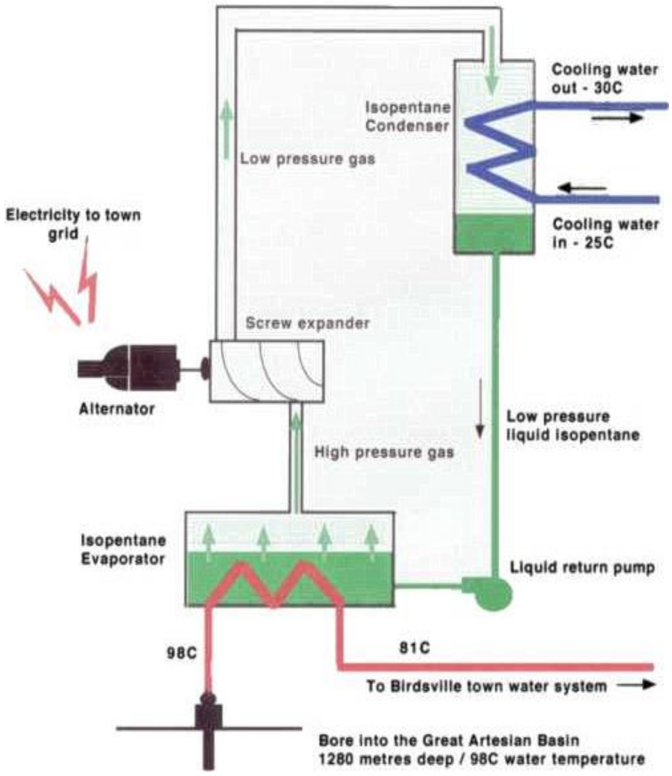




14.1.1 The Organic Rankine Cycle

The Organic Rankine Cycle (ORC) uses a heat source to heat an organic motive fluid. The motive fluid is evaporated and is then expanded through a turbine, which generates power. The motive fluid is then condensed in a heat exchanger using a cooling medium (cooling water, seawater, air) from where it is pumped back to the evaporator. The overall net ORC cycle efficiency comparing extracted power to available thermal energy typically ranges from around 3% (with 90 °C source water) to 7% (120 °C source water) of the available thermal energy. Turbine efficiencies are around 80% and the electrical generator efficiency 90%.

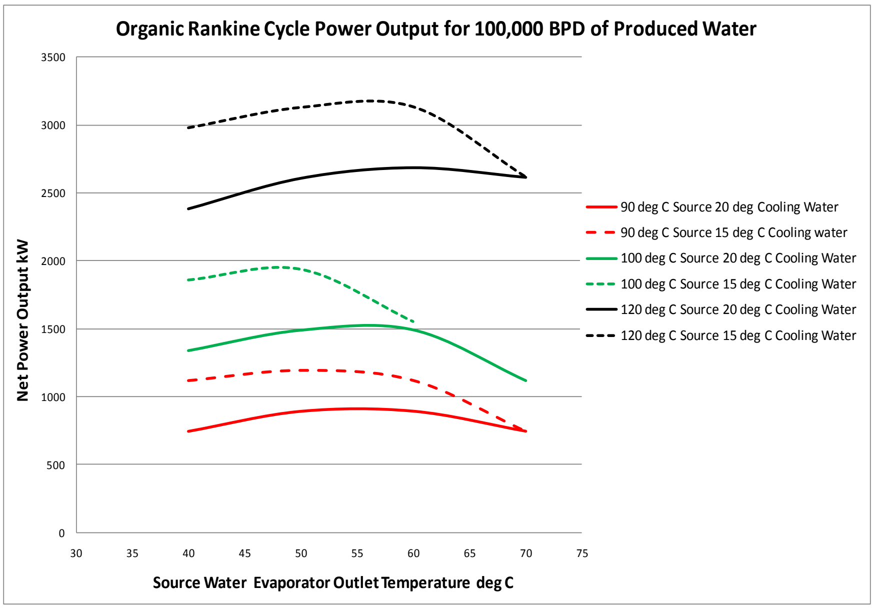
This system is analogous to the steam/water Rankine cycle used in power stations to generate power; the main difference being that the motive fluid has to boil at a lower temperature than water. A schematic of an ORC used for Geothermal Recovery is shown.



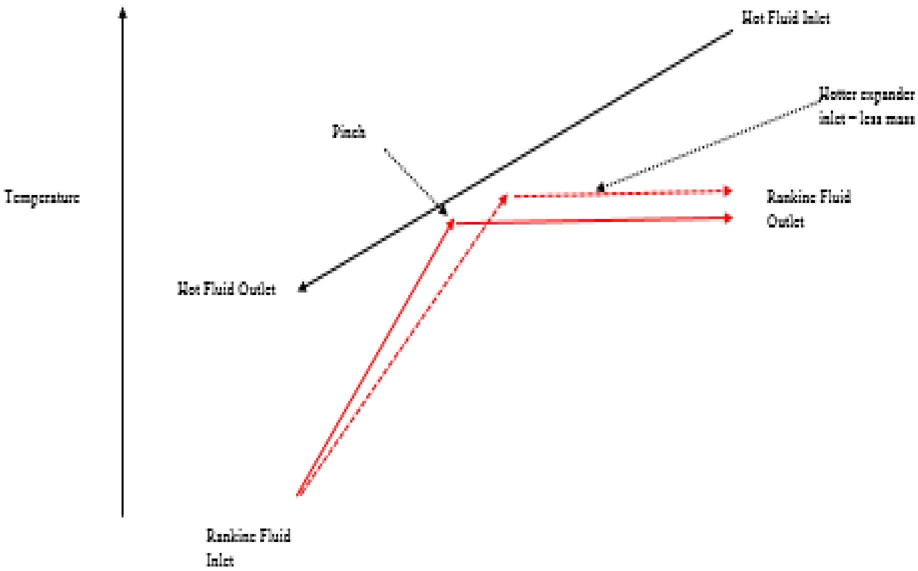
The power recovery in an ORC is dependent on a number of factors:

- Motive fluid type
- Temperature and flowrate of the source water
- Temperature and flowrate of the cooling water
- Outlet temperature of the source water from the evaporator

The influence of the above factors on power output is illustrated in the following figure is for energy recovery from hot produced water.



The Expander/Turbine inlet pressure and temperature requires to be optimised. The more heat that is extracted from the water the lower the water outlet temperature. This will vapourise more ORC fluid but it will be at lower outlet pressure and temperature. This is illustrated as follows.



Organic Rankine Fluids (ORC)

The motive fluid for an ORC must have the following properties:

A favourable vapour pressure curve – it must be able to condense out at 25^{0}C and evaporate at a temperature of 60 - 100^{0}C. The difference in vapour pressures at these temperatures must be sufficient to drive a turbine.

Ozone-friendly and non-detrimental to the environment.

The vapour pressure at 25^{0}C must not be so low that the system operates under extreme vacuum. This would not only lead to the risk of air ingress into the unit but also require large diameter pipes and equipment.

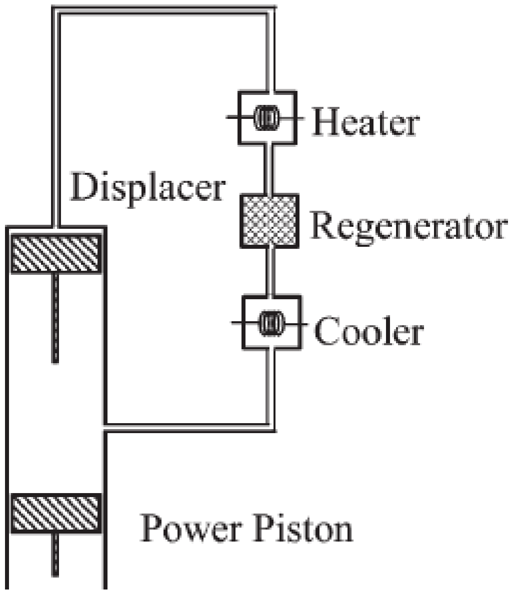
Flammability, volatility and toxicity are also important.

Commonly used ORC fluids are:

- Refrigerant R134a – 1,1,1,2 tetrafluoroethane
- Ammonia
- Butane
- Iso-pentane

14.2 Stirling Engine

Another means for recovering low grade heat which is receiving renewed interest is the Stirling Engine. A Stirling Cycle heat pump uses a heat source to expand a gas, usually helium, hydrogen or CO_2, without change of phase and then a cold sink to cool the gas, reducing its specific volume. The expansion and contraction of the gas is used to drive a piston backwards and forwards. In a similar manner to an internal combustion engine the piston movement can be used to generate power.



Thermodynamically, Stirling Cycles are more efficient compared to other types of heat engines as it based on the Carnot cycle rather than Rankine cycle to produce energy. The Carnot cycle is the most efficient of the thermodynamic cycles as there is no change in phase of the motive fluid. A typical power station, which uses a steam/water cycle, is based on the Rankine cycle. The Rankine cycle is thermodynamically less efficient as energy is lost through the cycle of vaporisation, condensing and circulation pumping of the motive fluid.

The main components of the Stirling engine are as illustrated.

Power piston : Performs the compression and the expansion of the working fluid.

Displacer piston: Shuttles the working fluid back and forth through the heater, regenerator, and cooler at constant volume. A displacer that moves to the cold space, displaces the working fluid from the cold space causing it to flow to the hot space and vice versa.

Regenerator: is a temporary heat store for the working fluid.

Hot side heat exchanger : Heat source of the cycle, continuously heated by the for example hot produced water.

Cold side heat exchanger: Heat sink of the cycle, continuously cooled by for example seawater.

Achieving a high efficiency using the Stirling cycle is difficult in practice because it involves reversible heat transfer in all the components, including the regenerator. For small temperature differentials, such as geothermal water, a large heat exchange area is required. Stirling engines have generally been restricted to very hot sources from combustion. Until recently Stirling cycles have been mainly of only theoretical interest.

WhisperGen a New Zealand company, has developed an "AC Micro Combined Heat and Power" Stirling cycle engine. These micro CHP units are gas fired boilers which generate power and hot water. These are now available in the UK through the major power suppliers and can be fitted as an alternative to domestic boilers. They claim efficiencies in excess of standard condensing boilers at around 95%.

Although not directly applicable to energy recovery from hot water, in the future these small scale Stirling Engines could be considered as an efficient generator to provide life support heating and power for control on for example on unmanned gas platforms. Emissions would be lower than from diesel generators or gas engines.

Deluge Inc a United States Company have developed a 16 cylinder CO_2 filled 250kW Stirling engine which operates on 4200 BPD of hot water at 80 °C. Cooling water is required at 26 °C at around 12000 BPD.

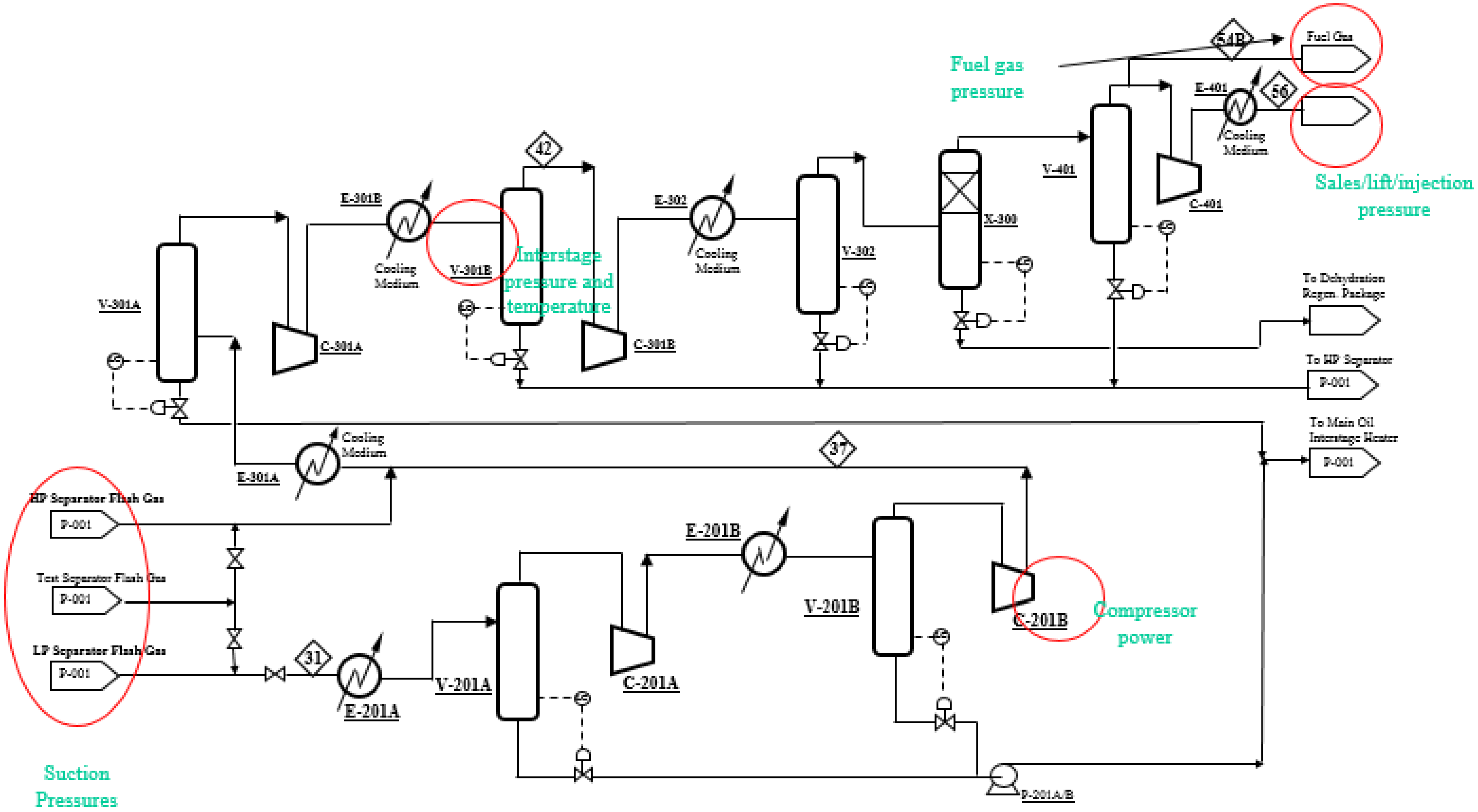
The temperature and flowrates are in an ideal range for produced water from oil and gas wells and seawater cooling source. Deluge Inc. claims an efficiency of up to 23%.

The Stirling machine was tested, funded by US government grants, at an onshore oil and gas test centre in Nevada 2005. The machine was used to pump oil wells using hot geothermal water. A commercial unit was subsequently supplied to Hawaii for installation in 2009.

A key issue is the complexity, size and weight of the Stirling Engine. A photograph of the unit is shown . The machine has a footprint of $73m^2$ for only 250kW of power and is likely to be heavy. The company say they hope to reduce the foot print to fit onto a standard lorry trailer in the future.

15 Compressors

The key energy issues associated with compression are shown in the following PFD.


Fuel gas pressure
Fuel Gas
54B
56
E-401
Cooling Medium
C-401
Sales/lift/injection pressure
V-401
X-300
42
E-301B
Cooling Medium
V-301B
Interstage pressure and temperature
C-301B
E-302
Cooling Medium
V-302
V-301A
C-301A
To Dehydration Regen. Package
To HP Separator
P-001
To Main Oil Interstage Heater
P-001
Cooling Medium
E-301A
37
HP Separator Flash Gas
P-001
Test Separator Flash Gas
P-001
LP Separator Flash Gas
P-001
31
E-201A
V-201A
C-201A
E-201B
V-201B
C-201B
Compressor power
P-201A/B
Suction Pressures

15.1 Compressor Power

Recap compressor thermodynamics;

$$\Delta h_{poly} = \frac{T_1 \cdot Z_a \cdot R}{\left(\frac{n-1}{n}\right) \cdot MW} \cdot \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$T_2 = T_1 \cdot \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$W = \frac{\phi_m \cdot \Delta h_{poly}}{E_{poly}}$$

$$\frac{n-1}{n} = \frac{k-1}{k \cdot E_{poly}}$$

Where;

Δh_{poly} Polytropic head (kJ/kg)

T Temperature (K)

Z_a Average compressibility

R Gas constant (8.314 kJ/kmol.K)

P Pressure (Pa)

n Polytropic coefficient

k Ratio of specific heats (C_p/C_v)

MW Molweight (kg/kmol)

W Gas power (kW)

ϕ_m Mass flow (kg/s)

E_{poly} Polytropic efficiency - Function of volumetric inlet flow varying from approx. 0.6 to 0.8 (compressor specific)

γ Gas relative density (-)

$_{1/2}$ Suction/Discharge

The thermodynamics indicate power is a function of suction temperature, efficiency, compression ratio and gas properties.

