http://maintenanceengineering.in/COMPRESSOR.htm

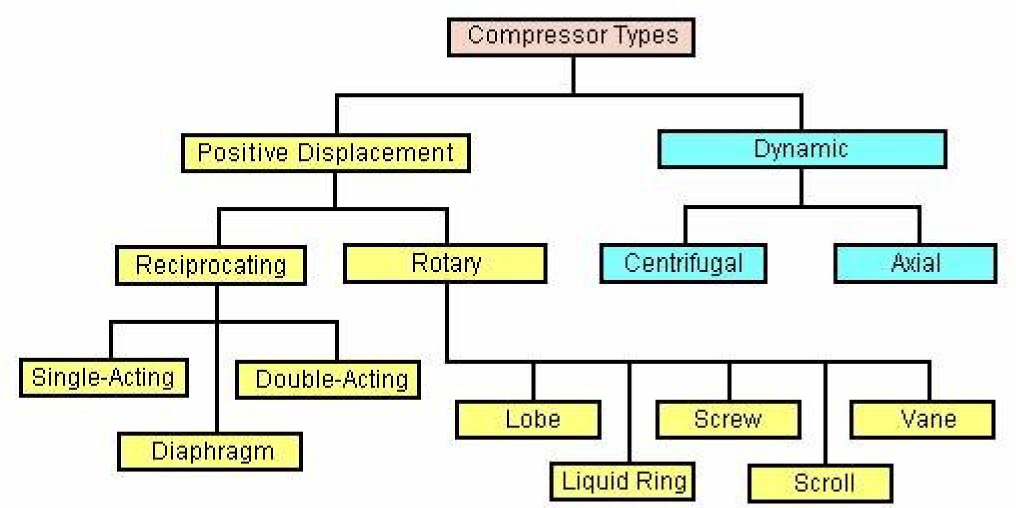
COMPRESSOR

The compressor is a machine, which compresses gases from low pressure to required high pressure. In compressors, mechanical energy is converted into kinetic energy and imparted as thermodynamic energy to the fluid in compression, and is exhibited in form of high pressure and temperature of the gases after the compression phenomena.

Compressors are broadly divided into two main classes

I) Positive displacement compressor (Reciprocating, screw compressors)
ii) Dynamic compressor. (Centrifugal Compressors)

Positive displacement compressors are further divided in two groups namely rotary type and reciprocating type
The classification tree is as below:



In case of positive displacement compressor, entrapped gas in some type of enclosure (cylinder or lobes) is pushed out from the enclosure with some mechanical device (Piston or screw) thus reduces the volume of the enclosure and increase the pressure.

In case of Dynamic compressor, mechanical action of rotating impeller or blades impart kinetic energy to the gases, which then is converted in to potential energy in form of high pressure through diffuser or volute casing.

In both the cases the flow of a compressor is directly related to the speed of the compressor and the intake pressure

RECIPROCATING GAS COMPRESSOR:

Reciprocating compressors are generally made as per API 618, for general purpose compression services in refinery and petrochemical industries. API 618 covers all general minimum requirement and specifications of the compressor parts for the specific services. Portable air compressors are excluded from this standard. This standard covers related lubrication system, controls, inter and after coolers, pulsation suppression devices and other auxiliary equipments.
In reciprocating gas compressor the fluid (air/gas) is compressed in a cylinder with the help of piston. This compression of fluid can be done in a single cylinder or through consecutive multi stage cylinders in series or parallel, depending upon the process requirement of final discharge pressure and flow.

Compressor Cycle: The energy supplied to a compressor through prime mover goes in creasing the pressure and temperature of the gas following the 'real gas law' of thermodynamics. However our main purpose of the compressor is to increase the pressure of the gas as per requirement using minimum possible power.
If the compression process is adiabatic, that is there is no heat transfer between the compressor and surrounding. Least work will be done in compressing the gases if the process is isentropic. Which mean that there are no losses of energy in the compressor. This is an unachievable goal, but still the compressor efficiency is given as the isentropic efficiency. The work done in a reversible isothermal process is less than done in an isentropic process.
In a reversible isothermal process, the temperature of the gas is maintained at the suction temperature by reversible heat transfer as the compression proceeds (by means of jacketed cylinder cooling and interstage cooler). Many compressors have a final discharge temperature that is much lower than the isentropic discharge temperature and the power required in such compressors get reduced.

Definitions of these related terms are given as below:

Isothermal - gas remains at constant temperature throughout the process. In this cycle, internal energy is removed from the system as heat at the same rate that it is added by the mechanical work of compression. Isothermal compression or expansion is favored by a large heat exchanging surface, a small gas volume, or a long time scale (i.e., a small power level). With practical devices, isothermal compression is usually not attainable. For example, even a bicycle tire-pump gets hot during use.

Adiabatic - In this process there is no heat transfer to or from the system, and all supplied work is added to the internal energy of the gas, resulting in increases of temperature and pressure.
Theoretical temperature rise is $T_2 = T_1 \cdot R_c^{((k-1)/k))}$,
with T_1 and T_2 in degrees **Rankine** or **kelvins**, and
***k* = ratio** of specific heats (approximately 1.4 for air).

R is the compression ratio; being the absolute outlet pressure divided by the absolute inlet pressure.

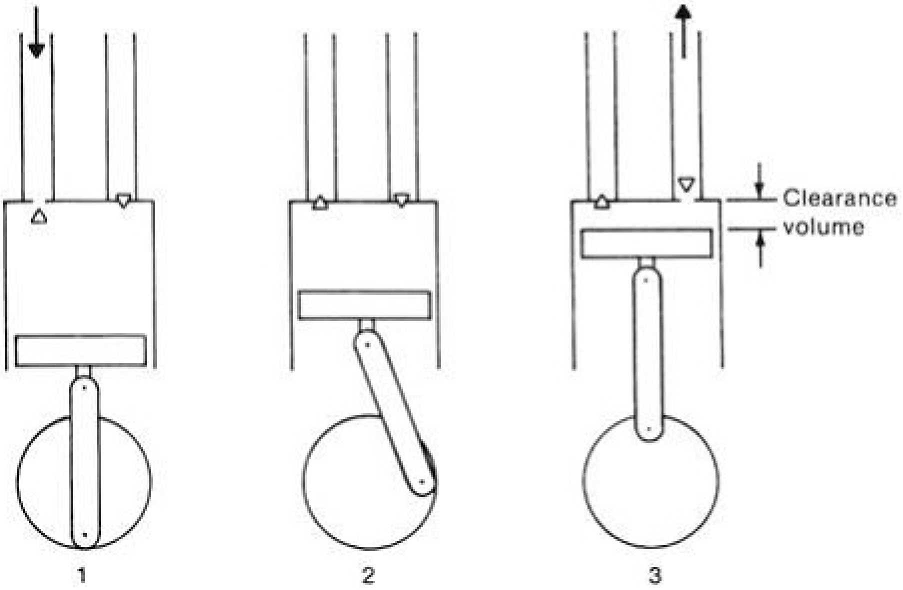
The rise in air and temperature ratio means compression does not follow a simple pressure to volume ratio. This is less efficient, but quick. Adiabatic compression or expansion is favored by good insulation, a large gas volume, or a short time scale (i.e., a high power level). In practice there will always be a certain amount of heat flow, as to make a perfect adiabatic system would require perfect heat insulation of all parts of a machine.

Polytropic - This assumes that heat may enter or leave the system, and that input shaft work can appear as both increased pressure (usually useful work) and increased temperature above adiabatic (usually losses due to cycle efficiency). Cycle efficiency is then the ratio of temperature rise at theoretical 100 percent (adiabatic) vs. actual (polytropic).

Sketch of Compression Cycle:

Ps Pd

& 4



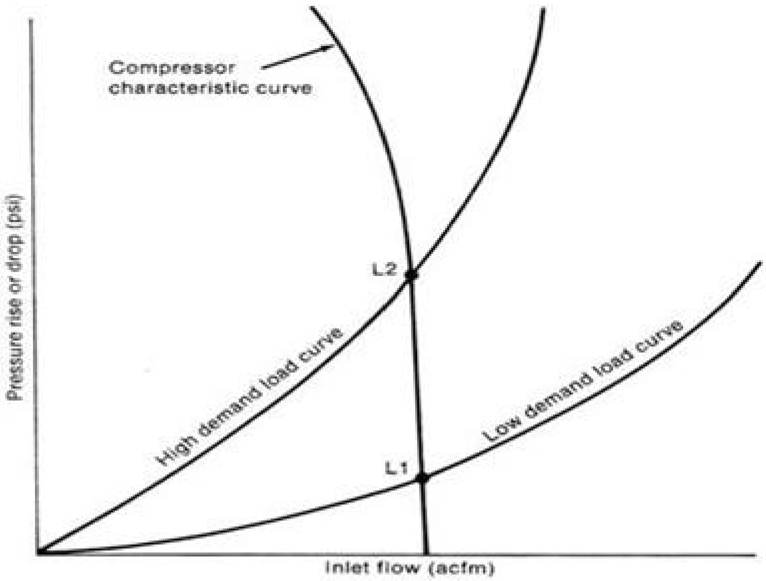
The attached sketch is of a single acting reciprocating compressor. This is a ideal compression cycle where ps is the intake or suction pressure and pd is the discharge pressure.

Point 1 is the piston dead center that gives the maximum cylinder volume and the gas in this position is at suction pressure **ps**.

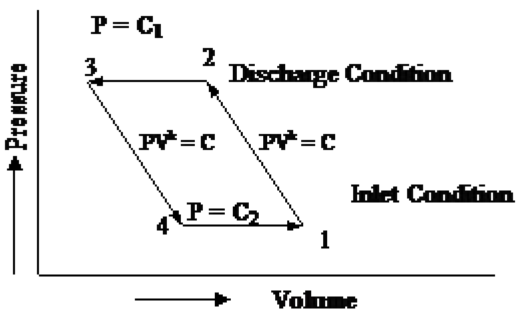
As the piston moves from point no. 1 to point no.2 the gas is compressed, volume get decreased and pressure & temperature increases as per gas law.

At point 2 the discharge valve open and compressed gas will pushed out from the cylinder at discharge pressure **pd**

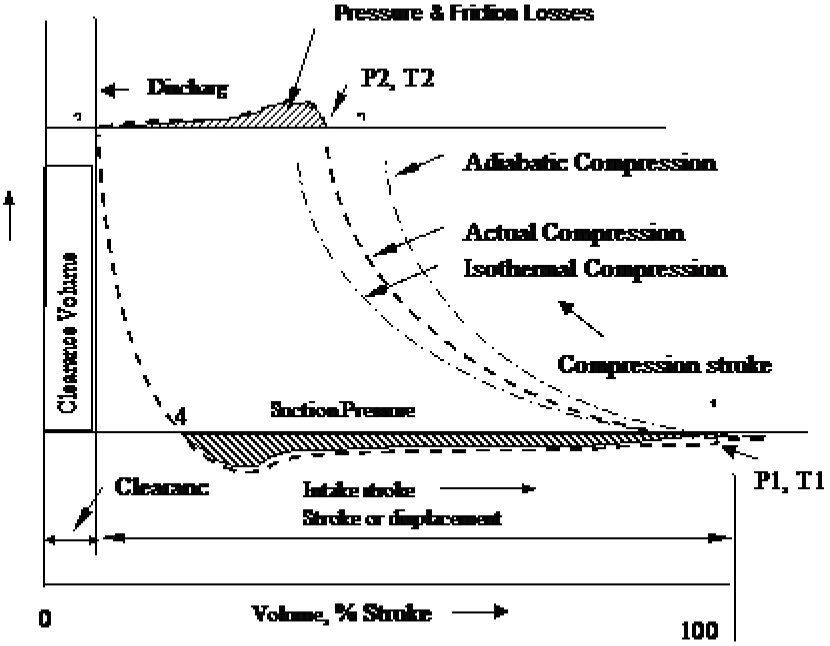
At point 3 the piston has reached the end of the travel and the cylinder is at its minimum volume.
At point 4, which is same as point 3 but the pressure og gas gets decreased again to equal to the suction pressure **ps.**
At point 4 the suction stroke starts and gets completed at point no 1.
The above compression cycle is ideal compression cycle where there is no valve losses and clearance losses have been considered. In actual machines some losses always take place through valves, valve covers, piston rod packings etc. Every reciprocating compressor cylinder are designed with head end and crank end piston end clearance. These clearances are very important as far as the efficiencies of the compressors are concerned.
Characteristic curves of the reciprocating compressors are attached in the fig. It is clear from a typical operating line that if we compare high demand curve with a low demand curve, the flow almost remains constant and only pressure gets reduced. At very high pressure characteristic shifts towards reduced flow due to slippage effect.



