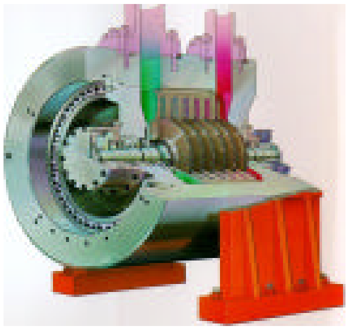
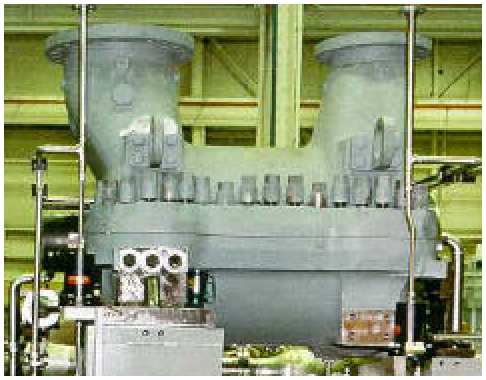
Rolls-Royce

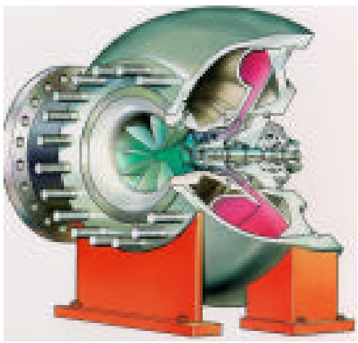
CENTRIFUGAL COMPRESSOR NOTES



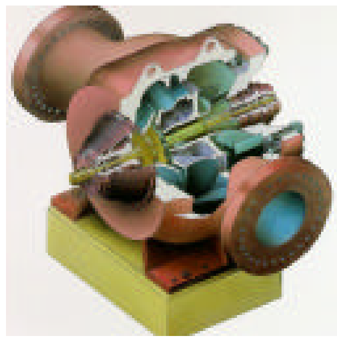
Barrel



Horizontal Split



Pipeliner – Axial Inlet



Pipeliner – Horizontally Opposed

Prepared by the

Rolls-Royce Customer Training Center

Mt. Vernon, Ohio
November, 2003
Revision 3

NOTICE

THIS MANUAL IS INTENDED FOR TRAINING PURPOSES ONLY.

THE EQUIPMENT MANFACTURERS TECHNICAL MANUAL SHOULD BE USED FOR ALL OPERATION AND MAINTENANCE PROCEDURES.

REFER TO THE OEM TECHNICAL FOR ILLUSTRATION OF SPECIFIC MAINTENANCE PROCEDURES.

NO REVISIONS TO INFORMATION CONTAINED IN THIS MANUAL WILL BE ISSUED.

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ROLLS-ROYCE CENTRIFUGAL COMPRESSSOR NOTES

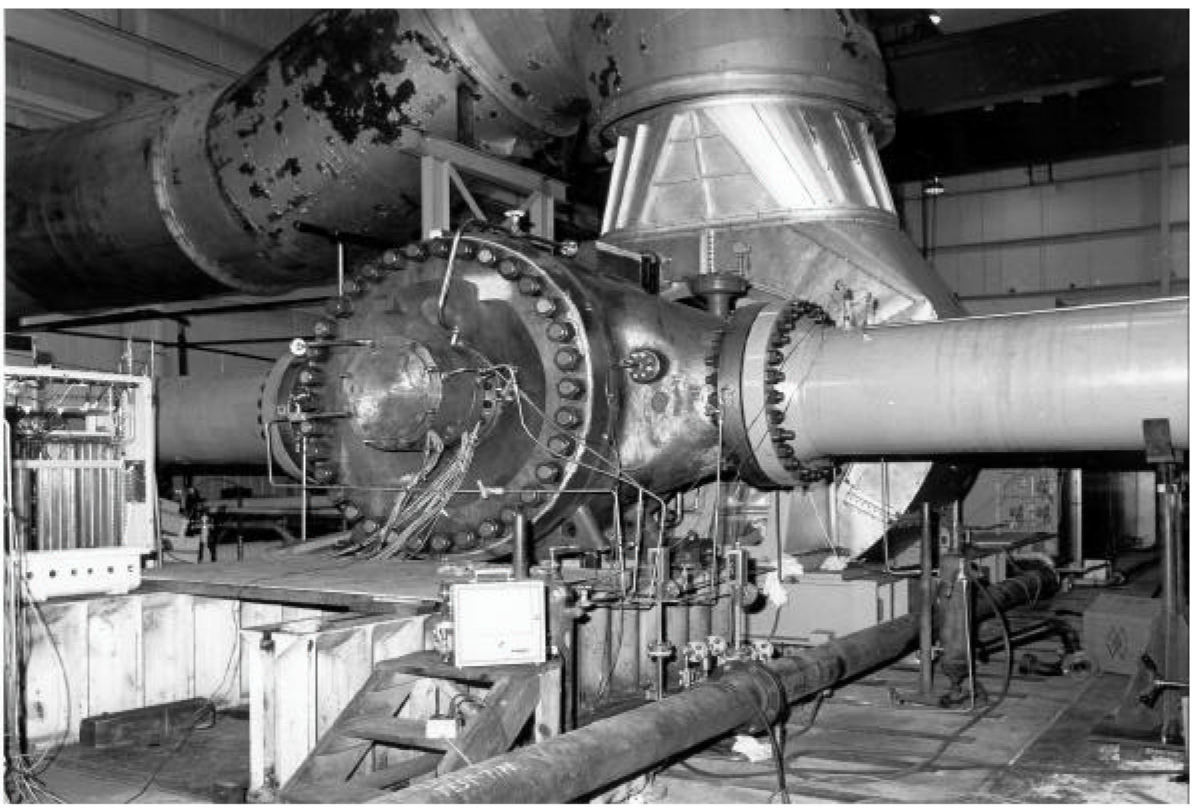
INTRODUCTION

Definition – *centrifugus*, from *centr* + *fugere* to flee

The use of Centrifugal Compressors dates back to the year 1900, when single and multi-stage units were applied for air compression service in mines and steel works in Europe.

During the 1930's centrifugal compressors found increasing application in refinery and petrochemical service, and in 1947 the first unit was applied to gas pipeline service.

The first gas turbine driven centrifugal booster was installed by the Mississippi River Fuel Corporation in Arkansas in 1949; in this installation, a single-stage compressor was directly coupled to a 1850 HP, 8750 R.P.M. Turbine.



TYPICAL EARLY CENTRIFUGAL COMPRESSOR GAS TRANSMISSION INSTALLATION

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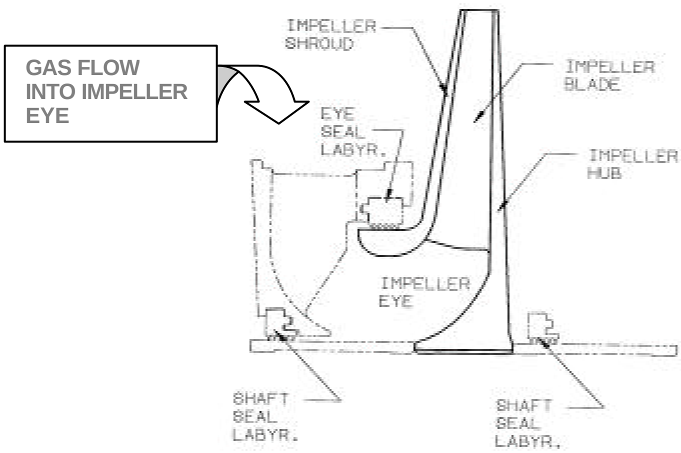
Basic Principles:

Compressor : A device that increases pressure by mechanically reducing volume.

The centrifugal turbocompressor, like the axial-flow compressor, is a dynamic machine, in contrast to positive displacement-type compressors. For example, in a reciprocating compressor a quantity of gas is drawn into the cylinder and trapped by the action of the valves and motion of a piston. As the piston moves in the cylinder, compression is achieved by direct volume reduction. Backflow or recirculation of the gas is prevented by check type valves, both during operation of the machine and at standstill.

By comparison, centrifugal and axial compressors achieve compression by applying inertial forces to the gas (acceleration, deceleration, turning) by means of rotating blades or bladed impellers, that continuously impact and perform work on the gas during operation.