15.2 Compressor Efficiency

15.2.1 Centrifugal

Centrifugal compressor efficiency is influenced by the same features as discussed in the previous pump section. Note that modern machinery design can deliver high machine efficiencies, efficiency though will come at a cost. Many Process Engineers will assume a 75% polytropic efficiency for system design. This value should be questioned to ensure energy management is optimised.

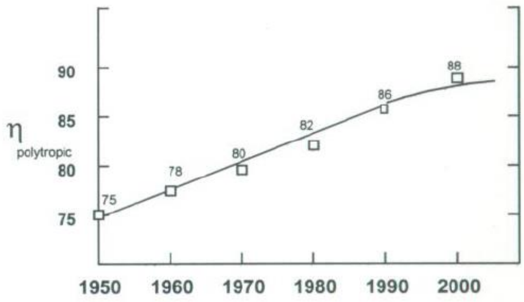


Figure 1. Flange to Flange Polytropic efficiency for large multistage centrifugal compressors.

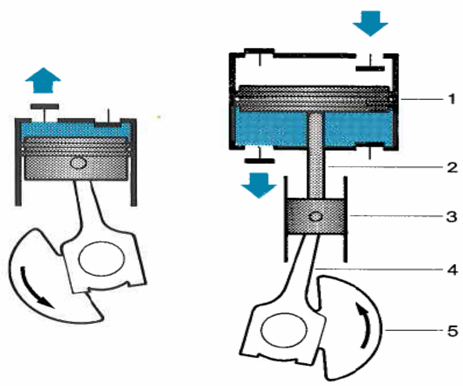
15.2.2 Reciprocating Compressor

Effects of clearance

When the piston has reached the end of the compression stroke, some gas is retained in the clearance space in the cylinder after the discharge valves have closed. When the compression cycle restarts, there is already a volume of gas in the cylinder which re-expands and reduces the available volume for suction gas. Obviously, as the clearance volume increases, the volumetric efficiency reduces.

Wiredrawing

Wiredrawing is defined as a restriction of area for a flowing fluid, causing a loss in pressure by friction without loss of heat or performance of work. As gas passes through the compressor suction valves a mild throttling takes place which means that the cylinder is at a slightly lower pressure than the suction line and thus less volume of suction gas is transferred than if the gas was at suction line pressure. This reduces the volumetric efficiency of the compressor as the actual volume of suction vapour transferred is reduced. Wiredrawing is independent of compression ratio being a function of gas velocity through the valves and passages of the compressor. As velocity increase the effects of wiredrawing increase. The gas velocity is dependent upon valve characteristics, the gas and the speed of the compressor. As the compressor rpm increases so does the piston



1- piston
2- piston rod
3- cross-head
4- connecting rod
5. crankshaft with counter-weight

displacement and velocity of the gas passing through the valves. This in turn amplifies the effects of wiredrawing.

Cylinder heating

The compression process generates heat as work is transferred into the gas from the piston. Some of this heat is retained in the cylinder walls. Heat is conducted from the cylinder walls to the gas entering from the suction line. Heat is also generated from friction by the turbulent movement of gas in the cylinder. The net effect of this is that suction gas is heated and expands in the cylinder. This reduces the amount of gas which can be transferred from the suction line. Cylinder heating increases as the compression ratio increases. This reduction in actual volume of gas in the cylinder reduces the volumetric efficiency of the compressor

Piston and valve leakage

Leakage back through suction or discharge valves or around the piston will decrease the gas transferred by the compressor. Back flow through valves is minimised by designing them to close promptly by spring assistance. However, spring tension increases wire drawing so the spring rating is critical. Back leakage is a function of compression ratio and compressor speed.

15.3 Interstage Temperatures and Pressures

Interstage conditions are set at the development of the heat and mass balances. If there is a large amount of condensate being recycled interstage conditions may require careful evaluation to ensure an optimal setting is achieved which maximises value. In this instance an optimiser could be deployed with the objective function of minimising compressor power. The adjusted variable being interstage conditions.

15.4 Pressure |Losses

Significant pressure drop can occur through pipes, valves and exchangers. At the design stage these are set to minimise CAPEX. However, use of larger pipe, low pressure drop heat exchangers and meters can deliver energy savings. In operation unnecessary losses across control valves and manual valves can contribute to poor energy management.

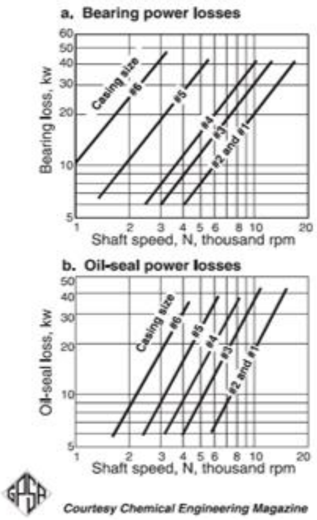
15.5 Compressor Control System

Standard recycle control for Surge management can be energy wasteful. Alternative schemes as presented in the Compressor module should be considered e.g. speed control. Again interstage pressures should be reviewed to establish the compressor efficiencies

15.6 Mechanical Losses - Seals and Bearings

Typical power losses are shown.

Casing Size	Max Flow (inlet m^3/h)	Nominal Speed (rpm)
1	12 700	10 500
2	33 900	8 200
3	56 000	6 400
4	93 400	4 900
5	195 000	3 600
6	255 000	2 800



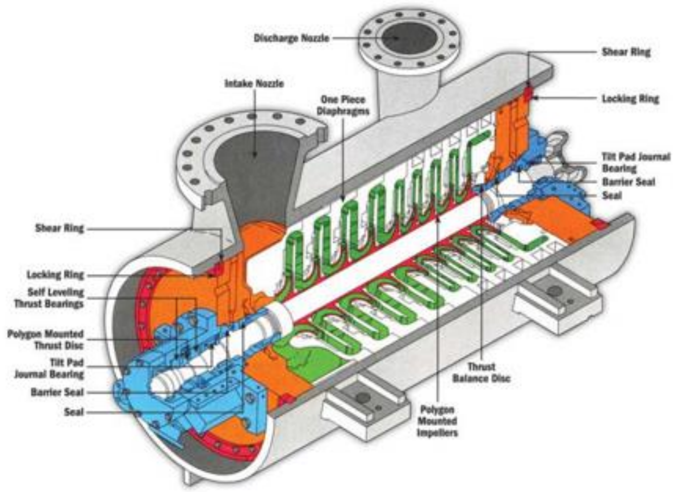


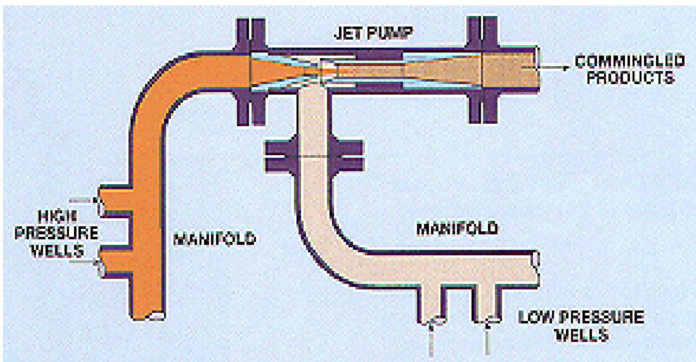
Figure 1. Major Components of Multistage Barrel-type Centrifugal Compressors (Dresser-Rand Co., Olean, NY)

15.6.1 Magnetic Bearings

By lessening parasitic loads, active-magnetic-bearing system affords the compressor a very low energy-consumption profile. With traditional mechanically lubricated bearings, oil viscosity creates frictional losses during initial start-up and also during normal operation. These losses can result in a substantial power penalty. On a magnetic bearing total system energy consumption can be reduced by 1% to 3% from the friction horsepower alone. Since the compressor rotor is levitated by electromagnetic force, the conventional bearing lube-oil system is eliminated along with the attendant oil reservoir, pumps, coolers, check valves and associated auxiliary lubrication components that draw power.

15.7 Eductors

If a high pressure gas source is available this can be used to compress gas from a low pressure. Such an application is seen in gas platforms where high pressure wells are used to enhance production from low pressure regions. The venture principle is again utilised here.

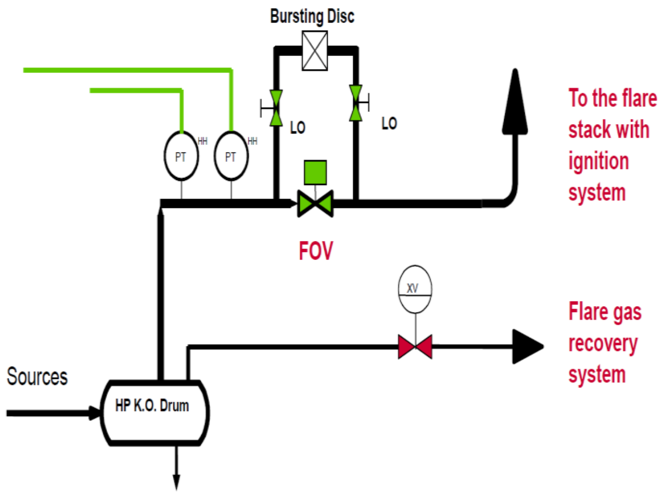


16 Flaring

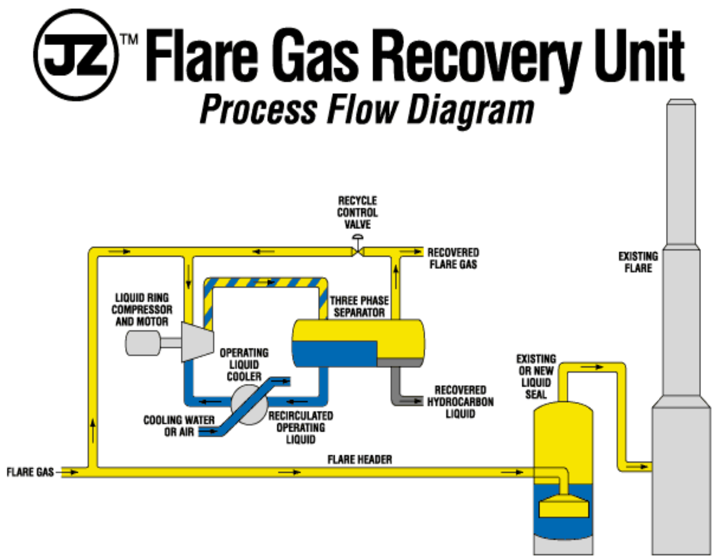
Continuous flaring is a clear contributor to CO_2 atmospheric emissions.

16.1 Flare Gas Recovery

Continuous flaring can be reduced to zero by installing a device to isolate the flare stack from the flare gas collection system. Normally a valve with a bypass bursting disc. A means of returning the gas normally flared back to the process is required. Under upset conditions, when large flows are sent to the flare, the rising gas flow is detected and the flare isolation valve is opened allowing gas to be safely discharged via the flare tip and burnt. This requires a system to reliably open the flare stack valve and an automatic ignition system. The automatic ignition system is an air driven ballistic pellet which ignites at a striker plate adjacent to the flare tip. Purging the flare stack with nitrogen to prevent air ingress and flame back during normal operation is also required.



An alternative to the flare isolation valve is a water seal. A flare system consists of a vapour header that collects the flare gases from various sources, a knockout vessel, a liquid seal vessel, and the flare itself. The flare gas recovery unit connection is typically located between the knockout vessel and the liquid seal. The primary control variable of the John Zink flare gas recovery unit is flare system pressure.



As the flare header pressure reaches the predetermined pressure control set point, a liquid ring compressor starts up and begins to compress the flare gas. The operating liquid is cooled in a shell-and-tube heat exchanger, evaporative cooler or air-cooled heat exchanger to control compressor discharge temperature.

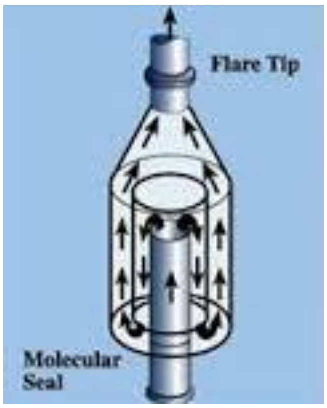
The compressor discharges the gas into a three-phase separator that separates the operating liquid from the flare gas and then the condensed hydrocarbons from the operating liquid. Instead of venting process vent streams into the flare system, the compressed gases are made available to the operating plant's fuel gas supply or possibly as a process feedstock. Integration and control of a flare gas recovery unit is of critical importance. For example, care must be exercised in the design of the recovery system to prevent application of a vacuum to the vapour header that might draw in air and create a flammable mixture in either the flare header or the fuel gas system.

When all compressors are operating at full capacity and if the process vent flow rate continues to increase, flare gas will begin to pass through the liquid seal and flow to the flare stack. Therefore, the safety function of the flare system is maintained in the event of process upset conditions.

16.2 Purge Gas

In normal operation a flare drum operates at a low pressure close to atmospheric pressure. This pressure is the built-up back pressure due to the continuous flared or purged gas flow in the flare stack. Pressure at the stack tip is atmospheric and back-pressure in the knock out drum is atmospheric pressure plus the frictional pressure drop from continuous venting/purging of the gas. If the flow of gas to flare stack stops or is very low there is a possibility of air ingress into the flare stack and into the flare KO drum and piping network. This can result in an explosive mixture of air and hydrocarbons in the vent/flare network, which can be catastrophic. To

prevent this a continuous flow of purge gas is used to prevent ingress – purge gas is often process gas, hence there is continual combustion of hydrocarbon gases.



A Molecular Seal can significantly decrease the amount of purge gas required for safe operation. The Molecular Seal design is based on the varying molecular weight between the air and the flared gas. Minimum purge-gas flow rates are allowed to pass through the Molecular Seal.

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16.3 Pilot Gas

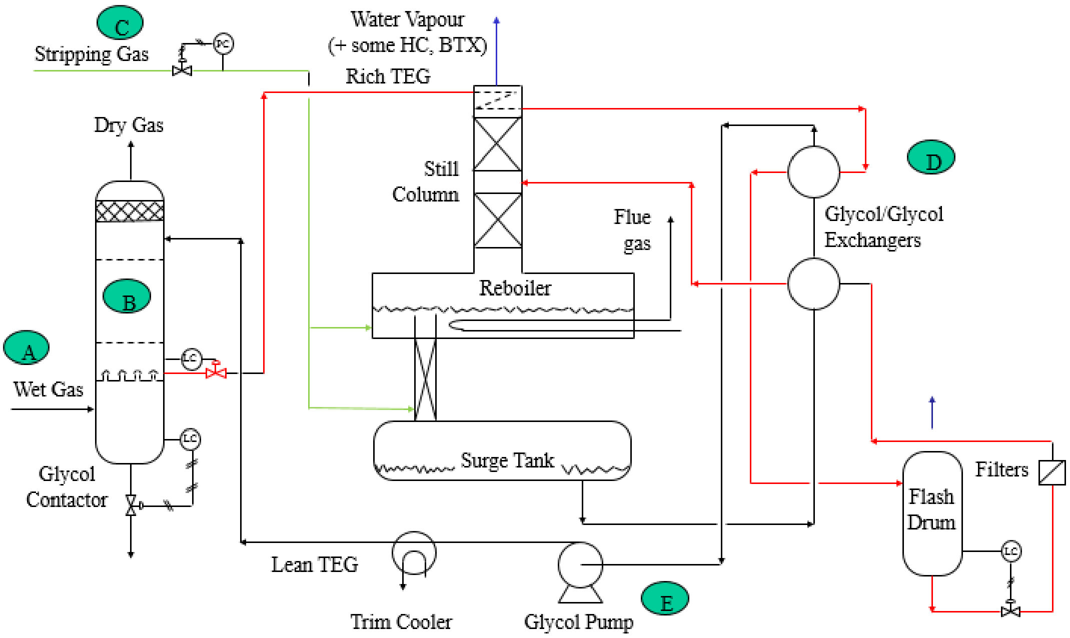
For most flare systems a continuous flow of process gas supplies the ignition system at the flare tip. This becomes a continuous combustion source.