Characteristic Curves for Reciprocating Compressors



Ideal Reciprocating Compression Diagram



Reciprocating compressor Actual compression diagram

(Ideal reciprocating compression diagram) This cycle starts from the emptied out cylinder at the end of discharge stroke at point no4. Line 4-1:The suction valve opens at point 4. As the piston travels toward BDC, the volume in the cylinder increases and gas flow in the cylinder. The pressure inside the cylinder is slightly less than suction line pressure. This small differential pressure allows the valve to open during the suction stroke.

Line 1-2:The suction valve closes as pressure across the valve equalize as the piston has reached BDC and changes direction at point 1.The cylinder volume decreases as the piston moves towards TDC raising the pressure inside the cylinder.

Line 2-3:At point 2,the pressure inside the cylinder has become slightly greater than discharge line pressure and the resulting differential pressure across the valve causes the discharge valve to open, allowing the gas to flow out the cylinder. The volume continues to decrease towards point 3,maintaining a sufficient differential across the discharge valve to hold it open.

Line 3-4:At point 3,the piston reaches TDC and reverse in direction and at this point pressure across the valve becomes equal and this allow discharge valve to close. The volume increases, resulting in a corresponding drop in pressure in the cylinder. The gas trapped in the cylinder expands as the volume increases towards point 4. At point 4, the gas pressure inside the cylinder become less than suction line pressure, creating a differential pressure that opens the suction valves. The cycle then starts over again.

PISTON DISPLACEMENT :

The piston displacement is the net volume actually displaced by the compressor piston as the piston travels the length of its stroke from BDC to TDC or TDC to BDC and is expressed in cubic feet or meter cubic

For single acting

PD = AHE (X) S (X) RPM   Where

PD =Piston Displacement
AHE = Cross section area of Head end of Piston
S =Length of stroke

For double acting

PD = (AHE (X) S (X) RPM) +(ACE (X) S (X) RPM) Where

ACE = Area of cross section of Crank end Piston - Area of croos section of Piston rod

PISTON ROD LOAD:

In a reciprocating compressor the piston rod is always in cyclic loading condition under compression and tension at discharge and suction strokes of the compressor. Tensile and compressive load cycles goes on repeating. Therefore it become very important to select the material of construction of piston rods which is suitable in this type of cyclic loading. Let us check the piston rod load:

CRL =(HEA (X) HEPd) – (CEA(X)CEPs)
TRL = (HEA (X) HEPs) – (CEA (X) CEPd)

where

CRL = compression Rod load
TRL = Tensile Rod load
HEA = Cross section area of Head end of the cylinder
CEA = Cross section area of crank end of the cylinder-Area of Piston rod
HEPd, HEPs = Head end discharge & suction measured line pressure
CEPd, CEPs = Crank end measured line pressure discharge & suction

CRL is always greater than TRL(Tension Rod Load)

INTERSTAGE PRESSURE:

Actual interstage pressure of a multistage compressor can be have calculated from the given formulae:

Two stage Compressor:

$P_2 = \sqrt{P_1 P_3}$ Where
P_1 = First stage intake pressure
P_2 =Interstage pressure

P_3 =Second stage discharge pressure

Three stage Compressor:

$$P_2 = \sqrt[3]{(P_1^2 P_4)}$$

P3 = $\sqrt[3]{(P_1 P_4^2)}$

P_1 = First stage intake pressure

P_2 =First Interstage pressure

P_3 =Second interstage pressure

P_4 =Third stage discharge pressure

LOSSES IN RECIPROCATING COMPRESSORS:

There are many type of losses in reciprocating compressors e.g. suction valve leak losses, discharge valve leak losses, piston ring leak losses, pulsation effects, valve and cylinder gas passage losses and high cushion losses etc.

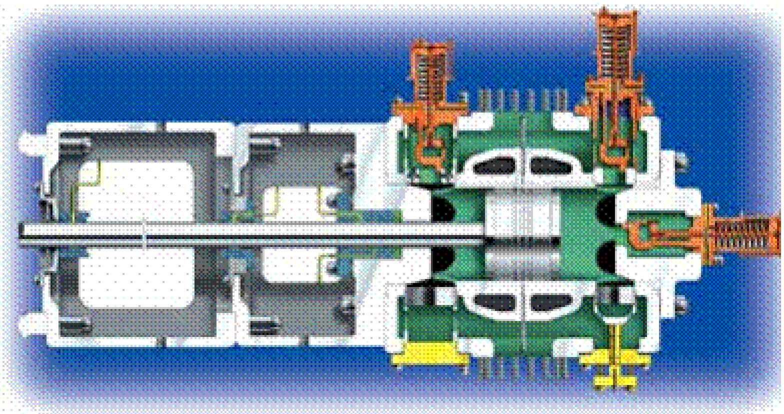
A general thumb rule is that valve and cylinder gas passage losses should not exceed 5% of IHP for particular cylinder end.

Clearance Losses:

Normally in every cylinder of a reciprocating compressor some clearance is maintained at the head end and crank end of the cylinder, between the heads and respective piston ends. When the piston reaches at the dead end of its stroke and has discharged all the gases, a small amount of gas remains undischarged in the clearance space between piston end and cylinder head. When the piston starts its return stroke, this clearance gas, at discharge pressure expands up to below the inlet pressure, before inlet valves gets opened. This way the clearance gas reduces the volume of the intake gas in the compressor cylinders and thus the efficiency gets reduces.

By increasing the clearance volume of any compressor, the compressor capacity and compression ratio can be reduced.

Generally the recommended clearance volume is 4% ~ 16% of the cylinder volume.



Cylinder assembly of reciprocating Compressor

Compressor Load Rating:

Each component in a compressor frame and cylinder are designed to certain design parameters and it should be ensured that these limits of parameters should not exceeded during actual normal operation of the compressors.

All reciprocating compressor components are subjected to alternating loads based on cylinder pressure and fatigue consideration.

Each cylinder stroke exerts a rod load on the running gear components and a frame load on the stationary components.

Frame Load = $P_{HE}\,A_P - P_{CE}(A_P - A_{ROD})$ where

P_{HE} & P_{CE} = Pressure in the head end and crank end

A_P & A_{ROD} = Area of piston and piston rod

The frame load will vary through out the cycle of the cylinder, depending on the pressure in the head end and crank end of that cylinder. These are the actual loads (Tensile load and Compressive load) the stationary components and bolting system of the component must withstand during the operation and shall be designed accordingly considering the required factor of safety for the system.

The rod load depends on many factors like the force exerted on the piston rod, cross head, cross head pin and crankshaft and this will be different for each compressor. It is the point of the frame where the frame load plus the inertia of all moving parts affecting that point, shall be of interest to a designer. For example the rod load at the cross head pin is the frame load plus the inertia of the piston with rings, the piston rod and the cross head.

The cross head pin bearing do not experience full rotary motion, rather the connecting rod oscillates on it through an arc. The load changes its direction from compressive to tensile in every cycle.

Capacity Controll of a Compressor:

In many applications it is required to vary the capacity of a compressor to meet changing process needs. There are several ways to accomplish this capacity control in reciprocating compressors e.g.

1) By variable speed of driver,
2) By manual un-loader valves,
3) By pneumatic or hydraulic finger un-loader valves,
4) By fixed volume & variable volume clearance pocket and
5) By opening the by-pass valve at the discharge of the initial stages.

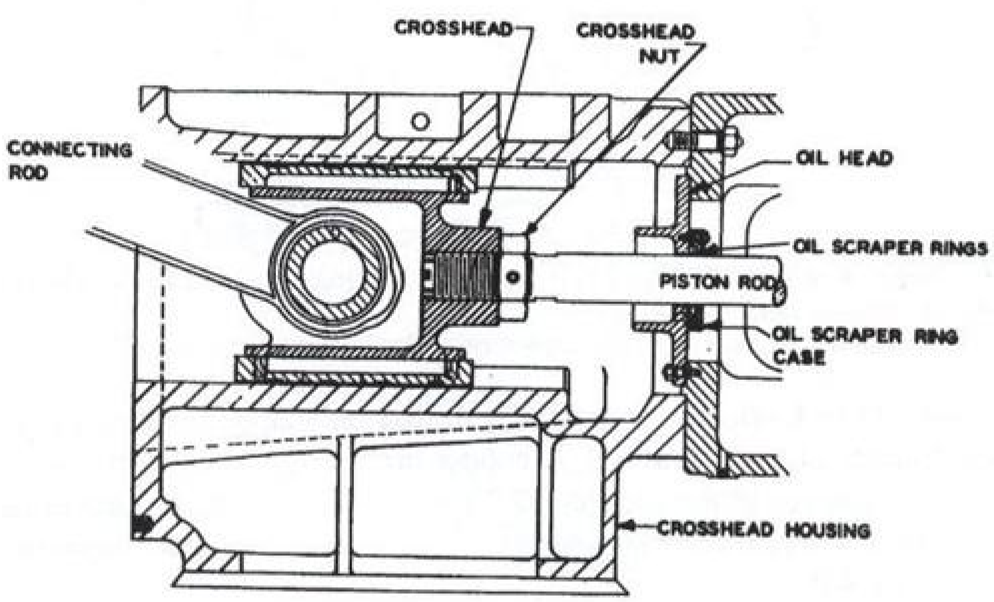
By any of the above procedure, capacity can be varied but lot of compressor power is wasted in all these methods except the 1st method but the variable speed drive for big compressors is very-very costly.

The best way to vary the capacity is by port or plug type un-loader valve. This type of capacity controllers saves the power against the un-utilised capacity of the compressor.

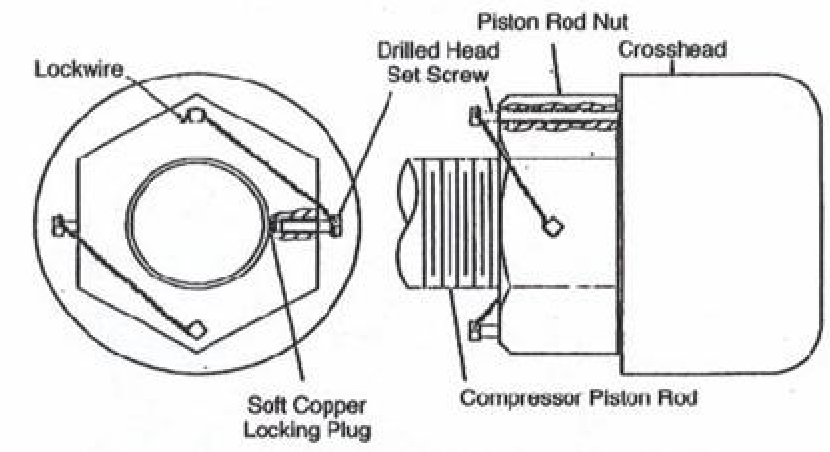
Finger type un-loader valve are very common in refrigeration industry but not used upto that level in petrochemical industries. A set of fingers or fork type components is fitted in the suction valve body and can be used in case of need from outside itself during running compressor. Fork or

fingers are driven by hydraulic or pneumatically. Fingers presses valves plate in the suction valve and thus allows the compressed gas to pass through it back to suction port. Thus capacity of the particular cylinder gets reduced.
In case of plug or port type un-loader valve a plug is operated to make the valve to function as suction valve or allow the gas to acquire the volume of the un-loader valve pocket. A ball type assembly having calculated volume is mounted on the valve assembly and is known as pocket. The movement of the plug is done through this pocket assembly. By providing this pocket assembly over the suction valve effectively increases the end clearance volume and thus the suction capacity of the compressor gets reduced.

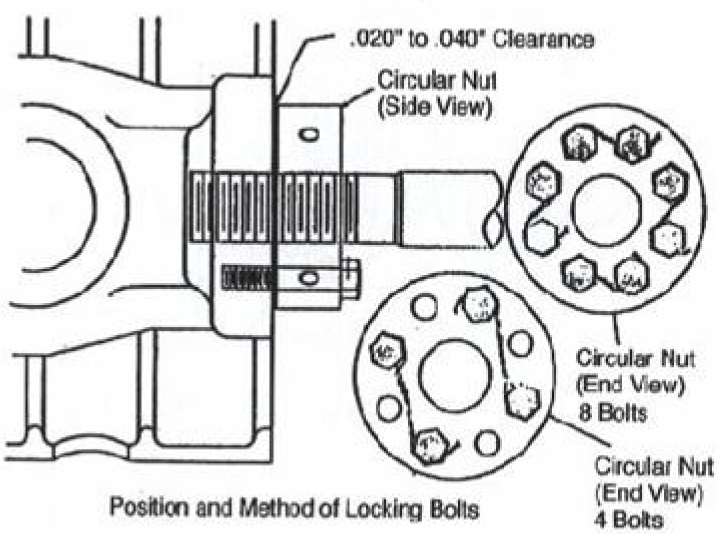
General Crosshead and Piston rod arrangement with Oil wiper rings



Conventional Pistonrod Locking arrangement with Crosshead
In this arrangement a hexagonal nut of same thread of piston rod is tightened against the cross head face and then locking pin or dowel is inserted along and perpendicular to the piston rod axis to prevent rod rotation



Multi-Bolt Type Modified Pistonrod locking arrangement
In this arrangement circular nut/bush with same thread of piston rod is used instead of conventional hexagonal nut and multiple torque bolts as per torque required is used to fasten the round nut/bush with the cross head face.These bolts actually locks the piston rod.It is very easy to assemble and dismantle the piston rod as torque bolts are very very small as compare to conventional hexagonal cross head nut.