**TYPICAL IMPELLER
AND ASSOCIATED COMPONENTS**

<u>Function</u>: Conversion of rotative shaft energy into gas velocity energy & pressure

<u>Method</u>:

- Efficient acceleration of gas flow
- 90 degree turning of gas flow from axial into radial direction
- Efficient deceleration of gas to convert velocity into pressure

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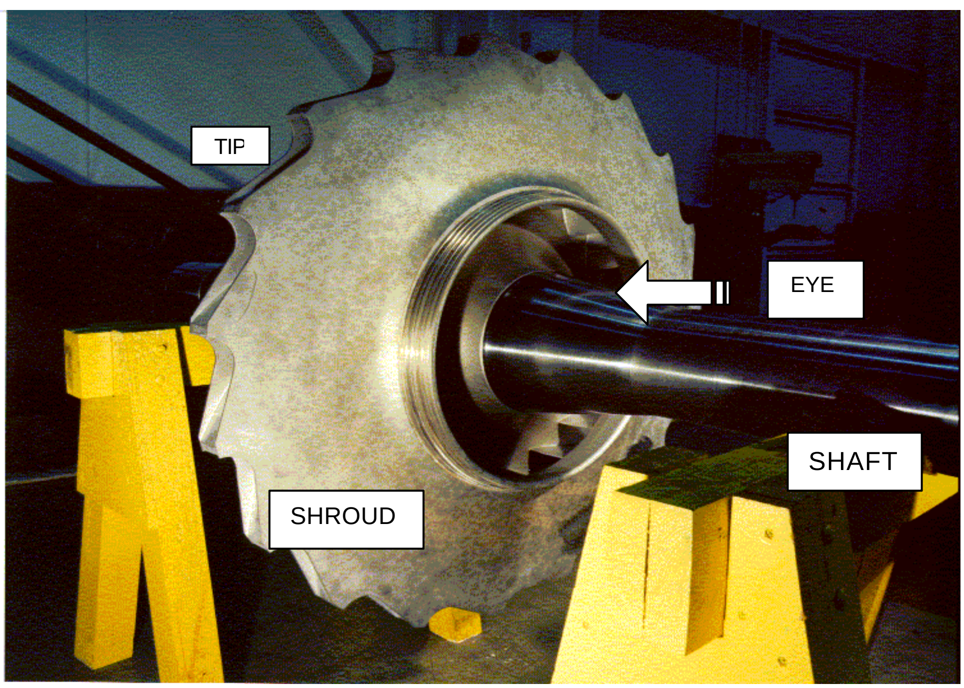
In the case of centrifugal compressors, the gas enters the impellers at the eye, moving in an axial direction. The flow is then changed to a radial direction and accelerated in a peripheral direction as it moves through the impeller from the eye to the tip. Gas then exits into some type of diffuser (flow decelerator). In a single-stage compressor the gas is then discharged to the process. In a multi-stage compressor the flow must be returned to the eye of the next impeller. About two-thirds of the pressure rise occurs in the impeller with the remaining increase taking place in the diffusion (velocity reduction) process. Flow passages are open throughout the compressor. There is no mechanical means of preventing backflow in the design of the unit, and backflow can occur at standstill or reduced compressor speeds unless some check valve is used externally.

Note that the impeller is the only means of adding energy to the gas and all the work on the gas is done in this element.

The stationary components such as diffusers, guide vanes, return channels designed to control the overall aerodynamic flow of gas, convert velocity energy into pressure energy and incur losses. These components common to centrifugal compressor design do not perform work on the fluid medium flowing through a centrifugal compressor.



TYPICAL CENTRIFUGAL COMPRESSOR IMPELLER

CENTRIFUGAL COMPRESSOR TERMINOLOGY

Numerous references have been utilized in preparing this chapter. Information is organized to develop relatively simple, straightforward and consistent understanding of terms and procedures.

A clear understanding of the following basic terms will greatly assist an individual in developing a working knowledge of centrifugal compressors, how they function, and how they differ from other types of compressors.

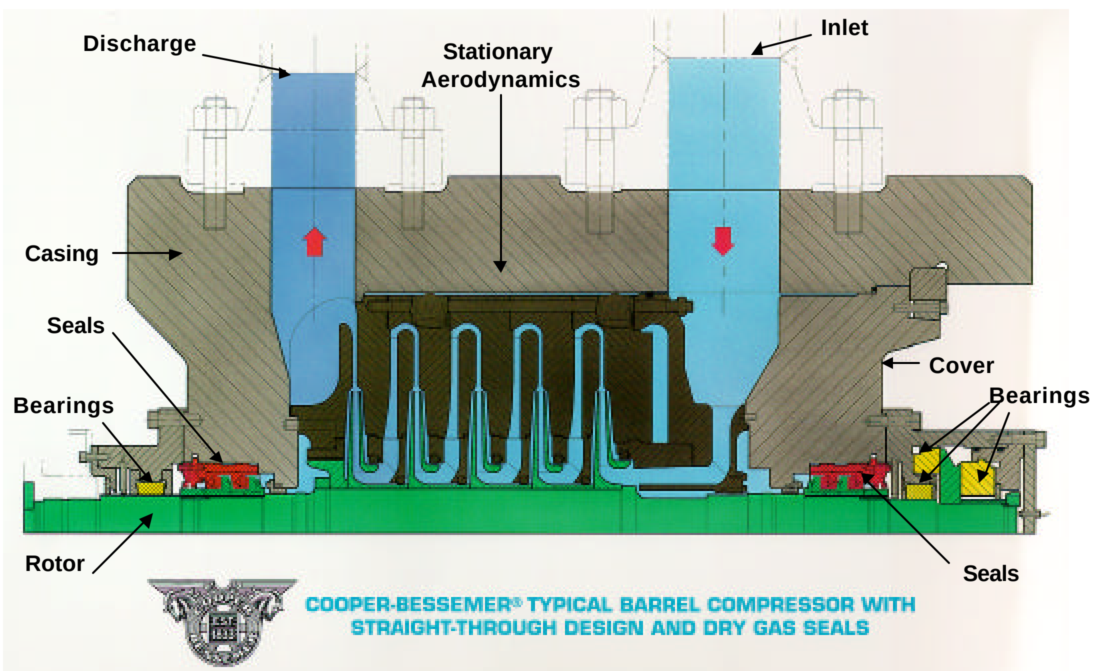
The Compressor Consists of Three Major Components:

1. Casing – Case and Mounting Feet
   Pipe Flanges

2. Aerodynamic – End cover and inlet scoop
   Vane support – Adjustable vanes
   Diaphragm
   Inlet guide
   Discharge scroll
   Labyrinth holder and Labyrinth seals
   Impellers

3. Quill Assembly – Impeller shaft
   Thrust bearings
   Thrust collar
   Seals [Dry Gas or Oil Seals]
   Journal bearings

Anatomy of a Centrifugal Compressor



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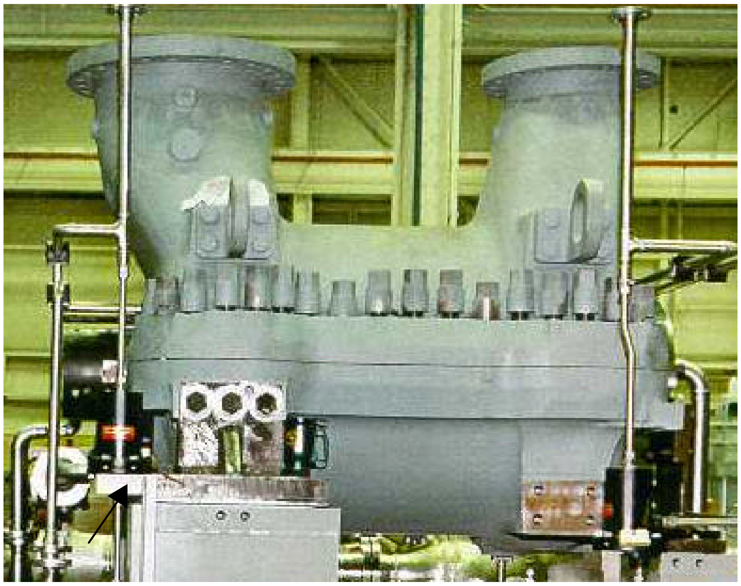
CASING

The casing is the pressure-containing stationary element which encloses the rotor and associated internal components and includes integral inlet and discharge connections. Casings are manufactured in three broad categories – Horizontally split, Vertically split barrel and Pipeliner-type compressors.

Compressor casings are steel casting, steel forgings, or fabrications from steel plate, which are designed to withstand high velocities and pressure. Removable covers allow access for servicing and disassembling of the internal components, and leveling screws are provided in the base of the casing for alignment purposes. Drains in the bottom of the casing allow draining of any excess oil or liquids from the casing. Sensors are connected to the discharge side of the casing to protect the compressor against excessive discharge pressure and temperature.

HORIZONTALLY SPLIT CASING

The horizontally split casing is split along the horizontal centerline and is normally used for low to moderately high pressures.



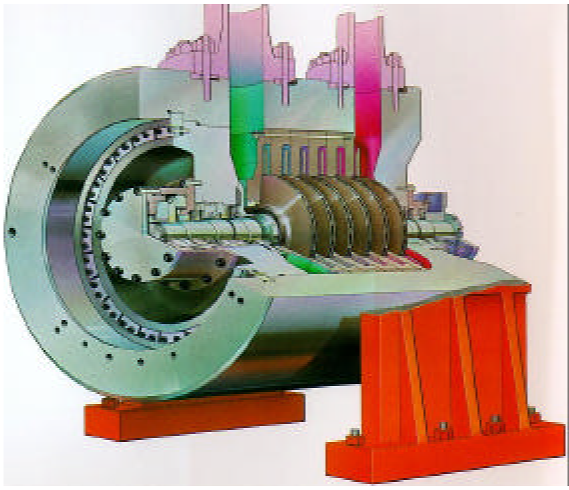
Horizontally split casings, with a bolting flange at the shaft level, permit removal of the upper casing and stator half for a convenient check on all the rotor and stator elements, their relative locations and clearances.

Sealing around the horizontal flange [split case compressors], which is often of irregular outline, must usually be achieved without gaskets by lapping the two halves together. This and the sealing of the stator inserts (diaphragms) present a problem at high pressures or with low molecular weight gases.

The first Cooper-Bessemer multi-stage centrifugal compressor was produced in 1956 – the RC5S air compressor. Through the years, Cooper-Bessemer produced several different size compressors for many different applications.

VERTICALLY-SPLIT (BARREL) CASING

A vertically split (Barrel) casing is split in the vertical plane and can be applied in services for high pressures or low molecular weight gas.