17 Gas Line Pack

Many operators will 'pack' a gas pipeline. This involves taking the gas pipeline to pressures higher than needed for delivery. The reason for this is to provide a storage of gas should the plants feeding the pipeline shutdown. The packed pipeline provides a means of continuing to supply gas to the receiving site whilst the shutdown plants are restarted. This might be commercially attractive from a contractual viewpoint but line packing is energy wasteful.

18 TEG Dehydration

Key energy considerations are illustrated in the accompanying PFD.



Notable features are;

A – The colder the gas the lower the water content hence lower reboiler duty

B – Pressure drop across bed – larger pressure drop will require increased compression power

C – Stripping gas sent to flare – if the gas is a hydrocarbon gas this is energy wasteful

D – Heat exchanger temperature approach – the more heat recovery the lower the reboiler duty but at a cost of larger exchangers

E – Pump power – use HP rich glycol to power pump via a turbine

19 Power Generation

In addition to the items of equipment and unit operations being energy efficient, the efficiency of the energy production plant is equally important. Energy efficiency here will clearly reduce CO_2 emissions but also associated emissions such as NOx, SOx, CO will also be reduced.

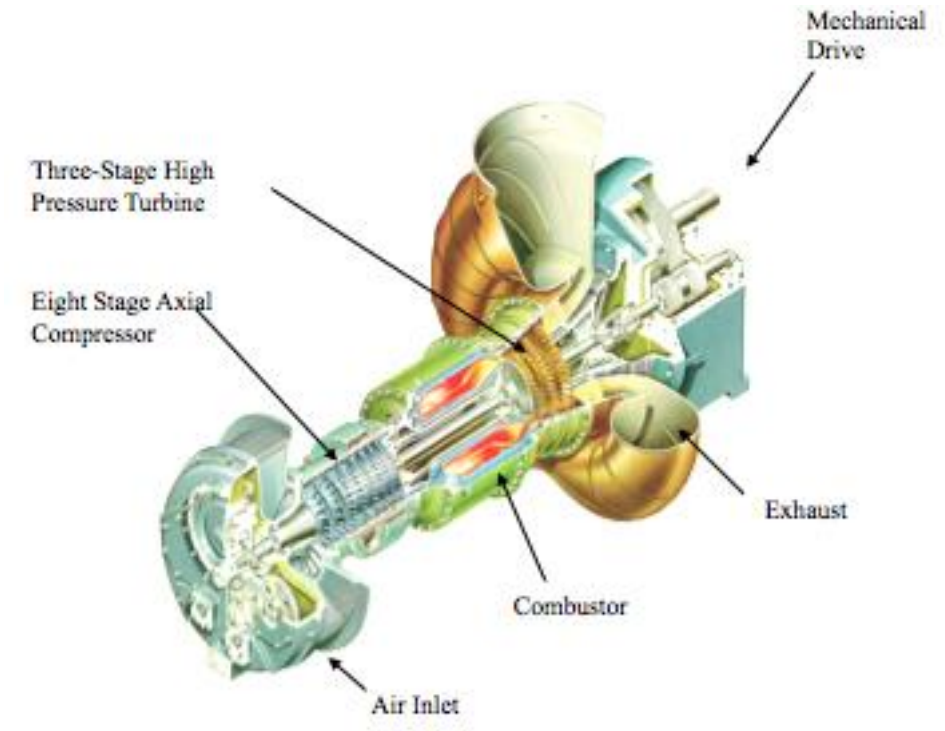
The main power generation system on many onshore and offshore installations are gas or gas/diesel (dual fuel) turbines with diesel engines for emergency power generation. Large individual consumers such as compressors often have their own power supply.

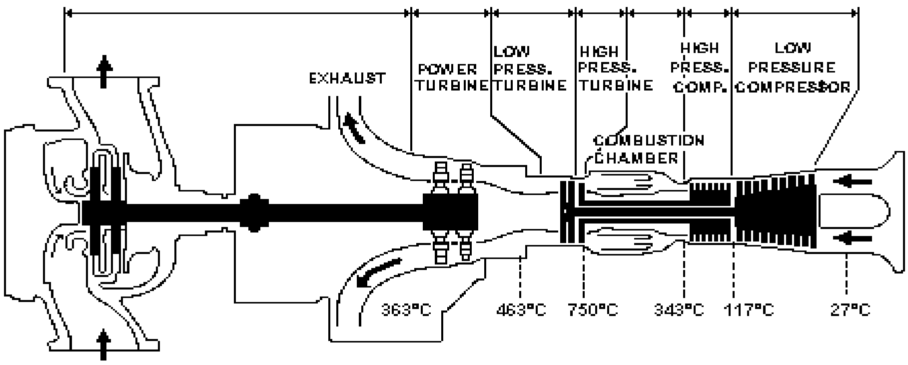
Gas turbines are normally fired by fuel gas taken from the production train, when this is not available, during start-up for example, the turbine can operate on diesel fuel

The gas turbines are virtually self contained units, complete with their own controls, switchgear, lubricating and hydraulic oil system, compressed air system, air filters, inlet and exhaust silencers

The operating principles of a gas turbine are as follows:

- Air is drawn through inlet filters and silencers and adiabatically compressed in stages of axial compression, the air then passes to the combustion chamber
- Some of the air is fed directly to the fuel burners, the majority being used to cool the outer surfaces of the combustion chamber
- At higher combustion temperatures, the efficiency of the turbine increases however the turbine blades would have a reduced operating life, therefore an economic compromise is required
- The hot gases then expand adiabatically through the power turbine section which drives the alternator/rotating plant.



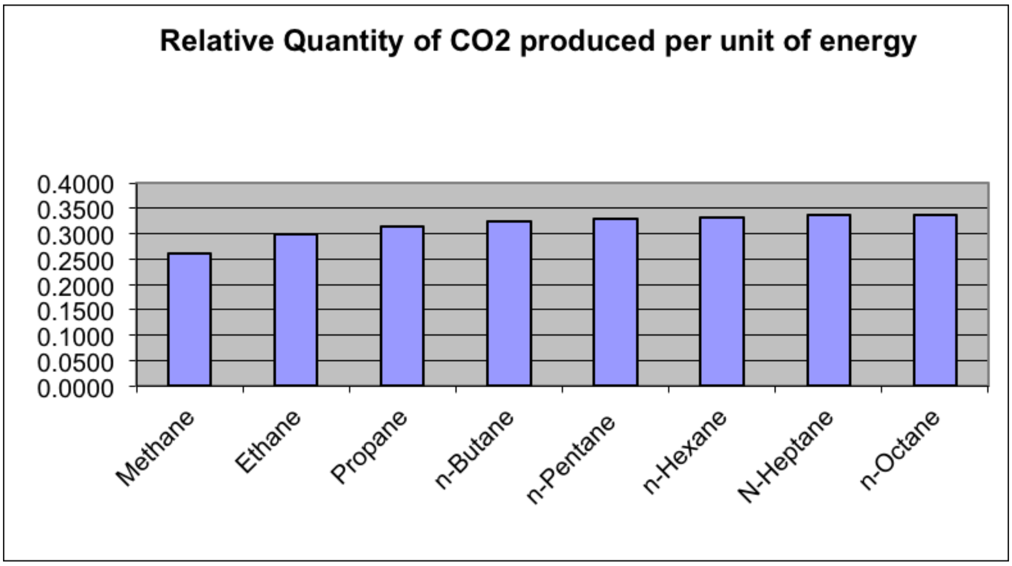


19.1 Combustion Chemistry

A review of the combustion reactions clearly indicates that the most efficient fuel is Methane.

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$$

$$C_8H_{18} + 25/2O_2 \longrightarrow 8CO_2 + 9H_2O$$



19.2 Combustion Temperature

When natural gas is burned with air under stoichiometric conditions the resulting temperature is around 1940ºC (3500ºF) depending on the temperature of the combustion air. It is necessary to utilize excess of air in the combustion step as excess air acts as a thermal diluent reducing the temperature of the combustion products, this temperature is set by the material limitations used in the turbine parts exposed to the hot gas.

From a cycle efficiency and engine specific power output (kW per kg/s of suction air flow) standpoint, it is important to minimize the amount of cooling air as well as the excess combustion air.

Air is also used for turbine blade cooling. This creates a large parasitic load on the cycle, since compression of the air requires mechanical energy, thus reducing the net power produced from the system, as well as reducing the overall efficiency of the system.

19.3 Gas Turbine Basics

A schematic diagram for a simple-cycle, single shaft gas turbine is shown in Figure 2. Air enters the axial flow compressor at point 1 at ambient conditions. Since these conditions vary from day to day and from location to location, it is convenient to consider some standard conditions for comparative purposes. The standard conditions used by the gas turbine industry are 59 ^{o}F/15 ^{o}C, 14.7 psia/1.013 bar and 60% relative humidity, which are established by the International Standards Organization (ISO) and frequently referred to as ISO conditions.

Air entering the compressor at point 1 is compressed to a higher pressure. No heat is added; however, compression raises the air temperature so that the air at the discharge of the compressor is at a higher temperature and pressure. Upon leaving the compressor, air enters the combustion system at point 2, where fuel is injected and combustion occurs. The combustion is at constant pressure.

Although high local temperatures are reached within the primary combustion zone, the combustion system is designed to provide mixing, burning, dilution and cooling. Thus, by the time the combustion mixture leaves the combustion system and enters the turbine at point 3, it is at a mixed average temperature. In the turbine section of the gas turbine, the energy of the hot gases is converted into work. This conversion takes place in two steps. In the nozzle section of the turbine, the hot gases are expanded and a portion of the thermal energy is converted into kinetic energy. In the subsequent bucket section of the turbine, a portion of the kinetic energy is transferred to the rotating buckets and converted to work. Some of the work developed by the turbine is used to drive the compressor, and the remainder is available for useful work at the output flange of the gas turbine.

As shown in Figure 2, single-shaft gas turbines are configured in one continuous shaft and, therefore, all stages operate at the same speed. These units are typically used for generator drive applications where significant speed variation is not required.

A schematic diagram for a simple-cycle, two shaft gas turbine is shown in Figure 3. The low pressure or power turbine rotor is mechanically separate from the high-pressure turbine and compressor rotor. The low pressure rotor is said to be aerodynamically coupled. This feature allows the power turbine to be operated at a range of speeds and makes two shaft gas turbines ideally suited for variable speed applications.

All of the work developed by the power turbine is available to drive the load equipment since the work developed by the high-pressure turbine supplies all the necessary energy to drive the compressor. On two-shaft machines the starting requirements for the gas turbine load train are reduced because the load equipment is mechanically separate from the high-pressure turbine.

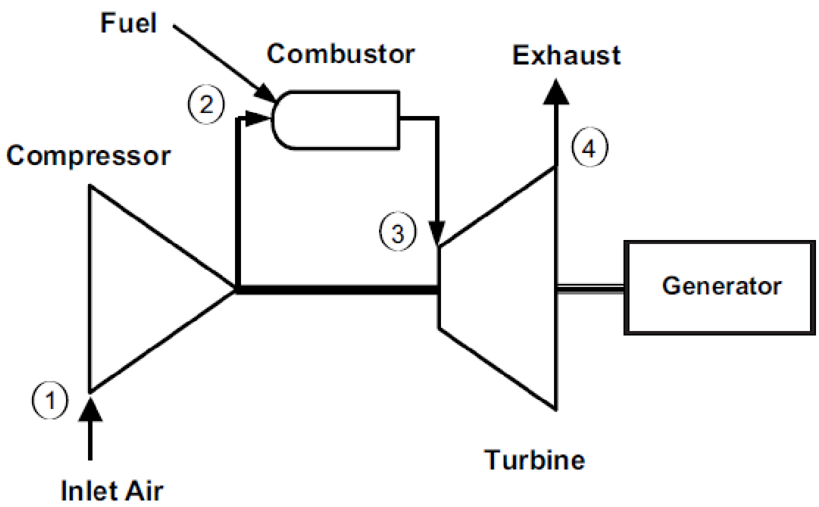


Figure 2. Simple-cycle, single-shaft gas turbine

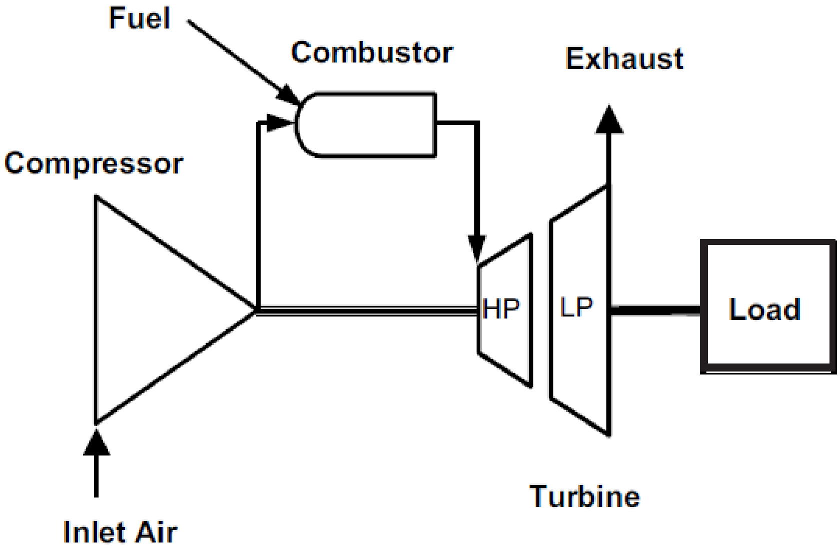


Figure 3. Simple-cycle, two-shaft gas turbine

19.4 The Brayton Cycle

The thermodynamic cycle upon which all gas turbines operate is called the Brayton cycle. Figure 4 shows the classical pressure-volume (PV) and temperature-entropy (TS) diagrams for this cycle. The numbers on this diagram correspond to the numbers also used in Figure 2. Path 1 to 2 represents the compression occurring in the compressor, path 2 to 3 represents the constant-pressure addition of heat in the combustion systems, and path 3 to 4 represents the expansion occurring in the turbine. The path from 4 back to 1 on the Brayton cycle diagrams indicates a constant-pressure cooling process. In the gas turbine, this cooling is done by the atmosphere, which provides fresh, cool air at point 1 on a continuous basis in exchange for the hot gases exhausted to the atmosphere at point 4. The actual cycle is an “open” rather than “closed” cycle, as indicated.

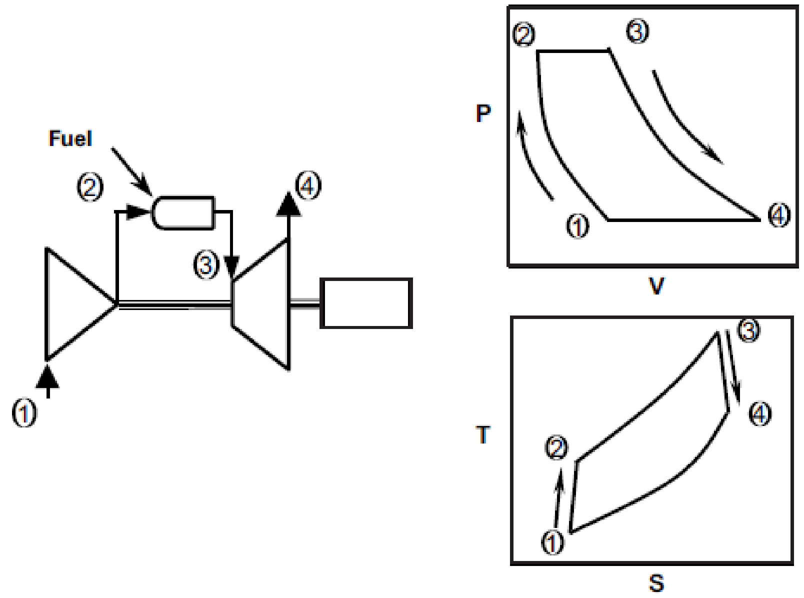


Figure 4. Brayton cycle

Every Brayton cycle can be characterized by two significant parameters: pressure ratio and firing temperature. The pressure ratio of the cycle is the pressure at point 2 (compressor discharge pressure) divided by the pressure at point 1 (compressor inlet pressure). In an ideal cycle, this pressure ratio is also equal to the pressure at point 3 divided by the pressure at point 4. However, in an actual cycle there is some slight pressure loss in the combustion system and, hence, the pressure at point 3 is slightly less than at point 2. The other significant parameter, firing temperature, is normally defined as the mass-flow mean total temperature at the stage 1 nozzle trailing edge plane. (note differing manufacturers have differing definitions).