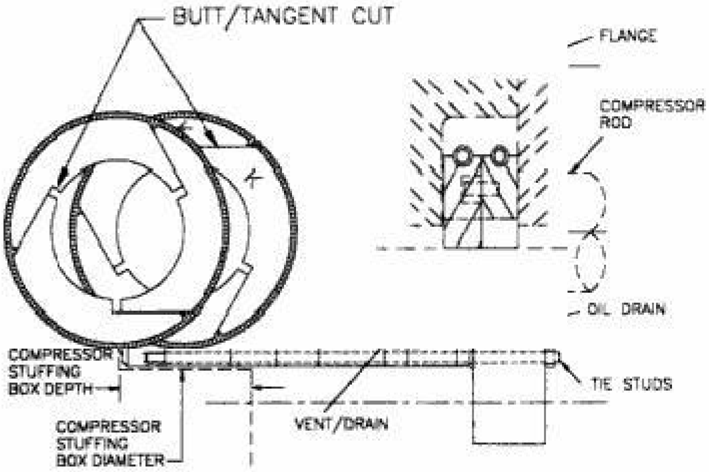


Sealing of Reciprocating Compressor:
Compressor sealing consists of a series of packing/sealing elements which is installed into piston rod gland packing housing and each packing/sealing elements restrict the flow of gas one after another and thus prevent gas leakages into atmosphere. No compressor sealing is 100% seal proof. There will be some gas leakages which is collected through vent rings and again may be recycled to suction depending upon the back pressure.The sealing/packing elements are held in separate cups within a packing case.The each sealing rings seals in two directions i.e. against the piston rod and against the packing cups perpendicular to the

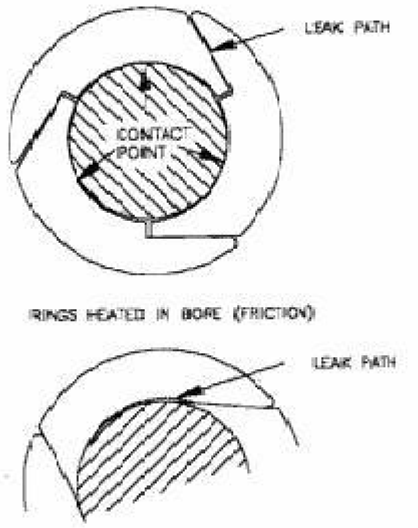
piston rod axis.Seal rings are free to move laterally along with the rod and free to float in the packing cups.
Conventional piston rod packing consists of following things

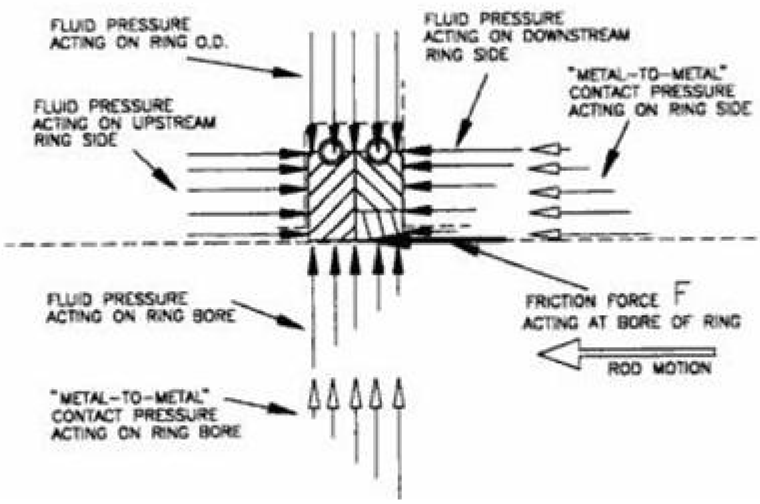
- Pressure Breaker which function as a flow restricter or to break the initial pressure rather than sealing just like barrier in case of Water Dam or Barrage which breaks the intial water tide from the water gate.Another important function of the pressure breaker is to restrict rapid expansion of gas from the packing case into the cylinder during the suction stroke as in the suction stroke gas contained in the packing case(leakages through packing elements during compression stroke and accumulated into packing case) tend to flow back into cylinder where the pressure is dropping rapidly to suction pressure.If this back flow of gas is not restricted an exploding action of the sealing/packing elements may occur which may cause premature packing failure.Pressure breaker are not generally required when pressure is below 300 psi.
- Number of Sealing/packing elements (Actaul number of sealing rings depends upon the suction and discharge pressure of the compressor)which actually seals the leakages
- Vent Ring which stop the leakages of gas from the last sealing rings into the atmoshphere.

Compressor sealing assembly may be lubricated ,water cooled or may not depending upon the application/services.
Presure drop is highest across sealing rings nearest pressure side when the sealing rings are new and as sealing rings/packing rings wears, the downstream rings are experienced more pressure drop as the path of leakages increased with wear.A reverse drop exists across some rings during suction stroke i.e. gas will flow back into the cylinder from the packing case.
Conventional piston rod packing consists of one metallic radial cut rings and one metallic tangential cut rings i.e. one set of radial cut and tangential cut rings installed in one packing cups and there may be several packing cups. Vent ring consists of two tangential cut rings.
In case of Sandwich packing each packing set consists of one radial cut metallic rings, one tangential cut non metallic rings and one metallic radial cut back up or anti extrusion ring.
In some cases both the radial cut and tangential cut may be non metallic rings but, the anti extrusion ring must be metallic.
The material of metallic rings may be bronze, cast iron babbitt etc and the non metallic rings may be of carbon graphite, PTFE, PEEEK or other plastic materials.



General Piston Rod Packing Assembly

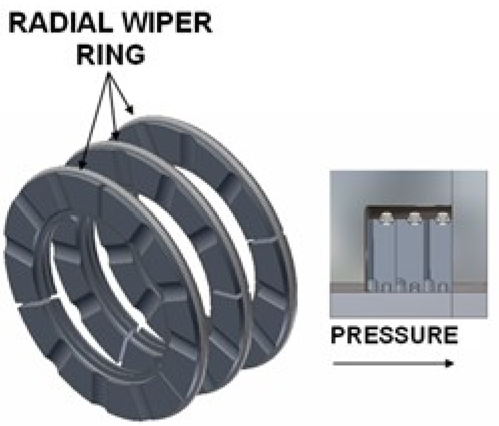




Construction of Tangentail Rings and Fluid Pressure on Packing/Sealing Elements

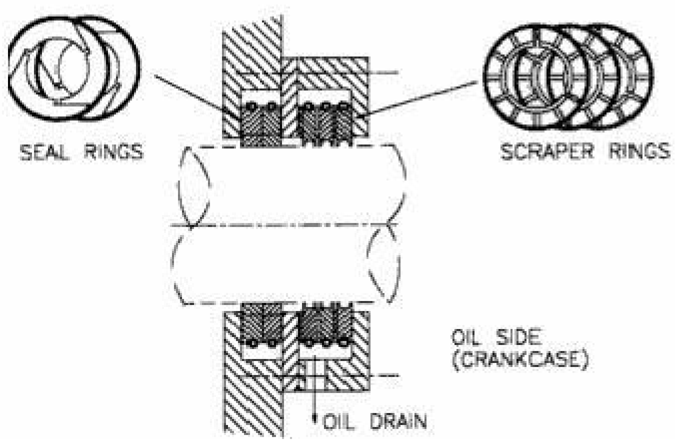
Wiper Ring or Oil Scrapper Ring or Diaphragm Packing:

To scrap out the oil from the moving piston rod, oil scrapper ring or Oil wiper rings are installed in the Diaphragm assembly, between the cross head guide distance piece and the Cylinder block and the scrapped oil is drained back to crank case frame. One set of Seal ring (Tangential ring set) is also installed with oil scrapper ring in the diaphragm for the breathing action of the crosshead.
These wiper rings and seal rings are generally made of bronze material and are locked by Garlok springs.
These rings should be free to slide with each other but should not have high axial clearance



Wiper Ring set without tangential seal rings **Wiper Ring set with Tangential seal rings**

These are the basic and most commonly used wiper ring combinations. The only function of these rings is to wipe the rod and drain the fluid away. The three ring configuration is doweled to prevent alignment of cuts with respect to each other.



Assembly of Oil Wiper Rings

Diaphragm assembly

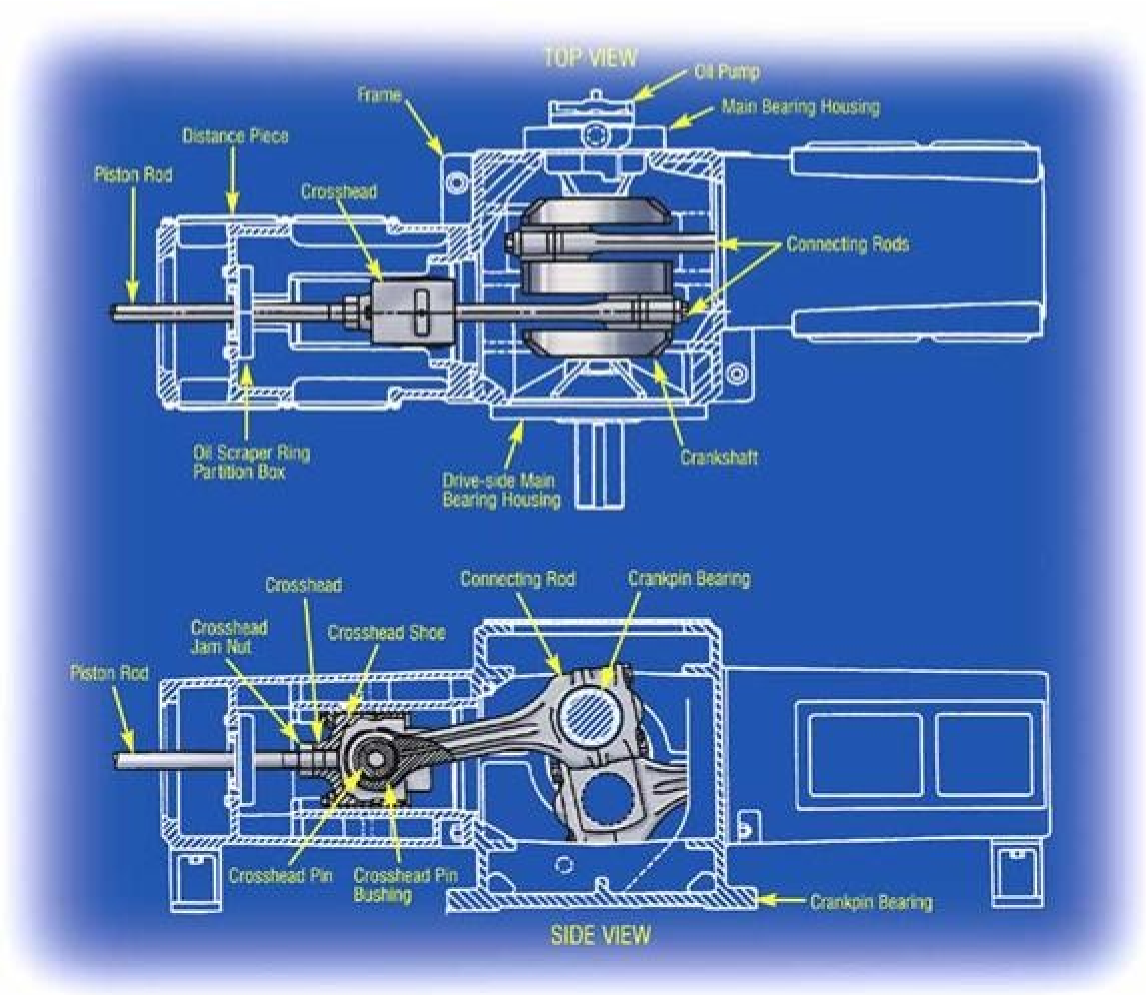


Figure 2 A. Two-throw HSE Frame and Running Gear

Maintenance Activity of Reciprocating Compressor:

Maintenance is the field where reciprocating compressors get minimum marks. “Wait until destruction” type of negative thinking is very bad and extreme.Maintenance should be condition monitoring type so that unwanted equipment downtime can be avoided and it can be done in planned way. For condition monitoring equipment opearational and maintenance history and analysis is very much required.