The barrel-type casing consists of circular vertical retaining flange at the casing perimeter or ends that can be sealed with ring-type gaskets of metal, metal asbestos, neoprene, or silicone o-rings.

Barrel casing construction is simple and accessible on smaller single and two-stage machines. On multi-stage compressors, accessibility is enhanced by a dual casing design where a horizontally split inner housing, or, “bundle” assembly is assembled separately with the rotor enclosed. This assembly is inserted into the barrel casing for operation.

The casing, either fabricated from thick, high quality steel plate or machined from a thick-walled steel forging, is cylindrically shaped with one or two end covers and is very conservatively designed to contain the gas up to its specified maximum working pressure. Toward the end of its manufacturing cycle, the casing must pass a hydrostatic test at 150% of its maximum working pressure with no indication of distortion or leakage, followed by a very careful inspection certifying complete dimensional conformation to all engineering requirements.

BARREL COMPRESSOR PRODUCT SYMBOLS

“R” will be the first character in the designation and stands for **“Rotating Compressor”.** Vertically Split Barrel and Horizontally Split Multi-Stage Compressors:

The second character will be a letter ***“B”*** through ***“E”*** and will denote the relative frame size.

The third (and possibly the fourth) character will be a numeral of **“2"** or higher and will denote the maximum number of stages that the compressor can accommodate. This numeral may extend to the fourth character in the event that the compressor can accommodate **“10"** or more stages.

The next character will be a dash **“-“**. This is to separate the former number denotation of maximum stages with the next numerical character.

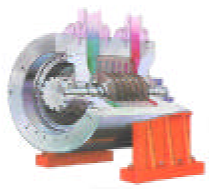
The next character will be a numeral of **“1"** or higher and will denote the actual number of stages that the compressor contains. This numeral may be two characters long in the event that the compressor has **“10"** or more stages.
The last character will denote the type of split utilized by in the compressor casing. The letters **“B”** or **“S”** will be used to denote a vertically split casing or a horizontally split casing respectively.

Barrel Compressor Examples

RF6-4S

- R — Rotating Compressor
- F — Frame size [approx. 42/12 “ impeller used]
- 6 — Max 6 Stage
- S — Horizontal Split

RB8B



- R — Rotating Compressor
- B — Frame size [approx 16 ½ Inches Impeller used]
- 8 — 8 Impellers
- B — Vertical Split

Frame sizes :			
	B	Approximate impeller size	16 ½”
	C	Approximate impeller size	20 ½”
	D	Approximate impeller size	26 ½”
	E	Approximate impeller size	34 “
	F	Approximate impeller size	42 ½”

Cooper-Bessemer Multistage Barrel Centrifugal Compressors

MODEL	RBB	RCB	RDB	REB
Maximum working pressure, psig (bar)	6500 (450)	3200 (221)	2000 (140)	1200 (85)
Number of stages	1 - 9	1 - 9	1 - 9	1 - 9
Design speed range, rpm	9,000 - 13,800	5,000 - 11,000	4,500 - 7,500	3,500 - 6,500
Maximum design flow, acfm (m^3/h)	6,000 (10,200)	13,500 (23,000)	22,000 (37,400)	35,000 (59,600)
Weight, lb (kg)	25,000 (11,300)	45,000 (20,400)	77,000 (34,900)	135,000 (61,200)

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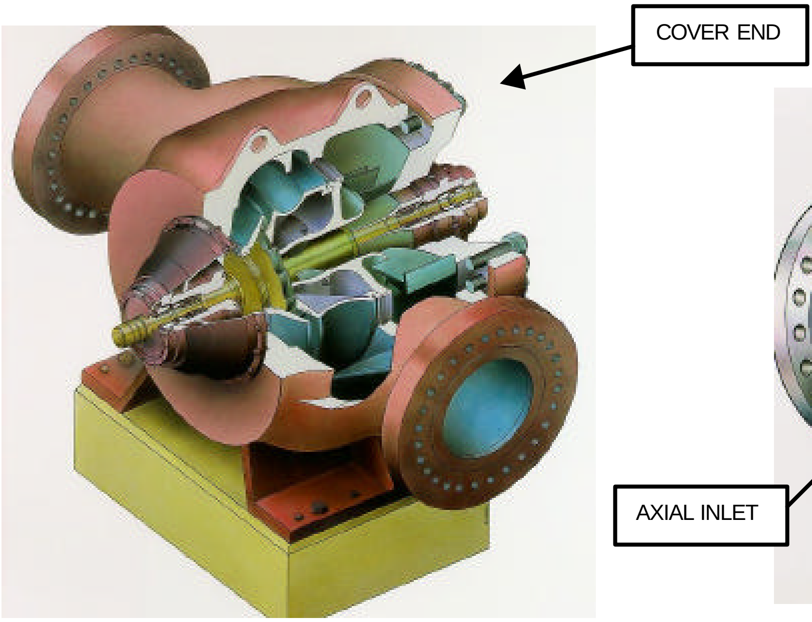
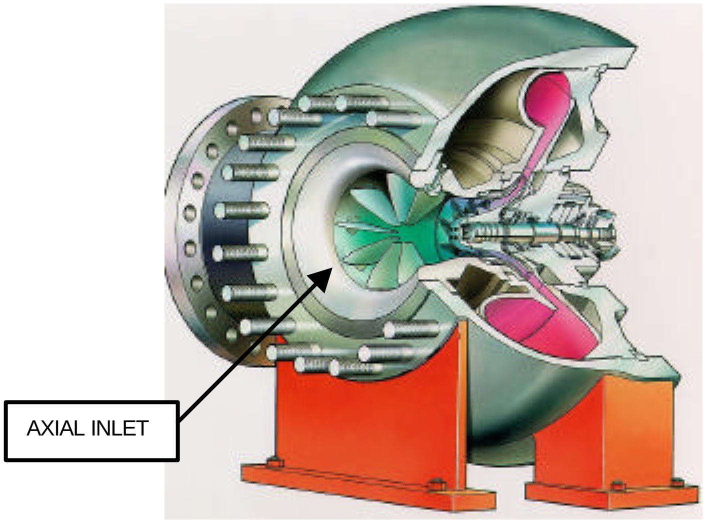
PIPELINERS

The “beamstyle” pipeliner-style casing illustrated is a vertically split casing with in-line flanges which provides the ability to replace all compressor internals without removing any station piping.



TYPICAL PIPLINE BOOSTER COMPRESSOR



TYPICAL SINGLE STAGE OVERHUNG RFA AXIAL INLET COMPRESSOR

All pipeline boosters are of the barrel-type casing design with a circular vertical bolting flange. The casings are steel castings up to eight feet in diameter with walls 2 ½ to 5 inches thick. Assembly and servicing of the internal components is done through a circular front opening using large special tool assembly fixtures, or, in combination with bearing and rotor removal from the discharge side of the compressor as in the case of overhung axial inlet compressors. Casing inserts, diffusers, diaphragms, and guide vanes are made of high-grade cast iron or steel.

Casing mounts must be strong enough to transmit nozzle or pipe loads to the foundation. Mounting must maintain an acceptable compressor shaft alignment regardless of casing loads which are due to heat, pressure, torque, or nozzle loads.

Pipeliners are made with the following rotor designs:

- Single Stage – overhung short shaft [shown to the right]
- Two or More Stages – Beam-style shaft/bearing arrangement.

PIPELINE COMPRESSOR PRODUCT SYMBOLS

Pipeline Compressor Designations are identified by the last two characters of the product label. These characters are the same as the nominal ANSI pipe sizes to which the compressor will flange. They include the following sizes: “12", “14", “20", “24", “30", “36" and “42".

The second character will be a letter **“C”** or **“F”**. The use of the character ***“C”*** only applies to the **12** to **14** inch pipeline frames. (This practice goes back to the days when frame sizes were denoted by their normal impeller diameter. This practice ceased when the RF20 was introduced.) Note that the “F” can be found in the second character position of a horizontally split compressor (e.g., RFS), therefore it alone will not designate that the compressor is a pipeliner.

The third character will usually denote the number of stages. Note the RFB24 and RFB30 compressors were offered during the 1960's and utilized an added “B” in the third position. This practice was dropped later to allow the “B” to denote the bearing support configuration as explained below.

NOTE: The only positive differentiation of an old RFB24 or RFB30 compressor compared to a RF24 or RF30 is to examine the detail drawings.

The next character(s) denotes the inlet flange orientation and type of bearing support. There are three options and they are:

1) The character “A” denotes that the compressor is an axial inlet flange orientation and an overhung support configuration.

2) A single “B” denotes that the compressor is a side inlet flange orientation and an overhung support configuration.

3) The characters “BB” denote that the compressor has a side inlet flange orientation and a beam bearing support configuration.

Maximum design flow rates [ACFM]:	**RC14**	6,640	**RF20**	12,725
	RFA24	18,000	**RF30**	30,800
	RFBB36	45,400	**RFBB42**	62,800
	RFA36	60,500		

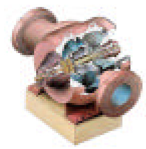
To date, Cooper-Bessemer has built in excess of 1,525 centrifugal compressors, of which about over 840 have been pipeline boosters, 687 have been barrels, and the remainder have been other applications. Note: This information is dated and subject to change and latest information available.

