Upstream Process Engineering Course

5. Gas Compression

Units and Standard Conditions

Common Gas Units

mmscf, MMSCF, MMCF - one million standard cubic feet

BE CAREFUL!

In English units MM means million

In metric units M means “mega” or 10^6

Standard Conditions (Pressure & Temperature)

English	P= 14.7 psia,	T = 60° F (15.56° C) = 520° R
Metric	P = 101.325 Kpa (1 atm),	T = 15 ° C = 288 K
Normal	P = 101.325 Kpa (1 atm),	T = 0° C = 273K

Volume to Mass conversions

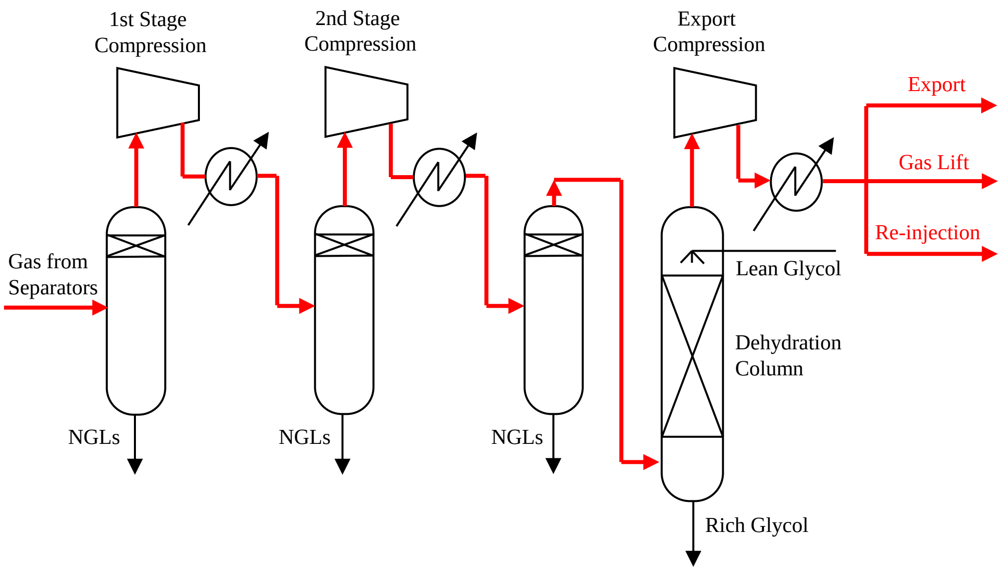
English:	379 ft^3/lb-mol	2626 lbmol/MMscf
Metric:	23.96 m^3/kmol	41470 kmol/10^6 std m^3

1 MMscfd = 110 lbmol/hr

1 x 10^6 std m^3/d = 1739 kmol/h

Gas Handling

- Gas Management
  - compress to export gas to market
  - reinject for reservoir support
  - reinject to improve recovery
    - miscible
    - WAG (Water alternating gas benefits some reservoirs)
  - compress to recover valuable NGLs
  - use as fuel
  - Flare
  - reinject into separate formation

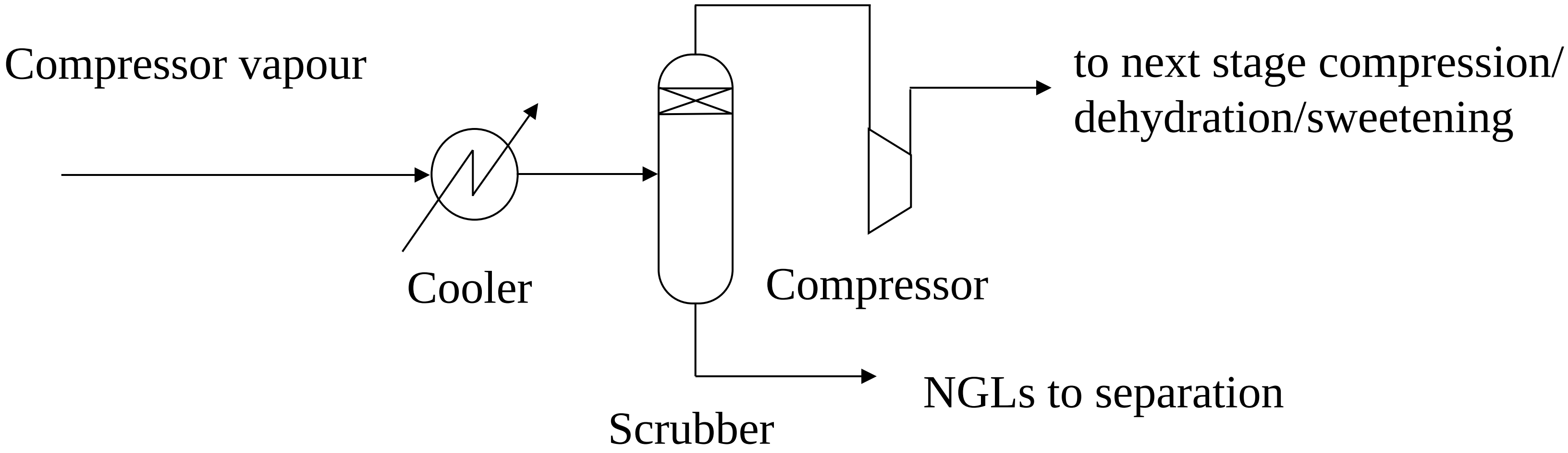


- Configuration is application specific

NGLs – Natural Gas Liquids/Condensate

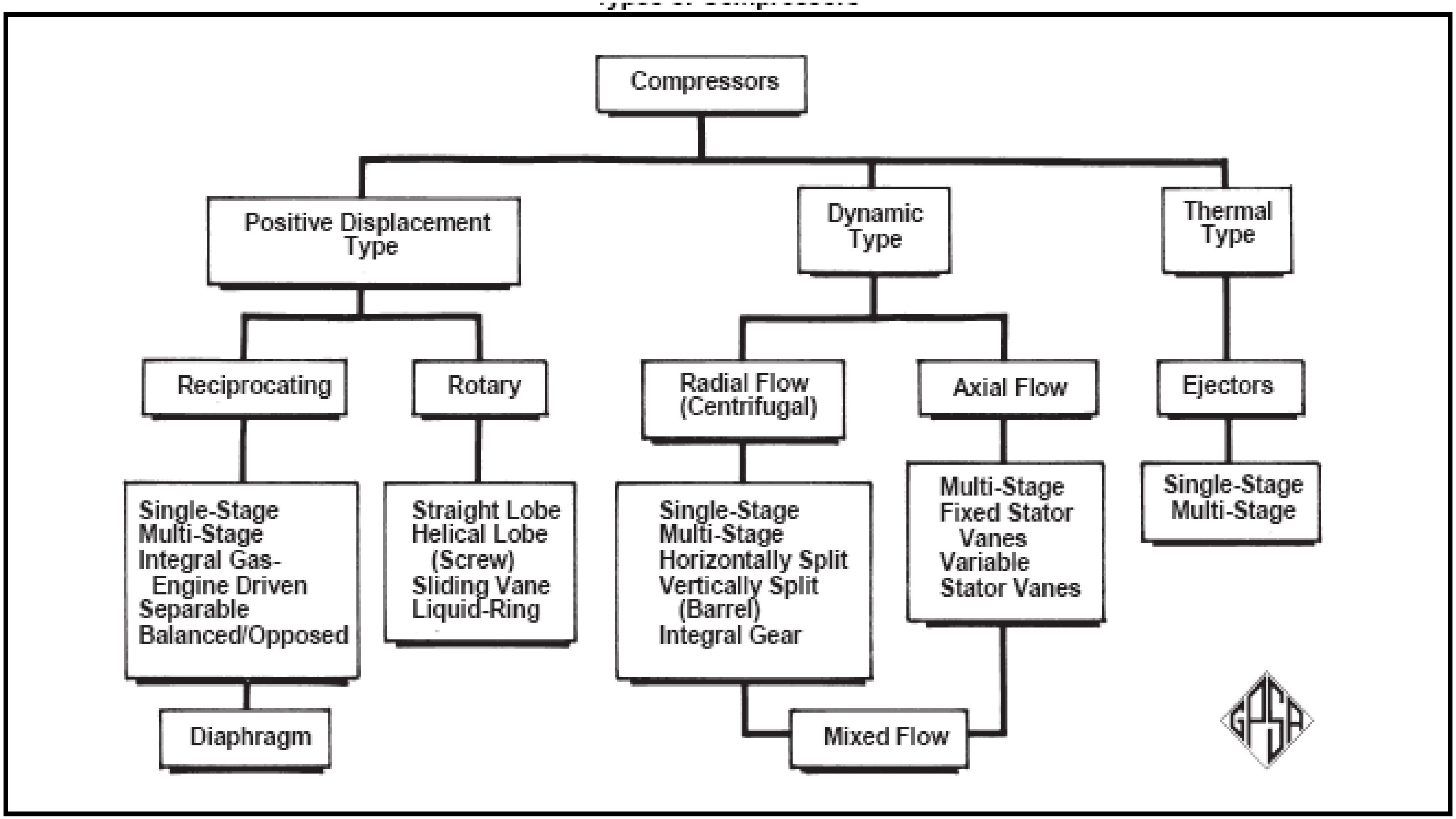
Don’t confuse with LNG – Liquefied Natural Gas

Cooling / Liquid Knock Out / Compression



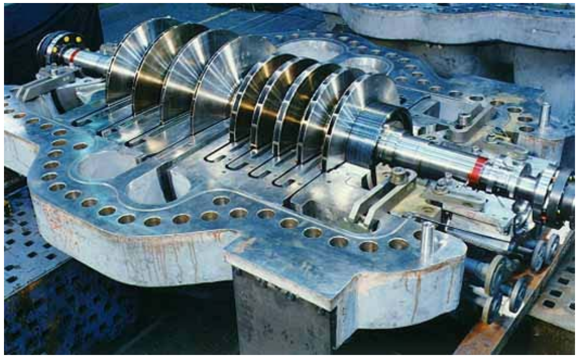
- Compression will heat the gas and therefore requires (interstage/after) cooling to reduce power and also to avoid exceeding discharge temperature limits
- Cooling of the saturated vapour to compressor inlet conditions causes condensation of heavier components and water
- Leaner gas from scrubber with reduced cricondenbar
- Cooler outlet temperature typically 25-30 DegC set by hydrates, cooling medium temperature, and heat of compression from upstream compressor
- NGL (Natural gas liquids) recycle can significantly affect system heat and mass balance

Types of Compressors

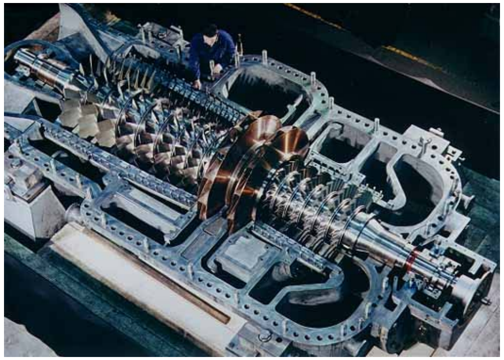


Compressors

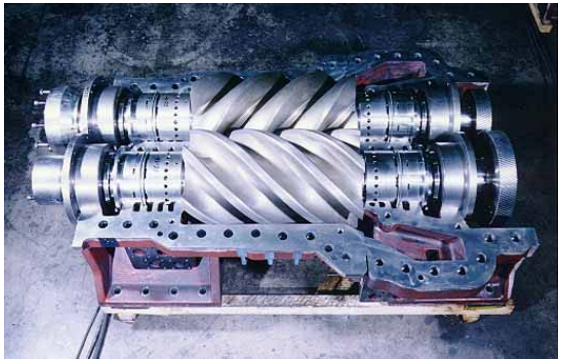
Four most common types



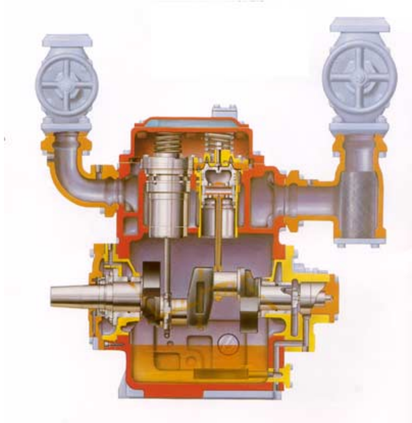
Centrifugal Compressor



Axial Compressor



Screw Compressor



Reciprocating Compressors

Compressors

Rotary compressors cover lobe-type, screw-type, vane-type, and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

The dynamic types include radial-flow (centrifugal), axial flow, and mixed flow machines. They are rotary continuous flow compressors in which the rotating element (impeller or bladed rotor) accelerates the gas as it passes through the element, converting the velocity head into static pressure, partially in the rotating element and partially in stationary diffusers or blades.

Ejectors are “thermal” compressors that use a high velocity gas or steam jet to entrain the inflowing gas, then convert the velocity of the mixture to pressure in a diffuser.

Centrifugal Compressors

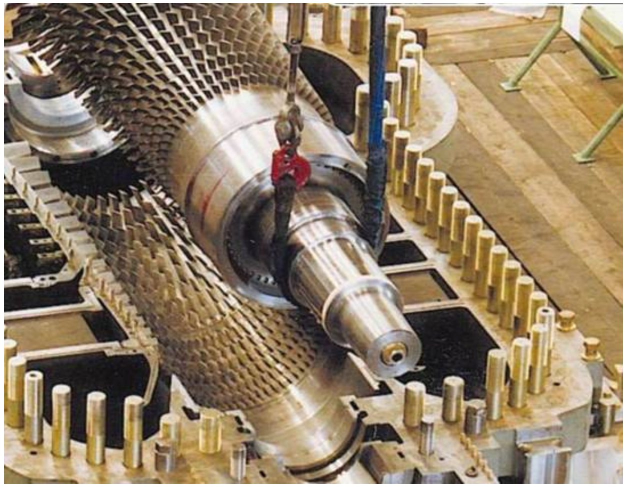
Centrifugal compressors use a rotating disk or impeller in a shaped housing to force the gas to the rim of the impeller, increasing the velocity of the gas. A diffuser (divergent duct) section converts the velocity energy to pressure energy.

They are primarily used for continuous, stationary service in industries such as oil refineries, chemical and petrochemical plants and natural gas processing plants.

Their application can be from a few kW to multiple 1000kW. With multiple staging, they can achieve extremely high output pressures greater than 70MPa.

Axial-Flow Compressors

Axial-flow compressors are dynamic rotating compressors that use arrays of fan-like aerofoils to progressively compress the working fluid. They are used where there is a requirement for a high flow rate or a compact design.



The arrays of aerofoils are set in rows, usually as pairs: one rotating and one stationary. The rotating aerofoils, also known as blades or rotors, accelerate the fluid. The stationary aerofoils, also known as stators or vanes, decelerate and redirect the flow direction of the fluid, preparing it for the rotor blades of the next stage. Axial compressors are almost always multi-staged, with the cross-sectional area of the gas passage diminishing along the compressor to maintain an optimum axial Mach number

Axial compressors can have high efficiencies; around 90% polytropic at their design conditions. However, they are relatively expensive, requiring a large number of components, tight tolerances and high quality materials. Axial-flow compressors can be found in medium to large gas turbine engines, in natural gas pumping stations, and within certain chemical plants.

Reciprocating Compressors

Reciprocating compressors use pistons driven by a crankshaft. They can be either stationary or portable, can be single or multi-staged, and can be driven by electric motors or internal combustion engines.

Reciprocating compressors well over 1,000 hp (750 kW) are commonly found in large industrial and petroleum applications. Discharge pressures can range from low pressure to very high pressure (>18000 psi or 180 MPa).