Nozzles are cooled to keep the temperatures within the operating limits of the materials being used. The two types of cooling currently employed are air and steam.

Air cooling has been used for more than 30 years and has been extensively developed in aircraft engine technology, as well as the latest family of large power generation machines. Air used for cooling the first stage nozzle enters the hot gas stream after cooling down the nozzle and reduces the total temperature immediately downstream.

Steam-cooled first stage nozzles do not reduce the temperature of the gas directly through mixing because the steam is in a closed loop. Steam is a much more effective coolant. As shown in Figure 5, the firing temperature is increased without increasing the combustion exit temperature.

Figure 6 shows how the various temperatures are defined.

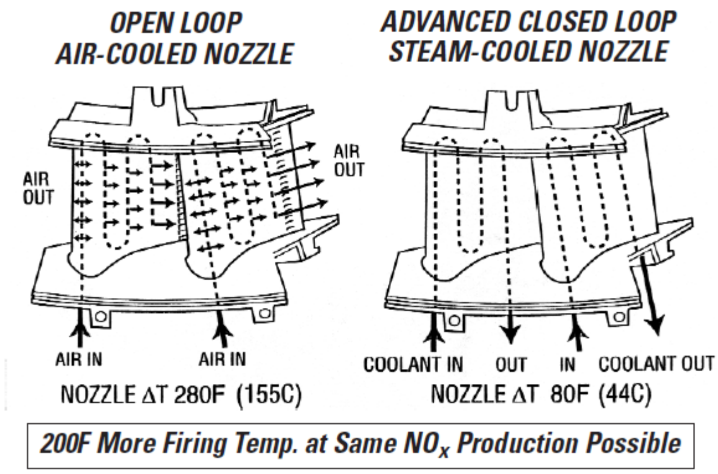


Figure 5. Comparison of air-cooled vs. steam-cooled first stage nozzle

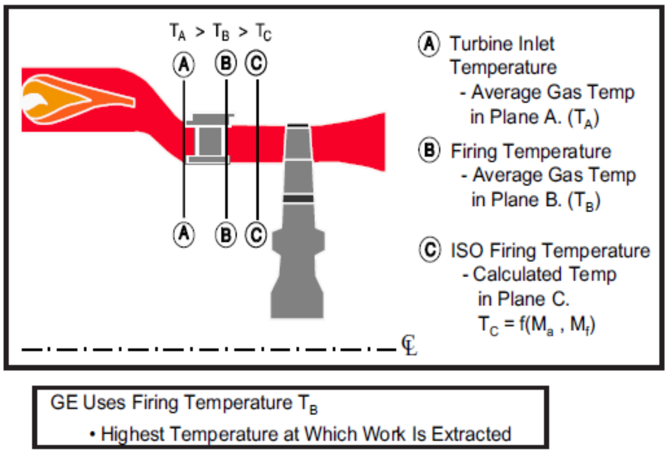


Figure 6. Definition of firing temperature

Classical thermodynamics permit evaluation of the Brayton cycle using such parameters as pressure, temperature, specific heat, efficiency factors and the adiabatic compression exponent. If such an analysis is applied to the Brayton cycle, the results can be displayed as a plot of cycle efficiency vs. specific output of the cycle.

Figure 7 shows such a plot of output and efficiency for different firing temperatures and various pressure ratios. Output per mass of airflow is important since the higher this value, the smaller the gas turbine required for the same output power. Thermal efficiency is important because it directly affects the operating fuel costs. Figure 7 illustrates a number of significant points. In simple-cycle applications (the top curve), pressure ratio increases translate into efficiency gains at a given firing temperature. The pressure ratio resulting in maximum output and maximum efficiency change with firing temperature, and the higher the pressure ratio, the greater the benefits from increased firing temperature. Increases in firing temperature provides power increases at a given pressure ratio, although there is a sacrifice of efficiency due to the increase in cooling air losses required to maintain parts working lives.

In combined-cycle applications (as shown in the bottom graph in Figure 7), pressure ratio increases have a less pronounced effect on efficiency. Note also that as pressure ratio increases, specific power decreases. Increases in firing temperature result in increased thermal efficiency. The significant differences in the slope of the two curves indicate that the optimum cycle parameters are not the same for simple and combined cycles. Simple-cycle efficiency is achieved with high pressure ratios. Combined-cycle efficiency is obtained with more modest pressure ratios and greater firing temperatures.

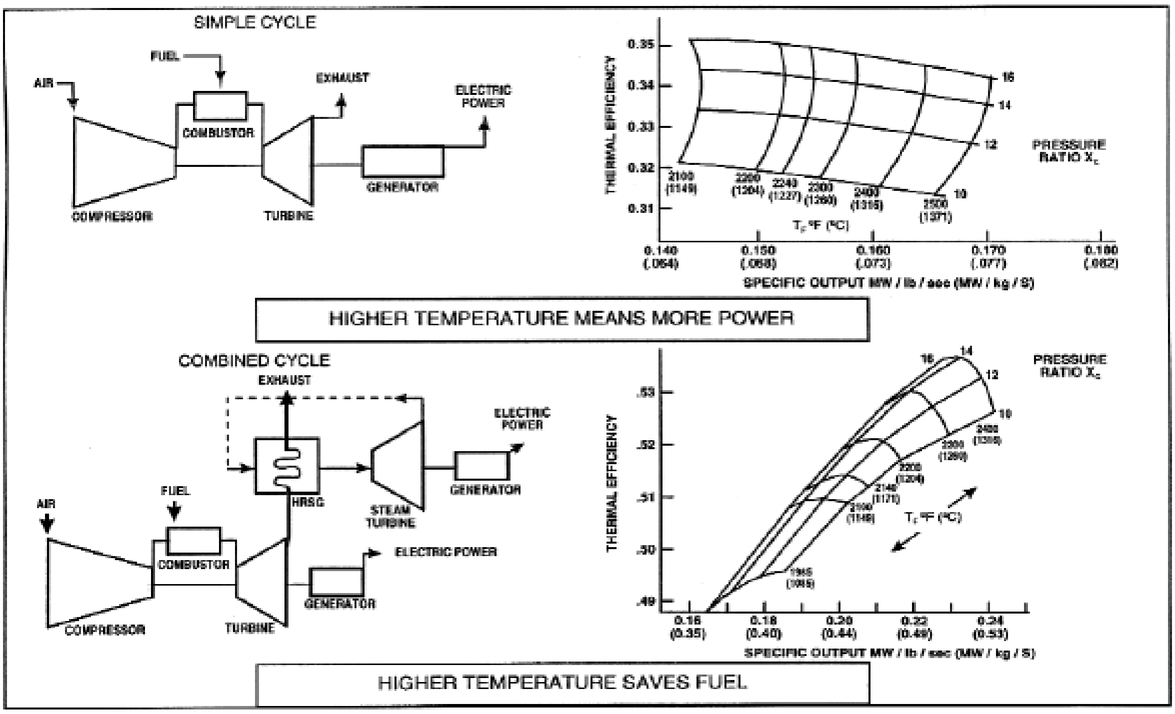


Figure 7. Gas turbine thermodynamics

19.5 Combined Cycle Gas Turbines (CCGT)

A typical simple-cycle gas turbine will convert 30% to 40% of the fuel input into shaft output. All but 1% to 2% of the remainder is in the form of exhaust heat. The combined cycle is generally defined as one or more gas turbines with heat-recovery steam generators in the exhaust, producing steam for a steam turbine generator, heat-to-process, or a combination thereof.

Figure 8 shows a combined cycle in its simplest form. High utilization of the fuel input to the gas turbine can be achieved with some of the more complex heat-recovery cycles, involving multiple-pressure boilers, extraction or topping steam turbines, and avoidance of steam flow to a condenser to preserve the latent heat content. Attaining more than 80% utilization of the fuel input by a combination of electrical power generation and process heat is not unusual.

Combined cycles producing only electrical power are in the 50% to 60% thermal efficiency range using the more advanced gas turbines.

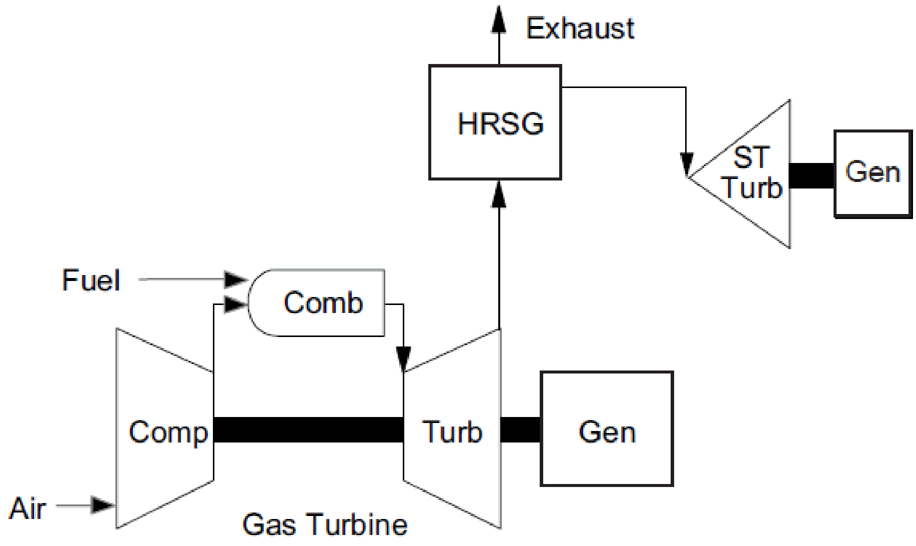
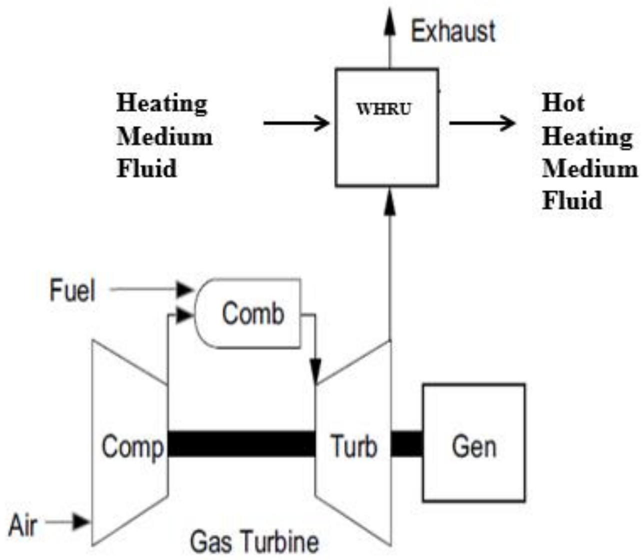


Figure 8. Combined cycle

It is interesting to note that there are no CCGTs deployed in the UK offshore sector. However, there are a few units in Norway as a result of the CO_2 tax structure.

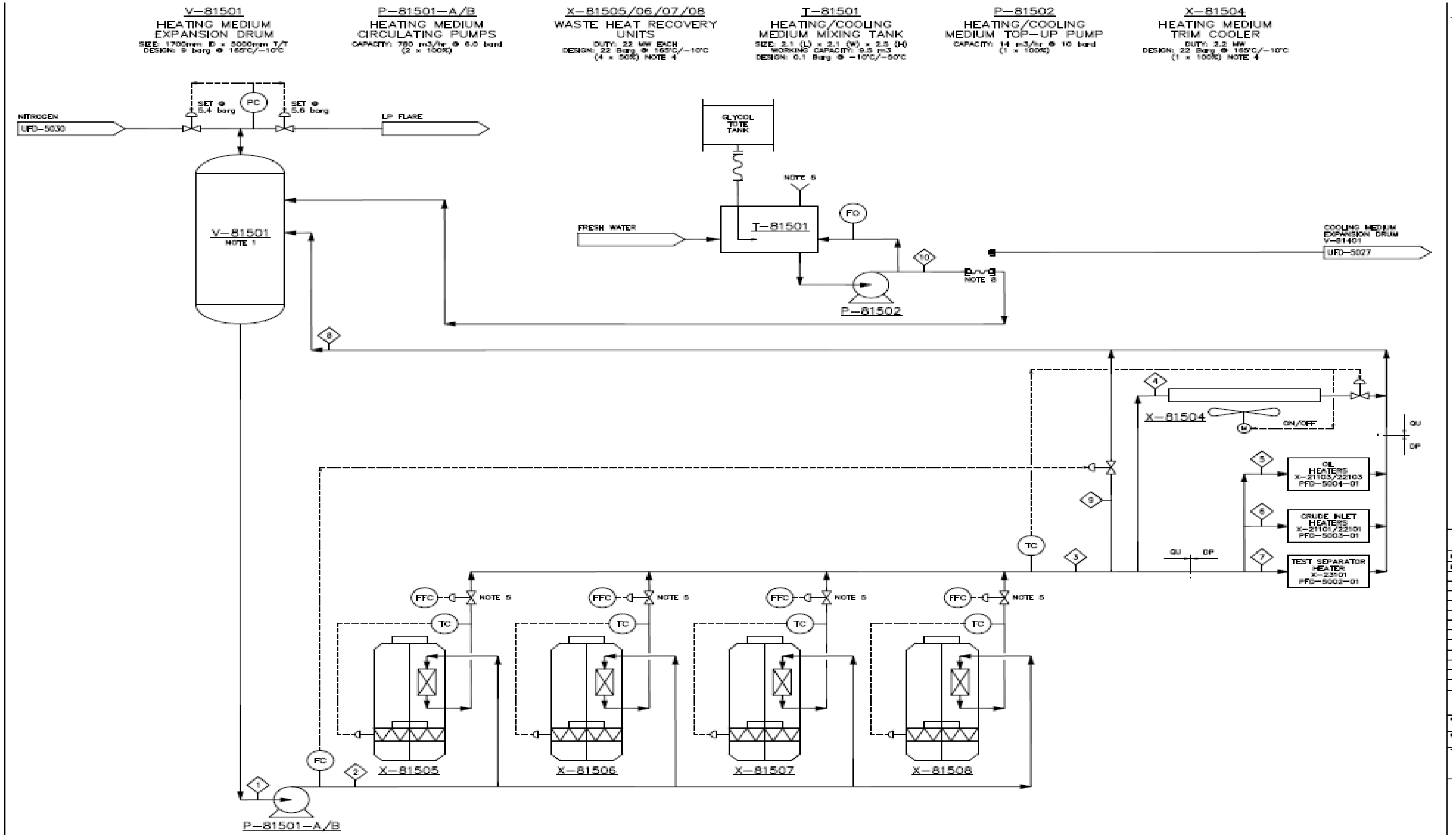
19.6 Combined Heat and Power

As an alternative to additional power generation the exhaust gases can be used to deliver heat to the process via a heating medium, for example separation heating system and glycol unit reboilers. As a rule of thumb heat available is around 1.25 times the turbine power rating.



The heating medium is often a Triethylene glycol water mixture. The glycol prevents the water from freezing.

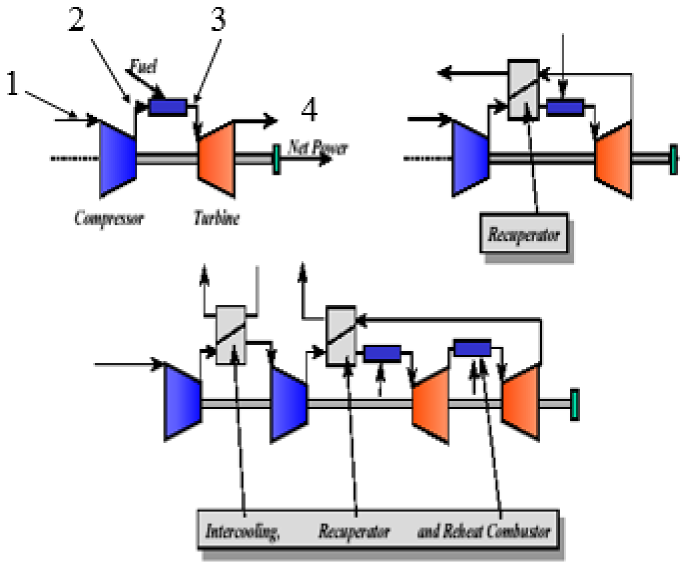
A typical heating medium PFD follows.