Pipeline Compressor Examples

RFA-36

- R — Rotating Compressor
- F — Up to 45,000 Bhp
- A — Axial Inlet, Overhung
- 36 — 36 Inches Nominal Flange Diameter

RF3/1BB-30



- R — Rotating Compressor
- F — Up to 45,000 Bhp
- 3/1 — Max 3 Stage using 1
- BB — Side Inlet Beam style
- 30 — 30 Inches Nominal Flange Diameter

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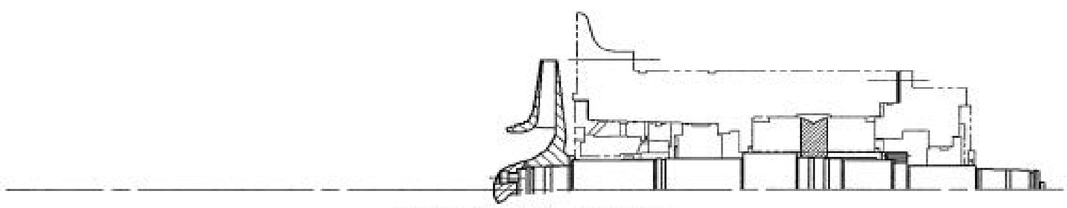
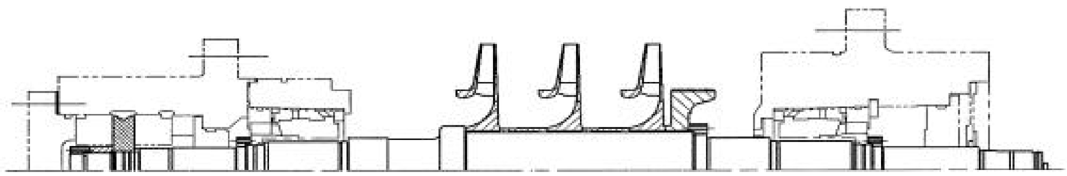
ROTOR ASSEMBLY

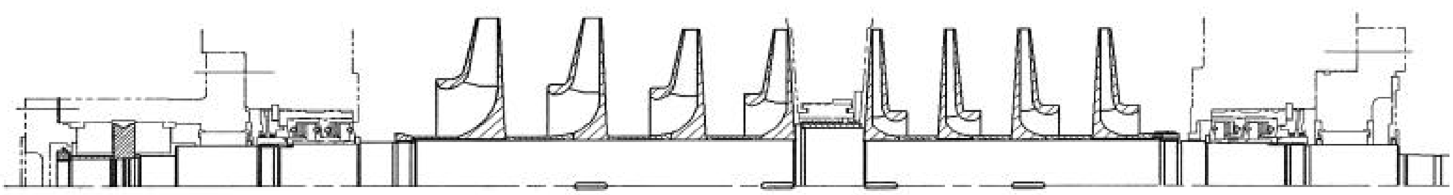
The rotor assembly is the rotating element which includes the impellers, shaft, balance piston, thrust disc, spacers, keys, etc. Generally the rotor assembly represents the highest investment in engineering, tooling and manufacturing efforts.



OVERHUNG ROTOR



BEAM STYLE ROTOR



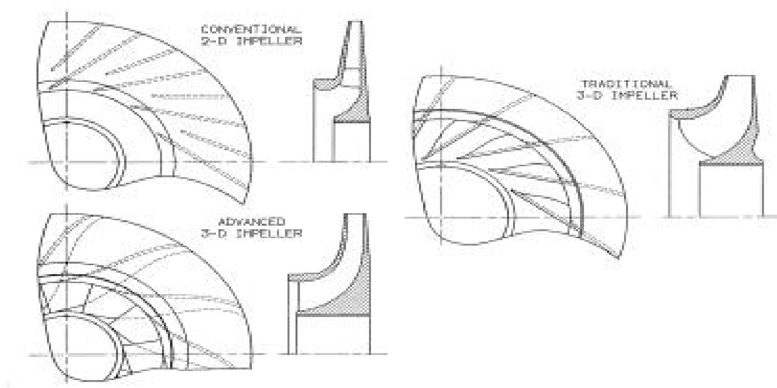
BACK TO BACK ROTOR

The rotor is supported in two journal bearings and a double-sided thrust bearing. The journal bearings are of the tilting-pad, self-aligning type, each containing five babbitt-lined bearing pads. The thrust bearing consists of two tilting-pad bearings (in which each bearing could have six, nine, or twelve pads), separated by a steel, precision-lapped thrust collar. The bearing pads are self-leveling and will equally divide the thrust load on all pads. Multi-stage compressor thrust during starting is absorbed by the inner thrust bearing and running thrust by the outer thrust bearing.

IMPELLERS

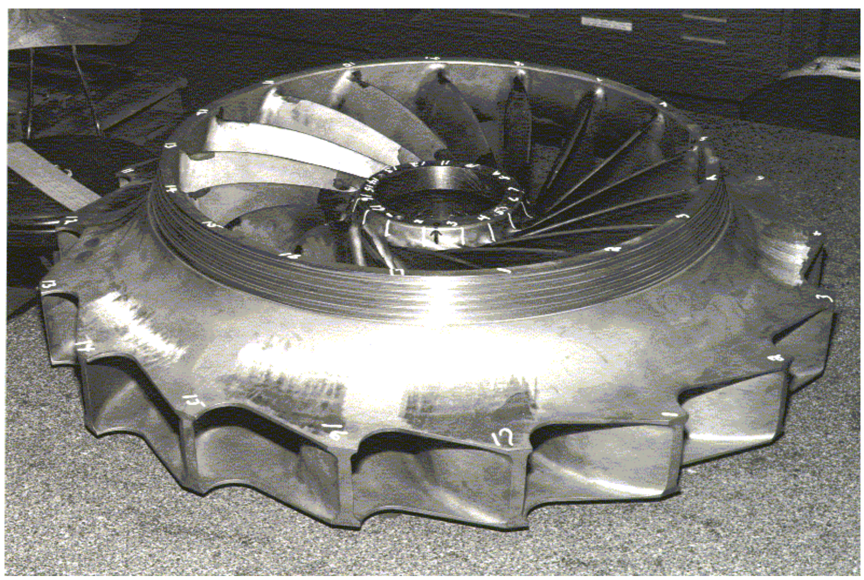
Impellers are the only components of the rotating element which impart energy to the gas by means of centrifugal force. Impellers normally utilized in natural gas and process applications are closed, “two-dimensional” or “three-dimensional” impellers with backward-leaning vanes.

2 - D vs. 3 - D Impellers



Impellers must be designed for a specific aerodynamic performance while incorporating adequate structural strength, and must be balanced statically and dynamically in two balancing planes, to extremely close tolerances.

The impellers and shaft are balanced separately with the impellers having been spin-tested at 115% of their maximum continuous operating speed before they are re-inspected, assembled, and balanced as an assembly with the shaft.



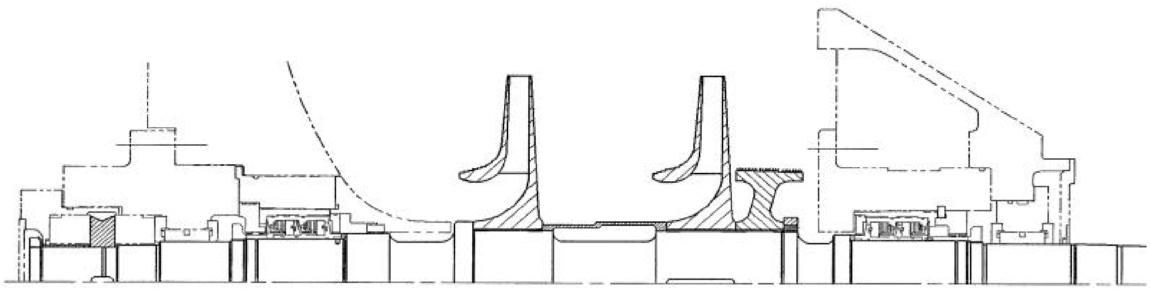
TRADITIONAL 3-D IMPELLER

SHAFT

The shaft is the part of the rotating element on which the rotating parts are mounted and by which energy is transmitted from the prime mover.

Shafts must be designed for maximum stability throughout the compressor's operating speed range. The "critical speeds," that is, the rotor speeds coinciding with the natural frequencies of lateral or bending vibration, must be known in advance and placed safely below or above possible operating speeds – usually with a 15 – 20% safety margin. Short rotors for single, two, or three-stage compressors can usually be operated below the first critical speed. Long rotors, with four or more stages, are normally designed to run between the first and second critical speed.