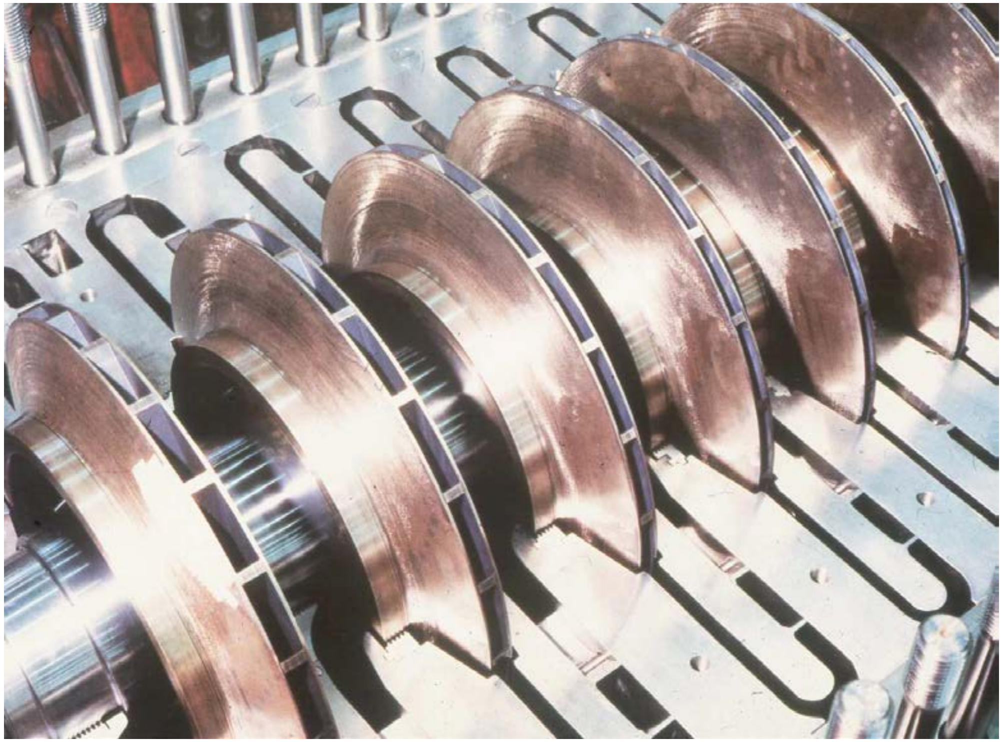
In certain applications, such as air compression, multi-stage double-acting compressors are said to be the most efficient compressors available, and are typically larger, and more costly than comparable rotary units

Rotary Screw Compressors

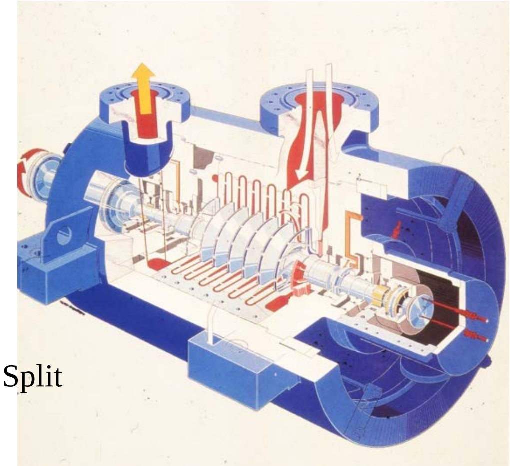
Rotary screw compressors use two meshed rotating positive-displacement helical screws to force the gas into a smaller space. These are usually used for continuous operation in commercial and industrial applications and may be either stationary or portable.

Rotary screw compressors are commercially produced in Oil Flooded, Water Flooded and Dry type.

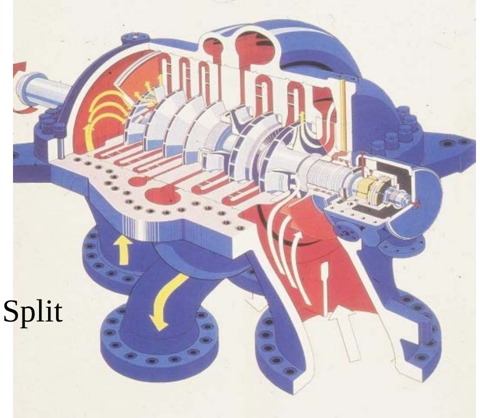
Centrifugal Compressor



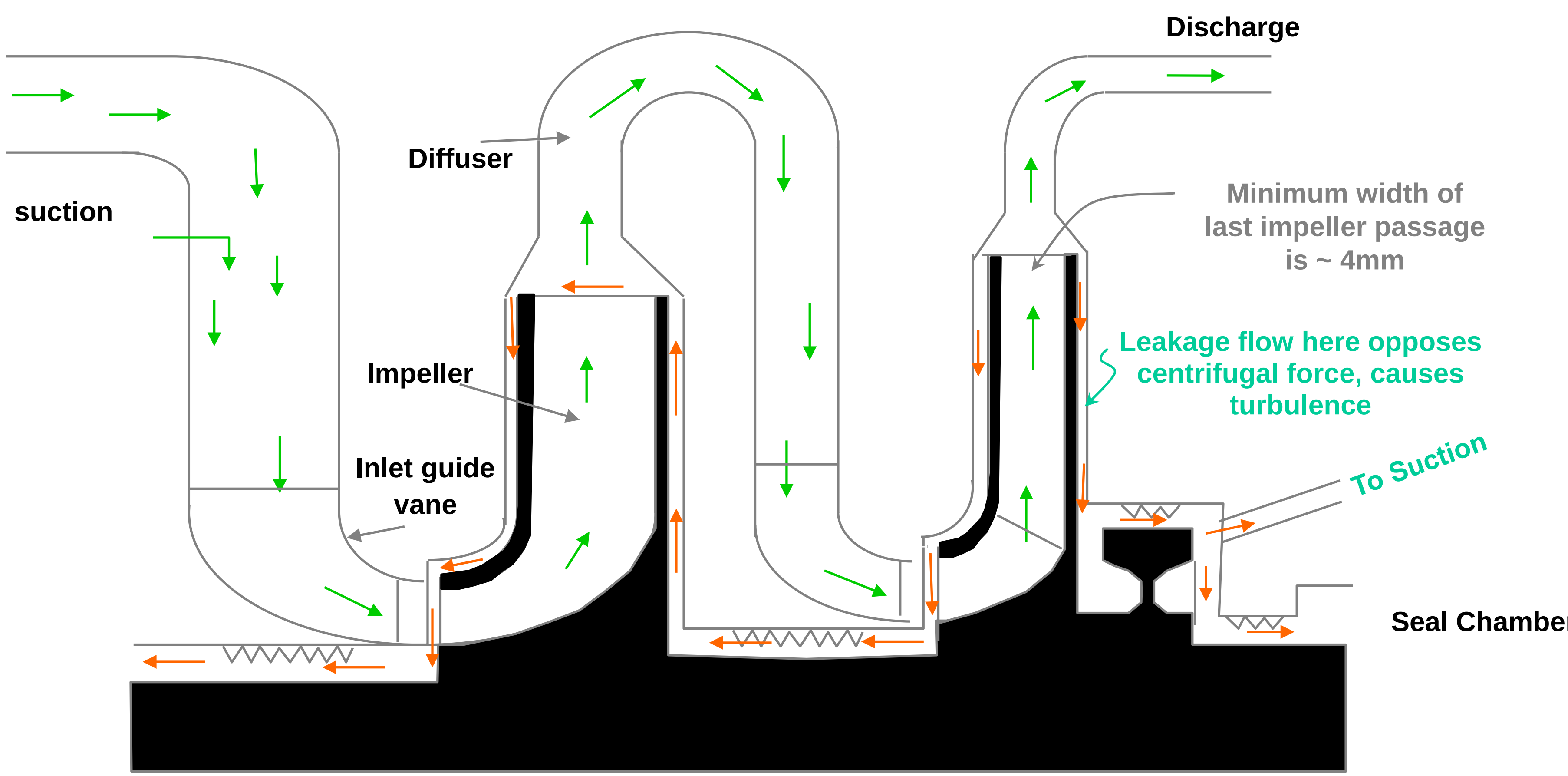
Vertical Split



Horizontal Split



Flow through Centrifugal Compressor



End thrust generated by Impellers **IS** **Opposed by the balance Piston**

Compressor Basics

Compression Ratio

The absolute ratio of discharge to suction pressure is known as the compression ratio. The discharge temperature will increase with increasing temperature ratio. The discharge temperature will also be a function of the compressed gas properties, particularly the molecular weight. Acceptable discharge temperatures are limited by the mechanical design of the compressor – materials and seals. **A rule of thumb is approximately 165 DegC.**

Intercooling

By inspection of the equation relating to the power requirements of a compressor it is evident that the colder the gas the lower the pressure. Other factors such as compressibility are important but the main benefit from cooling is reducing the volume of the gas to be compressed. Where large compression ratios loads are required interstage cooling will be required. Interstage cooling will often result in condensation of natural gas liquids (NGLs). The NGLs often contain valuable liquid products such as propane and butane hence they are recycled back to the associated separation system.

Compressor Basics

Compressor Selection

Selection of the compressor type and number of stages requires consideration of a number of factors – flow rate, compression ratio and physical properties. A rule of thumb is given in the following chart. In general reciprocating compressors are better suited to low flow, high pressure/head applications.

Centrifugal Compressors

Casing

The casing houses the rotating and static parts of the compressor. It provides pressure containment and two basic arrangements are designed;

Horizontally split and end split. The end split arrangement generally allows for higher pressure applications. To maintain the rotating elements the rotor and other components are withdrawn along the machine axis. Hence sufficient space has to be provided when laying the system out.

Horizontally split arrangement has a horizontal flange connecting the upper and lower halves of the compressor. Maintenance access in this configuration is from above hence lifting arrangements have to be considered.

Compressor Basics

Rotor

This is a steel shaft with the required arrangement of impellers.

Impellers

Impeller design requires precision aerodynamics. The rotational load of a machine turning at thousands of revs per minute requires the use of high strength steels. Like all rotating plant the system has to be designed to ensure that resonant frequencies are not experienced during normal operation.

Flow Path

Flowing in from the inlet nozzle the gas will pass through inlet guide vanes. These direct the gas to the first stage impeller with at efficient flow angles. The impellers impart velocity to the gas as the gas moves radially outwards along the blades. The gas leaves the impeller and enters a profiled diffuser passage where the velocity (kinetic energy) in the gas is converted to pressure. The gas is returned to the next vane/impeller/diffuser arrangement and pressure is increased at each stage to achieve the required discharge pressure. At the last stage the gas is delivered to a discharge volute and exits via the discharge nozzle.

Compressor Basics

Bearings
Centrifugal compressors are fitted with journal bearings to support the weight and to position the rotating assembly within the static parts. A thrust bearing will be provided to accommodate any axial loads resulting from pressure forces.

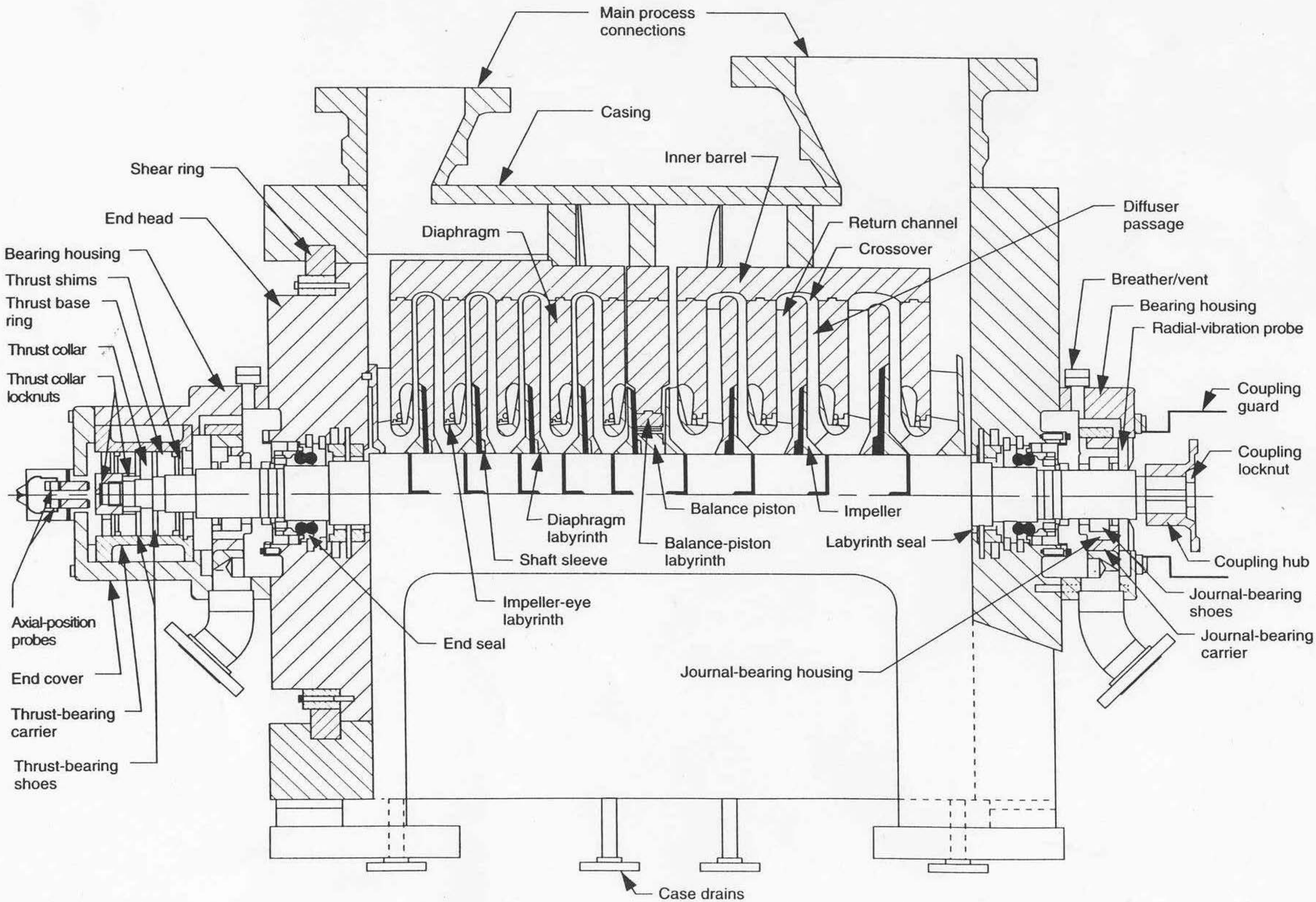
Labyrinth Seals
These minimise recirculation losses within the compressor. They can be static or dynamic and are generally a series of wedges. Sealing action is by repeated throttling across each wedge. The adjacent parts – wedge teeth and rotating part are made of soft and hard material.
Liquid Film Seals
Two adjacent seal rings are fitted to each end of the compressor rotor, a sealing fluid is introduced between the seals at a pressure higher than the duty gas. A minimal amount of seal oil flows into the compressor thus preventing process gas from escaping along the shaft. Seal design is a highly specialised area and is a critical part of a compressor design.

Compressor Basics

Dry Gas Seals

These seals are a relatively new design. Dry gas seals are non-contacting, dry-running mechanical face seals consist of a mating (rotating) ring and a primary (stationary) ring. When operating, grooves in the rotating ring generate a fluid-dynamic force causing the stationary ring to separate and create a gap between the two rings. Dry gas seals are mechanical seals but use other chemicals and functions so that they do not contaminate a process. These seals are typically used in a harsh working environment such as oil exploration, extraction and refining, petrochemical industries, gas transmission and chemical processing.

The dry gas seal has spiral grooves, with provides for lifting and maintaining separation of seal faces during operation. Grooves on one side of the seal face direct gas inward toward a non-grooved portion of the face. The gas that is flowing across the face generates a pressure that maintains a minute gap between the faces, optimizing fluid film stiffness and providing the highest possible degree of protection against face contact.