Generally in reciprocating compressor the frequency of replacement of piston rings, piston rod packing, valve assembly, piston, piston rod, cylinder liner is high. In nominally well lubricated machines, the main bearing, big end bearings, cross head pin bearing, cross head pin, cross head shoe, connecting rod, crank shaft are very rare and generally replaced once in a while due to some major problem in the system. The running maintenance of reciprocating machines is of very high importance as this directly effects the production of the plant. Any minor variation in replacement part will effect the maintenance time and hence the machine performance.

During repair/ replacement of all frequent consuming spares dimensional measurements and material of construction shall be selected carefully and care shall be taken to avoid frequent failure of such parts. Root cause analysis of repeated failures shall be studied and modifications should be done. Important parameters, which shall be noted during maintenance of compressor and taken care, are as below:

-ID of cylinder liner
-OD of piston
-Grooves of piston rings and rider rings on piston
-End and side clearances of piston rings / rider rings
-Perpendicularity of piston ring groove walls
-OD of piston rod at various locations.
-Depth of packing cups
-Floating of packing rings
-Perpendicularity of packing cups to rod axis
-Cross head pin bush clearance
-Cross head shoe clearance
-Deflection of piston rod
-Seat of valve housing in valve port
-Lift of valve plates
-Surface of valve seat
-Diaphragm packing etc.

The above measurements shall be taken on the protocol format and shall be compared with the original dimensions. Dimensions of parts shall be within permissible limits of variation otherwise replacement of the affected part shall be planned. Weared out parts if used may fail without giving significant life and may deriorated other mating parts also. Such failure may arise without any explanations.

The end gap and side clearances of piston rings/rider rings should be maintained according to OEM or manufactures recommendations. Too much side gap will break the piston rings due to high relative motion in the piston ring grooves and higher end gap will blow off the piston rings. Excessive tight clearance will seize the piston rings in the piston ring grooves and no compression will be achieved in that cylinder. These clearances are kept considering the difference of coefficient of thermal expansion of the rings at operating temperature with respect to piston. These values depend upon the material of construction of piston rings and piston.

The following malfunctions can occur to a compressor cylinder regardless of the gas pumped and whether or not it is double acting or single acting,large or small diameter , multistage or single stage.

-Exceeding assigned rod load

-Accelerated wear and scuffing

a)Piston to liner

b)Piston rings

c)Piston Rod Packing

-Valve breakage

-Knocks,noises and vibration

Exceeding Assigned Rod Load-It is essential that operators and mechanics should understand rod load.Most major casualties such as broken piston rods,damaged crossheads,cross head pin,broken cylinder to distance piece or distance piece to crankcase or frame failure are generally caused by exceeding the maximum rod load.These failure does not occur instantly exceeding the rod load but ,after prolong operation in over rod load condition.The frightening aspect of this that the failure can happen within just few revolutions after the infraction or after a period which slowly detoriate the machine condition and atlast failure.

By explanation when the piston moves towards head end the discharge pressure force(Pd)on the piston ends tends to compress or buckle the piston rod.At the same time in the CES gas is entering into the cylinder behind the piston at suction pressure(Ps) and putting suction pressure force at the back of piston.The two force are opposite in direction but since discharge pressure is higher than suction pressure the net force tends to compress the rod which is called “Rod Load Compression”.So it is basic that if the suction pressure decreased or discharge pressure increased the net compression load on the rod increases. So it is very necessary that there should not be too much pressure deviation in suction and discharge pressure.

Again when the piston moves toward crank end and compressed gas the net force of the suction and discharge pressure results in tension load on the rod which is called “Rod Load Tension”.

Although the tension and compressive forces are absorbed by the rod , other parts such as head bolts, piston, connecting rod and bolts, crosshead, pin bushing, frame are likewise stressed.

Loading and unloading cylinders of multi stage compressor changes interstage pressure and so the compressor should be loaded and unloaded very carefully and sequentially and also start up of compressor to avoid any abrupt change in interstage pressure which can exceed the rod load.

Knock , noise and Vibration:Knock noises and vibration are good indications of trouble.The maintenance people should have enough knowledge about the knock and noises and it should not be misinterpreted which can create panic.A common type of knock is caused due to hitting of piston at the CES or HES caused by improper clearance.Another type of knock sound may come due to loosening of piston nut.This is the nut that secures the piston with the piston rod .If it becomes loose by 0.003 inch it will knock very loudly.Other type of knocking sounds are due to loose valve assemblies, liquid carry over, loose piston rod packing assembly.

Crank Shaft Deflection:The crank shaft web deflection can be measured with connecting rod assembly and without connecting rod assembly but, it is adviseable to measure/check without connecting rod as this will give the exact true value.With connecting rod the deflection can be taken from the given formula.When measuring without connecting rod the web gauge can be installed in position A and in case of with connecting rod the web gauge should be positioned at point B with special deflection gauge attachment and with the following calculation.If the deflection gauge is positioned at point B which is out on the counterweights, the deflection recorded there would be twice the actual deflection measured at point A.

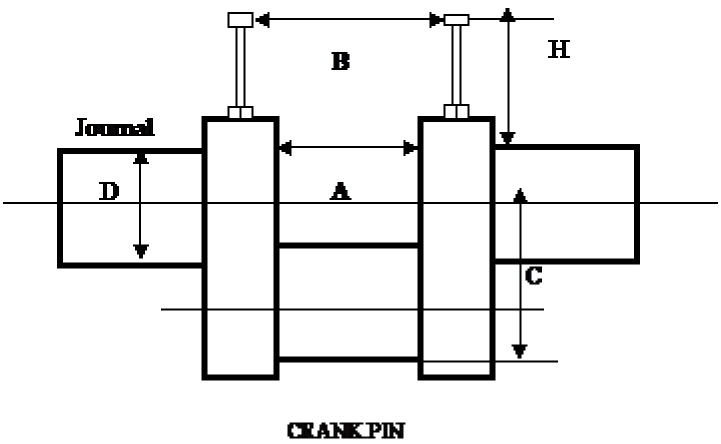
Measuring point A: Normal (Measure without connecting rod)
Measuring point B: Extension (Measure with connecting rod)
When measure deflection at B point, calculate to A value.

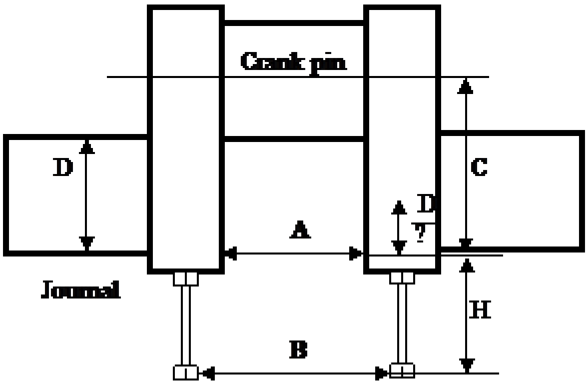
Deflection A = B X C/(H+C)

B= Actaul Reading Value at B Point

C= Stroke/2 + D/2

D= Journal Diameter





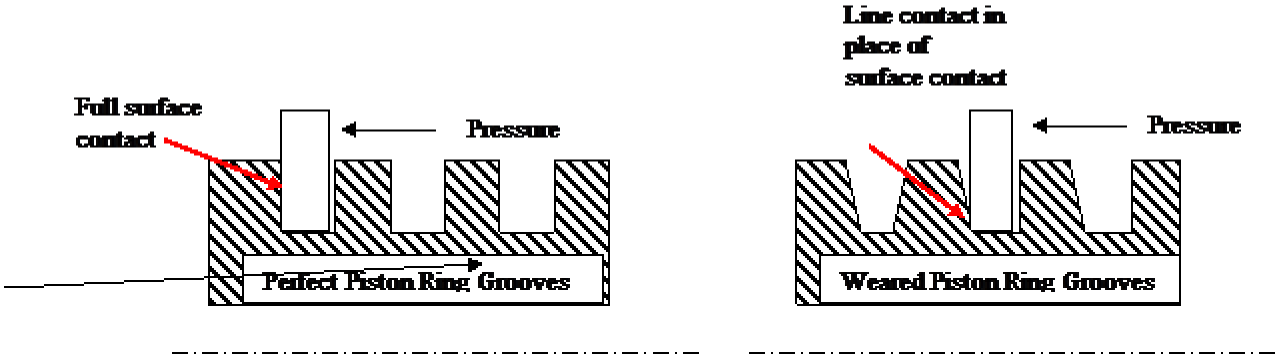
Suppose the crank pin is at down position and this is designated as Zero or starting position.Let us assume that the main bearing journal to the right is low due to bad or wornout main bearing.The crank web would then be spread apart.The web gauge is set to zero at this position and when the crank is rotated to the up 180 degree position the web move inwards which registered a minus (-) reading on the dial.If the starting point sets to crank pin up position then the reading will be reversed i.e.when the dial gauge is set to zero position at cranpin upwards and then after rotation 180 degree rotation the webs would spread and the dial would register a plus(+)movement.The magnitude of deflection in both cases for the same cause (bad bearing or lower pedestal) but, the sign (+), (-) of the dial gauge would be reversed.

Generally deflection at 0 degree and 180 degree i.e. up and down position of crankpin is taken to judge whether the clearances of main bearing or ok or not , or whether the crank shaft is sagged or bowed due to bad bearing or bearing pedestal lowered .The 90 degree and 270 degree positions are normally used to determine whether the main bearings are out of alignment in ahorizontal plane. If the bearing saddles are out of alignment in a horizontal plane , the signs at 90 degree and 270 degree would be reversed but if the 180 degree position has excessive deflection caused by journal being low , the reading may carries upto 90 degree and 270 degree.So it requires experience to correct the deflection.Some times the reading will be high at 0 degree and 180 degree but the bearing clearances may be found ok and this is due to lowering of frame pedestal ,So in this case the level of frame should be corrected.The severity of deflection depends upon the length of stroke and RPM of the crank shaft.As the same deflection may be tolerable in low RPM and the same deflection may be unacceptable at high RPM as in case of high RPM the frequency of web inwards and outwards will be more than low RPM.

EFFECT OF WEARING OF MOVING PARTS

Piston Ring Grooves / Piston Rings :

The grooves wall surface of the piston rings in the piston should be perpendicular to the piston axis and should be within tolerance limits of dimension. Generally after prolong use of piston the piston rings, grooves size gets changed and some times may become taper as shown below. This variation in groove size may increase the gas leakages along the piston rings. Due to damaged or tapered piston ring grooves the piston rings does not get full surface contact with the piston groove wall. In such cases, due to line or point contacts, piston ring malfunctions in the compression stroke and may failed early.

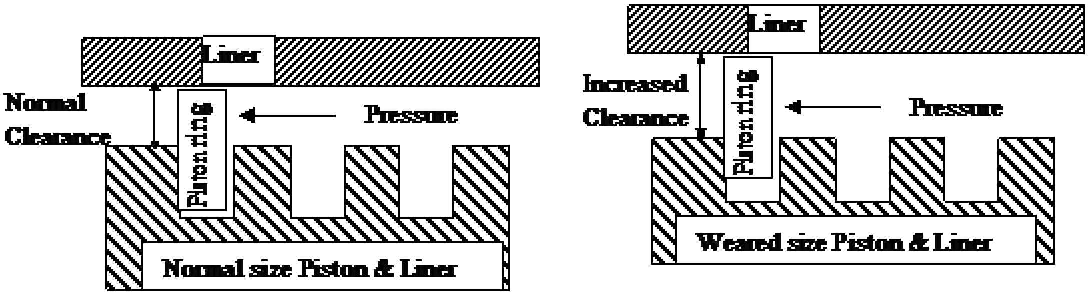


Effect of Wearing of Piston Ring Grooves.

Piston / Cylinder Liner:

If the piston OD get reduced due to worn out or Cylinder Liner ID gets enlarged, then piston to liner clearance will be more, the projection of the

piston rings will become more than the desired/designed values. In such case, Piston rings will be subjected to higher thrust load and similarly piston rings groove wall will also be experiencing increased thrust load. Weak material in this case will break away, it may be piston ring or piston rings groove wall.
With increased piston-liner clearance, piston will also experience jumping during the motion. This jumping is very deteriorating for piston rod, Cross head nut etc.