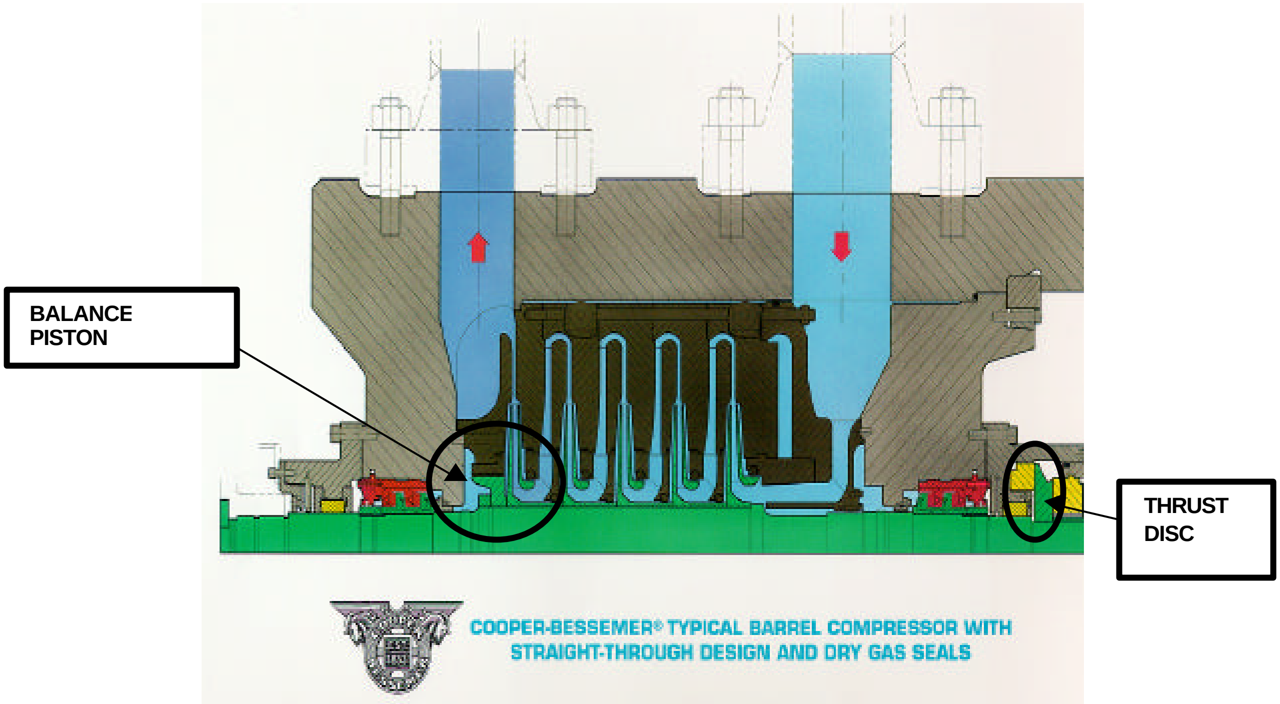
The first critical speed is the one at which the shaft deflects between two nodes located at the two journal bearings. At the second critical speed, the deflection line has a third node located between the bearings. All the critical speeds are affected not only by shaft or rotor flexibility, but also by the flexibility of bearing mountings, casing, etc.



TUNED/ UNDERCUT ROTOR

BALANCE PISTON

A balance piston is used on all machines except the pipeliner models which incorporate the overhung shaft. In the past, the balance piston seal, like other interstage and impeller "eye" seals in the aerodynamic assembly, normally consisted of a stationary aluminum labyrinth ring located adjacent to a rotating cylindrical steel surface separated by the minimum clearance sufficient to prevent contact under normal operating conditions. In contrast to impeller eye and aerodynamic shaft seals, the balance piston seal is exposed to a much higher differential pressure which results in a proportionally higher leakage rate despite the balance piston seal's greater length which accommodates a greater number of labyrinth teeth. Since these higher balance seal losses directly affect the compressor efficiency, the old arrangement has been replaced by a new design which incorporates rotating labyrinth teeth machined into the steel balance piston, which now run at closer clearances against a layer of stainless steel honeycomb cells. This redesign has effectively reduced balance seal losses by about 50% without any loss of operating reliability.



THRUST DISC

The thrust disc or thrust collar, like the impellers and balance piston, is interference fit and keyed to the shaft. The rotor thrust is transmitted to the thrust bearing through this rotating thrust disc or collar; the positioning of the thrust collar on the shaft also determines the axial location of the impellers within the stationary diaphragms. This positioning is achieved with a small spacer ring located inboard of the thrust collar, whose final length is established toward the end of the compressor assembly process vital to proper centering of the rotor in the bundle assembly.

INLET GUIDE VANES

The inlet guide vanes, located on the inlet side of the first stage impellers, reduce turbulence in the gas stream and direct the flow to the impeller. Labyrinth seals prevent gas compressed by the impeller from re-circulating back into the guide vane/impeller eye area.

On multi-stage compressors, the guide vanes are stationary elements which direct the gas flow into the eyes of the impellers. Guide vanes are set to direct the flow against the direction of rotation (counter-rotation), perpendicular to the direction of rotation (radial), or in the direction of rotation (pre-rotation).

[Reference Anatomy of a Centrifugal Compressor illustration - page 6]

DIAPHRAGM

The diaphragm is a stationary aerodynamic element which serves as the dividing wall between individual impellers of a multi-stage compressor.

DIFFUSER

The diffuser is a stationary passage surrounding an impeller in which velocity imparted to the gas by the impeller is converted into static pressure. The diffuser is formed by the backside of the diaphragm of the preceding stage and the front side of the diaphragm of the following stage.

INTERNAL SEALS

The internal seals are devices used between rotating and stationary parts to separate and minimize gas leakage between areas of unequal pressure. Almost without exception on natural gas and process applications, the internal seals are labyrinth type; typically, this term covers interstage shaft seals and impeller eye seals.

SHAFT SEALS

The shaft seals are devices used to isolate the gas stream from the atmosphere and from the journal bearings. Shaft seals come in many forms, but there are three general categories that high-speed shaft seals fall into:

1. Labyrinth seals
2. Bushing or oil film seals
3. Mechanical or face seals
4. Dry gas seals [most common application today]

None of these types are a “perfect” seal as they are all controlled leakage devices. That is, they must allow a certain amount of process gas or seal fluid to leak past the seal points or faces to avoid mechanical contact or to permit lubrication and cooling of the contact faces of mechanical or face seals.

LABYRINTH SEALS

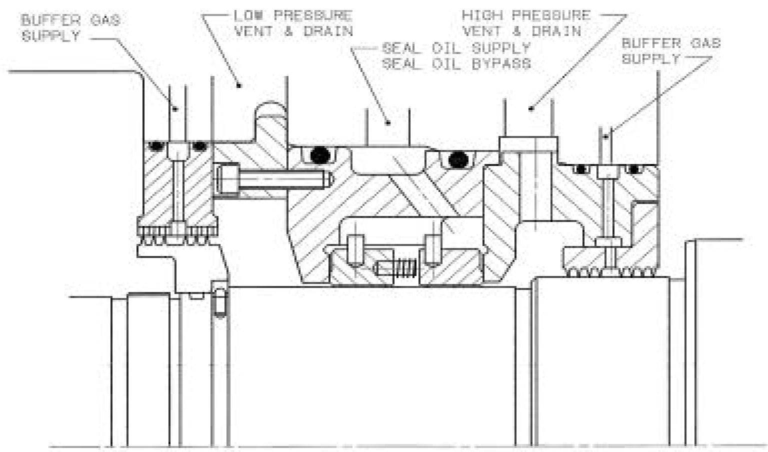
The labyrinth seals are among the oldest types of seals; they are simple to build and relatively trouble-free. Their leak-rate is relatively high (1 – 2% of compressor flow on small units), but easy to calculate. They are universally at interstage seal points, at the shaft ends of air blowers and compressors, and in other cases in which a small loss of process gas or of a seal gas (air, nitrogen, etc.) may be tolerated.

Many successful variations are in use. The multiple labyrinth lands or teeth may be part of a stationary ring (bronze or aluminum) dovetailed into casing grooves, or the teeth may be of hard material (steel) rotating with the shaft and running against a stationary sleeve of softer material. In either case, occasional light contact should increase the clearance without further harm.

OIL FILM SEALS

Oil film seals are used to positively eliminate leakage of the compressed gas to atmosphere. Since the leakage of any natural gas in even minute quantities is unacceptable, a positive seal is accomplished by a very thin, high-pressure oil film which is maintained under a free-floating cylindrical seal ring.

BUSHING/OIL FILM SEAL



TYPICAL BUSHING SEAL WITH HP & LP BUFFER

The typical Rolls-Royce oil film seal arrangement on any multi-stage barrel compressor consists of two radially free-floating, cylindrical rings that are held against rotation, but will otherwise easily follow any shaft motion. The seal rings are made of steel with a babbitt or aluminum lining and are machined to provide a very small, precisely controlled diametral clearance to the shaft which varies from 0.0030 inch to 0.0085 inch, depending upon operating conditions. Axially, the two rings are separated by springs and oil pressure and seal against ground and lapped seal housing faces which have a sufficiently smooth surface to permit the elimination of the seal face o-rings which had been used for many years.

The seal oil is supplied to the cavity between the two seal rings at pressures exceeding the compressor case pressure by only 5 – 7 PSI. This pressure differential is sufficient to generate an adequate oil flow between the inner seal ring and the shaft (usually not exceeding 20 GPD) which provides a reliable seal. The much greater flow (2 – 4 GPM) across the outer seal ring is subjected to a substantial pressure drop from case pressure down to atmospheric pressure levels and provides the cooling needed by both rings. To satisfy this seal oil requirement in emergency situations, a fairly sizeable overhead seal oil retention tank is needed to permit an orderly shutdown without loss of gas.

To accommodate gas compositions containing significant quantities of H_2S or other oil contaminating components, buffer gas provisions inboard and outboard of the seal assembly are available.

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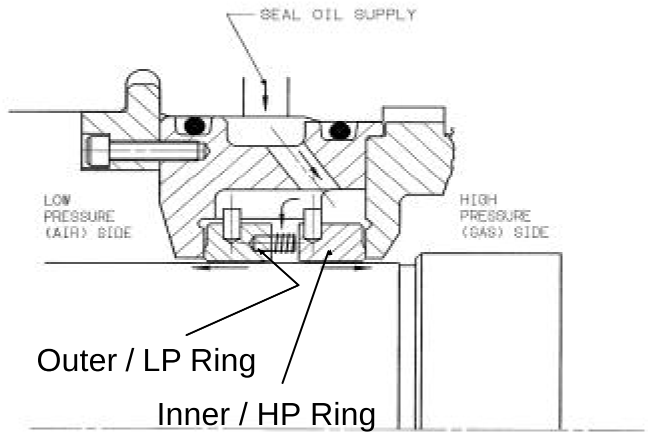
BUSHING/OIL FILM SEAL



BUSHING SEAL OPERATING PRINCIPLE

An inboard buffer is used to separate the gas from the oil system entirely; depending on operating conditions, this can be a mandatory feature. It requires, however, a constant supply of high-pressure buffer gas (usually Nitrogen). A low-pressure outboard buffer is used to separate the lube oil from the seal oil system. This arrangement is often used to prolong bearing life.