Centrifugal Compressor Nomenclature

Impeller Design - Head Generation

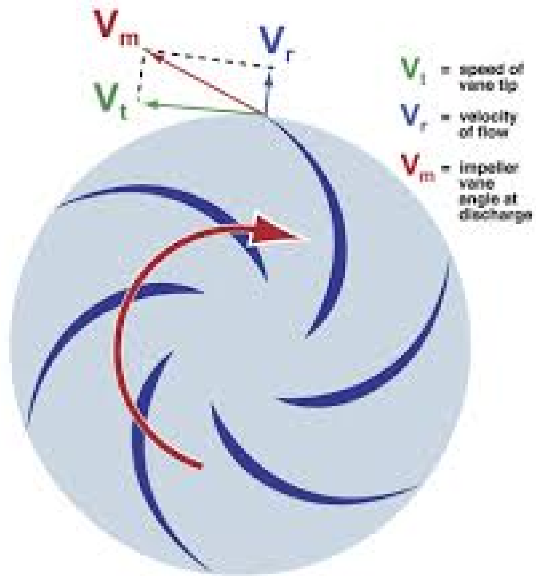


Fig. 4. Impeller discharge angle vectors

Head Generation

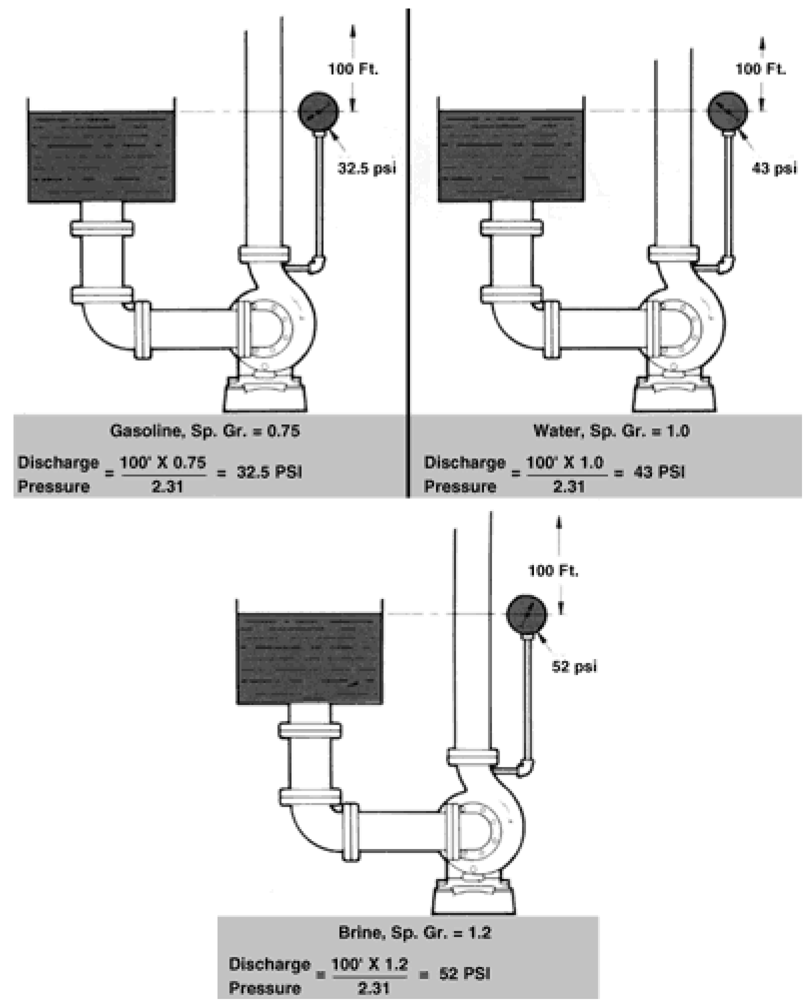
- Impeller imparts velocity
- Diffuser/Volute converts velocity to pressure head – kinetic to potential energy

Fluid Head

The pressure at any point in a liquid/gas can be thought of as being caused by a vertical column of the fluid which, due to its weight, exerts a pressure equal to the pressure at the point in question. The height of this column is called the static head and is expressed in terms of feet of fluid.

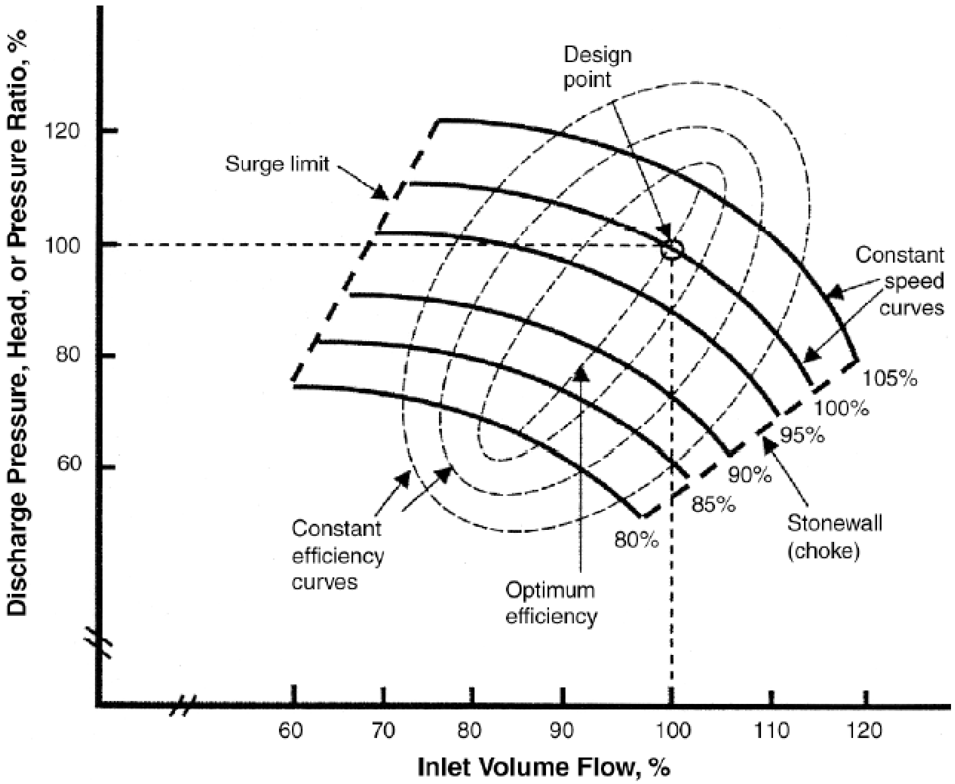
A Centrifugal compressor or pump imparts velocity to a fluid. This velocity energy is then transformed largely into pressure energy as the fluid leaves the compressor/pump. Therefore, the head developed is approximately equal to the velocity energy at the periphery of the impeller

A given compressor or pump with a given impeller diameter and speed will raise a liquid to a certain height regardless of the weight/density of the liquid.



Compressor Characteristics

The performance characteristic of a centrifugal compressor is graphically presented in the form of a family of curves – collectively known as a performance map or operating envelope. Normally the inlet volume is plotted along the x axis and the head or pressure ratio is plotted on the y axis. The surge line is shown on the left of the map. The right hand side shows the Stonewall or choke limit. This is shown for a range of compressor speeds. The slope of the curve varies with the number of stages, becoming steeper with increasing number of stages. Also indicated are compressor efficiencies. The design point is at 100% speed, optimum efficiency and has a safe margin from Surge and Stonewall.



Flow Limits

Two conditions associated with centrifugal compressors are Surge and Stonewall (choked flow). At some point on the compressor's operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. An aerodynamic instability is brought about by flow reduction, which causes stalling. Stalling can occur at the inlet to the impeller, the radial portion of the impeller and in the discharge volute. This is the surge point. Without discharge flow, discharge pressure drops until it is within the compressor's capability, only to repeat the cycle. The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor, causing damage to the rotating element, casing and bearings.

"Stonewall" or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor cannot be increased further without internal modifications.

Centrifugal Compressor Control

Centrifugal compressor controls can vary from the very basic manual recycle control to elaborate ratio controllers. The driver characteristics, process response, and compressor operating range must be determined before the correct control can be selected.

Recycle

A recycle loop is provided to adjust the suction volume flow rate to avoid surge.

Speed

One of the most efficient way to match the compressor characteristic to the required output is to change speed.

Adjustable Inlet Guide Vanes

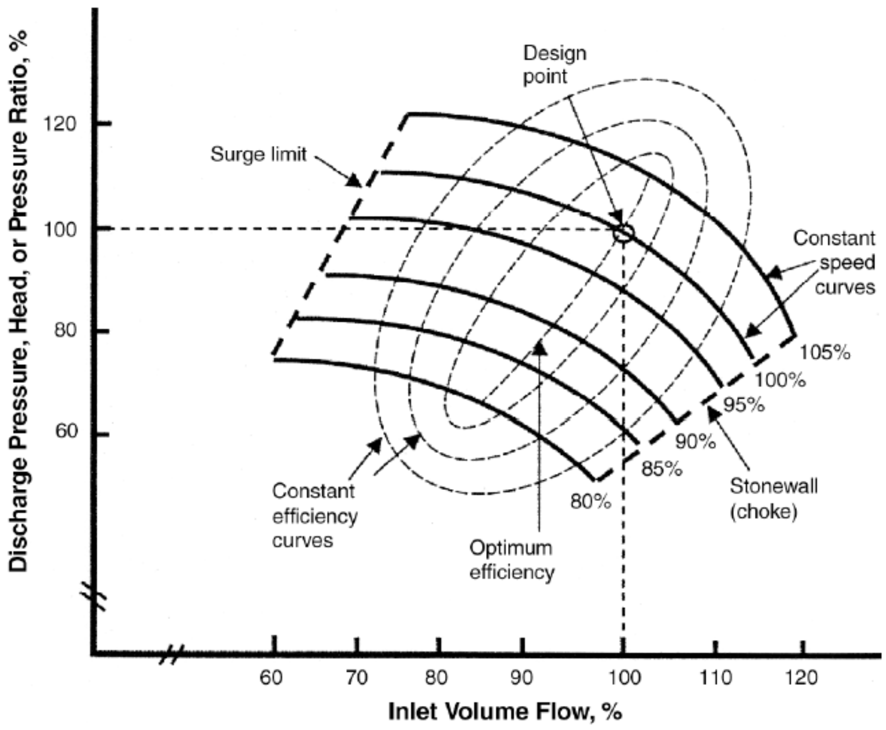
The use of adjustable inlet guide vanes is the most efficient method of controlling a constant speed compressor. The vanes are built into the inlet of the 1st stage, or succeeding stages, and can be controlled through the linkage mechanism either automatically or manually.

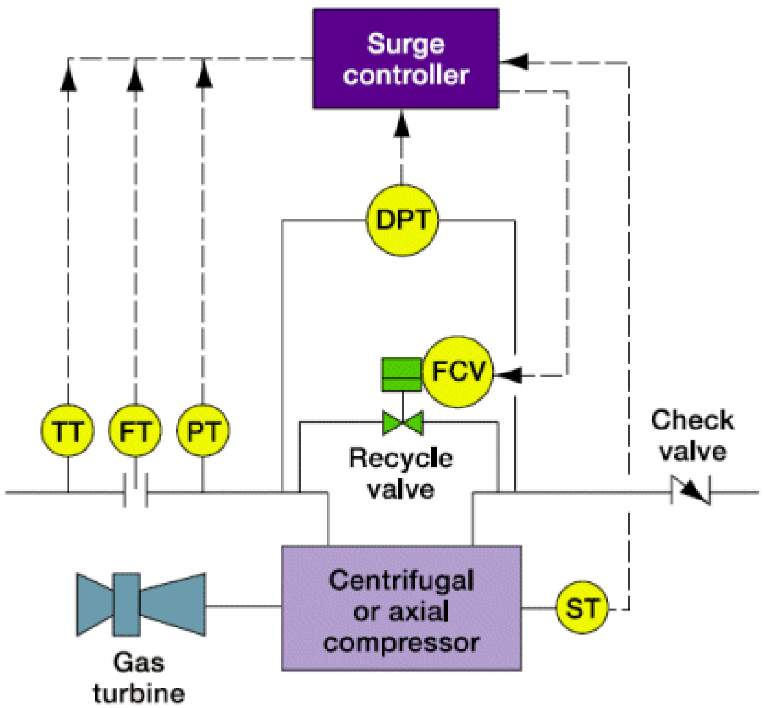
The vanes adjust the capacity with a minimum of efficiency loss and increase the stable operating range at design pressure. This is accomplished by pre-rotation of the gas entering the impeller which reduces the head-capacity characteristics of the machine.

Suction Throttling

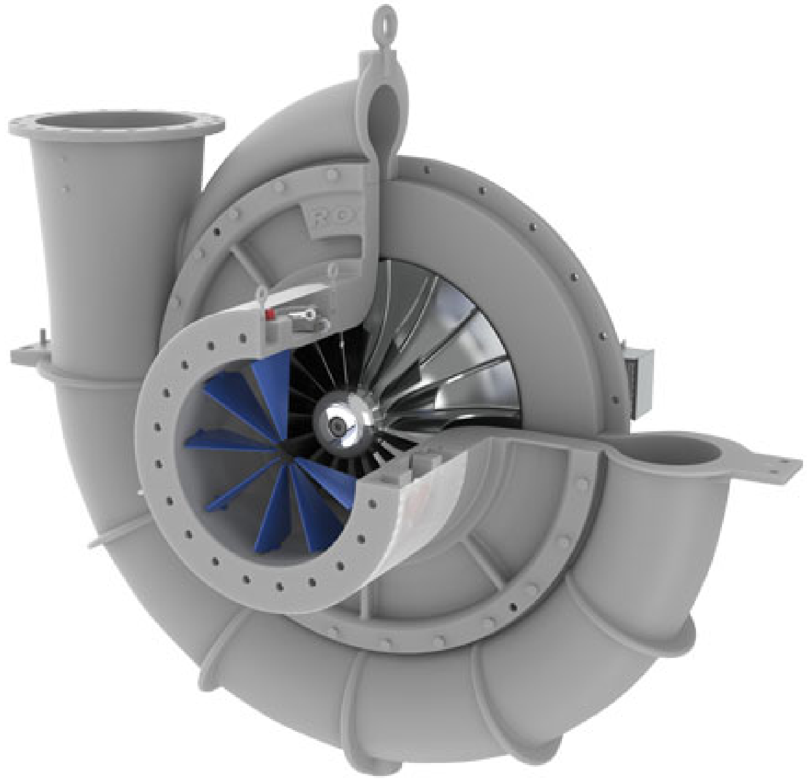
A valve is fitted in the inlet pipework enabling suction pressure and volume regulation.

Basic Anti-Surge Control





Inlet Guide Vanes



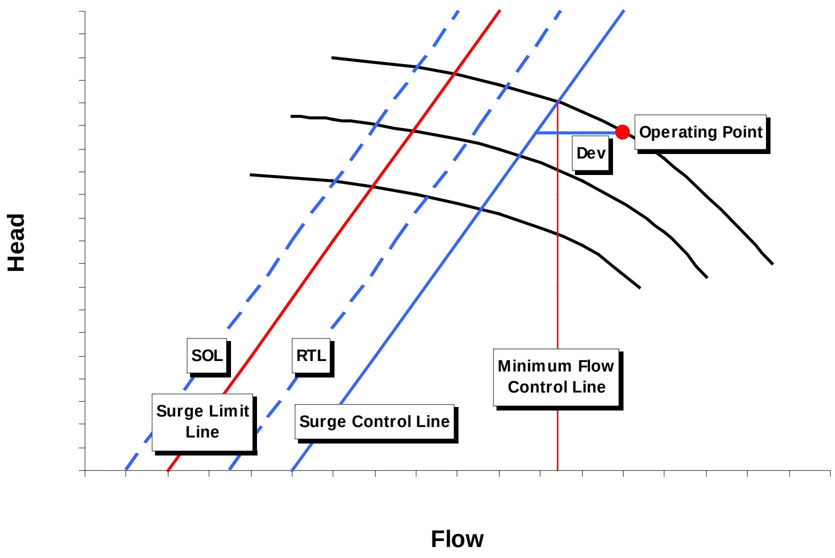
Anti-Surge Control

Anti-Surge Control Systems

Minimum Flow Control. Suitable for fixed speed machines only. Impractical and wasteful for variable speed.