Stream No.	1	2	3	4	5	6	7	8	9	10
Pressure (barg)	9.0	15.0	9.5	10.5	11.5	9.5	10.5	5.5	9.5	10.0
Temperature (°C)	93-149	93-149	149	152	149	149	149	93-149	149	AMB
Flowrate (m3/hr)	780	780	820	NNF	272	438	105	780	NNF	NNF
Design Flowrate (m3/hr)	780	780	820	36	272	438	105	780	390	10
Notes	9	7,9						9		

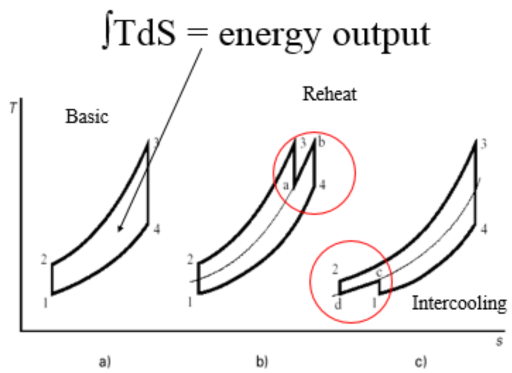
19.7 Brayton Cycle Variants

Regeneration involves the installation of a heat exchanger (recuperator) through which the turbine exhaust gases pass. The compressed air is then heated in the exhaust gas heat exchanger, before the flow enters the combustor.



Intercooling also involves the use of a heat exchanger. An intercooler is a heat exchanger that cools compressor gas during the compression process. For instance, if the compressor consists of a high and a low pressure unit, the intercooler could be mounted between them to cool the flow and decrease the work necessary for compression in the high pressure compressor. The cooling fluid could be atmospheric air or water (e.g. sea water in the case of a marine gas turbine). It can be shown that the output of a gas turbine is increased with a well-designed intercooler.

Reheating occurs in the turbine and is a way to increase turbine work without changing compressor work or melting the materials from which the turbine is constructed. If a gas turbine has a high pressure and a low pressure turbine at the back end of the machine, a reheater (usually another combustor) can be used to "reheat" the flow between the two turbines. This can increase efficiency by 1-3%. Reheat in a jet engine is accomplished by adding an afterburner at the turbine exhaust, thereby increasing thrust, at the expense of a greatly increased fuel consumption rate.



19.8 Factors Affecting Turbine performance

19.8.1 Air Temperature and Site Elevation

Since the gas turbine is an air-breathing engine, its performance is changed by anything that affects the density and/or mass flow of the air

intake to the compressor. Ambient weather conditions are the most obvious changes from the reference conditions of 59 ^{o}F/15 ^{o}C and 14.7 psia/1.013 bar. Figure 9 shows how ambient temperature affects the output, heat rate, heat consumption, and exhaust flow of a single-shaft.

Each turbine model has its own temperature-effect curve, as it depends on the cycle parameters and component efficiencies as well as air mass flow.

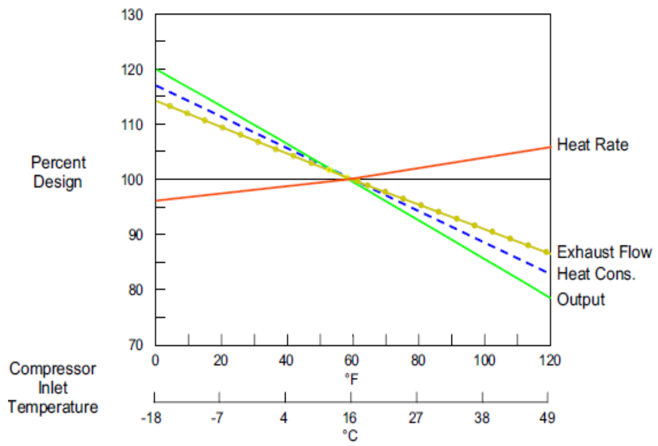


Figure 9. Effect of ambient temperature

Correction for altitude or barometric pressure is more straightforward. The air density reduces as the site elevation increases. While the resulting airflow and output decrease proportionately, the heat rate and other cycle parameters are not affected. A standard altitude correction curve is presented in Figure 10.

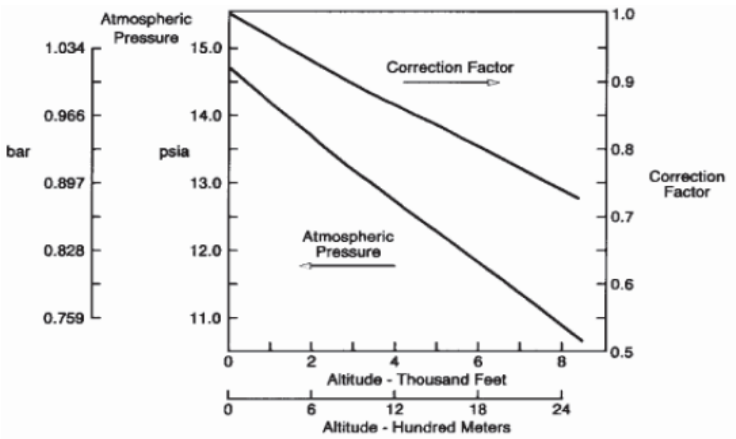


Figure 10. Altitude correction curve

19.8.2 Humidity

Similarly, humid air, which is less dense than dry air, also affects output and heat rate. This effect was considered negligible however, with the increasing size of gas turbines and the utilization of humidity to bias water and steam injection for NOx control, this effect now has greater significance. It should be noted that this humidity effect is a result of the control system approximation of firing temperature used on heavy-duty gas turbines. Single-shaft turbines that use turbine exhaust temperature biased by the compressor pressure ratio to the approximate firing temperature will reduce power as a result of increased ambient humidity. This occurs because the density loss to the air from humidity is less than the density loss due to temperature. The control system is set to follow the inlet air temperature function. By contrast, the control system on aeroderivatives uses unbiased gas generator discharge temperature to approximate firing temperature. The gas generator can operate at different speeds from the power turbine, and the power will actually increase as fuel is added to raise the moist air (due to humidity) to the allowable temperature. This fuel increase will increase the gas generator speed and compensate for the loss in air density.

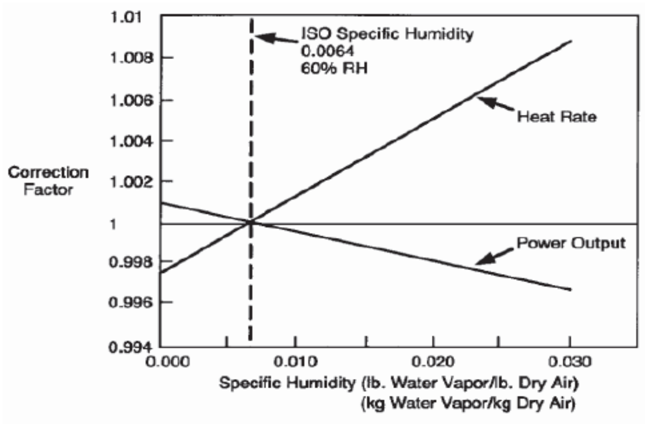


Figure 11. Humidity effect curve

19.8.3 Inlet and Exhaust Losses

Inserting air filtration, silencing, evaporative coolers or chillers into the inlet or heat recovery devices in the exhaust causes pressure losses in the system.

19.8.4 Fuels

Several side effects must be considered when burning this kind of lower heating value fuels:

Increased turbine mass flow drives up compressor pressure ratio, which eventually encroaches on the compressor surge limit.

The higher turbine power may exceed fault torque limits. In many cases, a larger generator and other accessory equipment may be needed

High fuel volumes increase fuel piping and valve sizes (and costs).

Lower-Btu gases are frequently saturated with water prior to delivery to the turbine. This increases the combustion products heat transfer coefficients and raises the metal temperatures in the turbine section which may require lower operating firing temperature to preserve parts lives.

As the calorific value drops, more air is required to burn the fuel. Machines with high firing temperatures may not be able to burn low calorific value gases

As a result of these influences, each turbine model will have some application guidelines on flows, temperatures and shaft output to preserve its design life. In most cases of operation with lower calorific value fuels, it can be assumed that output and efficiency will be equal to or higher than that obtained on natural gas. In the case of higher calorific value fuels, such as refinery gases, output and efficiency may be equal to or lower than that obtained on natural gas.

19.9 Gas Turbine NOx Management

NOx gases react to form smog and acid rain as well as being central to the formation of fine particles and ground level ozone, both of which are associated with adverse health effects. Hence NOx emissions from gas turbines are a key aspect of environmental management. NOx can be controlled using water injection, low NOx burners and by exhaust gas catalytic control.

19.9.1 Water or steam injection

Injecting water or steam into the turbine combustion chamber limits NOx formation by reducing the average combustion temperature. The typical rate of steam or water injection is 50-100% of the fuel input rate. Although this procedure has the additional advantage of slightly increasing turbine output, there are disadvantages, notably a minor reduction in CHP system efficiency and a possible increase in carbon monoxide (CO) levels in the exhaust as water or steam levels are increased – the result of partial quenching of the flame.

To avoid turbine damage, the water or steam injected must have a high degree of purity. Even so, injection tends to reduce the life of some of the turbine components and has operating cost implications for the turbine and its associated systems. Gas and gas-oil firing systems can usually incorporate water or steam injection, and virtually all gas turbines can be fitted with this facility.

The equipment needed to treat and inject water or steam into a gas turbine increases the capital cost of a CHP scheme by around 2-3%. Operating costs typically rise by 1-2%.

19.9.2 Low-NOx burners

Low-NOx burners, sometimes known as dry low-NOx (DLN), have been developed for large gas turbines and are now available for most smaller turbines. The burners are designed to operate with a lower temperature flame to reduce NOx emissions. At present, some of these burners can burn only gaseous fuels in low-NOx mode and have no capability to burn gas-oil/diesel as a stand-by fuel. This limits their application on sites that use interruptible gas tariffs and, therefore, have a dual-fuel requirement.

19.9.3 Catalytic control

Current research is seeking to incorporate catalysts into turbine combustion systems but has yet to establish the feasibility, durability and cost-effectiveness of the catalytic control of NOx emissions in gas turbines.

19.10 Gas Turbine Data

Data from GT manufacturers is shown.

MANUFACTURER	MODEL	TYPE	ISO RATING (MW)	EFFICIENCY (%)	DIMENSIONS L:W:H (m)	WEIGHT (tonnes)
Solar	Centaur 40	H	3.5	28.5	9.8:2.5:2.6	30
	Centaur 50	H	4.6	29.4	9.0:2.5:2.8	32
	Taurus 60	H	5.5	30.5	11:2.5:3.1	32
	Taurus 70	H	7.5	33	12:2.8:3.1	55
	Mars 90/100	H	9.4 - 10.7	31.5	15:2.8:3.8	70
	Titan 130	H	14	33.5	14:3.1:3.8	75
Alstom Power - (formerly Ruston and EGT)	TA1750	HDI	1.3	18	9.0:2.6:3.1	23
Lincoln Design and Manufacture	TA2500	HDI	1.9	20	9.5:2.4:2.8	25
	TB5000	HDI	4.0	25	9.7:2.4:2.4	34
	Typhoon	LWI	4.35 - 5.25	30.5	8.0:2.4:3.2	36
	Tornado	LWI	6.75	31.5	11.0:2.4:3.3	56
	Tempest	LWI	7.9	31.2	10.75:2.4:3.6	57
	Cyclone	LWI	12.9	35	13.5:2.7:3.9	75
Alstom Power - (formerly ABB Stal)	GT35	H	15 - 20	32	25.0:4.0:4.2	160
Finspong Design and Manufacture	GT10B	LWI	24.8	34	20.5:4.5:5.3	185
	GT10C	LWI	29.1	36	20.5:4.5:5.3	185
	GTX100	LWI	43	37	22.0:4.5:6.0	275
Rolls-Royce/Allison	Avon	A/D	14.58	27	-	165
	RB211	A/D	23.0 - 30.8	36	-	185
	501-K	A/D	3.95 - 5.27	29.5	-	27
	601-K	A/D	6.5 - 7.92	31	-	63
GE	LM1600	A/D	13.74	35	-	-
	LM2500	A/D	22-30	38	-	-
	LM6000	A/D	42	41.5	-	-
	MS5001E (Frame 5)†	HDI	26.3	28.5	21.5:3.28:4.0	288
	MS6001B (Frame 6)	HDI	42.1	32.5	22.6:3.28:3.8	276

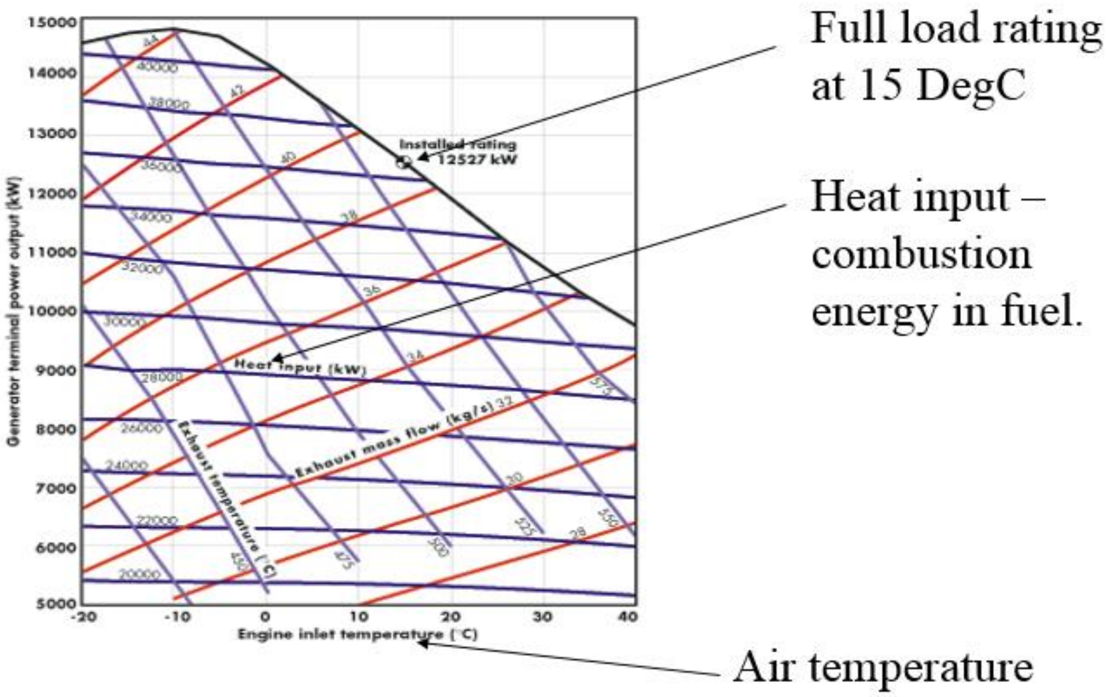
- HDI = Heavy Duty Industrial
- A/D = Aeroderivative
- H = Hybrid (mixture of industrial and aeroderivative)
- LWI = Light Weight Industrial

† Frame 5 is a relatively old design which accounts for the lower power/weight ratio than the Frame 6

Analysis of this data shows that not surprisingly that aero derivatives are more efficient than industrial variants. Also the larger the machine the better efficiency

19.11 Gas Turbine Rating Charts

A typical rating chart is shown. For a given set of conditions the machine output and efficiency can be determined.