FACE SEALS

Face seals consist of a stationary seal face placed perpendicular to the shaft axis, a facing rotating member of a compatible material, a spring, bellows, or magnet to hold one member against the other, and a secondary axial seal between the loose seal member and the case (or shaft). Seal face materials may be carbon or ceramics running against steel, stainless steel, stellite, or hard ceramics. Even though these are referred to as face seals, microscopic film lubrication must be established at running speed to avoid rapid wear. Consequently, even the face seals show some leakage, but at a far lower rate than bushing or labyrinth seals. Sealing at standstill with a face seal is practically perfect; this is one of the greatest assets of face seals.

A combination of labyrinth and oil film seals provides a very reliable seal system. Since the entire centrifugal compressor is designed for long-life, maintenance-free operation, contact or face seals with components constantly rubbing against shaft surfaces would require frequent maintenance and servicing.

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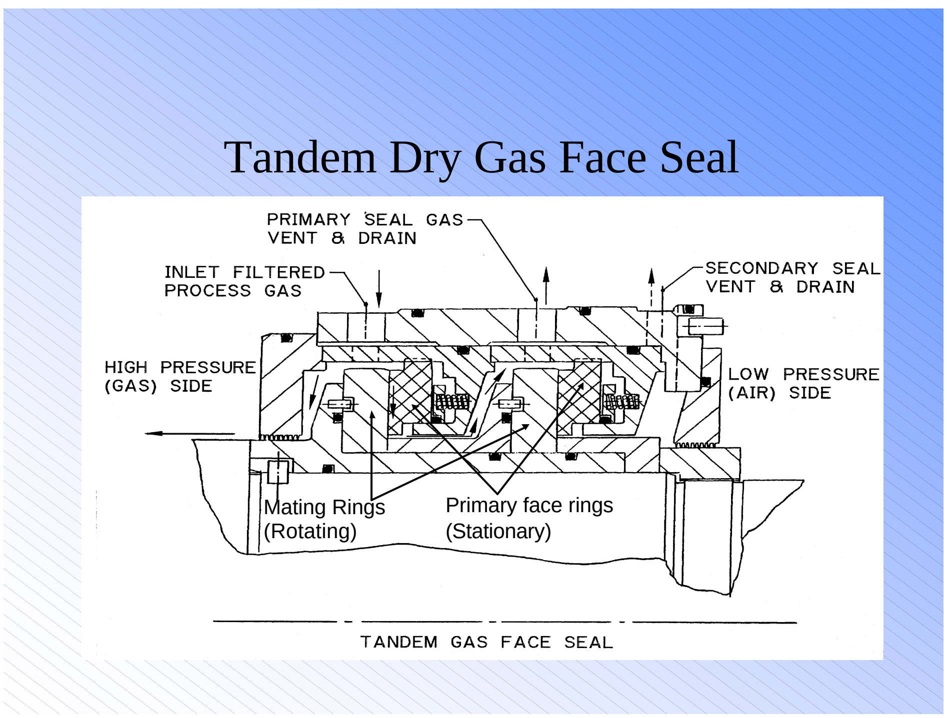
DRY GAS SEALS

Today's business requires cost effective solutions be considered for sealing against leakage of gas in centrifugal compressors. Thus, the use of dry gas seals is popular, and, most common. The dry gas seal eliminates the use of liquid sealing medium [seal oil] and its control. Dry seals are contained in an capsual inserted into one or both ends of the compressor depending on the type and model of compressor. The dry gas seal utilizes discharge gas from the compressor discharge for a continuous supply of sealing medium during operation. During operation, the gas is filtered and controlled at low pressure where it is introduced to the seal between a rotating **[mating]** seal ring and a stationary **[primary]** ring, both of specialized carbon material. The mating ring rotates with the compressor shaft assembly to induce a pressurize dam of gas that flows between the mating and primary carbon ring overcoming the force of spring tension in springs located behind the primary seal ring. This cools the sealing surfaces and provides a flow of approximately 15% of the gas to vent. Approximately, 85% of the sealing gas flows back into the compressor process. Care of the dry gas seal system revolves around the maintenance of the dry seal gas system and compressor control maintaining proper flow rates and eliminating sudden pressure differences across the compressor case much as possible during operation. Dry gas seals also prevent the ingress of oil [as results from oil seals] into the compressor discharge and into the gas transmission pipe line; reducing maintenance and extending the effective life of the pipe line operation and maintenance. This, plus the utility and operational savings make dry gas seals a favorable choice for sealing of gas in centrifugal compressors with discharge pressures of 2000 psig or lower.



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JOURNAL BEARINGS

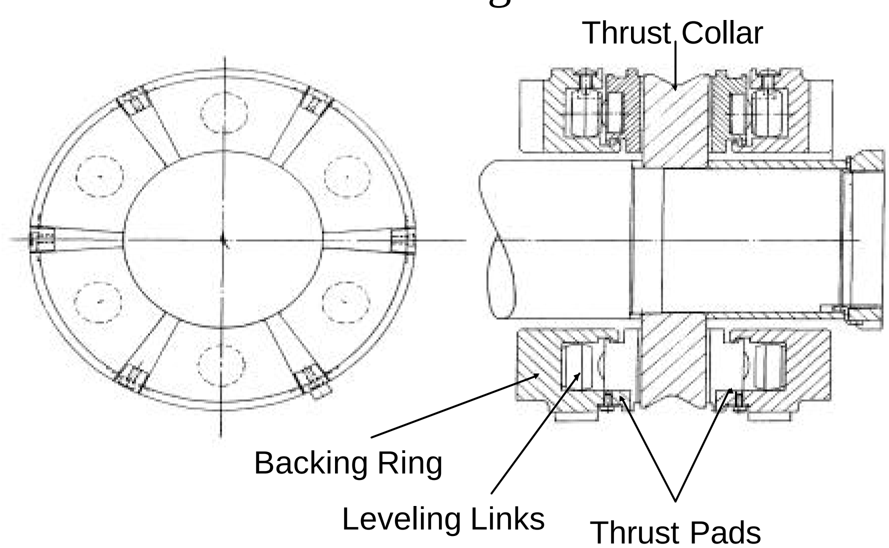
The journal bearings are utilized to support the rotor load (steady state and dynamic), provide stiffness and damping, and to control rotor position. For years, journal bearings were of the pressure pad or "anti-oil whip" type; however, with the advent of higher speeds and longer shafts, almost all current centrifugal process or natural gas compressors are designed using tilting pad "pivoted shoe" bearings.

Journal bearings in a centrifugal compressor have lower levels of loading than those of most reciprocating machinery, gears, motors, etc., because shaft diameter is usually made large for adequate rotor stiffness. Oil film lubrication is readily obtained throughout the operating speed range; the designer's problem therefore, is one not of lubrication, but of cooling because of the high velocities involved. To provide adequate oil flow for cooling, a pressurized lube oil supply system is used. Wear rate with such a system is or should be almost nil during start-up and operation.

THRUST BEARING

The thrust bearing is utilized to prevent axial motion of the rotating shaft, thereby maintaining the axial position of the rotor assembly within the compressor. Most thrust bearings utilized in centrifugal compressors are double-acting, tilting-pad (shoe) type.

Typical Double Acting Fluid Film Thrust Bearing



Under normal operating conditions, the thrust bearing in Cooper-Bessemer centrifugals is lightly loaded (below 100 PSI) compared to its maximum capacity (450 – 550 PSI, depending on bearing size and operating conditions). This is accomplished despite potentially high impeller thrust loads through the proper design of the balance piston and seal. In most high-pressure applications, the balance piston and seal are sized to overcompensate for the impeller thrust, which provides additional operating safety. This overcompensation transfers the normal thrust

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load from the outboard to the inboard side of the thrust bearing which provides compensation for labyrinth wear over long duration operation.

NOTE: What causes thrust? – All single flow closed impellers inherently develop an axial thrust in the direction of the eye of the impeller. The total amount of rotor thrust is a function of the sum of the pressure differentials across each of the impellers on the rotor.

CENTRIFUGAL COMPRESSOR OPERATION

To understand the operation and control of a centrifugal compressor, we must understand some of the basic principles of how it works. The following is provided for a better understanding:

It is helpful to once again provide the simplest definition of a centrifugal compressor:

Centrifugal Compressor : ***A device that increases pressure by mechanically reducing volume.***

We know that ***acceleration*** of gas is involved in the gas compression process. Acceleration is accomplished by work performed on the gas by the impeller(s) as they rotate at high speed with the compressor rotor and impact on the gas inside the compressor. There is, then, a relationship between the gas, it potential energy as it enters the impeller, and acceleration of the gas, within certain limits, in order for compression of the gas to occur.

We can understand ***acceleration*** as <u>an object's change in velocity (v) in unit time</u>. The acceleration that takes place during a time interval (t) is:

$$a = \frac{V2 - V1}{T}$$

Where V2 is the final velocity and V1 is the initial velocity. Because velocity involves both speed and direction of movement, an object accelerates if either of these changes.