Flow ΔP Control. Series of control points at differing speeds gives surge control line of the form y = mx + c. Recycle valve modulated to ensure operating point is not to the left of this line. Simple and robust.

CCC (major control system supplier) uses two additional lines Allows more efficient (closer to surge limit line (SLL)) and responsive control

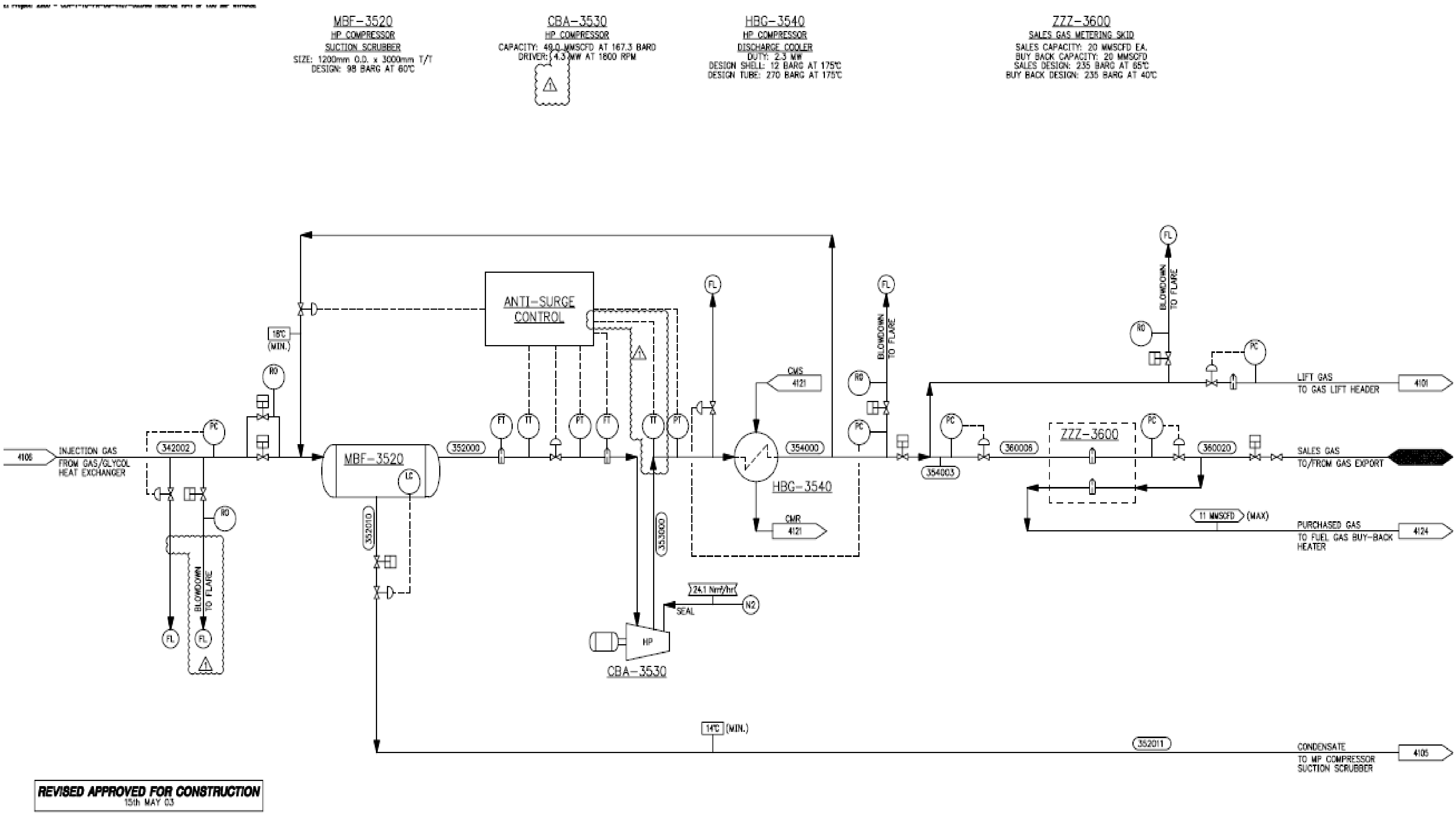


Surge Control Line (SCL). Uses PI control to manage normal load changes.

Recycle Trip Line (RTL). Aggressive control, acts as flow approaches surge line. Recycle valve opened in steps by timer delay until RTL is re-crossed when PI control is reactivated. Should never be re-set.

Safety On Line (SOL). If SOL is crossed surge is assumed to have occurred. Incident is logged and the controller moves all control lines to the right. Can be re-set (after investigation into incident cause).

Typical Anti-Surge Control



Reciprocating Compressors

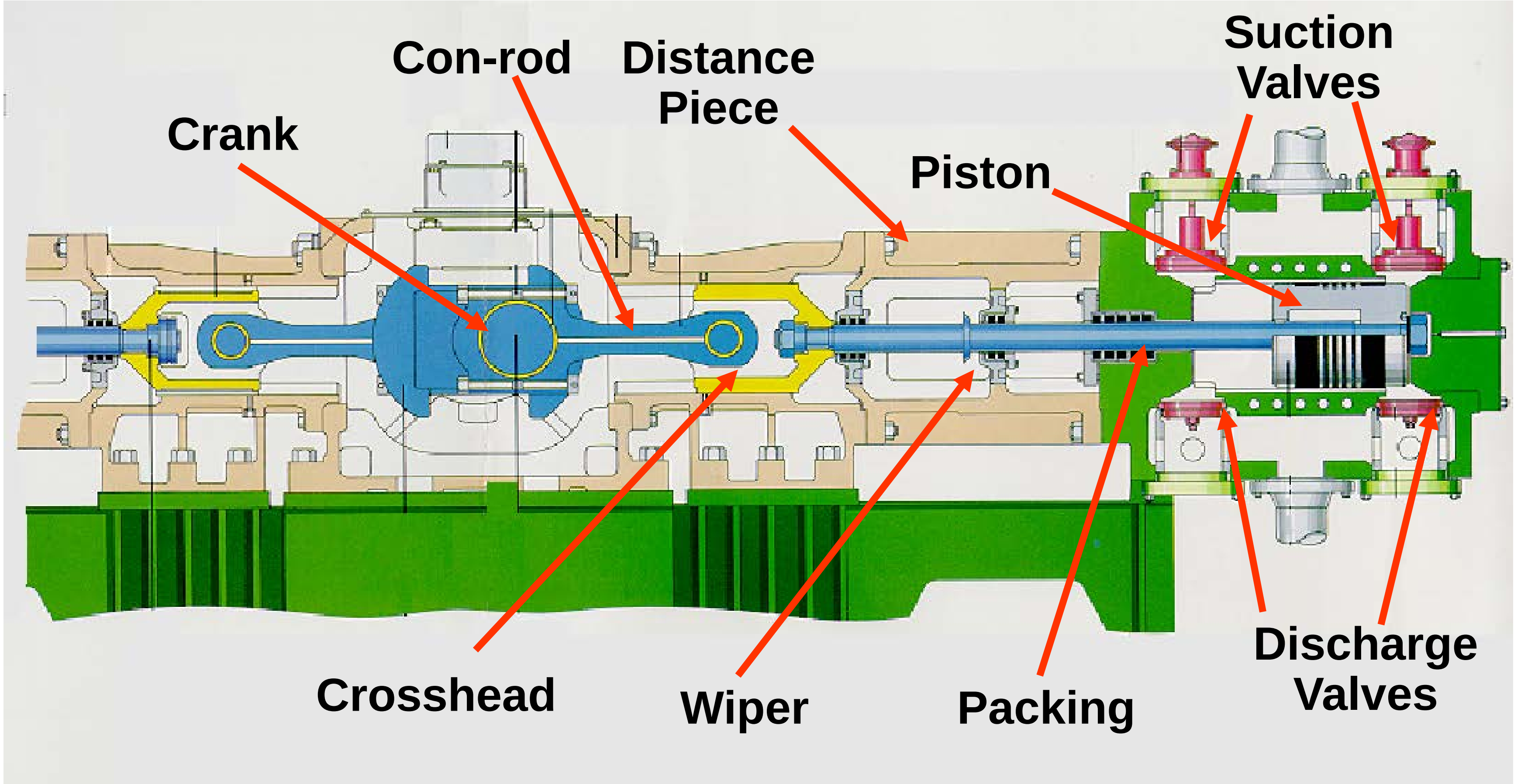
Piston reciprocating compressors are the most common type of positive displacement compressors. They are available both in single and double-acting design in various configurations.

Gas is drawn into the cylinder usually through a self-acting valve which is opened and closed by pressure difference. After compression, the gas leaves via a self-acting discharge valve. The valve is comprised of a seat, valve guard, plates and springs. The plate moves between the guard and seat aided by the springs which help to accelerate closure. The valve is fully open when held against the guard and fully closed when held against the seat. Some less common designs have cam-controlled or rotary slide valves. More detail on valves is given later.

Reciprocating compressors can be supplied as lubricated or oil-free designs. Oil-free compressors have piston rings and wear bands fitted. Trunk type oil-free compressors have dry crank cases with permanently lubricated bearings. Crosshead types have lengthened piston rods which keep oil wetted parts away from the compression space.

Other types of positive displacement compressor are oil-free labyrinth piston compressors (no piston rings are fitted, the cylinder wall to piston seal is achieved by labyrinth seals) and diaphragm compressors.

Reciprocating Compressor



Reciprocating Compressors

The compression cycle

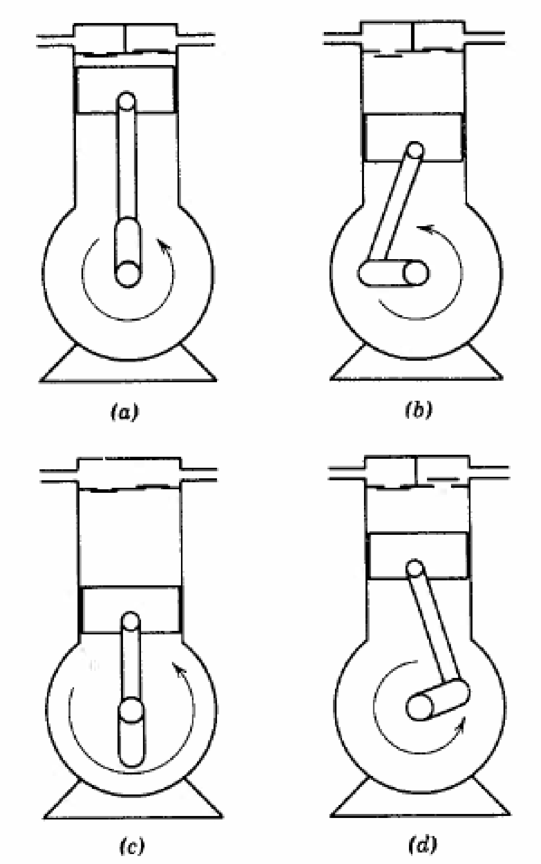
The compression cycle is shown in the adjacent figure.

a) The piston is at top dead centre with suction and discharge valves closed.

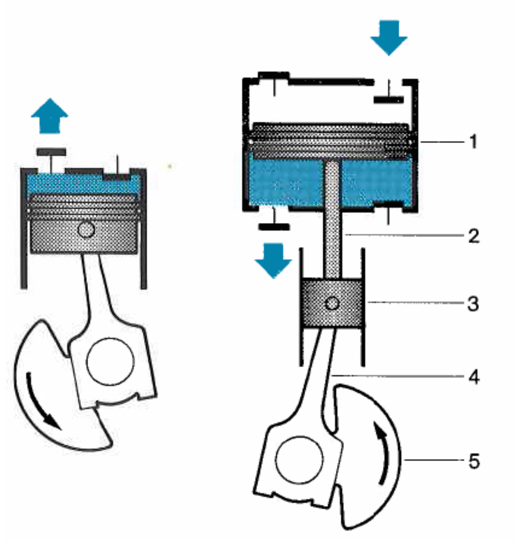
b) The piston has travelled down the cylinder and the suction valves open.

c) The piston is at bottom dead centre with suction and discharge valves closed.

d) The piston has travelled back up the cylinder and the discharge valves open.