MAINTENANCE OF PISTON ROD PACKINGS:
Piston rod packing is very important element for the reciprocating compressors and plays crucial part in efficiency of the compressor. There are many types of packings available in market for different services however common features are lubricated or non lubricated packings. In the packing assembly lot of heat is generated and packing elements may loss their properties at high temperature if heat is not removed from the assembly. Generally cooling water connection is done in a way to indicate flow of water through packing assembly. It may be with flow indicator or return line is kept open to atmosphere.
During replacement of piston rod packing the installation of packing elements in the piston rod packing cups plays an important role. The packing elements should be as per sequence recommended by manufacturer, i.e. radial cut rings should face towards pressure side and tangential cut or seal ring should face cross head side. In case of sandwich packing the radial cut metallic rings faces pressure side, tangential cut non metallic rings(Sandwich) works as sealing rings which actually seals the leakages and anti extrusion ring or back up metallic rings after seal ring towards cross head side. The back-up ring is generally larger in ID as compared to seal ring. The back up rings takes the heat generated by the radial cut ring and tangential cut ring and dissipated the heat.
The floating or axial clearance of packing ring elements in the packing cups should be as per OEM or manufactures specification and this also depends upon the material and type of packing i.e the depth of packing cups grooves should be more than the thickness of the packing rings. Generally the floating should be 0.15 ~0.20 mm.It actually depends upon the thermal expansion of the packing elements during actual operation

The packing cups face should be properly lapped before assembly to prevent any leakages and should be perpendicular to the axis of the piston rod. all elastomers should be of proper dimension and material and of required shore hardness. If the packing is lubricated and water cooled then the passages for lubrication and cooling water should be clear and in sequence.
The performance of piston rod packing also depends upon the deflection of the piston rod and deflection also depends upon the piston OD , cross head shoe liner clearance and ID of the cylinder liner. The axis of the piston rod, seal housing, cylinder liner and cross head should be in perfect alignment.

<u>VALVES</u>:

Valves are the elements which allows the gas to flow inside the compressor and from the compressor to the high pressure system. These are called inlet or suction valve and outlet or discharge valves respectively. For reciprocating compressors, valves are the most venerable part for the maintenance. Normally life could not be predicted for the valve assemblies however good quality valves gives quite good life.
Different manufacturers make different type of valves. Major categories are as below:

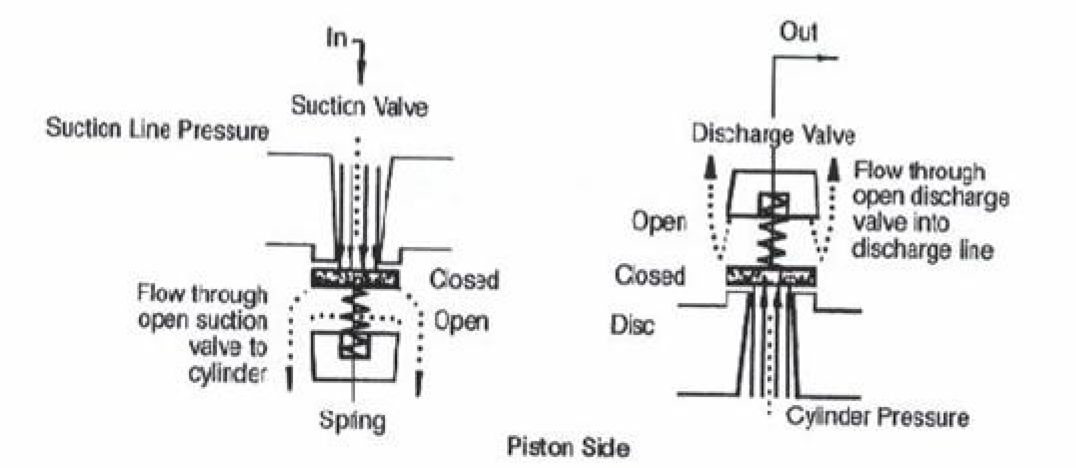
1) Damp plate type valves
2) Channel valves
3) Puppet type Valves
4) Bullet type puppet valves

Under the above categories, further variations are also available under modified categories, which have been developed based on further R&D.
During maintenance of valve assembly following parts shall be checked carefully for better performance.

i) Valve seats,
ii) Valve plates
iii) Springs and lift of valve plates in the guard seat
iv) Valve port where the valves are installed over gaskets,

Some times during valve inspection maintenance, valves are found OK but due to pitting or erosion or corrosion on valve seat in the compressor valve port / housing gas by-pass take place giving wrong indication of defective valves.
Some times it is also observed that the lift of valve plates in valve assembly is less then the desired lift, in such cases compressor capacity will decreased due to obstruction of flow (hence less flow). Also if the lift of valve plates is more then designed values, the tendency of failure of valve plates will be more.

<u>Actual action of valve plates and springs during suction and discharge and flow of gas</u>

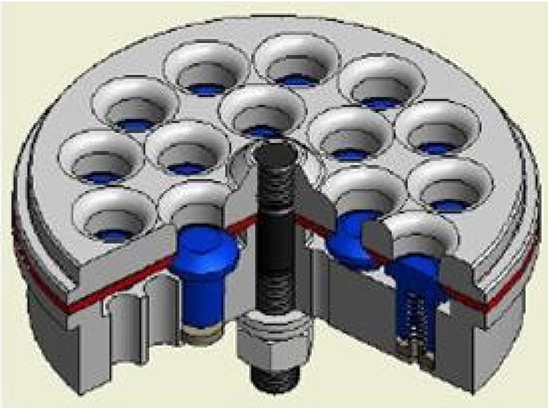


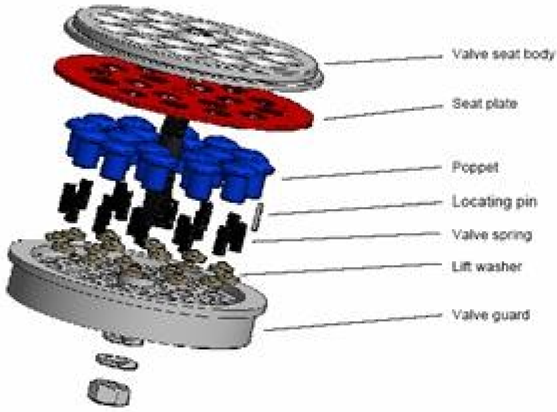
Damp plate type valves   Modified Damp plate valve

Poppet type Valves:

These are very efficient, easy to assemble and cost effective valves. However the presence of condensate in the inlet gas damages these valve badly. The disadvantage of these type of valves is the flow of suction valve damaged puppets to discharge valves making discharge valve also defective.

These valve have further been modified by changing the design of puppets to bullet shape. These are also known as bullet valves.





MAJOR OVERHAULING OF COMPRESSORS:

Major over hauling of big reciprocating compressors are done as per recommendations of manufacturer. Generally 6 to 8 years interval between two major overhauls is considered to be a good schedule.

During major overhauling of compressor complete machine is stripped off and is assembled back as if new machine is being assembled. Each and every part is marked during the disassembly and is stored at the floor in sequence. Measurements are taken and recorded. Measurements are compared with protocols and replacement of parts required is listed out. New parts are also brought to the floor and cleaned thoroughly and again measurement of new parts is also recorded.

Before assembly, crank case level is checked and if required correction is done by changing / modifying sole choke plates. Foundation bolts are tightened and final level readings are recorded.

Main bearings and crank shaft is assembled and clearances of main bearings are recorded. Crank web deflection is recorded. Connected rods are assembled and crank pin bearing, small end bearing and cross heads are assembled. Trueness of connecting rod bore, cross head pin shall be recorded.

Distance pieces with pedestals are assembeled and foundation bolts are tightened after leveling.

Cylinder blocks are assembeled and piston assemblies are mounted as per procedure. Piston rod defelections shall be recorded and shall be corrected if required.

Alignement of compressor with motor/turbine/gearbox is done and coupling bolts are tightened.

In last cylinder valves are assembled.

Through cleaning is done manually and then by dry air and crank case covers are fitted. Lubricating oil is charged in the frame.

Compressor freeness is checked by manual baring and then motorized barring is done for some time. In general barring time is recommended by OEM.

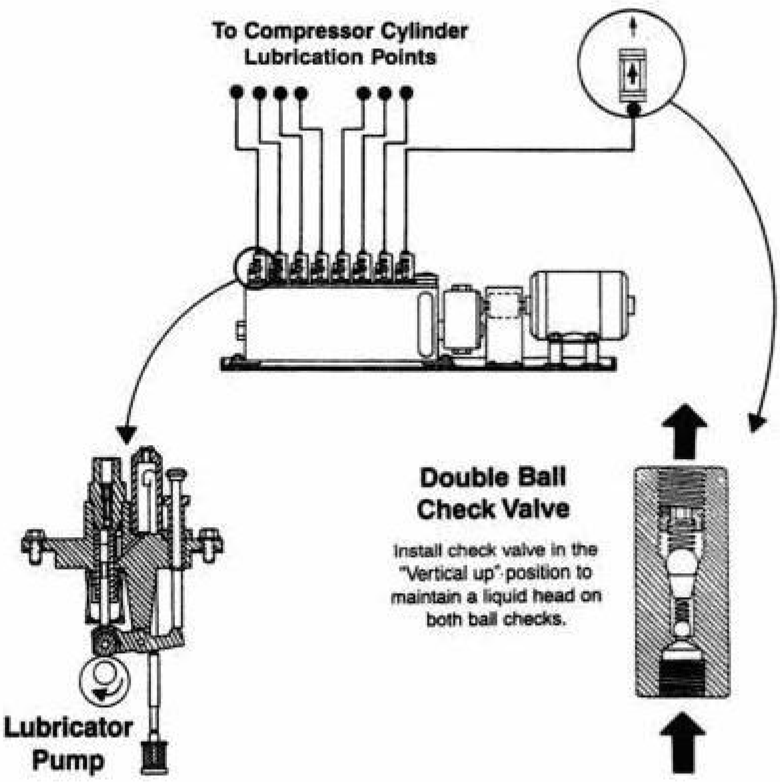
Finally compressor is run on idling.

Loading is done in systematic way as per procedure.

Minor maintenance jobs of the reciprocating compressors are done on as and when required bases during the opportunities.

Lubrication System of Compressor Cylinders and Packings

In reciprocating compressor main bearings, big-end bearings, cross head pins etc are lubricated by Gear type or Lobes type Oil pump driven by either directly through crankshaft(directly coupled with the crankshaft)or through separate motors or any prime movers.
In case of lubricated cylinders and packings separate plunger type lubricators of low pressure or high pressure is used which is driven through cam shaft arrangement and discharge of lubricators is fed into cylinders and packings through non return double ball type check valves. The function of these NRV is to fed lubricating oils into cylinders and to restrict gas coming out.
In normal case if the NRV is functioning OK then the NRV will not become hot but, as soon as NRV started malfunctioning hot discharge gas will back flow through this NRV and it will become hot and it should be replaced immediately as oil will not flow into cylinders which may damage piston rings, cylinder liners, packings etc.



Material of various componenets

The material construction of reciprocating compressors plays an important role as if any critical components fails not only equipment will be stopped but serious damages may occur also. During import substitution this material of construction and many other factors should be considered.

Here given some critical components material of construction:
Crank Shaft:The crank shaft should be forged carbon steel as this transmit rotating power and in some small machines cast iron also used. Forged carbon steel are ASTM 668,AISI 1020,AISI 1045 .

Connecting Rod:This should be also forged steel but in some small compressor it may be made of cast steel also

Piston Rod:The piston rod is under compression and tensile load constantly.The material of construction is forged steeli.e.EN41B,AISI 4140,AISI 1037,AISI 420,AISI 410.The surface of the piston rod should be hardened by nitriding, induction hardening and should be polished finished.The hardness shall be minimum 45~55 HRC.As per latest API-618 the surface hardness should be minimum 65 HRC which can be achieved through tungsten carbide coating done through HVOF.The coating should be such that it should not be pilled off during operation

Piston: The material of construction of piston can be nodular cast iron , alumunium alloy or steel in case of small bore with higher compression. In case of nodular cast iron ASTM A536,in case of alumunium alloy LM3 and in case of steels ASTM A351 .

Cylinder Liner: Cylinder liner should be grey cast iron with alloy ASTM A 278 and hardness shall not be less than 200 BHN. In case of very high pressure hardened cast steel may be used.

Packing Cups:Piston rod packing cups shall be of AISI-410 forged or from bar stock.The piston rod packing cup face should be lapped and surface hardness should be 35~40 HRC.