Worked Example

Compare one machine on full load with two

Machines at 50% load

Point 1 (100% full load)

Ambient 10°C

Generator Power 13200 kW

Heat Input 38000 kW

Efficiency 13200/38000 x 100 = 34.7%

Point 2 (50% thermal load)

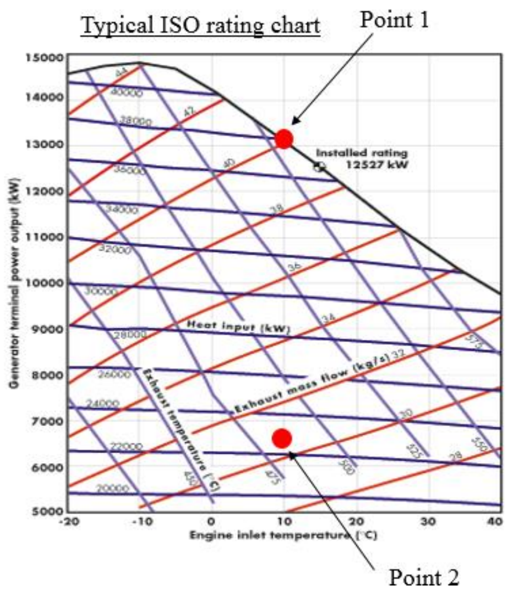
Ambient 10°C

Generator Power 6600 kW

Heat Input 22750 kW

Efficiency 6600/22750 x 100 = 29.0%

Result of 50% loading is;



20% increase in fuel mass flow for same total load. Hence it always better to run a machine at full load from an efficiency standpoint. Turbine operators though may wish to run with spinning reserve – two machines at part load. This is done because of machine trip concerns: with two machine running it is much easier for the other to pick up the load than restart an idle machine is a trip occurs.

19.12 Technology Developments

Designed to work in a "combined cycle" power plant the *H System*™ will be the most efficient power generation system in the world. It will be the first gas turbine to top the 60 percent efficiency threshold -- the "four minute mile" of turbine technology. When the US Energy Department began its advanced turbine development program in the early 1990s, the best turbines available had efficiencies of about 50 percent. The efficiency gains have been achieved because the turbine fires natural gas nearly 170 ^oC hotter than conventional turbines, reaching temperatures of 1430 ^oC. Advanced cooling techniques and new alloys were developed to handle the hotter temperatures. The turbine also employs the world's largest single crystal aerofoils, making the turbine blades much more resistant to high temperature cracking than the multi-directional crystal design currently used.

20 Electrical Load List

It is important to identify the utilisation voltage of each piece of equipment in order to properly design the electrical system.

An electrical load list is the best way to determine the overall system requirements in terms of the operating load. The load list should include the power rating of each individual consumer and whether it is run continuously, intermittently or as standby for another consumer.

Once this has been accomplished, local loads may be grouped to be served from switchgear which in turn are served from transformers.

When generating power, the starting loads need to be considered in addition to the total steady state load. Note that starting loads of three to five times full load are often required and the design of the GT configuration has to incorporate any requirement for large motor starts.

A typical load list is shown.

Description	Power (kW)	
	Continuous	Intermittent
MOL pumps	1800	-
Crude export pumps	-	3200
Drilling	-	3160
Separation	382	-
Compression	5000	600
Gas dehydration	350	-
Water injection	2500	-
Seawater lift pumps	380	190
Seawater treatment	50	50
Fresh water	-	80
Electrochlorinator	250	-
Fuel gas	150	-
Cooling medium	160	-
Heating medium	150	-
Relief system	25	25
Closed drains	20	-
Instrument air	360	180
Fuel oil	50	-
Chemical injection	100	150
Miscellaneous	2000	1000
Total (MW)	13.727	8.635

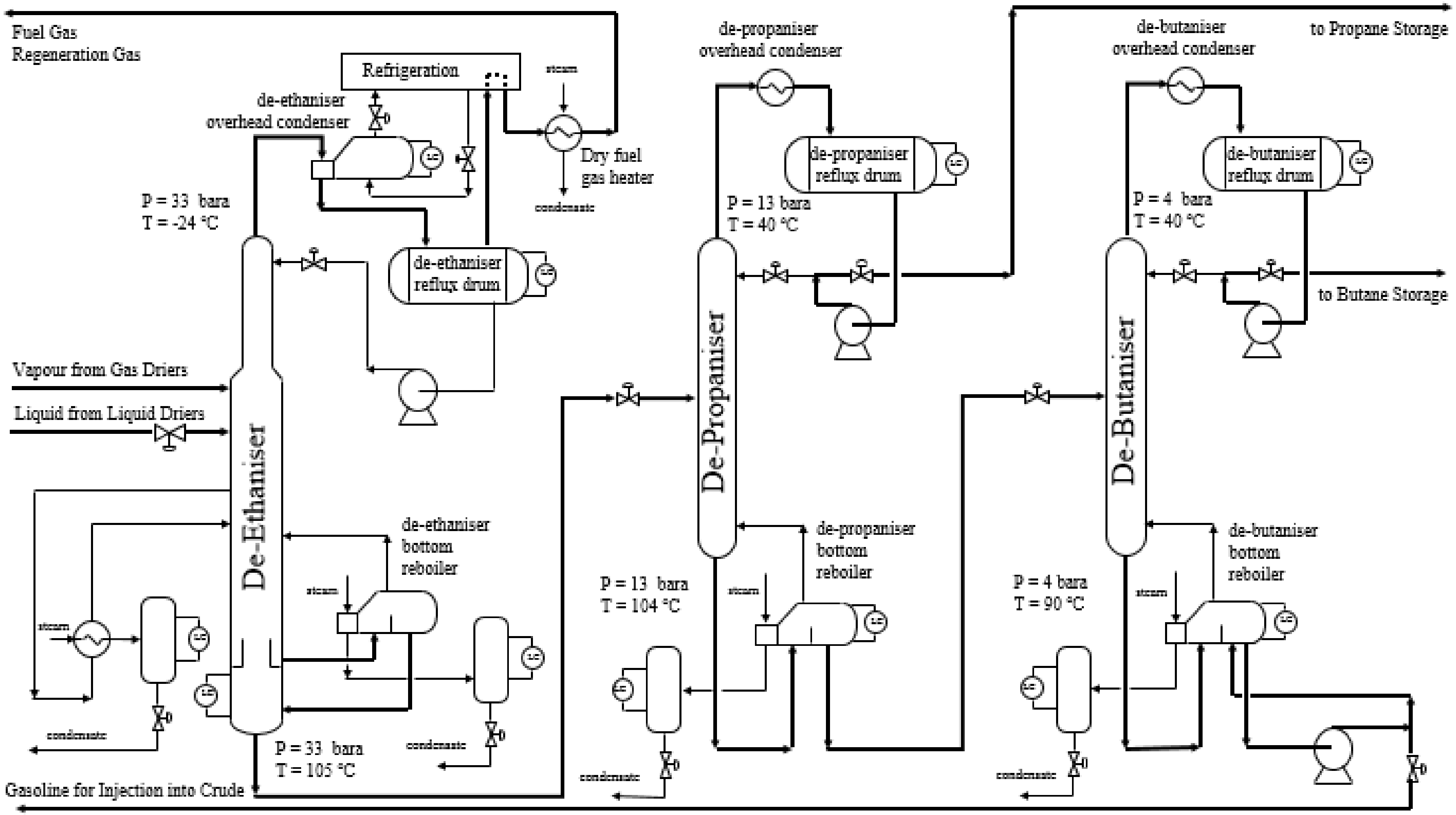
21 Fractionation

Fractionation processes are commonly seen at oil and gas terminals and refineries. Its purpose is to recover valuable light-end constituents of the gas mixture such that the required LPG specifications can be met.

Fractionation is the one of the most common separation techniques. It consumes significant amounts of energy, in terms of cooling and heating requirements and may contribute to as much as 50% of plant operating costs.

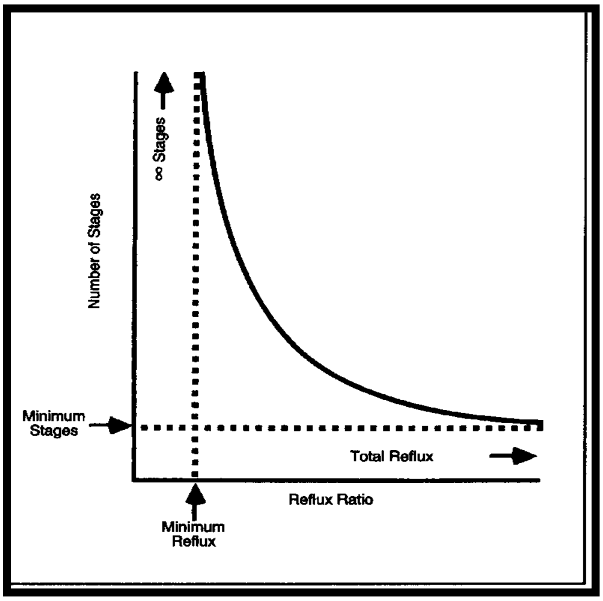
Therefore the best way to reduce operating costs of existing (or new) units, is to improve their efficiency and operation via process optimisation and control. To achieve this improvement, a thorough understanding of fractionation principles and column design is essential.

A typical sequential LPG fraction train is shown


Fuel Gas
Regeneration Gas
Refrigeration
steam
de-ethaniser
overhead condenser
Dry fuel
gas heater
condensate
P = 33 bara
T = -24 °C
de-ethaniser
reflux drum
Vapour from Gas Driers
Liquid from Liquid Driers
De-Ethaniser
de-ethaniser
bottom
reboiler
P = 33 bara
T = 105 °C
Gasoline for Injection into Crude
de-propaniser
overhead condenser
de-propaniser
reflux drum
P = 13 bara
T = 40 °C
De-Propaniser
de-propaniser
bottom
reboiler
P = 13 bara
T = 104 °C
de-butaniser
overhead condenser
to Propane Storage
de-butaniser
reflux drum
P = 4 bara
T = 40 °C
to Butane Storage
De-Butaniser
de-butaniser
bottom
reboiler
P = 4 bara
T = 90 °C

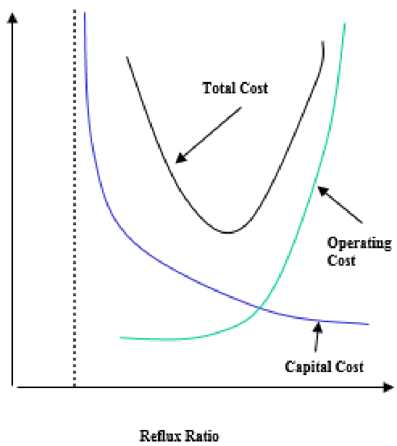
Key variables influencing operability, efficiency and costs are operating temperatures and pressures together with reflux and reboil ratios. Other aspects to note are

- The type of cooling medium available determines the operating pressure range of the column
- To maximise the relative volatility low pressure operation is favoured. However reducing the pressure may lead to a more expensive cooling medium
- Increasing pressure may exceed the critical temperature of the bottom product. This results in the failure to achieve product specification



- Cooling medium could be water, air, water or refrigerant. The choice of medium depends on the required temperature within the unit
- The reflux ratio and the number of stages are the primary parameters in the capital cost versus energy efficiency
- Reflux ratio is defined as the ratio of molar rate of reflux liquid divided by the molar rate of net overhead product
- Reboiler duty is a direct function of the reflux ratio as the fractionator is designed to achieve an overall heat & mass balance
- Fractionator columns can only produce a desired separation between the limits of minimum reflux and minimum stages
- At minimum reflux an infinite amount of stages is required. At maximum reflux a minimum amount of stages is required

The designer is looking to find the minimal cost arrangement. This is undertaken using simulation packages such as Aspen or Unisim allied with cost estimation systems.



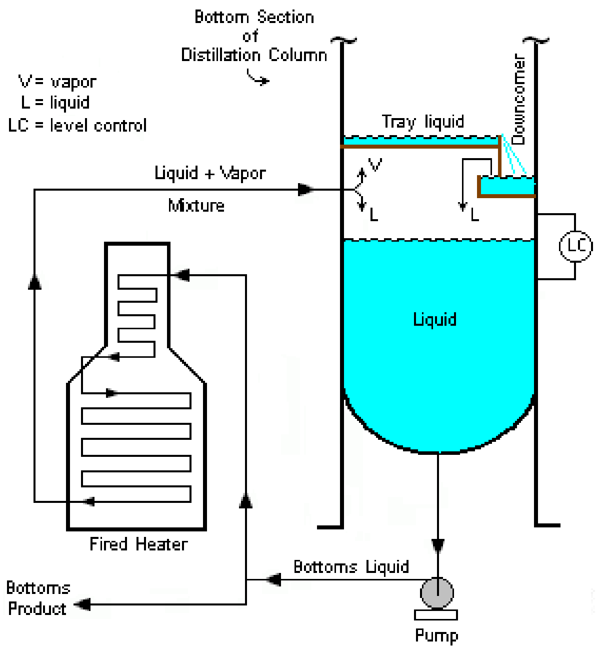
21.1 Fired Heaters

In many applications a fired heater is used for reboiler heating. If incorrectly designed, or set up, this unit can lead to poor carbon efficiency. Fired heaters are used to transfer heat directly to the process fluid, they generally have a large duty, but ranging from a few kW to 10s of MW.

There are two basic configurations, direct and indirectly fired

A fired heater consists of the following

- A combustion chamber lined with refactory material.
- The main burners.
- Tubes located within the combustion chamber where heat is transferred to the process fluid by radiation
- Tubes located external to the combustion chamber in a convection zone which is also lined with refactory
- Stack for disposal of flare gas
- Air supply system by fan or induced draft
- Instruments and controls



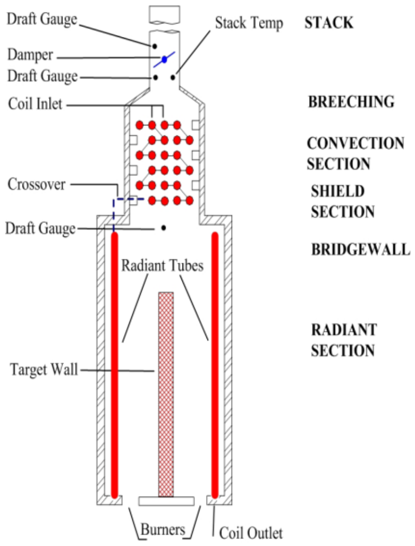
Typical applications of fired heaters include boilers, onshore glycol and amine regenerators and heating medium heaters.

The main sections are;

<u>Radiant Section</u>: The radiant tubes, either horizontal or vertical, are located along the walls in the radiant section of the heater and receive radiant heat directly from the burners or target wall. The radiant zone with its refractory lining is the costliest part of the heater and most of the heat is transferred here. This is also called the firebox.

<u>Convection Section</u>: The feed charge enters the coil inlet in the convection section where it is preheated before transferring to the radiant tubes. The convection section removes heat from the flue gas to preheat the contents of the tubes and significantly reduces the temperature of the flue gas exiting the stack. Too much heat picked up in the convection section is a sign of

too much draft. Tube temperature readings are generally taken in both convection and radiant sections.



Shield Section: Just below the convection section is the shield (or shocktube) section, containing rows of tubing which shield the convection tubes from the direct radiant heat. Several important measurements are normally made just below the shield section. The bridgewall or breakwall temperature is the temperature of the flue gas after the radiant heat is removed by the radiant tubes and before it hits the convection section. Measurement of the draft at this point is also very important since this determines how well the heater is set up. This is also the ideal place for flue gas oxygen and combustibles measurement.

Stack and Breeching: The transition from the convection section to the stack is called the breeching. By the time the flue gas exits the stack, most of the heat should be recovered. Measurement of stack emissions for compliance purposes is normally made here.

Other considerations are;

Efficiency: The thermal efficiency is simply E = Heat absorbed/Fuel calorific value

Excess Air: A higher combustion air rate is necessary than theoretically required for complete combustion of the fuel. This is caused by variations in the distribution of air and fuel to the individual burners, together with imperfect mixing of air and fuel in the burner and the flame. However no more air excess should be supplied to that actually required as any additional air must be heated up hence reducing efficiency. This is generally around 10% excess air.

Burners: Burners are classified according to the type of fuel combusted: gas, liquid or a combination. The air-fuel mixing efficiency is a function of the burner design. The better the mixing the better the efficiency. However this comes at a cost of more expensive burners.

Air Preheat: Efficiency can be improved by preheating the combustion air. This is accomplished by using the flue gas to heat the incoming air.