Reciprocating Compressors



Single-acting **Double-acting**

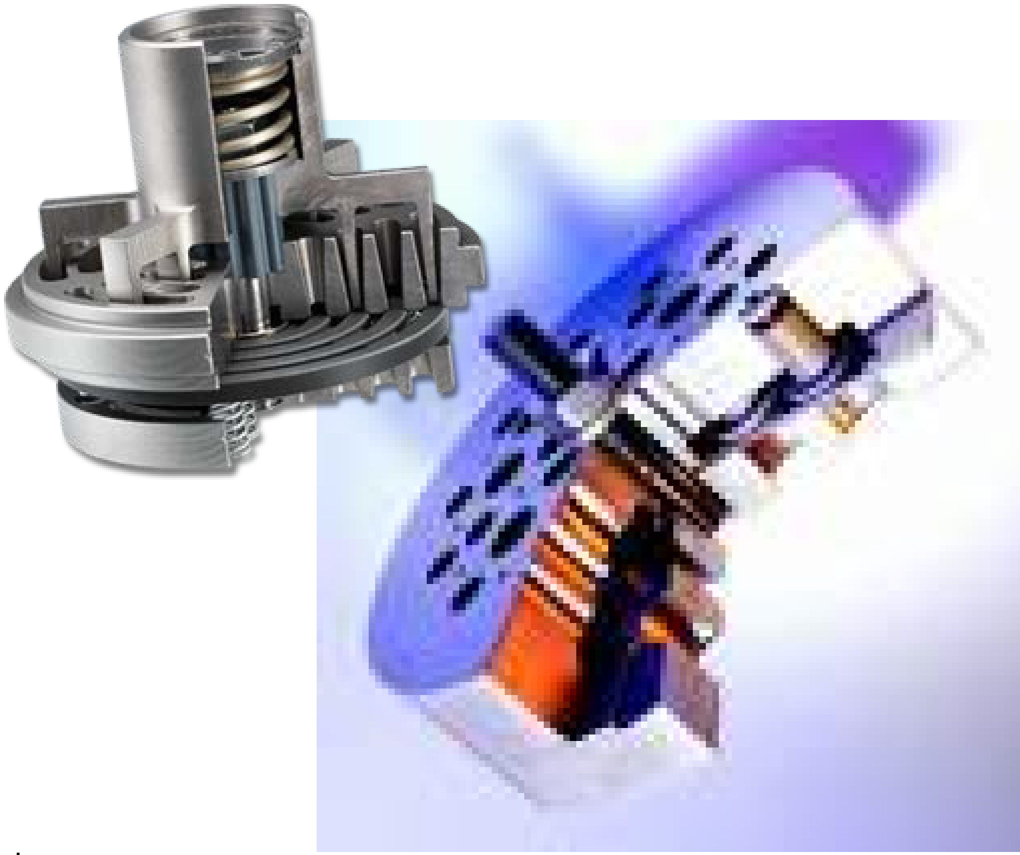
1- piston

2- piston rod

3- cross-head

4- connecting rod

5. crankshaft with counter-weight



Compressor valves

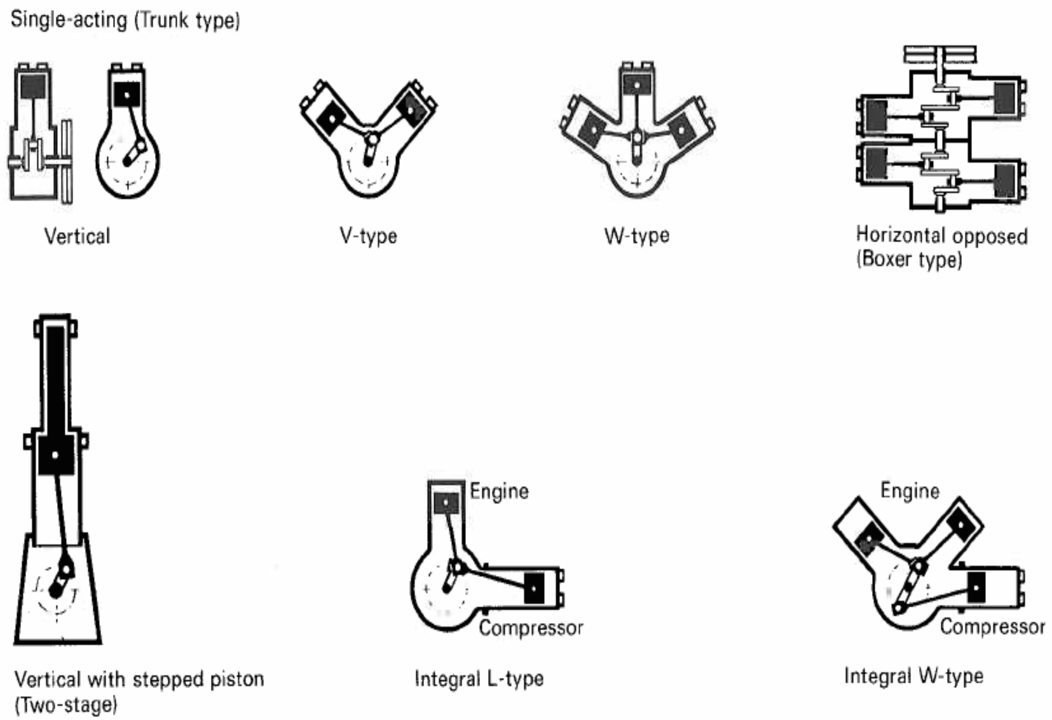
Reciprocating Compressors

Single-acting

Compression occurs only on one side of the piston and only once per revolution of the crankshaft. These machines are normally referred to as trunk type compressors.

Examples of cylinder layouts are shown. Single-acting compressors are usually of the enclosed type where the piston is directly driven by a connecting rod working off a crankshaft, both of which are enclosed in an externally pressure-tight crankcase.

Single-acting (Trunk type)

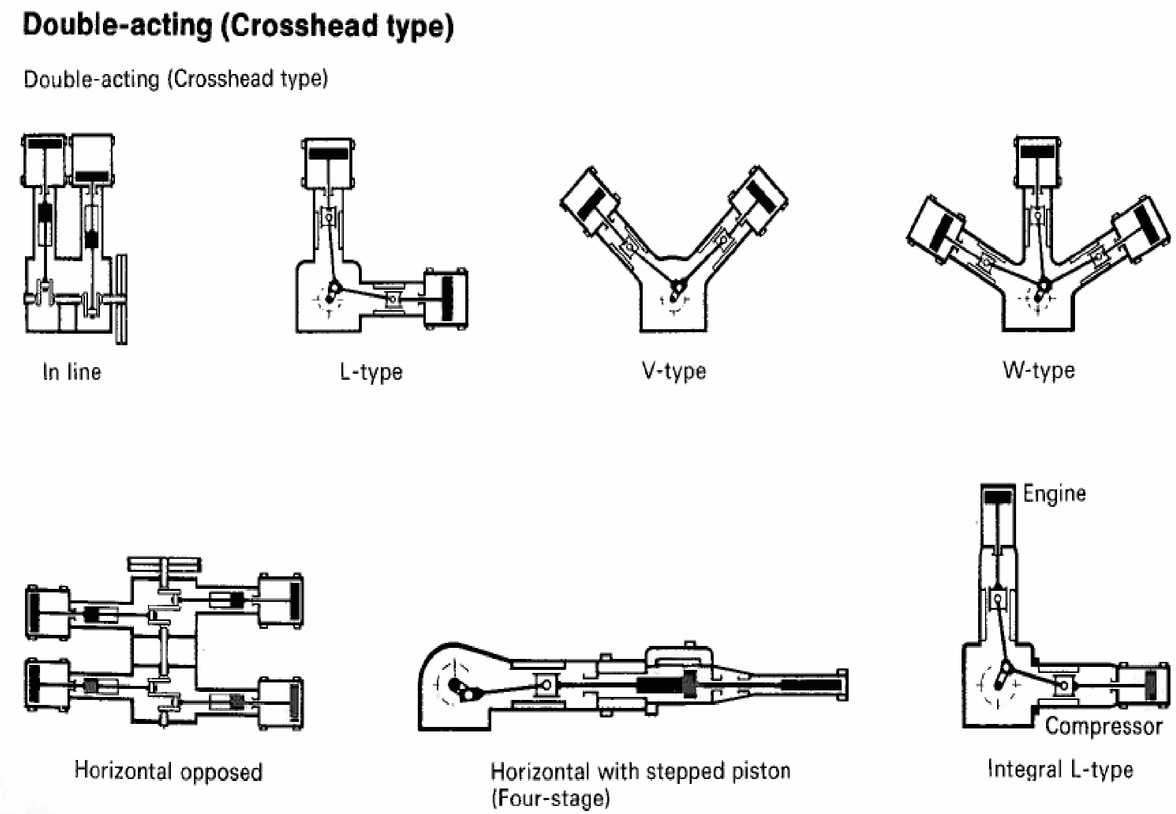


Reciprocating Compressors

Double-acting

Compression occurs alternately on both sides of the piston, twice during each revolution of the crankshaft.

These machines are normally referred to as crosshead type compressors. Examples of cylinder layouts are shown.



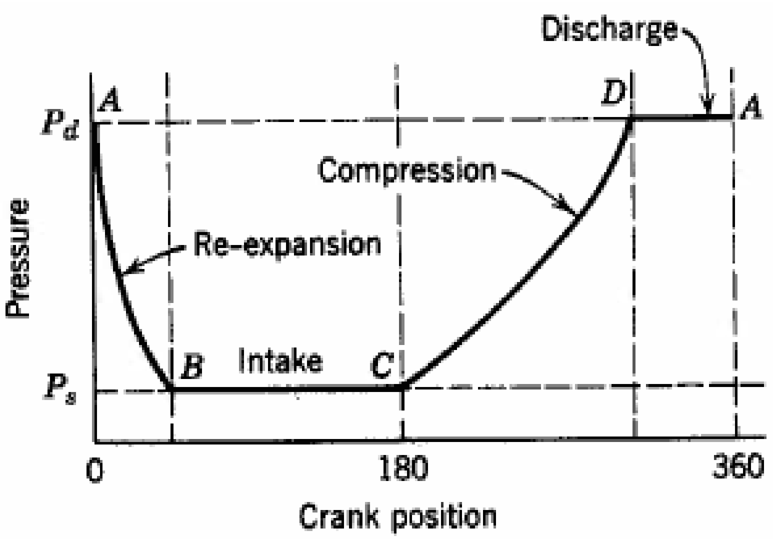
Reciprocating Compressors

The cycle is shown on the theoretical time-pressure diagram. At point A the piston is at top dead centre.

The pressure is maintained by the gas in the clearance space holding the valves closed. On the suction stroke from A-B the pressure is reduced as the gas expands to point B. Here, the gas in the suction line is at a higher pressure than the cylinder and the suction valves open.

From B-C the cylinder is filled with gas at suction pressure until point C when the suction valves close usually under spring action. Compression takes place from C-D.

At point D the pressure in the cylinder is higher than the gas in the head of the compressor and the discharge valves open. Gas flows until the piston reaches point A again and the cycle is complete with the crankshaft having completed one full revolution



Reciprocating Compressors Valves

The compressor valves are a critical component in a reciprocating compressor because of their effect on the efficiency (power and capacity) and reliability of the compressor. Compressor valves are basically check valves, but they are required to operate reliably for millions cycles, with opening and closing times measured in milliseconds, with no leakage in the reverse flow direction and with low pressure loss in the forward flow direction. Furthermore, they are frequently expected to operate in highly corrosive, dirty gas, while covered in sticky deposits.

Compressor valves affect performance due to the pressure drop caused by flow through the valve; the leakage through the valve in the reverse direction; and the fact that the valves do not close ideally.

Valve dynamics means

a) due to its inertia, the valve does not open instantaneously.

b) due to the springing, the valve does not stay at full lift for the full time it is open.

c) the valve does not close exactly at the dead centre.

The three most common suction and discharge valves in reciprocating compressors are

- The **poppet valve,** a cage serves as both a valve seat, stem guide and spring retainer. A spring, dashpot or bleeder arrangement is used to limit and damp valve travel. These are slow response valves and only used on slow speed compressors.
- The **ring plate valve**.
- **Flexing valves** vary in design but typically would consist of a seat, ribbon strips and a valve guard. This type of valve is the **feather valve.** The ribbon strips or reeds flex under pressure and as for the ring plate Valve operates under pressure difference

Reciprocating Compressor Control

There are a number of means of controlling compressor capacity

Variable speed shaft drive

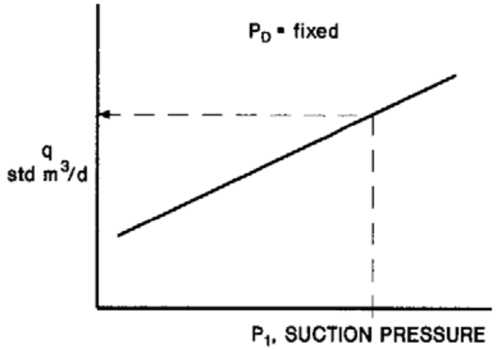
Compressor speed is adjusted to control capacity. Electric driven reciprocating compressors usually operate at constant speed.

Inlet throttling

Upstream control valve used to adjust suction pressure.

Fixed volume machine, so reduced suction pressure leads to reduced capacity.

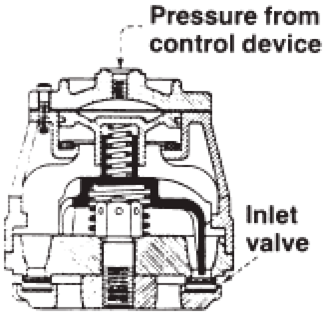
At low suction pressures there is a risk of high rod loading or high discharge temperatures. May require additional control provisions (valve unloading or clearance pocket control, see below) when the minimum suction pressure is reached.



Suction valve regulation (inlet valve unloaders)

A claw mechanism holds the valve plates open when gas demand is low

Gas is pushed in and out of the open suction valves, preventing compression.

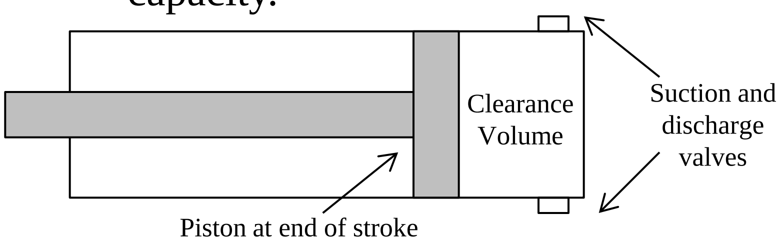


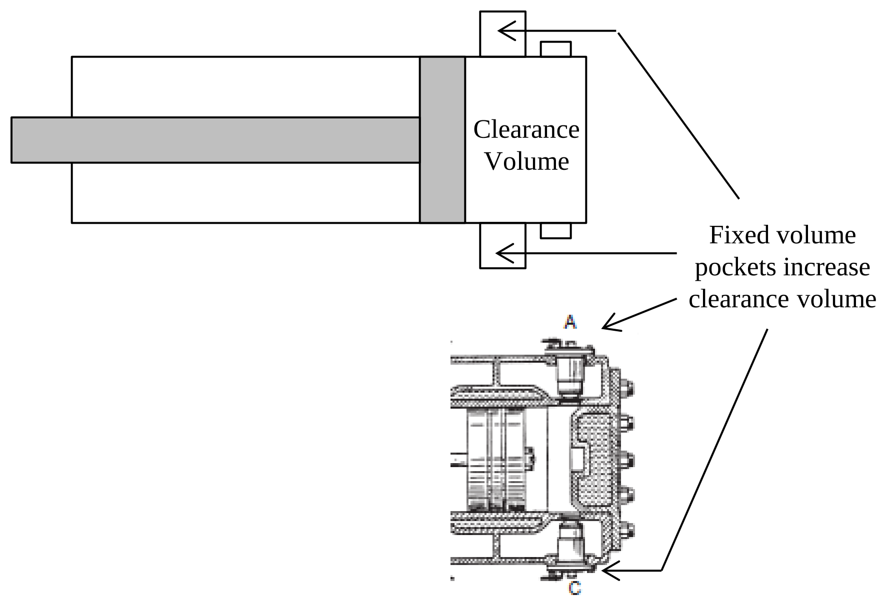
Clearance pocket control

All reciprocating compressors have some clearance volume between the piston and the cylinder head when the piston is at the end of the stroke. The gas contained in the clearance volume acts to reduce compressor capacity.

Clearance pockets can be used to control capacity. These are small pockets / reservoirs, provided with valves which can be opened when a reduction in capacity is desired.

Gas is compressed into the clearance pockets on the suction stroke, and expands into the cylinder on the return stroke. This reduces the intake of additional gas, reducing compressor capacity.

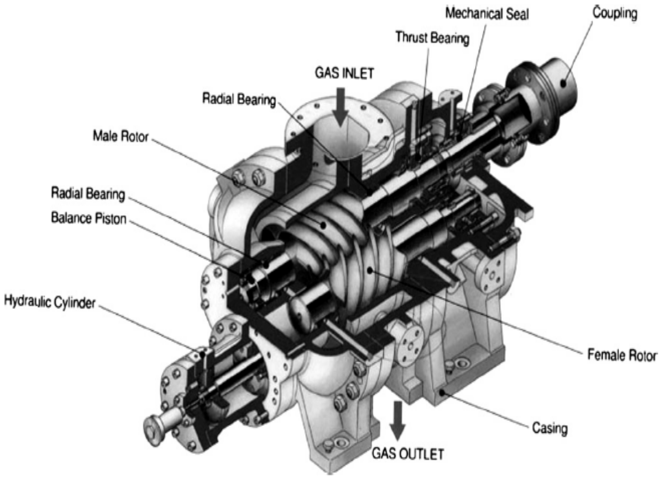




Screw Compressors

Screw compressors are positive displacement compressors. Twin screw machines compress gas between two meshing helically grooved rotors. Single screw machines have a single screw which meshes with two gate rotors. In twin screw compressors the male rotor is the driving rotor with a series of lobes, these mesh with matching flutes on the female non-driven rotor. In a dry running rotary screw compressor, timing gears ensure that the male and female rotors maintain precise alignment. In an oil-flooded rotary screw compressor, lubricating oil bridges the space between the rotors, both providing a hydraulic seal and transferring mechanical energy between the driving and driven rotor. Gas enters at the suction side and moves through the threads as the screws rotate. The meshing rotors force the gas through the compressor, and the gas exits at the end of the screws.