Piston Rings & Rider Rings:The material of construction of piston rings/rider rings may be Bronze or Cast iron in case of metallic piston rings and filled PTFE(Carbon filled, Graphite filled, Bronze filled,Ceramic filled,Glass filled etc)PEEK,Polymide etc in case of non-metallic.The MOC depends upon the service and application and some times it also depends upon the OEM and end users.

Main Bearing and Big End Bearings:The material of big end bearing and main bearings shell is of carbon steel or steel and the bearing lining shall be White metal of tin base or lead base or of tri-metal. The white metal lining is of WJ-3 standards

Cross Head Pin:The material of construction of cross head pin shall be forged steel.

Small End Bearing/Cross Head Bushing:The material of construction of cross head bushing may be of bronze or alluminium alloy or white metal lined

Valve seat and Guards: The material of construction of valve seal and guards shall be cast stell in case of low pressure cylinders and in case of high pressure cylinders material shall be forged steel AISI-410 or cast stainless steel of 316.The valve seat should be lapped and hardness shall be 40~45 HRC.

Valve Plates: The material of valve plate may be AISI-410 from bar stock with 13% chromium or PEEK in case of non metallic. The valve plate should be hardened and should be flat.

Valve Springs:The material of valve springs may be Spring steel, Inconel alloy ,Hastalloy or 17-4 PH depending upon the pressure and application.

Studs and Bolting material:All studs and bolting material shall be alloy steel of ASTM A193 and in case of non critical services carbon steel.

Screw Compressor:

Screw compressors are rotary positive displacement type compressor. This type of compressor employs a rotating action to compress and eject the entrapped gas. Screw Compressor consists of two mating, helically grooved rotors set, positioned in bearing at either end of the compressor casing. The rotor, which has generally four to six convex lobes is called the male rotor and the mating rotor having five or six concave lobes is called female rotor. The lobes profile is of special design having an unsymmetrical profile which gives high compression efficiency. Generally Rotors are forced lubricated type to avoid wear and the leading edges do not come in contact with the casing as there is some clearance maintained between rotor edges and compressor casing. Non lubricated screw compressors are also common.

The length and diameter of the rotors determine the capacity and the discharge pressure. The larger the diameter the greater the capacity and longer the rotors ,the higher the pressure.
The screw compressor are fitted with Mechanical seal of Conventional seal , Bellow type seal or Dry gas seal to prevent gas leakages from the drive shaft.

Suction phase: The pair of lobes unmesh on the suction port side creating space for the gas to occupy. Gas flows in the increasing volume formed between the lobes and the casing until the lobes are completely unmeshed.

Transfer phase: The trapped pocket of gas isolated from the inlet and outlet ports is moved circumferentially within the screws at constant suction pressure.

Compression phase: When re-meshing starts at the inlet, the trapped volume of gas starts reducing and the gas is gradually moved helically along the screw profile, while simultaneously being compressed towards the discharge end as the lobes mesh points moves along axially.

Discharge phase: Discharge starts when the compressed volume has been moved to the axial ports on the discharge end of the compressor and continues until all the trapped gas is completely purged in the discharge port.

Bearings: A sleeve type white metal lined bearings are generally used for main journal bearings. These bearings receive radial loads only.

Great amount of thrust load comes in the screw compressors. Individual thrust bearings (Thrust ball bearing or Thrust pad bearing) receives the axial loads of male and female rotors which results from the pressure of the gas and the interaction of the drive load and the helical configuration of the rotors. Thrust bearings are of very high importance in screw compressors.

Balance Piston:
During operation of the compressor, the male rotor runs with much high speed than the female rotor and so the thrust of male rotor is higher than female rotor. To compensate this extra thrust of male rotor a balance piston is provided on the male rotor and oil pressure is given to the balance piston from the opposite side of driving axial and to balance the axial load. Some compressors have balance piston on male rotor only and while some compressor may have balance piston on both the rotors but of different diameters depending upon the axial thrust. The balance piston can be mounted on either side, discharge end or suction end. Accordingly the direction of oil pressure on the balance piston is set in the casing.

Fig. 3

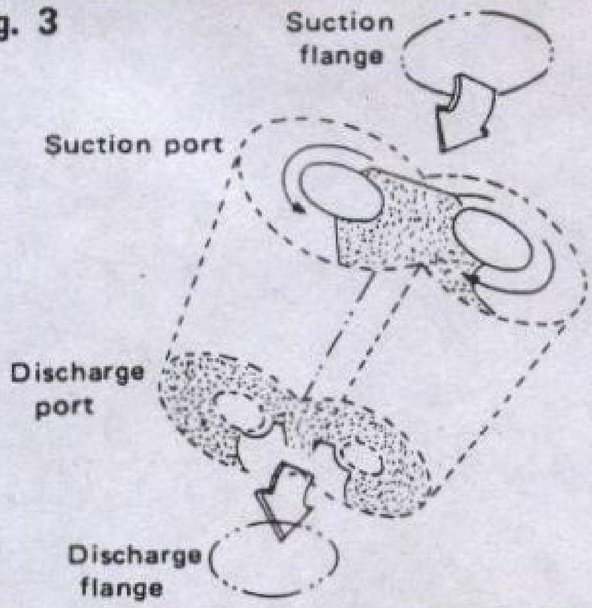

Suction flange
Suction port
Discharge port
Discharge flange

Fig. 4


Suction port
Suction phase

Fig. 5

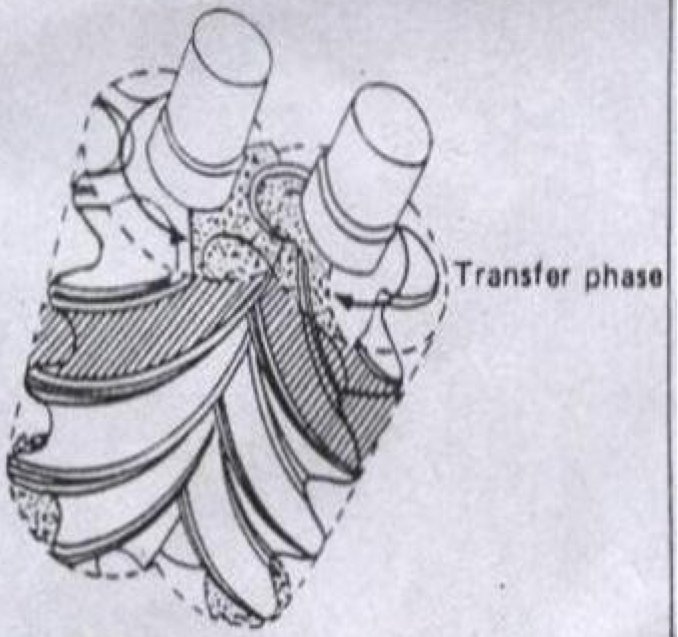


Fig. 6

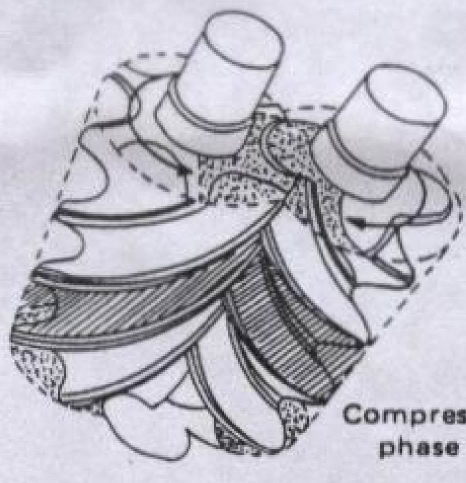
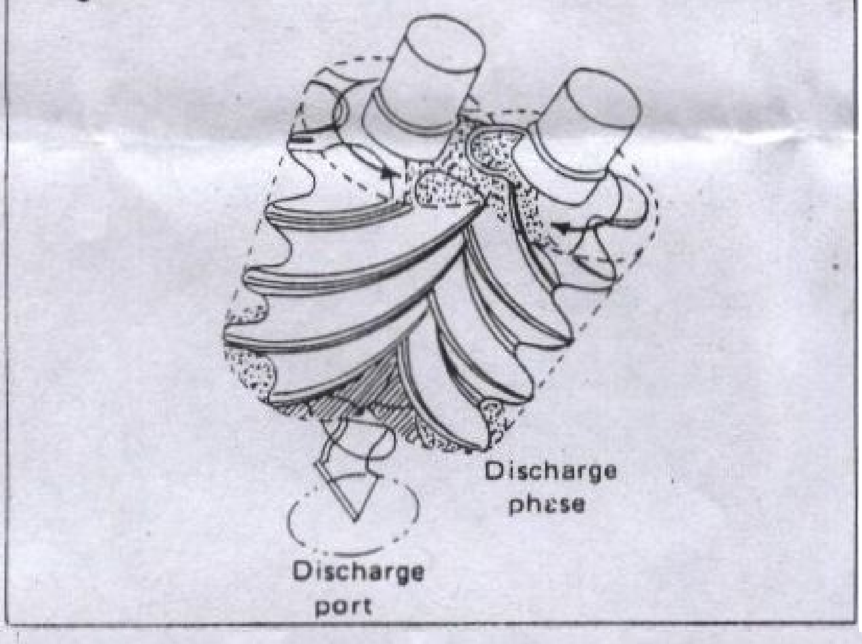
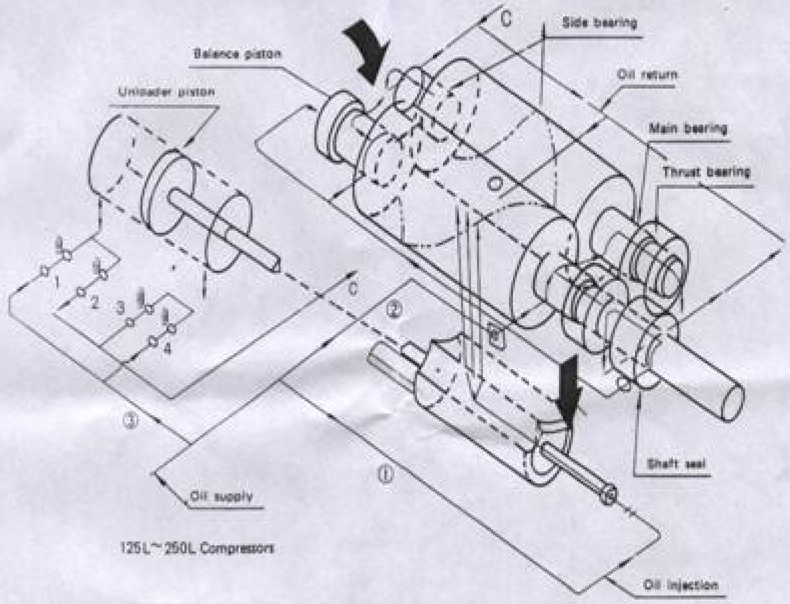