An object will accelerate if a force is applied. For an object of mass ***m***, the acceleration produced by a force ***F*** is:

$$A = \frac{F}{M}$$

Note: The magnitude of the acceleration of an object is the slope of its speed / time graph at any moment. Negative acceleration is sometimes called deceleration or retardation.

The above is helpful to know in relation to centrifugal compressor operation and control.

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In order to understand how a centrifugal compressor works to compress gas, we need to understand more about acceleration of a particle and how this occurs. To gain a better understanding of this phenomena we need to gain an understanding of a force called ***Centripetal Force***.

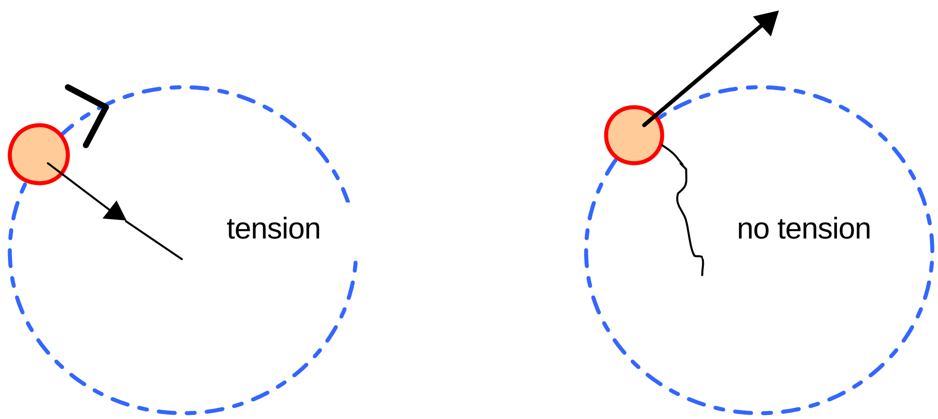
Centripetal force can be defined as ***a centrally directed force required to keep an object moving in a circle*** – that is, to provide the acceleration that is directed towards the center of the circle.

An object will move in a straight line at a constant speed unless there is a net (unbalanced) force acting on it. When this force acts at right angles to the direction of motion, the object will accelerate in the direction of the force and move in a circle.

The force acts at right angles to the tangent of the circle, and so is always directed towards its center.

Example: When a car turns a corner by moving in a circular arc the centripetal force is provided by road friction. Gravity provides the centripetal force keeping satellites and planets moving in their orbits.

The diagram shows an object moving in a circle at the end of a string. The tension in the string provides the centripetal force. If the string breaks, the object will not move outwards but will keep moving in a straight line which is a tangent to the circle.



CENTRIPETAL FORCE

CENTRIFUGAL FORCE

Centrifugal force is an apparent or fictitious force directed away from the center for an object moving in a circle. A force directed towards the center is required to keep an object moving in a circle, and if, for example, you were in a car taking a corner at sufficient speed you would feel this central force as the car body pressing against you. However, you interpret this as a (fictitious) force pushing you into the car body. It is best to avoid the term centrifugal force when discussing circular motion. However, we have to have an understanding of circular motion in order to assist in understanding the compression of gas in a machine called a centrifugal compressor in a process referred to as a ***dynamic***.

DYNAMICS - The study of the motion of objects and the forces causing changes in the motion.

What do we know about the operation of a centrifugal compressor?

- It is a device that transfers mechanical energy from a driver to a compressible medium, such as, a gas.
- It is sometimes called a ***dynamic*** compressor in relation to the energetic force that puts gas into motion quickly.
- Energy transferred results in higher gas pressure and temperature.
- The increase in gas pressure and temperature are a result of changes imposed on the gas by the compressor. In a centrifugal compressor, this results from a combination of centrifugal ***force*** and ***velocity*** changes imposed on the gas.
- A centrifugal compressor is about 75% efficient. About 75% of the energy provided by the driver is transferred to the gas for raising its pressure.
- About 25% is friction and internal losses.
- A portion of this loss enters the gas as additional temperature rise.

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- Note: Any problem with internal parts usually results and may be displayed as increased discharge temperature.

- In troubleshooting problems this is important to apply to diagnosis of problem cause (s).

- With suction temperature fairly constant, discharge temperature will give an indication of compressor performance.

- Temperature will rise:

  If the gas pressure is raised.

  If the machine looses efficiency because of component wear or fouling. If the “impeller” is not doing its work as designed.

GENERAL GUIDELINES FOR CENTRIFUGAL COMPRESSORS

- At a given speed discharge pressure varies inversely with flow.

- Volumetric flow is proportional to speed.

- Discharge pressure varies as a square of speed.

- Horsepower is proportional to speed cubed

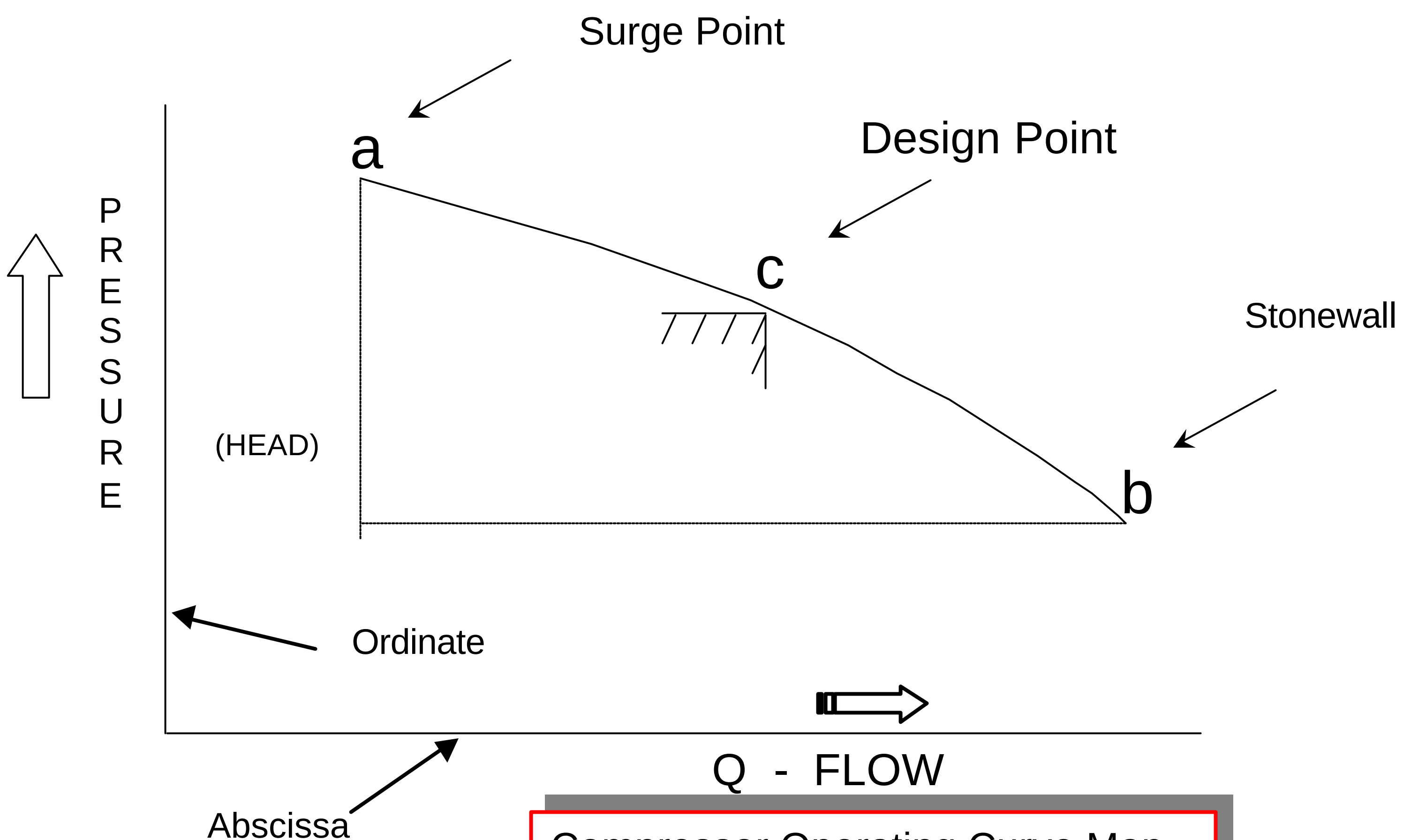
A centrifugal compressor has the capacity to perform work on a gas using an impeller(s) designed to impact on the gas, elevating the energy of the gas entering the eye of the impeller significantly increasing the velocity [speed + magnitude (direction)] and pressure of the gas. Approximately 2/3 pressure increase is achieved within the impeller. Gas exiting the impeller flows into a diffuser passage where it slows gaining an additional 1/3 increase in overall static pressure for a given stage of compression.

Three operating conditions:

- Design Point
- Stonewall
- Surge

Operation at Design Point

Every centrifugal compressor is designed to operate at a preferred optimum speed point relative to the impeller design. Impellers are designed to raise the gas pressure within limits that ensure the gas will flow at a desired production rate from the suction [inlet] of the compressor to the discharge of the compressor. This operating point is graphically defined along an operating curve and is referred to as "Design Point" operation.