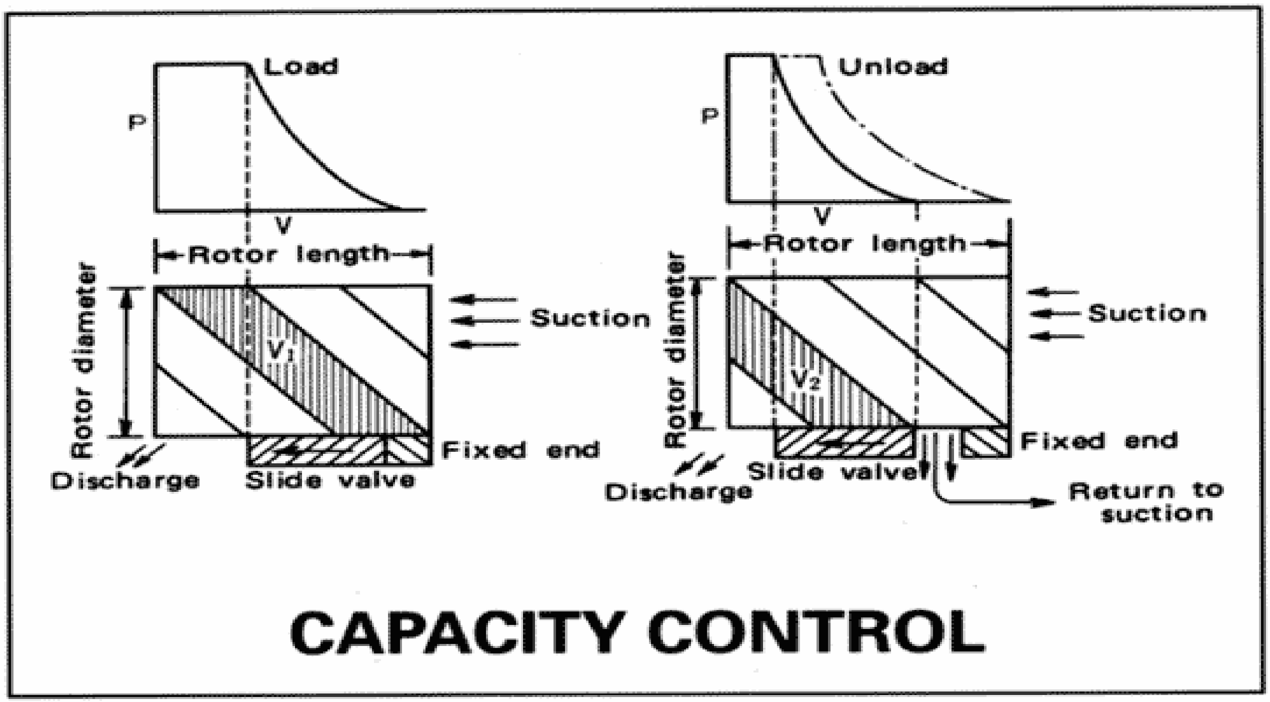
The effectiveness of this mechanism is dependent on precisely fitting clearances between the helical rotors, and between the rotors and the chamber for sealing of the compression cavities.



Screw Compressors

Capacity Control Mechanism

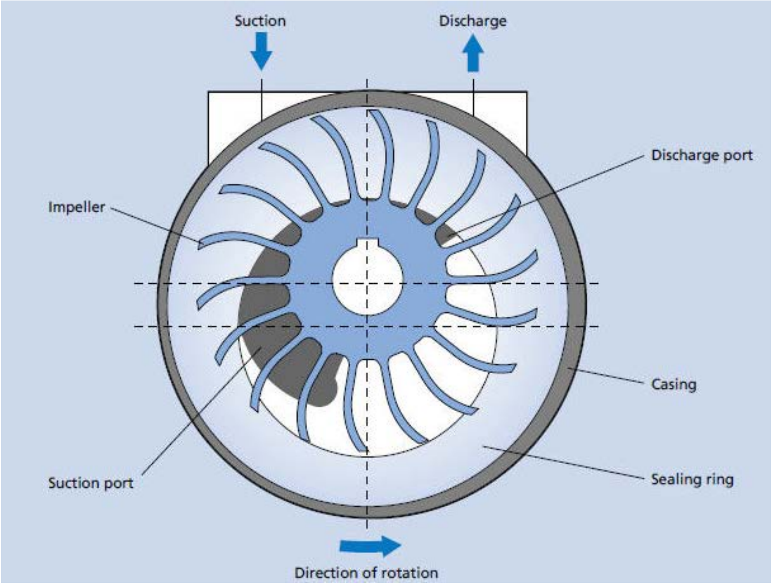
Capacity control is accomplished by a slide valve which moves parallel to the rotor axis and changes the area of the opening in the bottom of the rotor casing. This, in effect, lengthens or shortens the region of compression of the rotor and further acts to return gas to the suction side, while bypassing compressed gas. Variable speed drives, suction throttling and recycle can also control the capacity.



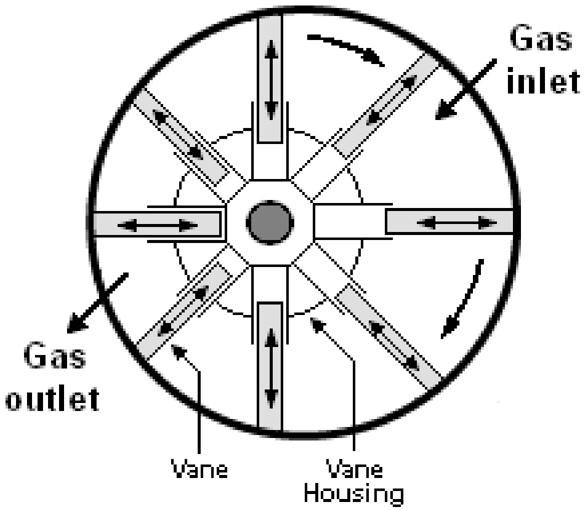
Liquid Ring Compressor

In a liquid ring compressor, an impeller is mounted eccentrically in a drum shaped casing between 2 end plates. As the impeller rotates the sealing liquid moves out by centrifugal force to the wall of the casing forming the liquid ring. The process gas is drawn into the pump by the expanding pockets trapped between the liquid ring and the impeller hub. The vapour is then discharged slightly above atmospheric pressure after being compressed as the pockets between the liquid ring and the impeller hub decrease.

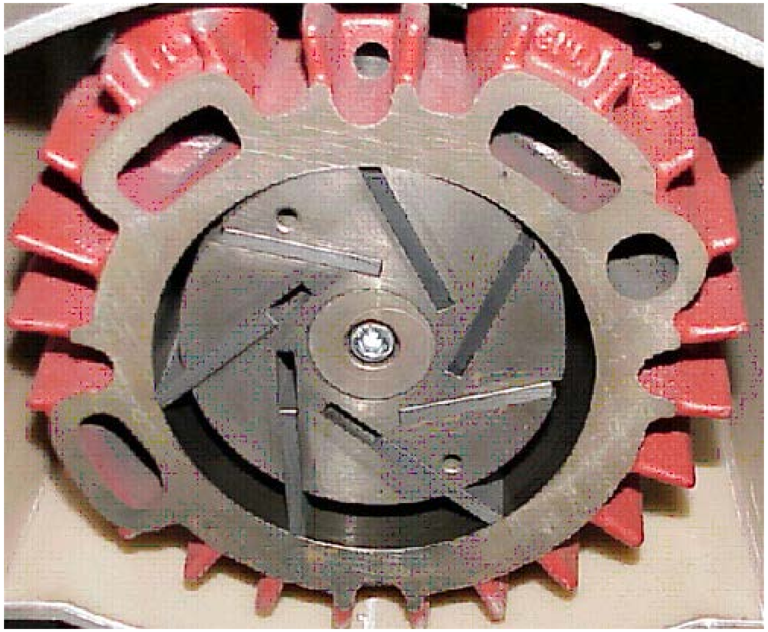
As a vacuum pump the liquid ring can generate pressures down to 0.05 bara and as a low pressure compressor with an atmospheric inlet will produce up to 3 bara at the discharge.



Vane Compressors



As the off centre shaft rotates the vanes slide in and out of the housing. The vanes maintain contact with the compressor casing wall. Gas enters the largest opening and the gas is compressed as the vanes rotate, exhausting at the smallest opening



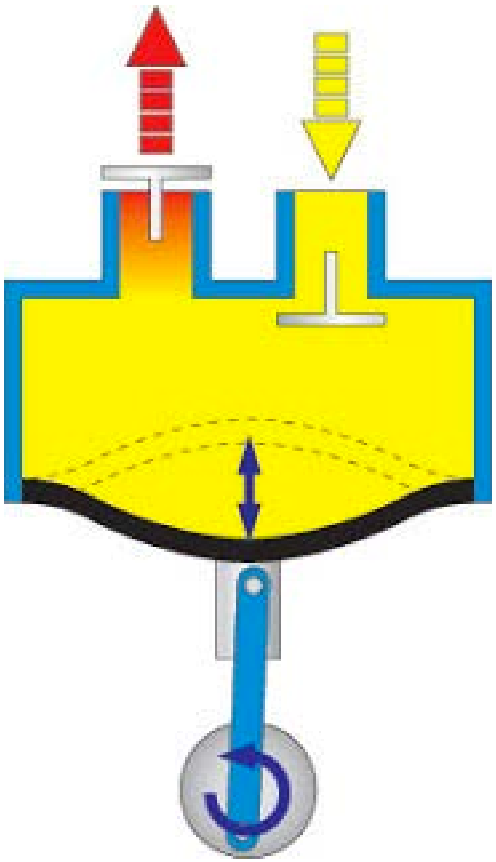
Lobe Compressors

As the lobes come out of mesh, they create expanding volume on the inlet side of the pump. Vapour flows into the cavity and is trapped by the lobes as they rotate.

Vapour travels around the interior of the casing in the pockets between the lobes and the casing—it does not pass between the lobes.

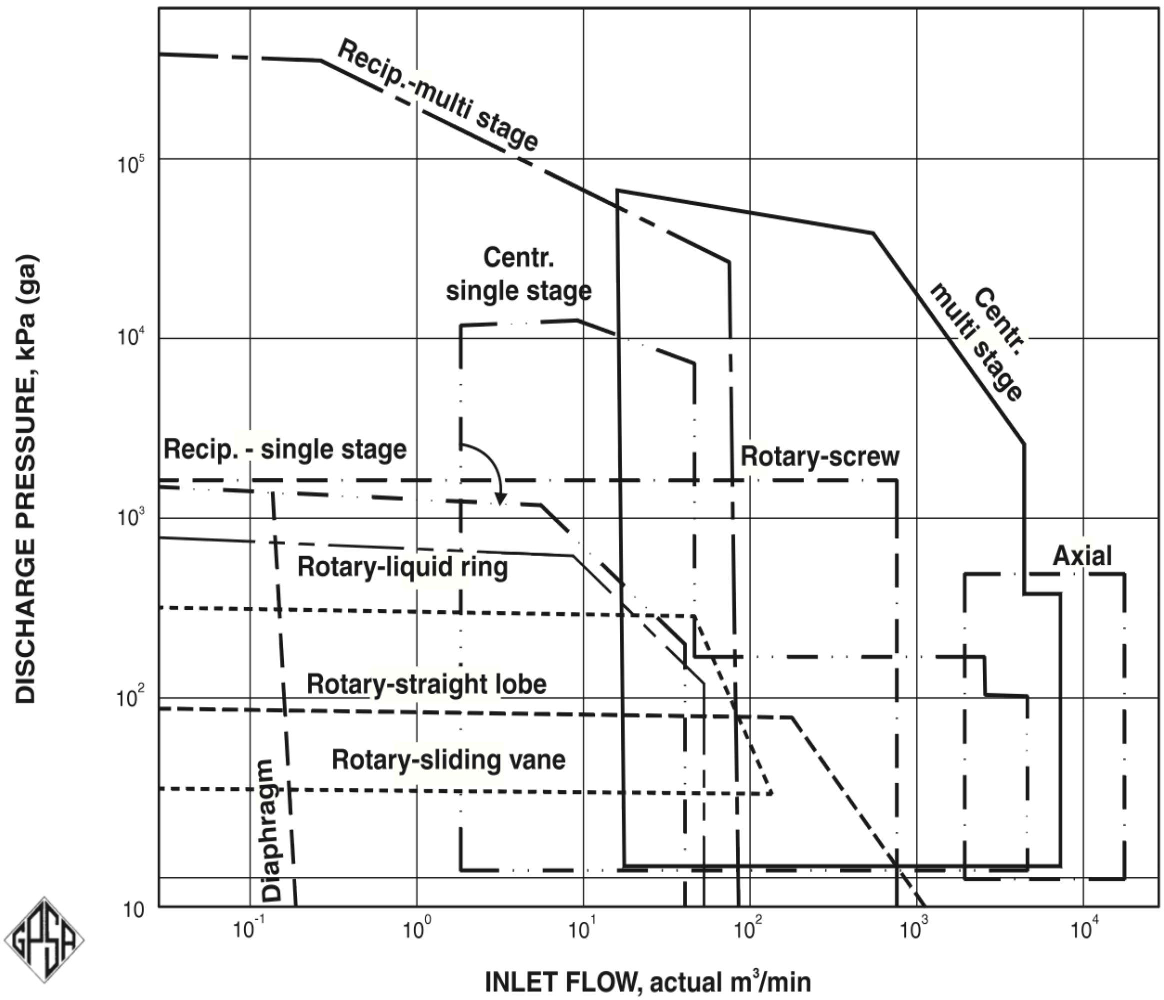
Finally, the meshing of the lobes forces vapour through the outlet port under pressure.

Diaphragm Compressors



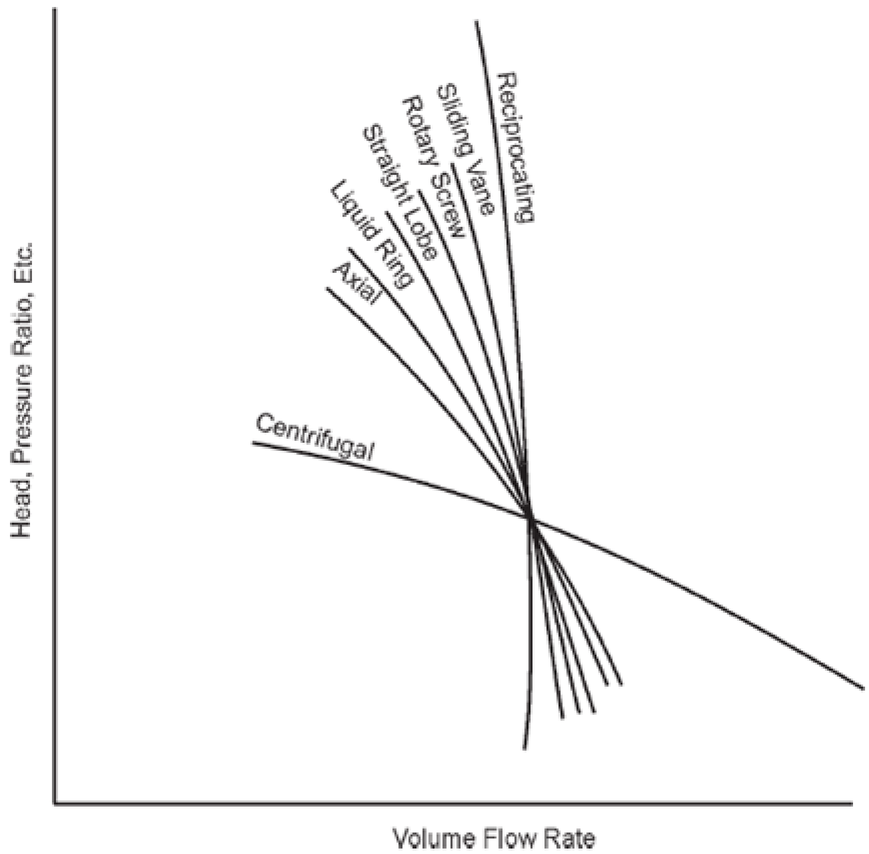
A rotating cam shaft flexes a diaphragm. On the downward stroke a suction valve opens allowing vapour to enter the expanding cavity. On the upward stroke a discharge valve opens and the gas is discharged.