Fig. 7





125L~250L Compressors

Fig. 11

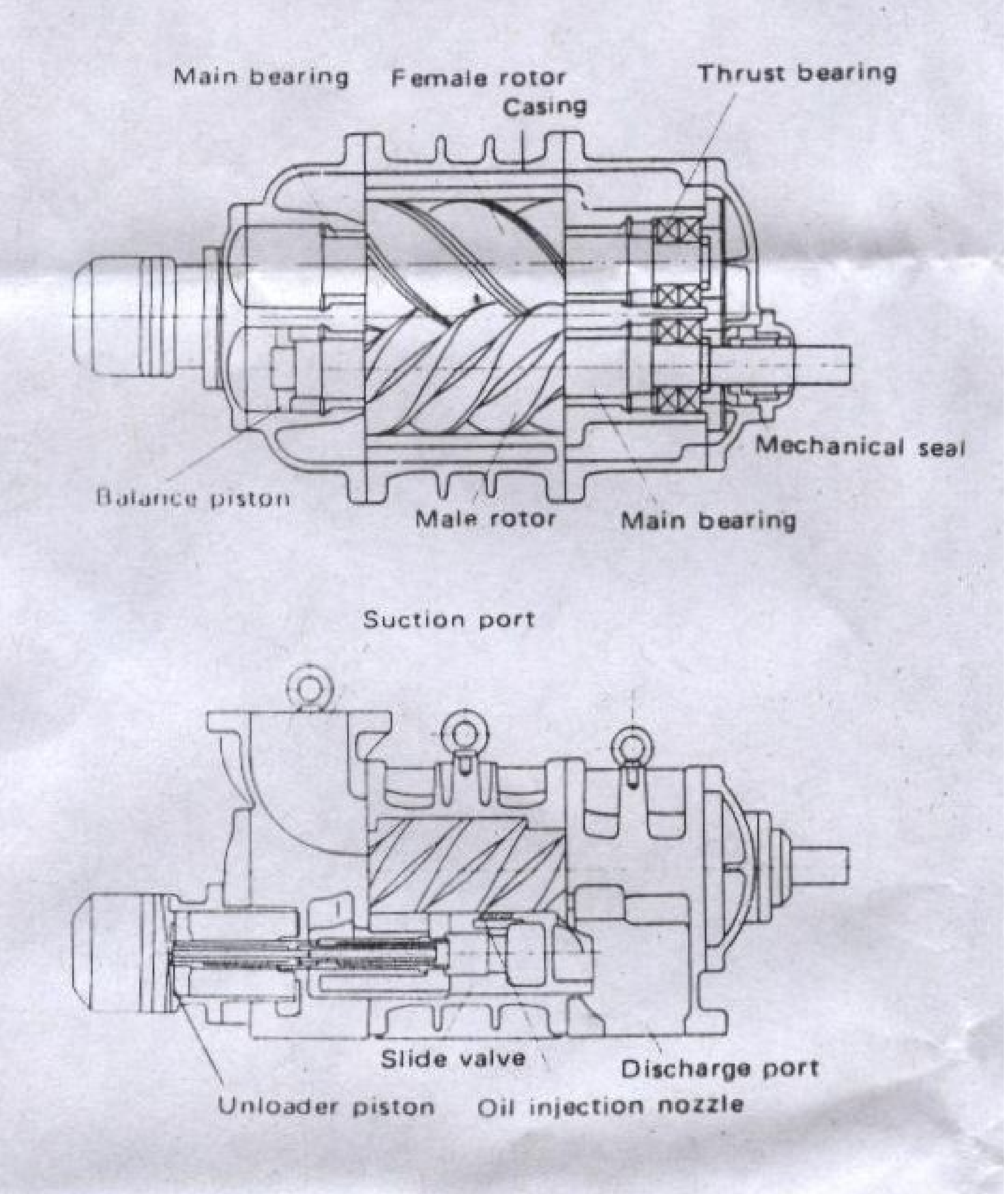


Fig. 1

Running Clearances Of Screw Compressor

Generally the running clearances of journal bearing is 1 thou(.001”) per inch diameter of journal but, different OEM recommends their own running clearances according to RPM, Oil viscosity and applications. Below given of some OEM running clearances

Diameter Clearance of Journal Bearing = 0.13~0.16 mm
Diameter Clearance of Balance Piston to Ring =0.04~0.12 mm
Rotor Discharge End Clearance =0.10~0.15 mm(When rotor push to suction side)
Rotor Suction End Clearance =0.55~0.72 mm

The clearance between Rotor pair is 0.003” and the clearance between rotor set and casing is 0.005”.

There are always some advantage and disadvantages of every type of compressors. All requirements cannot be fulfilled by any one type of compressor only. The comparison between Screw and reciprocating compressors can highlight the advantages of these compressors.

Compressor Comparisons

Reciprocating

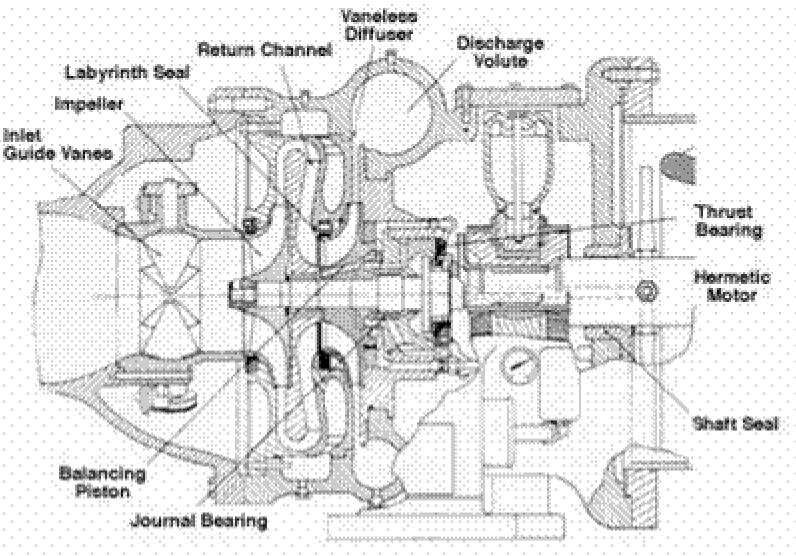
- Cost advantage as a single-acting, air cooled unit below 30 hp.
- Double-acting units used above 250 psig and in non-lubricated applications.
- Normally used for heavy-duty, continuous service.
- High overall efficiency.
- Operates efficiently at partial loads.
- Saves horsepower in no-load conditions.
- High initial and maintenance costs.
- Large sizes require heavy foundations.

Rotary Screw

- Used more in 150 psig, lubricated air systems above 30 hp.
- Used for constant-volume, variable-pressure applications.
- Oil or water is used for sealing and cooling.
- Must vent reservoir to lower power consumption when unloaded.
- Delivers high air volume in a compact space.
- Smooth, pulse-free output.
- Easy to install and maintain.
- Low vibration.

Roots blower is a positive displacement type device which operates by pulling air through a pair of meshing lobes not unlike a set of stretched gears. Air is trapped in pockets surrounding the lobes and carried from the intake side to the exhaust

CENTRIFUGAL COMPRESSOR:



A Two stage centrifugal compressor with inlet guide vane

Centrifugal compressors are fluid flow dynamic machines for the compression of gases according to the principals of dynamics. The bladed impeller with its continual internal flow serves as an element of energy transfer to the gas.
Pressure temperature and velocity of the gas leaving the impeller are higher than at the inlet. Diaphragms or diffusers arranged after the impeller helps in diverting the gas velocity, thus further increase in pressure and temperature is achived by the conversion of the kinetic energy into pressure energy.
During energy transfer in the impeller, the gas flows from the inside in an outward direction. It is therefore subjected to the change of the centrifugal field, through which the attainable pressure ratios are substantially higher than those of axial compressors.
The radial direction of flow in the impellers again requires radially arranged diffusers, which increases the outer diameters of the casing to about double the impeller diameters.
Generally centrifugal compressor are used for high capacity and low pressure and though initial cost is high but, lower maintenance and running cost places these compressors to compare with high efficiency reciprocating compressor. Manufacturing, testing and accessories for Centrifugal compressor for petro chemical plants are manufactured according to API 617 code.

General construction of centrifugal compressor:

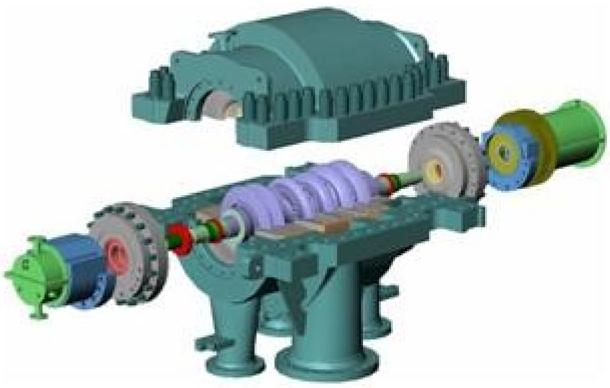
Centrifugal compressor are generally manufactured in two configurations:

1) Horizontally split Construction
2) Barrel type construction

Compressors of both the configurations are in use and the discharge pressure of the gas directs the designs. Centrifugal compressors are also manufactured in integral gear type construction, which are goverened, by API 672.
Centrifugal compressors are composed of outer casing which contains stator part called a diaphragm bundle, a rotor formed by a shaft with one or more impellers, a balance drum, thrust collar etc. The rotor is driven by means of coupling hub and is held in position radially by journal bearings and axially by a thrust bearing.
Sealing system in centrifugal compressors is very interesting and a bit complicated. Sealing of by-passing gas from one stage to another is required between all the inter stages and also at both the shaft ends as well. In general rotor and stators are fitted with labyrinth seal rings sealing. In toxic gas services 100% leak-proof seals are required which may be of different type depending on the service of the compressors. Oil deflectors and oil labyrinths are used as oil seal on both ends of the rotor.

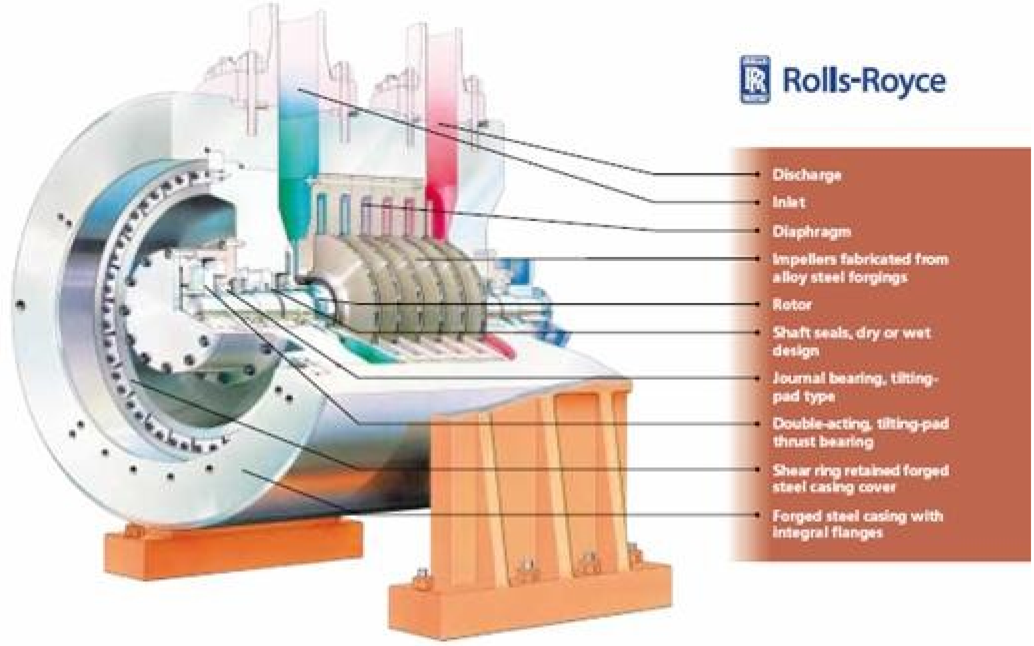
Gas is drawn into the compressor through suction nozzle of the compressor and enters in to an annular chamber called inlet volute, flowing towards the center of the impeller from all directions in a uniform radial pattern. The rotating impeller imparts energy through its wanes and pushes the fluid outwards raising its velocity (kinetic energy) and pressure as it passes through the impeller shroud. The outlet fluid leaves the impeller tangentially and then enters into another circular chamber called diffuser, where its velocity (kinetic energy) is converted into pressure energy. After this increase in pressure of the fluid in one stage again it enters into second stage impeller eye and cycle goes on till the final discharge of the pressurized gas from the compressor discharge nozzle.

Horizontally split Construction



Horizontally split casing consists of two halves joined along the center line by casing bolts. These type of compressors are generally made up to 40 kg/cmm2 discharge pressure. All the suction nozzle, discharge nozzle, lubricating lines and other connections are generally located in the lower half casing so that during maintenance / inspection, only the upper half casing can be removed easily and gain access to all internal components. The material of construction of casing depends upon the operating parameters like pressure, temperature, gas handled etc. API 617 governs this selection. Generally used material is meehanite grade cast iron, ASTM A216 WCA and ASTM A 351 Gr.CA15 steel in case of corrosive fluid.

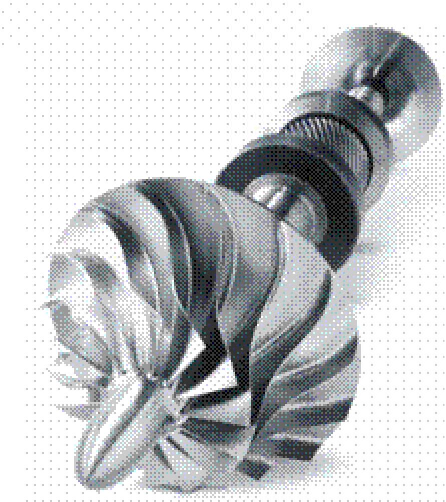
Barrel type construction



Axially split design is more suitable for high pressure services. Above 40 kg/cm2 discharge pressure this type of design is more common. The Outer casing is generally in form of a barrel which is closed by end cover. Internal casing may again be made in two designs called Axial split or horizontally split. Both type of designes are approved in API 617. Barrel casing may be designed with one side end cover of with both side end covers. Generally one side end cover design is more common and acceptable.
Material of construction of barrel and end covers is generally Forged carbon steel as per ASTM A105, however, forged alloy steel are also used depending on design requirements. Material of construction of internal casing depends on the design parameters however in general martensitic Stainless Steel is used.