21.2 Fractionator Efficiency Improvements

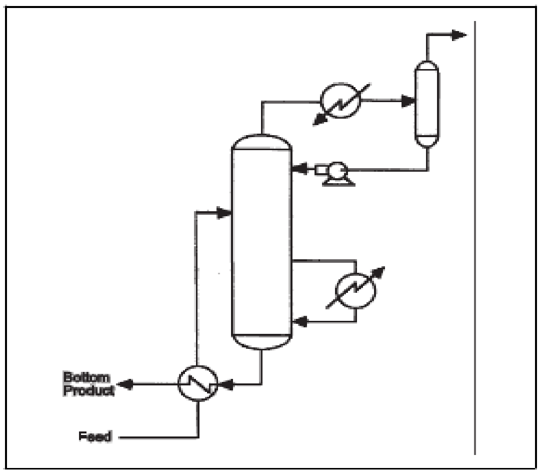
Fractionators require energy input in the form of heat to the reboiler. Regardless of the exact arrangement, fuel is often required and represents a major operating cost. Minimization of fuel usage is a common design goal.

To provide reflux for the fractionator, a utility is required to remove the heat to an appropriate heat sink. For columns utilizing air or water cooling, all the utilities use a common temperature heat sink. However, for columns using refrigeration, the temperature level is very important. A lower temperature refrigeration level increases both the capital and operating cost of a unit. If the condenser duty can be applied to a higher temperature system, considerable savings can be realised.

Feed/Product Exchangers

One of the simplest ways to reduce the reboiler fuel requirement is to preheat a liquid feed stream. This can be accomplished with a feed/product exchange as shown. In general this heat input will decrease the reboiler duty. However, since the feed is now partially vaporized, the overhead condenser duty will tend to increase. This increased condenser duty must be offset by reboiler duty. The net reboiler savings will be close to, but not equal to, the heat input to the feed. The net effect will depend on many system parameters; but feed/product exchange is generally an attractive heat conservation application.

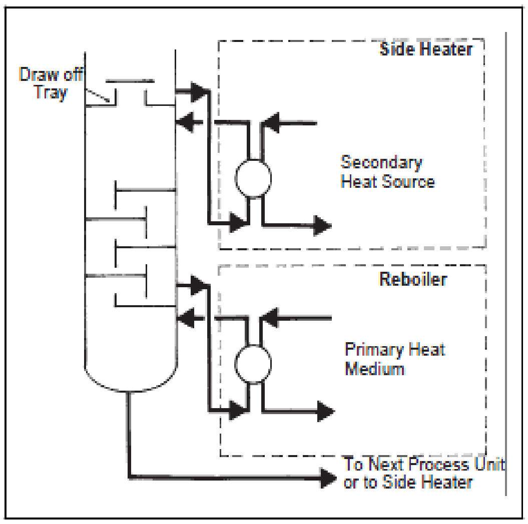
Example Feed/Product Exchanger



Side Heaters

Side heaters can be used to add heat to a tower several trays up from the reboiler. Because of the temperature gradient in the column, this heat is applied at a much lower temperature than the reboiler. The heat source for this side heater can be any stream which requires cooling and is at a high enough temperature level to be useful. Often, the bottom product is used to side-heat the

Example Side Heater

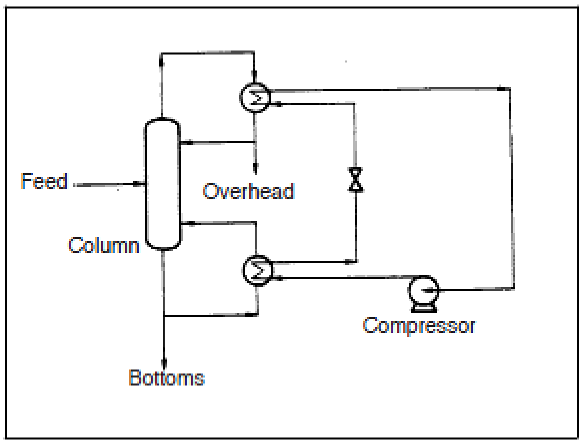


column. In cryogenic plants, the feed gas often supplies the reboil heat.

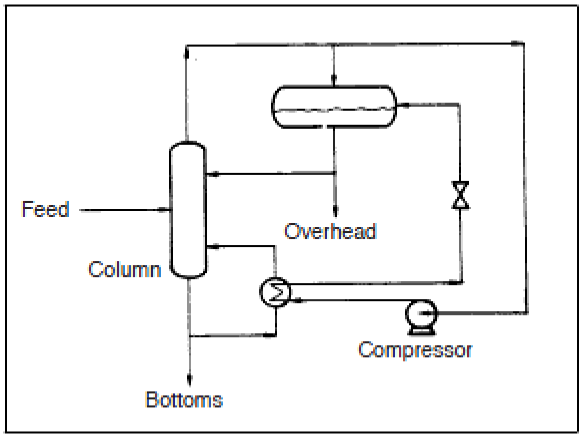
Heat Pumping

A technique for energy conservation in fractionation systems is the use of a heat pump. Heat pumping usually employs an external working fluid as shown. Compression is used to raise the temperature of the working fluid above that required for the reboiler. The fluid leaving the reboiler is then flashed and used to condense the reflux. The net result is that the heat absorbed in the condenser is used to reboil the column. The main operating cost then becomes the compressor rather than the normal heating and cooling utilities. An alternative to the basic heat pump is to use the column overhead as the working fluid. This alternative, vapor compression, eliminates the overhead condenser. It is often difficult to find a working fluid to reboil and condense in a single fractionator. However, often plants have several fractionators with condensers and reboilers at a variety of temperatures. It may be possible to link a condenser and reboiler from separate columns.

Heat Pumping



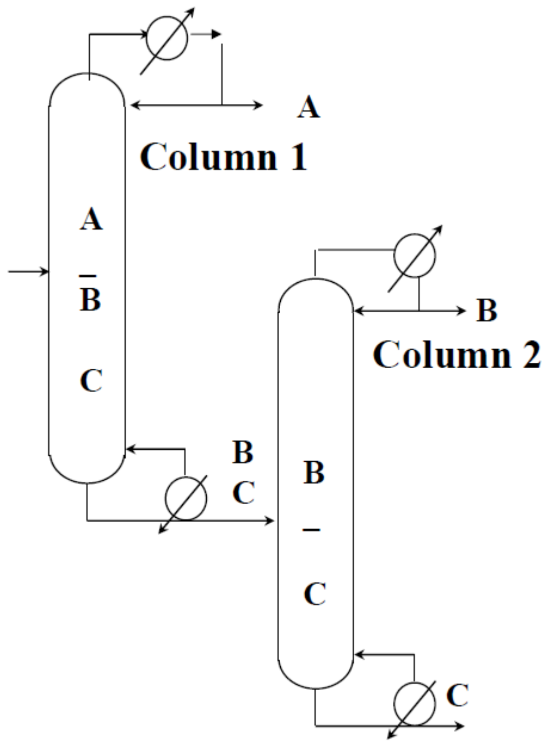
Vapor Recompression

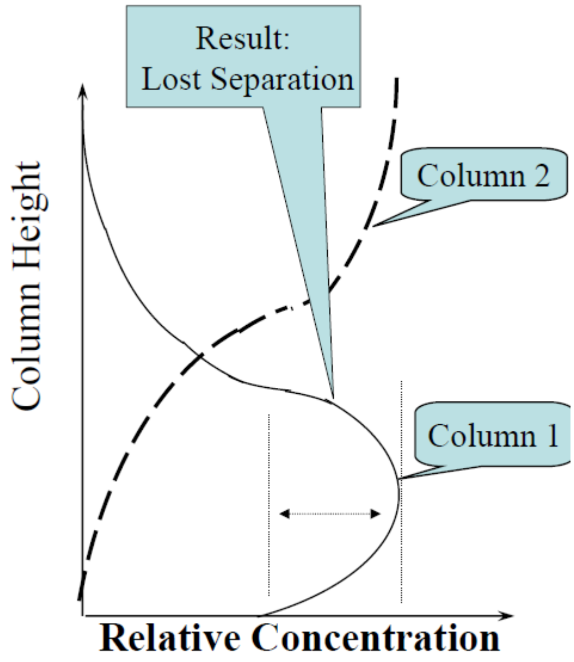


21.3 Divided Wall Columns

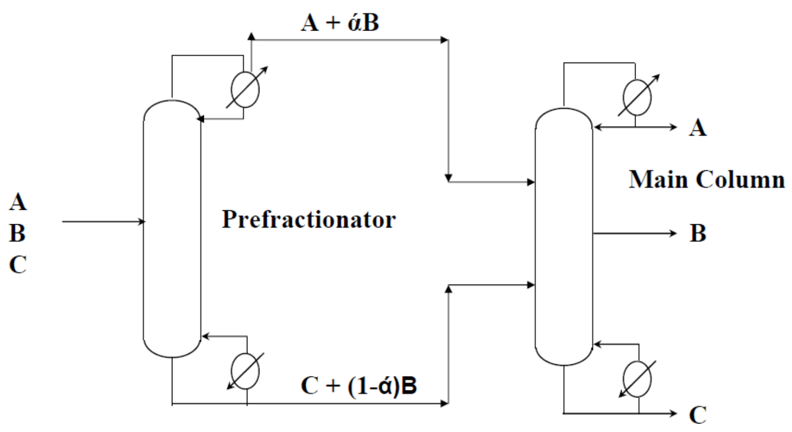
Consider the separation of three components, A, B and C, with A being the lightest, C the heaviest, and B the intermediate volatility component. Convention would be a direct sequence, removing A first with B and C going out the bottom. The following figures show this sequence, along with the relative tower concentration profiles for B.

In the first column, the concentration of B reaches a maximum somewhere in the tower but necessarily must be lower in the bottoms. This represents a thermodynamic loss that’s unavoidable in the sequential distillation scheme.

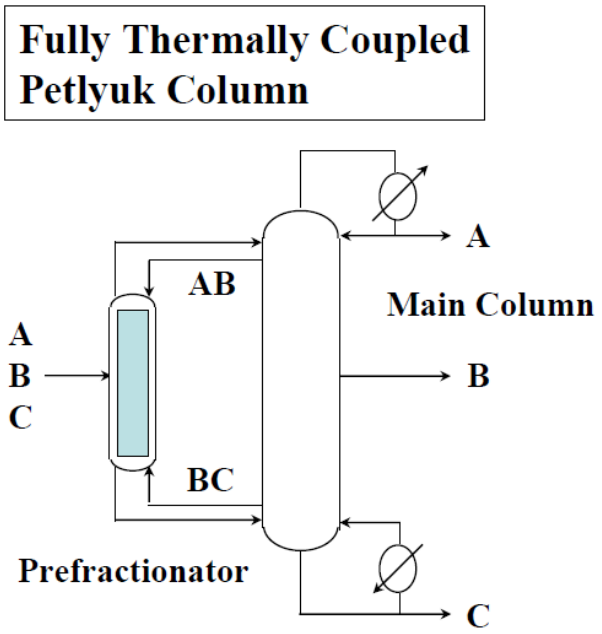


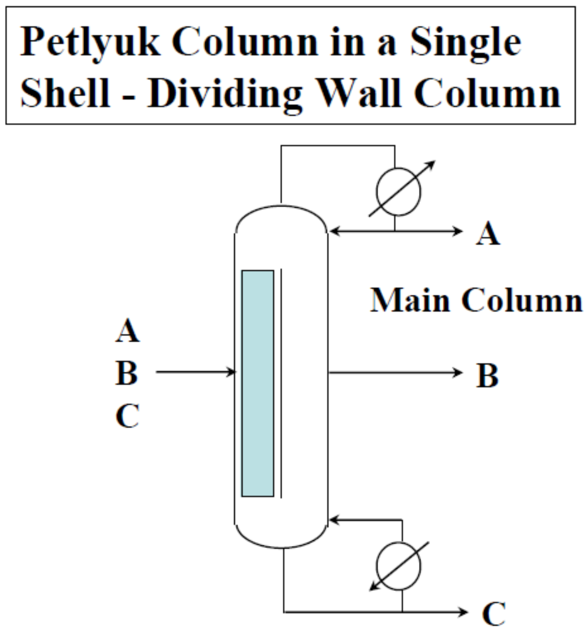


Now, consider an alternative two-column arrangement for the separations where in the first tower, rather than performing the sharp separation between component A and B, we pre-fractionate the components. In the first column, we take the entire amount of A and some fraction of B out of the top of the tower. The remaining fraction of B and the entire amount of the C exits the bottom of the tower.



The above shows the towers arranged with individual reboilers and condensers but this isn't required and alternative arrangements may be developed. If we thermally integrate the two towers by using a single reboiler and condenser, we arrive at an arrangement (Figure 3) often referred to as a "fully thermally coupled" or "Petlyuk" column, after the noted Russian academician who wrote extensively on the thermodynamics of separating multi-component mixtures. The DWC is the prefractionator inside the second tower so the prefractionator and the main column are a single vessel.





The thermal inefficiency has been eliminated, leading to a significant energy saving of about 30% for a typical design and can reach 50% or 60% for unconventional ones.

22 Key Learnings

Pump types

Fan laws

Inefficiency contributors to pump overall efficiency

Centrifugal pump characteristics – head, flow, efficiency, specific speed

Pump power estimation – know formula

Effect of drag reducers

Types of variable speed drive options

Expanders

Organic Rankine Cycle

Use of Mollier charts

Factors affecting efficiency of reciprocating and centrifugal compressors

Flare gas recovery

Optimisation applications

Energy efficiency associated with TEG dehydration

Gas turbine thermodynamics – Brayton Cycle

Simple, combined gas turbine cycles and combined heat and power

Factors affecting gas turbine efficiency

Use of gas turbine ISO rating charts

Life of field electrical load list

Fired heater principles

Energy usage in fractionators

23 Appendix – Fractionator Conceptual Design

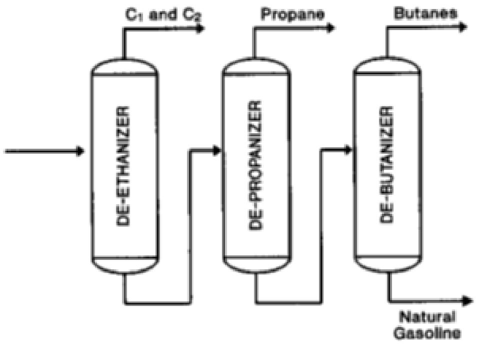


Figure 1: Series of Fractionator Towers

In hydrocarbon processing, it is often necessary to separate components in the raw feed stream into saleable products. This is can be achieved using process equipment known as fractionation columns. Each fractionation column can produce one refined component stream from a mixture of the others, therefore a number of sequential columns may be necessary depending on the number of components present in the initial feed (Figure 1).

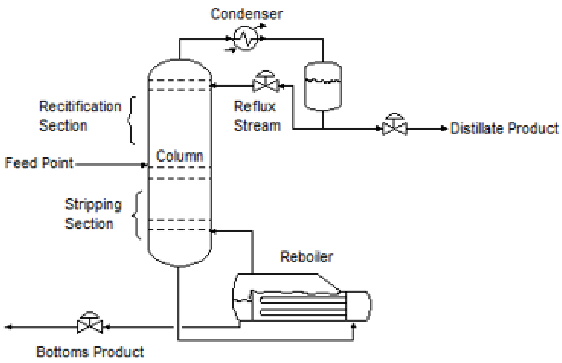


Figure 2: Elements of a Fractionator

A fractionation column consists of the following:

1.Column with trays: each tray is where vapour-liquid equilibration is allowed and where separation occurs.

2.Feed point: where the material to be separated enters, the position of this depends on the characteristics of the feed material.

3.Condenser: this condenses the vapour product from the column.

1. If the distillate product leaves as a saturated vapour it is a partial condenser (reflux is still condensed).
2. If the distillate product leaves as a saturated liquid it is a total condenser

4.Reflux Stream: this is condensed product which is sent back down the column to allow vapour-liquid contact at the trays, which in turn enhances separation.

5.Distillate Product: this is the top product of the column which can either be a saleable product or sent on for further processing.

6.Reboiler: this operates at the bottoms product bubble-point and sends vapourised components back up the column to allow for vapour-liquid equilibration and separation.

7.Bottoms Product: this is can be a saleable product or can be sent on for further processing.

The section of the column above the feed point is referred to as the rectification section, while the section below is known as the stripping section.