Compressor Selection



- Recip.-multi stage:
  - Gas export / reinjection
- Recip.-single stage:
  - Fuel gas compression
- Diaphragm:
  - Oil free (air) compression
- Rotary screw compressors:
  - Low pressure recovery
- Centr. Single stage:
  - Gas pipeline boosting
- Centr. Multi stage:
  - Gas export / reinjection
- Axial:
  - Pipeline boosting
  - Gas turbines
- Liquid Ring:
  - Vacuum systems

Compressor Head Flow Characteristics



Compressor Drivers

- Drivers:
  - Electric Motors
  - Gas Engines
  - Diesel Engines
  - Gas Turbines
  - Steam Turbines
  - Expansion Turbines

- Driver selection influenced by:
  - Compatibility with power load
  - Fuel availability
  - Weight & Volume limitations
  - Reliability/Availability Requirements
  - Cost

Compressor / Gas Turbine

Gas Turbines

Gas turbines are extensively used in all phases of the oil and gas industry as a source of shaft power. They are used to drive compressors, generators, and other equipment required to produce, process, and transport natural gas. The compact, lightweight design of gas turbines makes them ideally suited for offshore platform installations, portable generating sets, remote sites, or any application where size and weight are important considerations.

Maintenance

Once installed, the gas turbine requires a minimum of routine maintenance. It is important to monitor the operating parameters of the turbine (pressures, temperatures, speed, vibration levels, etc.). This can often be done by an operator at a location remote from the actual turbine installation. The gas turbine was first widely used as an aircraft power plant. However, as they became more efficient and durable, they were adapted to the industrial marketplace. Over the years the gas turbine has evolved into two basic types for high power stationary applications: the industrial or heavy-duty design and the aircraft derivative design.

Heavy Duty

The industrial type gas turbine is designed exclusively for stationary use. Where high power output is required, 26 000 kW and above, the heavy duty industrial gas turbine is often specified. The industrial gas turbine has certain advantages which should be considered when determining application requirements. Some of these are:

· Less frequent maintenance.

· Can burn a wider variety of fuels.

· Available in larger power sizes.

Aircraft Derivative

An aircraft derivative gas turbine is based on an aircraft engine design which has been adapted for industrial use. The engine was originally designed to produce shaft power and later as a pure jet. The adaptation to stationary use was relatively simple.

Some of the advantages of the aircraft derivative gas turbines are:

· Higher efficiency than industrial units.

· Quick overhaul capability.

· Lighter and more compact, an asset where weight limitations are important such as offshore installations.

Electric Motors

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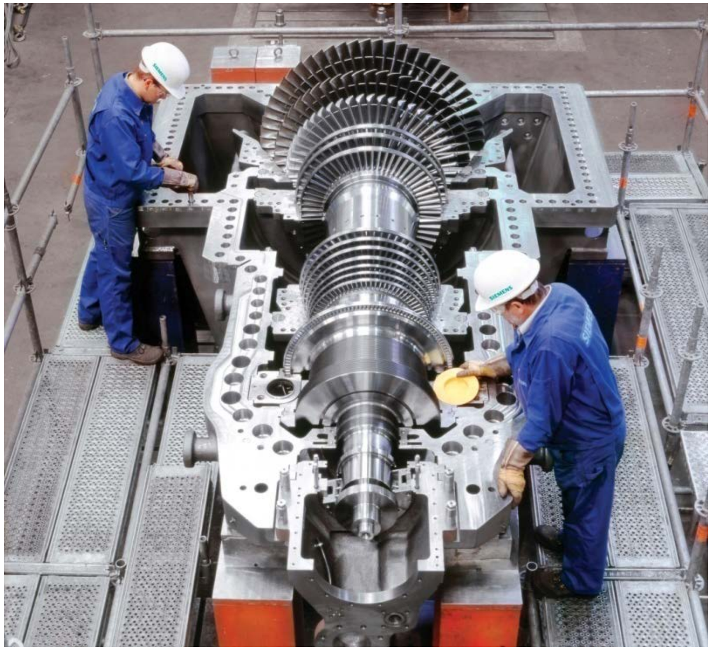
Electric motor drives offer efficient operation and add flexibility to the design of petroleum refineries, petrochemical plants, and gas processing plants. Electric motors can be built with characteristics to match almost any type of load. They can be designed to operate reliably in outdoor locations where exposed to weather and atmospheric contaminants.

Critical items to consider are load characteristics for both starting and running conditions, load control requirements, power system voltage and capacity, and any conditions at the plant site that could affect the type of motor enclosure.

Steam Turbines



Mechanical drive steam turbines are major prime movers for compressor, blower, and pump applications. They are extensively used in onshore chemical and petrochemical plant. Steam turbines are available for a wide range of steam conditions, power, and speeds. Typical ranges for each design parameter are:

Inlet Pressure, kPa (ga) 200 – 14 000

Inlet Temperature, ° C saturated – 540

Exhaust Pressure, kPa (ga) saturated – 4800

Power, kW 3 – 75 000

Speed, rpm 1800 – 14 000

Mechanical drive steam turbines are categorized by single stage/multi-stage, condensing or non-condensing exhausts, extraction/admission, and impulse/reaction.

Single Stage/Multi-Stage

In a single stage turbine, steam is accelerated through one cascade of stationary nozzles and guided into the rotating blades or buckets on the turbine wheel to produce power. Single stage turbines are usually limited to about 2000 kW although special designs are available for larger units.

Below 2000 kW the choice between a single and a multi-stage turbine is usually an economic one. For a given shaft power, a single stage turbine will have a lower capital cost but will require more steam than a multi-stage turbine because of the lower efficiency of the single stage turbine.

Condensing/Non-Condensing

The energy available in each kilogram of steam which flows through the turbine is a function of the overall turbine pressure ratio (inlet pressure/exhaust pressure) and inlet temperature. Condensing turbines are those whose exhaust pressure is below atmospheric. They offer the highest overall turbine pressure ratio for a given set of inlet conditions and therefore require the lowest steam flow to produce a given power. A cooling medium is required to totally condense the steam.

Non-condensing or back-pressure turbines exhaust steam at pressures above atmospheric and are usually applied when the exhaust steam can be utilized elsewhere.

Compression Thermodynamics

The basic thermodynamic equation for compression is:

$$\Delta H = \int V dP = -W_{theor} \qquad [1]$$

For the isentropic compression process:

$$P \cdot V^k = const = P_1 \cdot V_1^k \qquad [2]$$

Substitution of [2] into [1] yields:

$$\Delta H = P_1^{1/k} \cdot V_1 \cdot \int_{P_1}^{P_2} P^{-1/k} \cdot dP \qquad [3]$$

or (solving):

$$\Delta H = \frac{P_1^{1/k} \cdot V_1}{\left(\frac{k-1}{k}\right)} \cdot \left(P_2^{\frac{k-1}{k}} - P_2^{\frac{k-1}{k}} \right) \qquad [4]$$

Compression Thermodynamics

Equation [4] is rewritten as:

$$\Delta H = \frac{P_1 \cdot V_1}{\left(\frac{k-1}{k}\right)} \cdot \left(\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right) \qquad [5]$$

Substitution of the ideal gas law into [5] yields:

$$\Delta H = \frac{m \cdot Z_1 \cdot R \cdot T_1}{\left(\frac{k-1}{k}\right) \cdot MW} \cdot \left(\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right) \qquad [6]$$

Compression

Centrifugal Compressors

– Head equation for practical applications:

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

– Temperature relationship:

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

– Power relationship

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

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

For paraffin gases with a molecular weight less than air ($\gamma < 1$) k may be estimated from :
$k = 1.3 - 0.31 (\gamma - 0.55)$

Compression Thermodynamics

Equation [6] (previous slide) is usually written as:

$$\Delta H_{isen} = \frac{Z_a \cdot R \cdot T_1}{\left(\frac{k-1}{k}\right) \cdot MW} \cdot \left(\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right) \qquad [7]$$

which is the basic "head" equation for compressors, in which Z_a is the average compressibility $((Z_1+Z_2/2))$ and ΔH_{isen} is the isentropic head (in kJ/kg). For the isentropic head in meters:

$$\Delta H_{isen}(meters) = \frac{1000 \cdot \Delta H_{isen}(kJ/kg)}{g\ (=9.81)}$$

In a steady state situation the entropy change in a system is written as: $\Delta S = Q/T_b + S_p$ [8]

in which Q is the heat exchanged with surroundings, T is the absolute temperature of the system boundaries and S_p is the entropy production, reflecting the irreversibility of the process.

An industrial machine will not be isentropic. The machine will have irreversibilites such as friction and noise. The ideal (isentropic) machine is corrected for inefficiencies as indicated below.

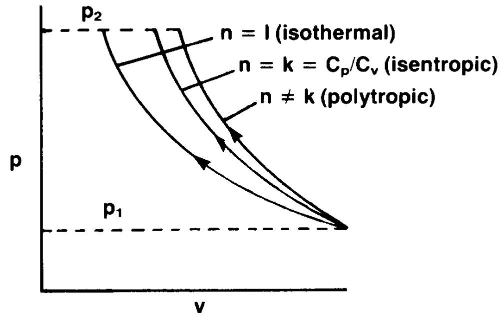
Isentropic Process: $\Delta S = 0$, reversible ($S_p = 0$) and adiabatic process ($Q = 0$)

Polytropic Process: $\Delta S \neq 0$, irreversible ($S_p \neq 0$), entropy is produced by internal friction in the system

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

n - Polytropic coefficient (n > k)

E - Polytropic efficiency



Compression Curves

$k = Cp/Cv$

Be wary of using the values in simulation packages.

Cp/Cv can be calculated from;

$k = \Sigma(y_i)(Cp_i)/(\Sigma(y_i)(Cp_i) - 8.314)$

y_i = mol. Fr of each component

Cp_i = molar heat capacity kJ/kmol ^oC

Calculation Example

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- 10 MMscfd of hydrocarbon gas (MW = 22.0 kg/kmol) is compressed from 4 bara to 15 bara. Calculate the required compressor power and discharge temperature. The polytropic efficiency of the compressor is 75%. Z_a = 0.98. $T_1 = 30^oC$

10 MMscfd = 0.283×10^6 sm^3/day = 492 kmol/hr

$\rightarrow$ Mass flow: 492 * 22 / 3600 = 3.0 kg/s

Specific gravity: 22.0 / 28.96 = 0.76

Ratio of specific heats (estimated):

k = 1.3 - 0.31* (0.76 - 0.55) = 1.23

Polytropic coefficient:

$$\frac{n-1}{n} \approx \frac{1.23-1}{1.23*0.75} = 0.25 \;\rightarrow\; n = 1.33$$

$$\Delta h_{poly} = \frac{303*0.98*8.314}{\left(\frac{1.33-1}{1.33}\right)*22.0} * \left[\left(\frac{15}{4}\right)^{\frac{1.33-1}{1.33}} - 1\right] = 175.5\; kJ/kg$$

or ΔH_{poly} = 175.5 * 1000 / 9.81 = 17890 m

$$W = \frac{3.0*175.5}{0.75} = 700\; kW$$

$$T_2 = 303*\left(\frac{15}{4}\right)^{\frac{1.33-1}{1.33}} = 421\,K = 148^o C$$

$$T_2 = 303*\left(\frac{15}{4}\right)^{\frac{1.33-1}{1.33}} = 421\,K = 148^o C$$

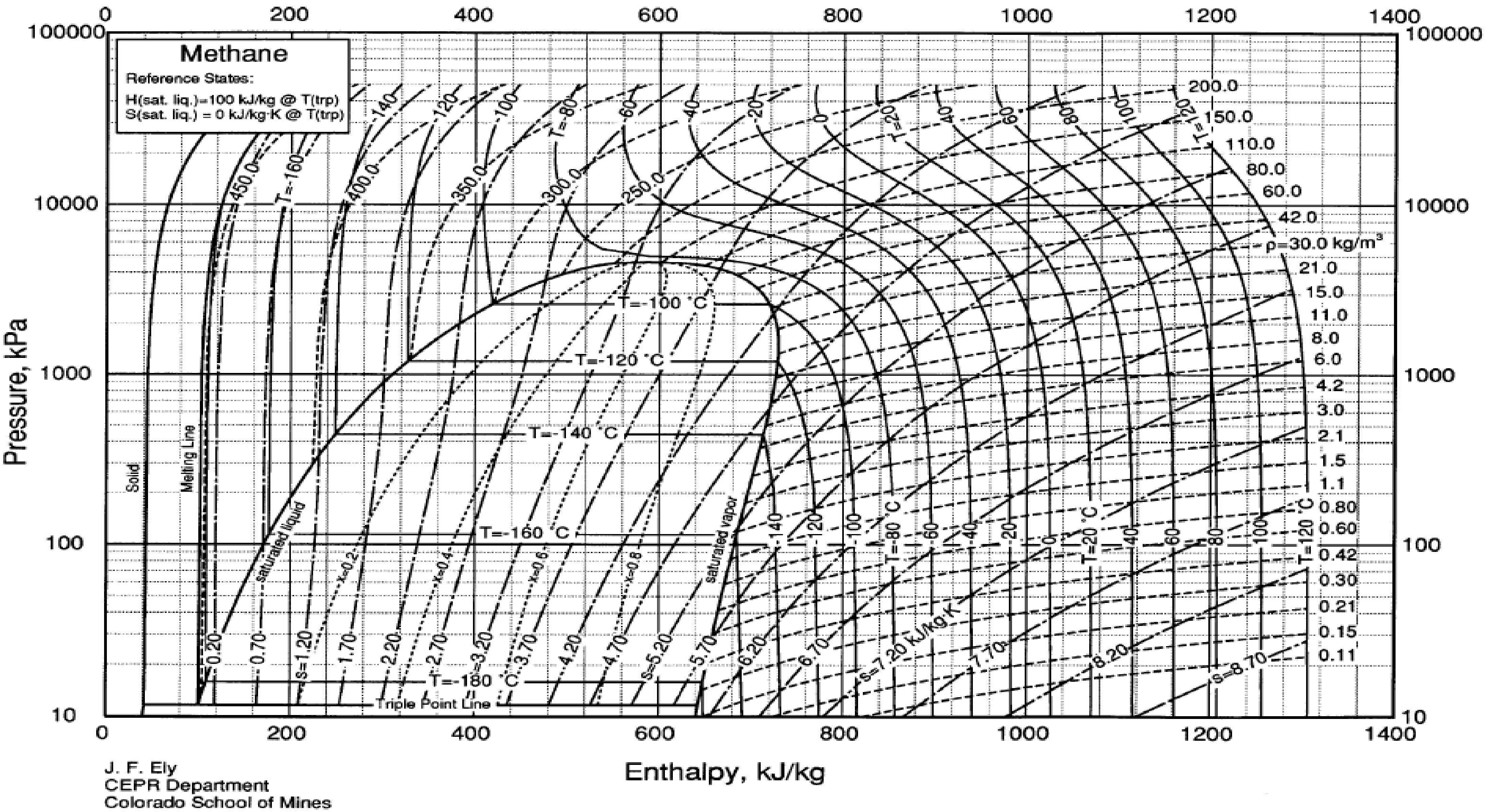
- If the same compressor were to compress a heavier gas, e.g. MW = 30, what would be the discharge pressure?
- The head is unaffected, because the head is only dependent on impeller (tip) speed:

$$\left(\frac{P_2}{4}\right)^{\frac{1.33-1}{1.33}} = \frac{175.5\cdot\left(\frac{1.33-1}{1.33}\right)\cdot 30}{303\cdot 0.98\cdot 8.314} + 1 \;\rightarrow\; P_2 = 22\; bara$$