Rotor:
The rotor of a centrifugal compressor consists of shaft, impellers, balancing device, thrust bearing collar, coupling hub, end seals, sleeves and spacer rings.
The material of construction of shaft may vary from forged plane are carbon steel to forged low alloy steel to forged martensitic stainless steel. Material shall be suitable for hardening and tempering so that toughness can be enhanced. Generally used grades are EN series grades 8, 9, 19, 24 and 56. Forged allow steels like SAE 4140, 4340 etc.

Material of construction of Impellers may vary from cast iron to stainless steel depending upon the design parameters and the economics of the machine. When Impellers are used in corrosive services, steel with higher chromium content is used. For higher strength and corrosive service Maraging (Manufacturer’s standard) steel is used. In old machines generally cast impellers were used. Then riveted impellers were very famous as the degree of freedom in manufacturing had enhanced the capabilities for making narrow and efficient impellers. As the demand for high pressure and small machines increased, welded impellers replaced the riveted impellers. Now a days, electron brazed impellers and electron beam welded impellers are famous for their quality and efficiencies. With improving technology, impeller design has improved from simple impeller to 3D impeller which has enhanced the compressor efficiencies many folds.



A Three Dimensional Impeller for Air Compressor

There are different levels of sealing systems required for centrifugal compressors. The end shaft seals, interstage seals, balancing device seal, oil seals etc. Depending on the service, end shaft seals are selected based on the toxicity of the gas to be handled. As per API 617, all toxic gas shall be sealed positively without polluting surrounding atmosphere. Positive seals are Mechanical seals which have been designed specifically for the high speed high pressure compressors. These may be pure mechanical seal with suitable mating seal combination, or gas seal or Entrapped oil dynamic seal which have been designed by different manufacturers.

For Non Toxic services Labyrinth seal are most common seal used in centrifugal compressors. Labyrinth seal design is again specific with different manufacturers however the principal of working is same. In side high pressure is allowed to let down successively in series of labyrinth rings so that very minimum gas pressure is generated in the outermost ring which discharges the gas to atmosphere. The sealing system is very simple, cost effective, but it is not positive sealing system.

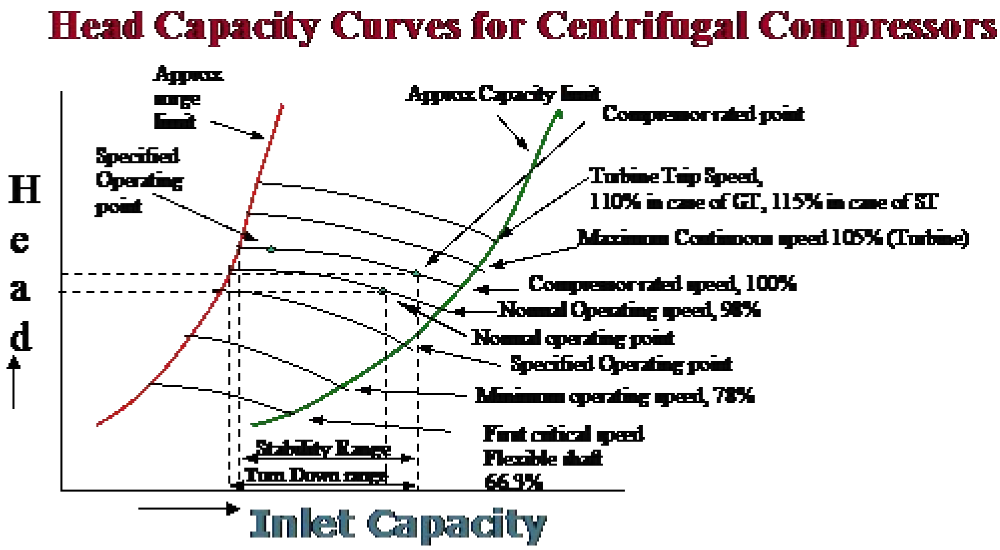
Similarly due to diffence in pressure rise across successive compression stages, to avoid the by passing seals are provided at the impeller suction eye, impellers' rear end and the corresponding surfaces of diffusers. The condition of these seals directly affects the compressor performance.

The material of construction of labyrinth seals should be resistant to corrosion and erosion. Generally used material for labyrinth seals are annealed aluminium alloy, stainless steel, bronze, babbit and martensitic stainless steels.

<u>Diffusers</u>:

In centrifugal compressors, the discharged gas from impellers enters into diffusers. The diffuser not only guides the gas to next stage but also helps in regaining pressure from the velocity of the gas. Since velocities are very high through the diffuser, surface finish and friction factor is very crucial for overall efficiency of the compressor. Centrifugal compressor losses their efficiencies in diffusers only. In many process services dirt or other inclusions gets stick to impeller and diffuser and fowling will not only narrow down the openings but also produce rough surface area in the fluid passages which can reduce the overall efficiency of the compressor drastically.

Material of construction of diffusers is generally cast iron, cast steels or cast stainless steels.



Operating curves of the centrifugal compressors are very critical for the operation of the compressors. The sketch shown above is not a real curve but to understand various important points of operation which are being defined as below.

Surge –

The term surge indicates a phenomenon of instabilit which takes place at low flow values and normally occurs at about 50% of design inlet capacity at design speed. It is a point on the operation curve of the compressor, at which the compressor cannot add enough energy to overcome the system resistance. This causes a rapid flow reversal (i.e. surge). As a result, high vibration, temperature increases, and rapid changes in axial thrust can occur. These cyclic occurrence of this phenomenon can damage the compressor rotor, seals, rotor bearings. Most turbomachines are designed to easily withstand occasional surging and surge control devices have become mendatory for these machines. However, if the turbomachine is forced to surge repeatedly for a long period of time or if the turbomachine is poorly designed, repeated surges can result in a catstrophic failure.

MCS

It is the maximum continuous speed at which any machine can run continuously.

TTS

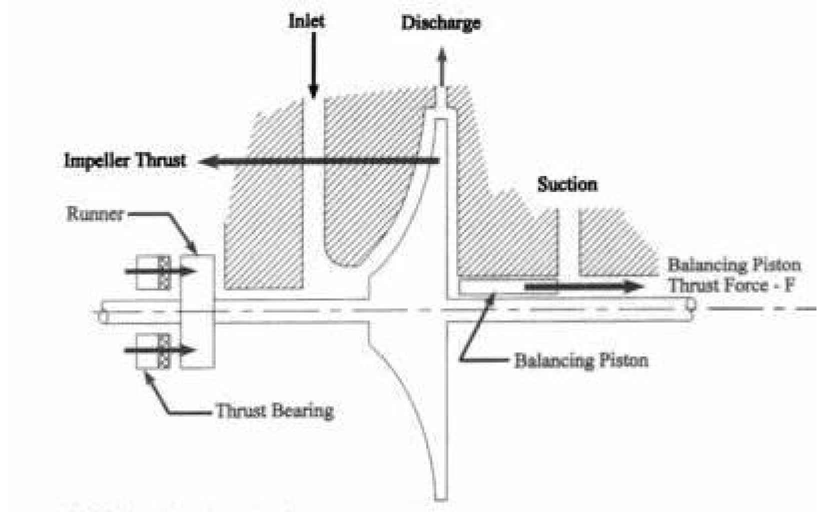
It is the safety feature of the system and at this speed, the drive turbine trips by the over speed protection device of the turbine.

Compressor rated Speed

It is the speed of the compressor at which it develops the required pressure at required flow. It is considered as 100% rated speed.

First Critical speed

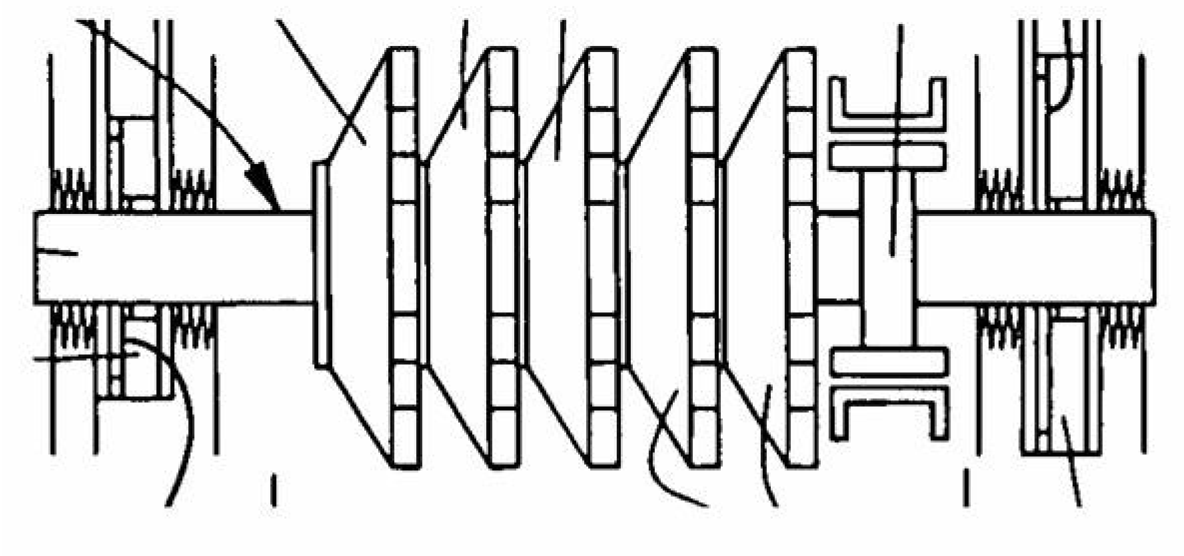
It is the rpm at which the natural frequency of the flexible rotor matches the system speed. This RPM should be crossed rapidly while increasing the speed of the system otherwise abnormal vibrations may set in the machine in this range of speed.



AXIAL THRUST

Discharge pressure

Suction Pressure



Within the compressor, the suction side of the impeller is partly exposed to suction pressure and partly to discharge pressure however the opposite side of the impeller, the whole area of cross section is exposed to discharge pressure only. This difference of pressure working on impellers creates axial thrust force in the rotor, which normally acts towards suction side. The total thrust produced by all the impellers is counter balanced by introducing a balance drum or balance disc or combined disc-drum, which is mounted on the shaft after the last stage impeller. Rotors shall be designed in such a way that minimum resultant thrust keeps the rotor stable on the active side of the thrust bearings. If the axial thrust of the rotor gets neutralized, rotor will remain floating during the operation.

For more details of balancing drum, balancing disc and combined disc-drum, please ref. to Centrifugal pumps.

Centrifugal versus Reciprocating Compressors

Integrally geared centrifugal compressors can operate at many times higher speeds than reciprocating compressors. The higher speeds ultimately result in smaller package sizes as compared to a reciprocating compressor. The operating speed of a reciprocating compressor is very slow due to mechanical and dynamic limitations.
Higher reliability is fully attainable with centrifugal compressors. The rotating aerodynamic components (impellers) have no physical contact with the stationary parts (inlet shroud). On the contrary, in the reciprocating compressor moving components such as the piston, cross head, connecting rod, piston rod etc. are having relative movement with each other. Similarly inlet and outlet valves are having parts having relative movement and physically in contact with each other. The physical contact and relative movement causes wear and tear of both moving and stationary components, which requires frequent and regular maintenance. However, a centrifugal compressor operates for many years with continuous service without overhaul maintenance, resulting in less plant down time. This eliminates loss of product, provides more profit, lowers risk, and results in lower maintenance cost.
Considering there is no physical contact between the centrifugal compressor rotodynamic components, except for the bearing lubrication, the need for lubrication within the compressor components is not required; thus it will not add oil or other contaminants to the process gas. However, a reciprocating compressor requires oil lubricant for the piston rings and other moving parts. This oil eventually ends up in the process gas or it has to be separated to get the oil free gas.

The dynamic loads placed on a centrifugal compressor foundation would typically be in the order of 10-lbf (44.5 N) as compared to 400-lbf. (1780 N) for a reciprocating compressor, in a similar services. The dynamic load is proportional to the unbalance weight and square of the speed; therefore, despite low operating speed the dynamic load is still very high on the foundation of the reciprocating machines.
Furthermore, the lower speed of reciprocating compressor lends itself to larger compressor size, heavier weight, and larger plot plan size. Whereas the centrifugal compressor with higher operating speeds results in smaller overall compressor package sizes such as smaller gearing, bearings, seals, lubrication system, and foundation. Smaller packages ultimately lend themselves to saving in lower overall installations as well as lower capital and spare parts costs.