Rigorous mechanical design of fractionators is a complex task due to the number of variables involved and specialty knowledge required. Therefore this is usually completed by the equipment vendor, however, specification by the operating engineer is made through more empirical, short-cut calculations. One such method is the Fenske-Underwood-Gilliland method, which is presented in the following slides. The overall procedure is given below:

1. Specify feed stream characteristics: composition, flow rate, temperature, pressure.
2. Carry out a mass balance on the column based on feed composition and product specifications.
3. Calculate condenser temperature.
4. Calculate bubble-point pressure of the distillate product.
5. Assume no pressure difference across column, and calculate bubble-point temperature of the bottoms product.
6. Calculate minimum number of trays and minimum reflux rate for specified product purities.
7. Calculate actual number of trays and reflux rate.
8. Carry out column energy balance, and determine condenser and reboiler duties.
9. Size the column.

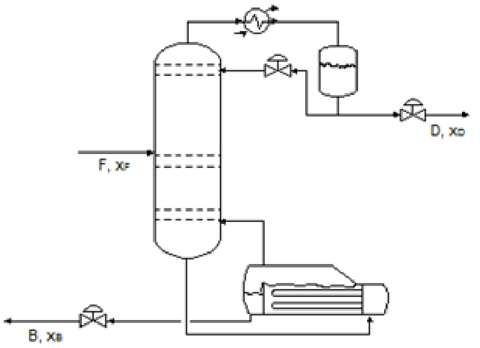


Figure 3: Mass Balance on Fractionator

Once the feed and product specifications are defined, a mass balance is completed on the fractionating column. From Figure 3, the following relationships can be derived (Equation 1 and 2):

$$\frac{D}{F} = \frac{x_F - x_B}{x_D - x_B} \qquad \text{Equation 1}$$

$$\frac{B}{F} = \frac{x_F - x_D}{x_B - x_D} \qquad \text{Equation 2}$$

As there are usually a large number of components, it is common practice to define the "heavy" and "light" keys for the system. The heavy key is the highest boiling component of the distillate stream, and the light key is the lowest boiling component present in the bottoms stream. An assumption is made that any components lighter than the heavy key is present in the bottoms and no components heavier than the light key are present in the distillate.

B Molar Flow of Bottoms Product
D Molar Flow of Distillate Product
F Molar Flow of Feed
x_B Component x Mole Fraction in Bottoms
x_D Component x Mole Fraction in Distillate
x_F Component x Mole Fraction in Feed

The next step is to determine the condenser operating temperature, the bubble/dew points of the bottoms and distillate streams, and the operating pressure of the column.

The condenser temperature is based on the cooling medium available (e.g. air, water) and an assumed temperature approach (e.g. 10° C) for the heat exchange equipment to be used.

Calculate the bubble/dew point of the distillate at condenser temperature

- For a liquid distillate product, the bubble point pressure is calculated at the condenser temperature from flash calculations.
- In the case of a vapour distillate product, the dew point pressure is calculated.

A margin of error for additional pressure drops is usually added to the calculated bubble/dew point pressure to give the operating pressure of the column.

Assuming no pressure drop across the column, calculate the bubble point of the bottoms stream at the operating pressure of the column.

The Fenske equation (Equation 3) can then be used to determine the minimum number of theoretical plates required for separation.

– For a total condenser, it is applied between the top plate of the column and the reboiler.
– For partial condensers, it applies between the distillate product and reboiler.

$$S_{\mathrm{m}} = \frac{\log\left[\left(\frac{x_{\mathrm{LK}}}{x_{\mathrm{HK}}}\right)_{\mathrm{D}}\left(\frac{x_{\mathrm{HK}}}{x_{\mathrm{LK}}}\right)_{\mathrm{B}}\right]}{\log \alpha_{\mathrm{avg}}} \qquad \text{Equation 3}$$

x_{LK} Mole Fraction Light Key
x_{HK} Mole Fraction Heavy Key
D Distillate
B Bottoms
S_m Minimum Number of Theoretical Stages
α_{avg} Relative Volatility at Average Tower Conditions

All the parameters of the equation are known apart from the relative volatility, α_{avg} – this is defined as the ratio of the K-value of the light key divided by the K-value of the heavy key at the average temperature of the column and is a gauge of the ease of separation. The average temperature for a total condenser is calculated as the average of the top plate temperature and the reboiler temperature.

In the design of a fractionating column, there is a trade-off between the number of stages and the reflux rate - This is shown graphically in Figure 5.

At minimum reflux rate, infinite theoretical stages are required for separation. At minimum stages, infinite reflux is required.

While these two points are not practical as operating points, it is useful to see the correlation between stages and reflux rate in designing the tower as there are infinite combinations which will be effective.

However, it is often desired to minimise the reflux rate to save on the heating and cooling costs of the reboiler and condenser – therefore a rule of thumb for reflux rate is around 1.25 × minimum.

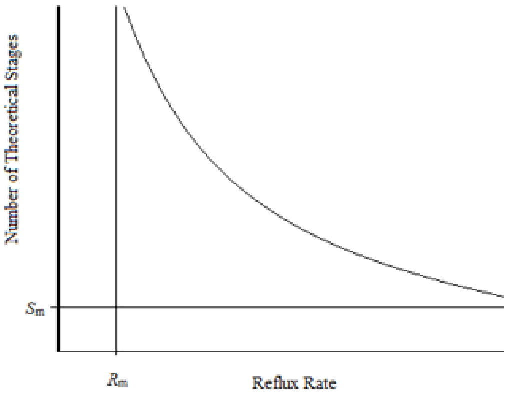


Figure 4: Theoretical Stages vs. Reflux

The next step is to calculate the minimum reflux rate required. There are two equations used in calculating the minimum reflux flow rate, Equation 4 and Equation 5:

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \varphi} = F(1-q) \qquad \text{Equation 4}$$

$$\sum_{i=1}^{i=c} \frac{\alpha_i d_i}{\alpha_i - \varphi} = L_m + D \qquad \text{Equation 5}$$

α_i Component Relative Volatility w.r.t. Heavy Key (K_i/K_{HK})
φ Constant
d_i Component Mole Fraction in Distillate
f_i Component Mole Fraction in Feed
L_m Minimum Reflux Rate
q Total Heat Required to Convert One Mole Feed to Saturated Vapour

- For bubble-point feed, q=1
- For dew-point feed, q=0
- For 2-phase feed, 0<q<1

Solving Equation 4 for a value of the constant φ is an iterative process, which requires use of a solver such as goalseek in Microsoft Excel. Once φ is found it can be used to solve equation 5 for the reflux rate.

The Gilliland correlation is then used to calculate the actual number of stages (and hence trays) required by the tower. This equation relates the number of theoretical stages to carry out separation with the actual reflux rate of any given tower. For tray-type towers, the equation takes the form given in Equation 6. The relation is shown graphically in Figure 5.

$$Y = 0.75(1 - X^{0.5668}) \qquad \text{Equation 6}$$

$$X = \frac{R - R_m}{R + 1}$$

$$Y = \frac{S - S_m}{S + 1}$$

R Actual Reflux Ratio

S Actual Number of Theoretical Stages

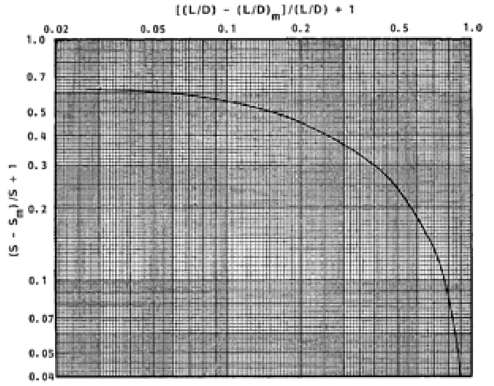


Figure 5: Gilliland Relation Chart

Assuming a value of the actual reflux rate of 1.25 times minimum, *X* can be calculated and hence Equation 6 can be solved for *Y*. Finally, the theoretical stages can be calculated from the value of *Y*.

The actual number of trays required for the fractionation column is determined by converting the theoretical stages calculated by the Gilliland method into an actual tray value using a tray efficiency value. A commonly used relation for the tray efficiency for fractionators and absorbers is that of O'Connell, given in Figure 6.

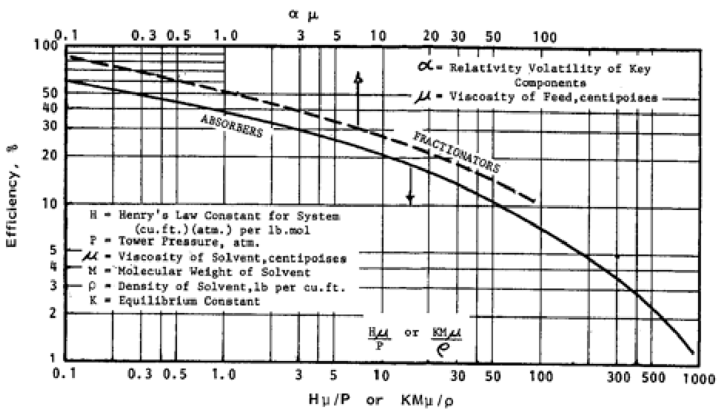


Figure 6: O'Connell Relation

The reboiler will usually count as the first stage of the tower, therefore:

$$T_n = \frac{S - 1}{\eta} \qquad \text{Equation 7}$$

T_n Number of Actual Trays

η Tray Efficiency

Finally, an energy balance is carried out on the tower to determine the duties of the condenser and reboiler. From Figure 3, the overall energy balance can be written as (Equation 8):

$$Q_B + Q_C = h_D D + h_B B + h_F F \qquad \text{Equation 8}$$

In this balance there are two unknown quantities: the duties of the reboiler and condenser. The duty for a total condenser is as follows (Equation 9):

$$Q_C = V_1(h_D - h_1) \qquad \text{Equation 9}$$

For a partial condenser, it is as follows (Equation 10):

$$Q_C = \mathrm{D}(h_D - h_1) + R(h_R - h_1) \qquad \text{Equation 10}$$

V_1 is the vapour leaving the top tower of the column, this can be calculated as the reflux flow plus the distillate product flow. The $(h_D - h_I)$ value represents the latent heat of the vapour exiting the column. For partial condensers the non-condensed distillate stream is also included in the balance.

The duty for the reboiler can then be calculated from the remaining values of the overall energy balance, all that is required is the enthalpy values of the feed, bottoms and distillate fluids. These can be calculated from specific heats.

To demonstrate use of the method, the following is a sample calculation. The problem statement is as follows:

A depropaniser with feed stream composition given in Table 1 is to be designed to recover 85% of the C2 fraction in the distillate product and 99% of the iC4 fraction in the bottoms product. Other design factors relevant to the design are given in Table 2. Specify the column parameters.

Table 1: Feed Composition

Component	Mole Fraction
C_2	0.4
C_3	10.5
iC_4	69.0
nC_4	20.1

Table 2: Other Design Factors

Ambient Air Temperature	30°C
Condenser Temperature Approach	5°C
Condenser Pressure Drop	300 kPa
Condenser Type	Total

The specification is to recover 85% of C2 at the top and 99% of iC4 at the bottom. As there are more than two components in the system we need to define the "heavy" and "light" keys. The heavy key is the heaviest component by volatility in the distillate product, while the light key is the lightest component in the bottoms product.

In this case we take isobutane to be the heavy key and propane to be the light key. It is assumed that the other two components partition perfectly in the distillate and bottoms

Using Equation 1 or Equation 2, the column product flows can be determined - the mass balance is as follows based on 100 mol·hr-1 of feed (Table 3):

Table 3: Fractionator Mass Balance

Component	Feed	Distillate, D		Bottoms, B	
	Moles	Moles	%	Moles	%
C_2	0.4	0.4	4	0.0	0.0
C_3	10.5	8.9	89	1.6	1.8
iC_4	69.0	0.7	7	68.3	75.9
nC_4	20.1	0.0	0.0	20.1	22.3
Total	100	10	100	90	100

The condenser temperature is dependent on the cooling medium – in this case we have cooling air at a temperature of 30° C and temperature approach of 5° C. Therefore the distillate product will be at 35° C when it leaves the condenser. Assuming a total condenser also, flash calculations with corresponding K-values are used to calculate the bubble-point pressure of the distillate at this temperature. Here process modelling software has been used to obtain the VLE data:

Condenser Temperature: 35° C
Distillate Bubble Point Pressure: 13 bar
The column operating pressure is largely dictated by the condenser pressure, however the pressure drop of the condenser is also known (300 kPa) and therefore must be added.
Column Operating Pressure: 13.3 bar
Thus the vapour leaving the top of the column (top plate) must be at its dew-point temperature at column pressure.
Top Plate Temperature: 38° C.
Assuming no column pressure difference, the reboiler temperature will be that of the bottoms bubble-point.
Reboiler Temperature: 85° C
Optimum separation is achieved when the feed enters at bubble-point, therefore:
Feed Temperature: 74° C

All the parameters of the equation are known apart from the relative volatility, α_{avg} – this is defined as the ratio of the K-value of the light key divided by the K-value of the heavy key at the average temperature of the column. The average temperature for a total condenser is calculated as the average of the top plate temperature and the reboiler temperature.

Top Plate Temperature: 38° C
Reboiler Temperature: 85° C

Average Temperature: $\frac{38+85}{2} = 61.5°\ \text{C}$

At average temperature and operating pressure of column:

$\mathbf{K_{C3}} = 1.66$

$\mathbf{K_{iC4}} = 0.88$

$\boldsymbol{\alpha}_{\mathbf{avg}} = \frac{1.66}{0.88} = 1.89$

Therefore using Equation 3,

$$S_m = \frac{\log\left[\left(\frac{x_{LK}}{x_{HK}}\right)_D \left(\frac{x_{HK}}{x_{LK}}\right)_B\right]}{\log \alpha_{avg}}$$

$$S_m = \frac{\log\left[\left(\frac{8.9}{0.7}\right)_D \left(\frac{75.9}{1.8}\right)_B\right]}{\log 1.89}$$

$$S_m = 9.9$$

Minimum of 9.9 stages for separation.

As we have previously assumed that the feed enters at bubble-point, we can take q=1 – therefore the RHS of Equation 4 becomes zero. The next step is to solve Equation 4 for a value of the constant φ.

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \varphi} = F(1-q)$$

The value of φ will always be between relative volatilities of the heavy and light keys, setting φ=1.6;

Table 4: First Iteration of Equation 4

Component	f_i	K-value @ 61.5 & 13.3. bar	α_i	$\alpha_i f_i$	$\alpha_i - \varphi$	$\frac{\alpha_i f_i}{\alpha_i - \varphi}$
C_2	0.4	3.8	4.2	1.7	2.6	0.653
C_3	10.5	1.6	1.8	18.9	0.2	94.5
iC_4	69.0	0.9	1.0	69.0	-0.6	-115
nC_4	20.1	0.6	0.67	13.5	-0.93	-14.5
					Total	-34.3

From Table 4, this value of the constant did not result in zero on the RHS of Equation 4, therefore the calculation must be repeated, however from using goalseek, φ=1.626.

We now need to calculate the actual number theoretical stages of stages required for the tower, this is determined by the Gilliland equation:

$$Y = 0.75(1 - X^{0.5668})$$

$$X = \frac{R - R_m}{R + 1} \qquad Y = \frac{S - S_m}{S + 1}$$

A typical value for the actual reflux rate is 1.25 times minimum:

$$R = 1.25R_{\mathrm{m}} = 10.45$$

Calculating X:

$$X = \frac{10.45 - 8.36}{10.45 + 1} = 0.182$$

Calculating Y:

$$Y = 0.75(1 - 0.182^{0.5668}) = 0.46$$

Knowing the minimum theoretical stages (S_{m}=9.9), we rearrange and solve for S:

$$0.46 = \frac{S - 9.9}{S + 1}$$

$$\boldsymbol{S = 19.2} \textbf{ actual stages}$$

We now wish to convert this value to a number of tower trays in reality.

A typical value for tray efficiency in a fractionating column is 65%, additionally the reboiler will usually count as the first stage of the tower, therefore:

$$T_{\mathrm{n}} = \frac{S - 1}{\eta}$$

$$T_{\mathrm{n}} = \frac{19.2 - 1}{0.65}$$

$$\boldsymbol{T_{\mathrm{n}}} = 28 \text{ trays}$$

The tower will require 28 trays.