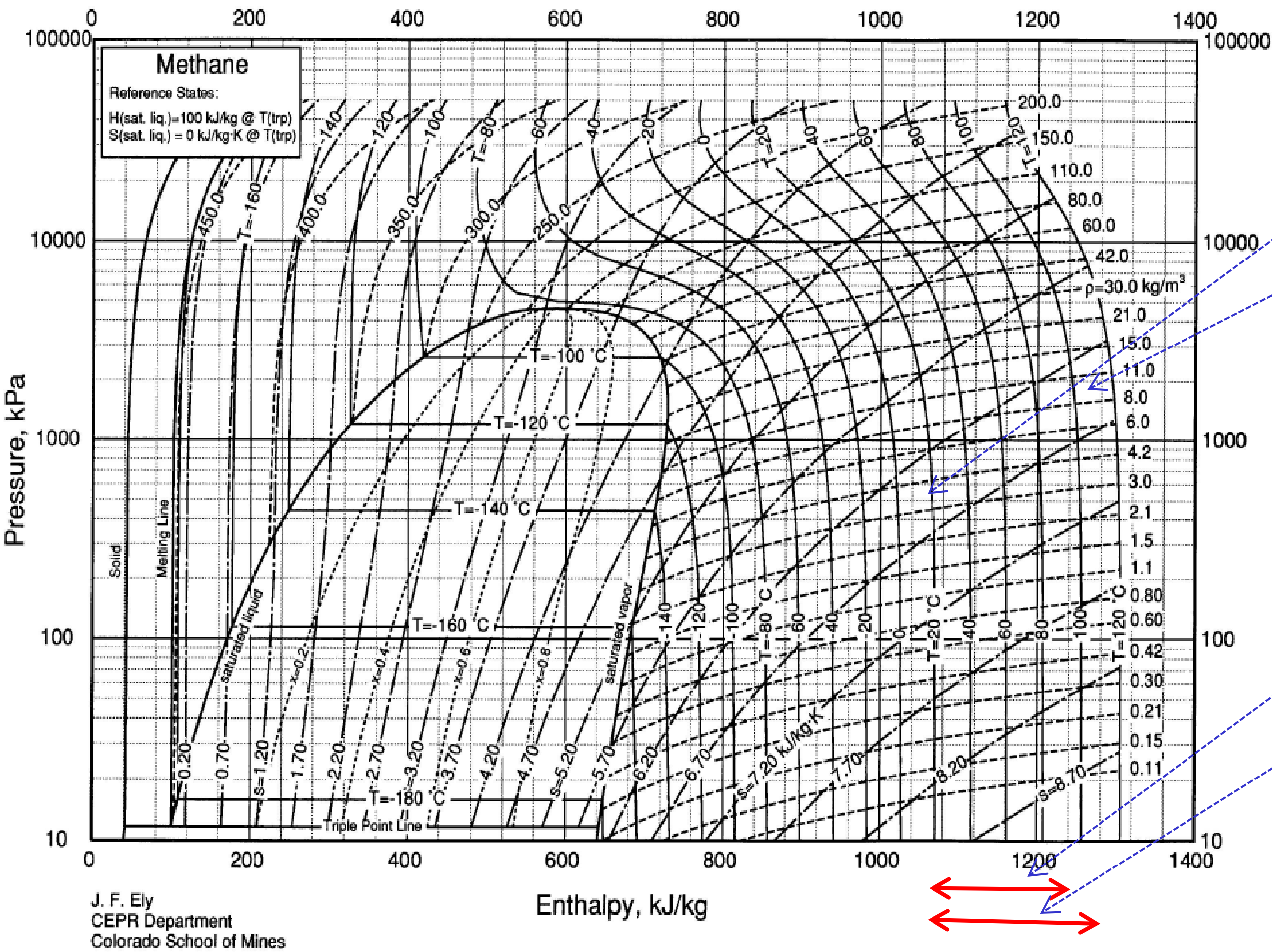
- The required power would be: 700 * (30/22) = 955 kW

:

The Mollier Chart



The Mollier Chart - Application



Compress Methane from 4 bar at 10 Deg C to 15 bar.

Compressor polytropic efficiency is 75%.

Isentropic compression

Enthalpy in – 1060 kJ/kg

Enthalpy out – 1260 kJ/Kg

Enthalpy change = 200 kJ/kg

Adjust for efficiency

Polytropic enthalpy change

= 200/0.75 = 266.7 kJ/Kg

Centrifugal Compressor Design

Mechanical design undertaken by machine supplier

Design will account for;

- Thermodynamics
- Aerodynamics
- Rotor Dynamics
- Stress/loads

Process engineer will specify;

- Number of stages
- Flowrate
- Gas Composition
- Inlet Pressure and Temperature
- Discharge Pressure
- Range in volumetric rate
- Range in molecular weights

Major Design Variables

- Speed
- Impeller Diameter
- Number of impellers
- Impeller Design
- Head per impeller

- Specific speed, N_s:

$$N_s = \frac{2.44 \times N \times q^{0.5}}{H^{0.75}}$$

N	speed (rpm)
q	volumetric flowrate (m^3/s)
H	head per stage (m)

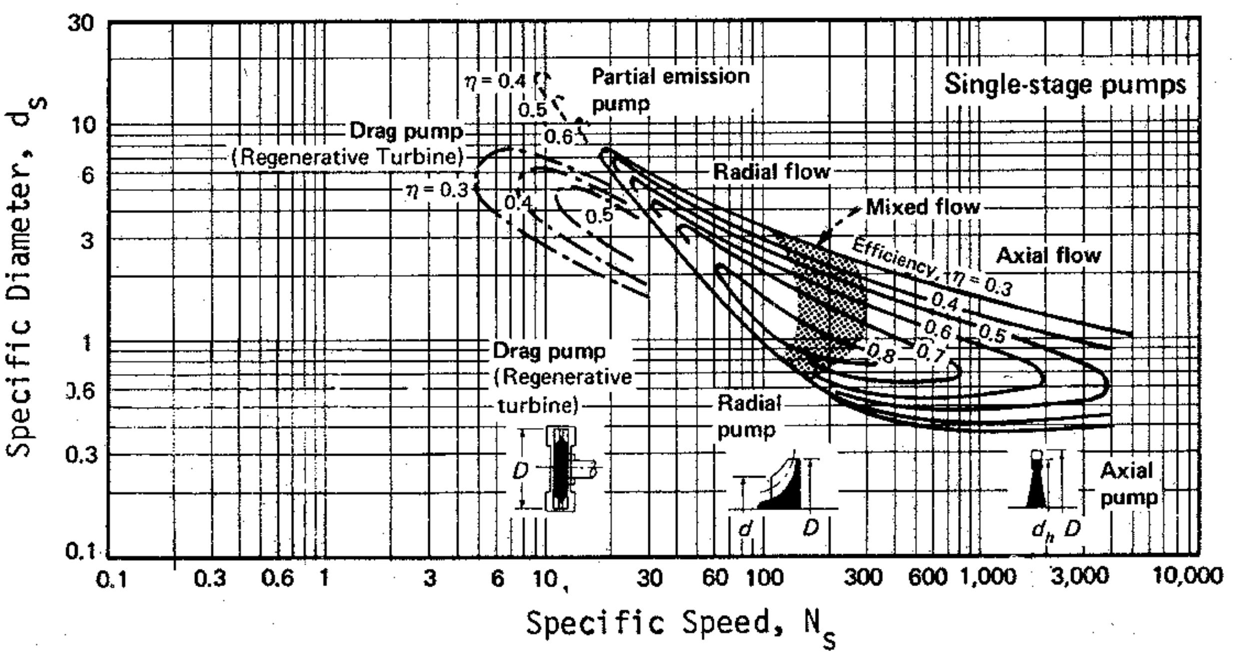
- Specific diameter, d_s:

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

Take care with specific speed and head in different units and use as defined by compressor/pump designer.

Specific Speed / Diameter

- The primary use of specific speed is in the classification of centrifugal pumps/compressors and it gives an indication of the efficiency
- The specific diameter is a concept, which can be used with specific speed to make a general choice of pump/compressor arrangement



- Specific speed:

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

N pump speed (rpm)

q liquid flowrate (m^3/s)

H pump head (m)

- Specific diameter:

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

d impeller diameter (m)

Conceptual Design
Head/Speed/Impellers

The energy imparted to the gas is related to the velocity change through the impeller. The equation that relates head and tip speed is;

$H = v^2/2g$ – v is the impeller tip speed.

For preliminary design purposes a reasonable estimate is

v = 250 – 300 m/s

Also, for a preliminary design head per impeller stage is approximately 3200 m

Impeller diameter and speed can be estimated from;

$d = \sqrt{(Q/(0.05v))}$ $\qquad\qquad$ $N = 60v/(d.\Pi)$

d = impeller diameter, m
Q = inlet flow to impeller, m3/s
v = impeller tip speed, m/s
N= speed, rpm

Centrifugal Compressor Preliminary Design

A centrifugal compressor has the following duty. Using this information calculate the compress polytropic head, power, discharge temperature, number of impellers, impeller diameter and shaft speed.

Volume flow, $q = 3.25 \times 10^6$ sm3/day

Suction Pressure, $P_1 = 2000$ kPa

Discharge pressure, $P_2 = 6000$ kPa

Suction Temperature, $T_s = 30$ DegC

$Z_{ave} = 0.9$

$Z_1 = 0.95$

$k = 1.25$

MWt = 19

Polytropic efficiency = 75%

R = 8.314 kJ/kmolDegK

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

$T_1 = 30 + 273 = 303\ °K$

$\dfrac{n-1}{n} = \dfrac{k-1}{k \cdot \eta} = \dfrac{1.25-1}{1.25 \times 0.75} = 0.267$

$\Delta h_{poly} = \dfrac{303 \times 0.9 \times 8314}{0.267 \times 19} \left[\left(\dfrac{6}{2}\right)^{0.267} - 1 \right]$

$= 152348\ J/kg$

$= \dfrac{152348\ J/kg}{9.81 \quad m/s^2}$

$\dfrac{N\,m}{kg} \cdot \dfrac{\cancel{s}^2}{m}$

$\dfrac{\cancel{kg} \cdot m}{s^2} \cdot \dfrac{m}{kg} \cdot \dfrac{s^2}{\cancel{m}}$

$= \underline{\underline{15530\ m}}$

2

2. Mass rate = volume flow × density.

Density, kg @ standard conditions $= \frac{19}{23.96}$ kg/Sm³

∴ Mass rate $= 3.25 \times 10^6 \times \frac{19}{23.96} \times \frac{1}{24 \times 3600}$

$\phi_m = 29.8$ kg/s.

Gas Power, $W = \frac{\phi_m \cdot \Delta h_{poly}}{3} = \frac{29.8 \times 152348}{0.75}$

$= 6.05 \times 10^6$ J/s.

$= \underline{\underline{6.05 \text{ MW}}}$.

3. Discharge Temperature

$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = 303 \times 3^{0.267}$

$= 406$ °K

$= 133.3$ °C

4. Number of impellers

Assume m/stage = 3200

∴ No of impellers = $\frac{15530}{3200}$ ≑ 5.

5. Impeller diameter.

Actual volume flow, $q = 3.25 \times 10^6 \times \frac{P_s}{P_1} \times \frac{T_1}{T_s} \times Z_1$

$= 3.25 \times 10^6 \times \frac{100}{2000} \times \frac{303}{288} \times 0.95$

$= 162,415$ m^3/day

$= 1.88$ m^3/s.

Assume tip speed = 250 m/s.

$d = \sqrt{\frac{Q}{0.05\,v}} = 0.39$ m.

6 Shaft Speed

$$N = \frac{60v}{d \times \pi} = \frac{60 \times 250}{0.39 \times 3.14}$$

$$= 12{,}250 \text{ R.P.M.}$$

Check efficiency.

Specific speed

$$N_s = \frac{2.44 \times 12250}{3200^{0.75}} \times 1.88^{0.5}$$

$$= \frac{40983}{425.5}$$

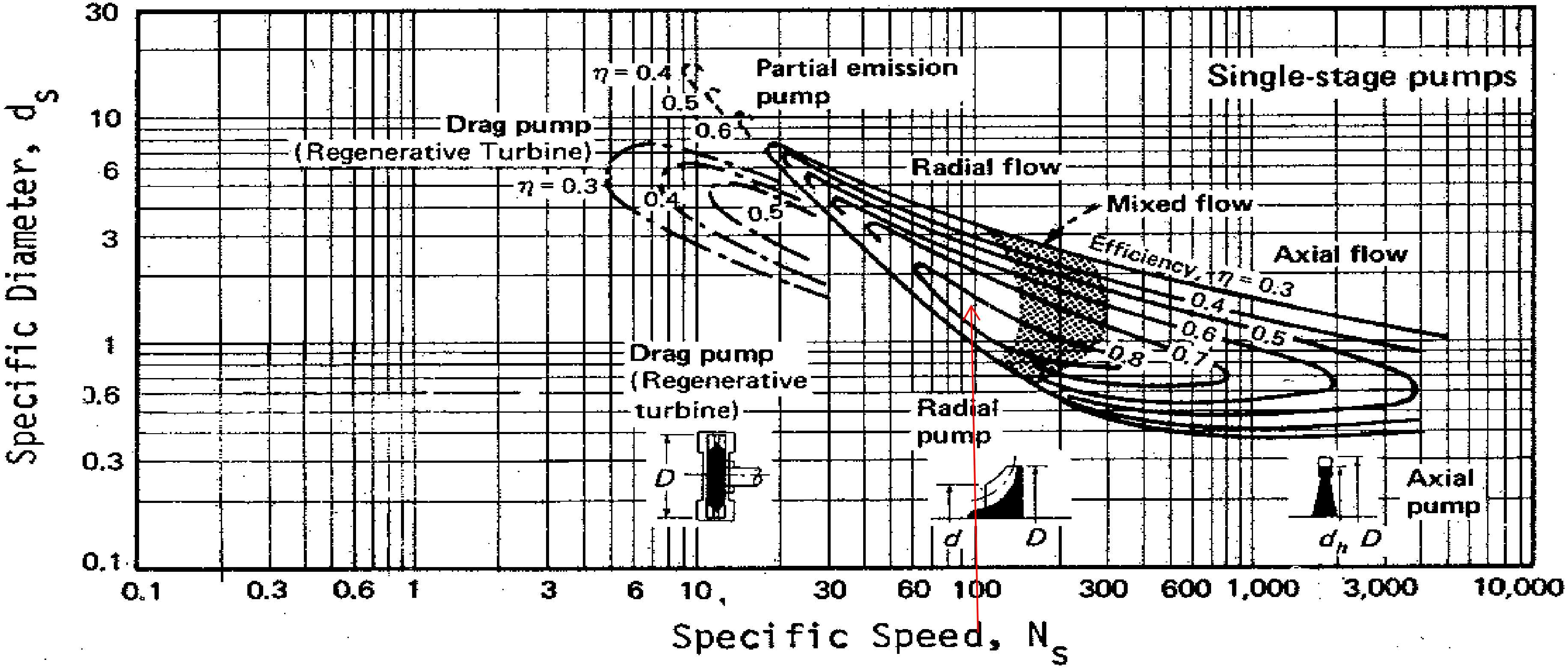
$$= 96.3$$

Specific diameter

$$d_s = 0.74 \times \frac{0.39 \times 3200^{0.25}}{1.88^{0.5}}$$

$$= 1.58.$$

Specific Speed / Diameter



$$d_s = \frac{0.743 \cdot d \cdot H^{0.25}}{q^{0.5}}$$

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

Key Learnings

- Compressor types and application
- Concept of fluid head
- Compressor head flow and efficiency curves/maps
- Centrifugal compressor characteristics and control – surge and stonewall
- Compressor Thermodynamics
- Application of polytropic head, power and temperature equations
- Use of Mollier Charts
- Specific head and speed

Additional Reading

- GPSA Technical Manual, Gas Processors Association
- Gas Conditioning and Processing, J Campbell