Compressor Operating Curve Map

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To avoid operation detrimental to the centrifugal compressor, the compressor must deliver enough gas flow to overcome system resistance and maintain the open position of the compressor discharge check valve positioned ahead of the main motor operated compressor discharge valve during normal operation. With gas turbine driven units this can be done by varying the driver speed at start up and during operation maintaining operation at or near design point operation shown on the compressor operating curve as point ‘C’. Operation at this point will flow the right amount of gas at about 90% speed point operation relative to the density of the gas being compressed. At this point the compressor impeller(s) will deliver the pressure increase required at each stage and across the compressor between suction and discharge. Flow is defined as ‘Q’ on the abscissa and pressure or “Head” on the ordinate axis of the compressor map. Design point is different for each compressor designed. Operation at design point should ensure safe operation in an area that satisfies production requirements without hazard to the compressor assuming there are no changes at suction or in the process that affect head and flow [i.e. changes to the gas composition, weight, temperature, etc.]

A compressor is designed to compress a fixed volume of gas from a designated suction pressure to a designated discharge pressure. In operating situations, the suction pressure, discharge pressure, and gas flow rate are seldom at design conditions. The function of the control system is to safely operate the machine at the lowest cost within safe operating limits, while keeping operating supervision time low.

Controls for a centrifugal compressor serve three distinct purposes:

1. Capacity control
2. Surge protection
3. Equipment protection

PRINCIPLES OF CAPACITY CONTROL

Centrifugal compressors driven by constant speed drivers, such as electric motors, are usually equipped with capacity control systems of varied type. If the suction and discharge pressure, and gas flow rate to a compressor remain constant, no control is required. The capacity control system provides the means for operating the unit when process conditions of flow and pressure are not constant.

The process condition which can vary are:

Suction conditions – temperature, pressure, and flow rate.

Discharge conditions - Temperature, pressure, and flow rate.

Compression ratio = $\dfrac{\text{Discharge Pressure (Absolute)}}{\text{Suction Pressure (Absolute)}}$

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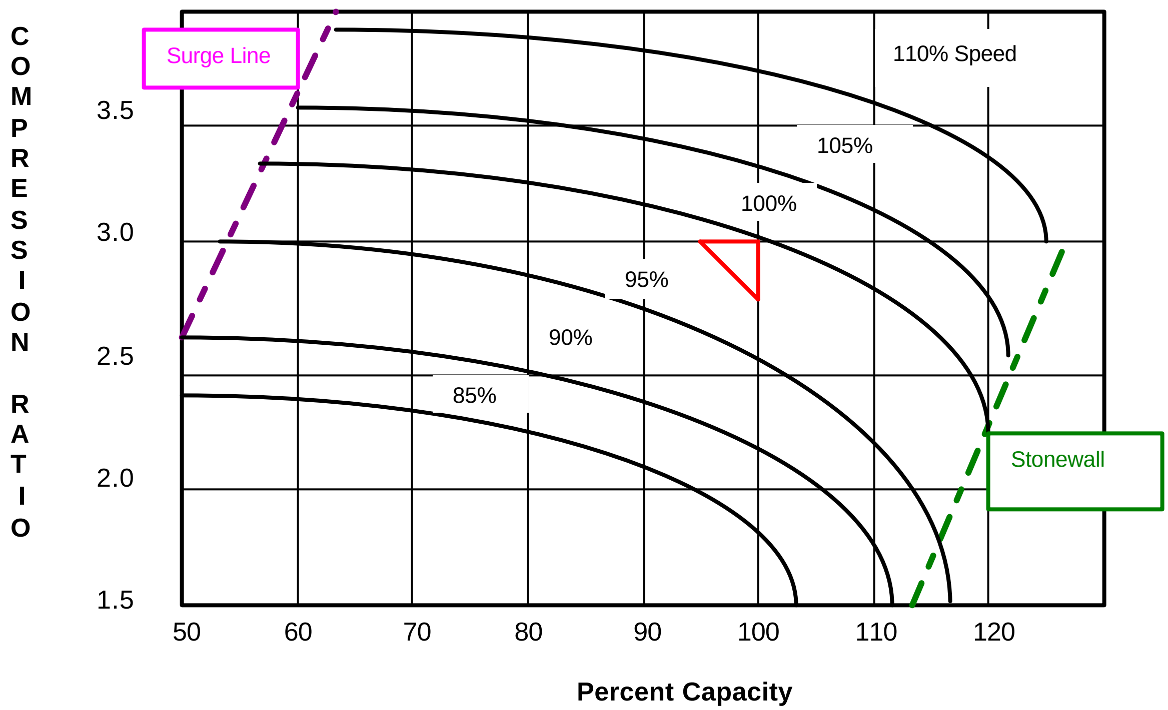
Gas Density – The pressure rise in a centrifugal compressor is higher with a gas having a high density than that of a low density gas.

The variables combine in a compressor to form its capacity characteristics, which are defined in the compressor operating curves or "map". At any compressor speed, the compression ratio developed at various gas flow rates [compressor capacities] are defined by the curve for that speed. The unit will operate only on a single curve at one speed.



For example, at a gas flow rate of 84% of design and a speed of 100%, the compression ratio this machine develops will be 3.2. Design compression ratio at 100% flow is 3.0. At 84% flow, the discharge pressure will be higher than design, or suction pressure will be lower than design. But, suppose suction and discharge pressure are at design conditions, which means that the compression ratio will also be at 100% design, which is 3.0. The compressor is going to deliver gas at compression ratio of 3.2 at 84% flow. The control system must get rid of the extra 0.2 of compression ratio that the unit will develop at 100% speed and 84% gas flow.

Assume that the gas flow is 84% of capacity, speed is 100%, and suction and discharge pressure are at their design points which require a compression ratio of 3.0. The compression ratio developed at 84% flow and 100% speed is 3.20, but operating conditions require a compression ratio of only 3.0. The capacity control system must get rid of the excess compression ratio. One way of doing this is with a throttling [capacity control] valve in the discharge line. The pressure drop across the throttling valve must be calculated properly to correct an excessive discharge at reduced capacity condition. Throttling valves can also be used in suction gas piping to get rid of excess compression ratio at reduced capacity and operation at 100% speed.

Another method of controlling capacity is by recycling some discharge gas back to suction to the compressor. Because this results in unwanted heat added to the compression process detrimental to the compressor, recycle gas is withdrawn from a discharge scrubber, for instance, which has been cooled and liquids removed from it.

In some compressors, adjustable guide vanes can pre-rotate the gas entering the impeller. This may load or unload an impeller depending on the pre-rotation of the gas in relation to impeller rotation which results in changing the capacity of the machine.

The four capacity control systems using only one method per unit for machines operating at constant speed are:

- Suction capacity control [throttling] valves
- Discharge capacity control [throttling] valves
- Recycle gas [not recommended as a long term operating solution]
- Adjustable guide vanes [varies with compressor supplier]

Speed Control

If the unit has a variable speed driver, the primary capacity control is that of varying the speed of the machine. Referring to the operating curve map on the previous page, the machine will develop the design compression ratio at 3.0 if the speed is reduced to about 97%. A throttling valve in the suction or discharge line is a secondary control for maintaining the desired compression ratio when speed alone will not provide the necessary capacity control.

Speed control is the simplest and most economical means of controlling the capacity of a compressor. The other control methods result in a waste of driver energy as a pressure drop is taken across a throttling valve or when gas is recycled around the unit. This waste is unavoidable in a constant speed machine. The recycle control results in the greatest energy consumption by the driver, so it is the least preferred method of control. Throttling the suction or discharge gas to maintain capacity control at reduced gas flow rates consumes less driver energy than recycling gas.

Summary of Capacity Control

1. The capacity of a compressor must be controlled so that it is the same as the volume of gas flowing to it.
2. Capacity is affected by speed and compression ratio.
3. Capacity is controlled on variable speed machines by adjusting speed and compression ratio.
4. Capacity is controlled on constant speed machines by adjusting the compression ratio. This is accomplished with:
    a. Control valve in suction line.
    b. Control valve in discharge line.
    c. Control valve in recycle line.
    d. Adjustable guide vanes at the compressor inlet [these are not supplied by all compressor manufacturers].

Note:
It should be noted that variable speed machines are limited by how much speed can be reduced. There is a limit to the amount speed can be reduced to control capacity and still develop the required compression ratio. Below that limit, gas recycle is required.

The operating range of a centrifugal compressor is restricted by a condition called ***surge*** at low flow rates, and a condition called ***stonewall*** or ***"choke"*** at high flow rates. The limitation of surge in a centrifugal compressor is the most critical of the two operating conditions.

STONEWALL OPERATION (Choke)

Maximum flow at a given speed is fixed by the ***stonewall*** limitation. Stonewall occurs at high flow rates when the impellers raise the velocity of gas to the point that it reaches sonic level speed. At this speed, more gas cannot be forced through the compressor, regardless of what we try, so we have hit a "stone wall" or "chock" condition. When this occurs, shock waves cause a rapid decrease of pressure and break down of flow pattern. Consequently, we want to stay away from the operating near the stonewall or high flow point. The compressor operating curve map will reflect maximum flow in acfm [actual cubic feet per minute] or as percent [%] capacity. It will appear at the far right of the operating curve map.

The stonewall effect is of minimal concern in most oil field and gas processing compressor applications where the gas is methane, ethane, propane, or a mixture of the three. The conditions of flow, speed, and compression ratio that result in stonewall seldom occur in most oil field operations. Consequently, controls to prevent stonewall are rarely used.

SURGE PHENOMENON AND PREVENTION

The operation of a centrifugal compressor is restricted by a condition called ***surge*** at low flow rates.

Inside the compressor there are no check valves or other mechanical devices to prevent discharge gas from flowing backwards towards the suction side of the unit. Flow passages are open throughout all impellers, diffusers, etc. Backflow can occur if two conditions prevail.

1) Low gas flow rate

2) High compression ratio

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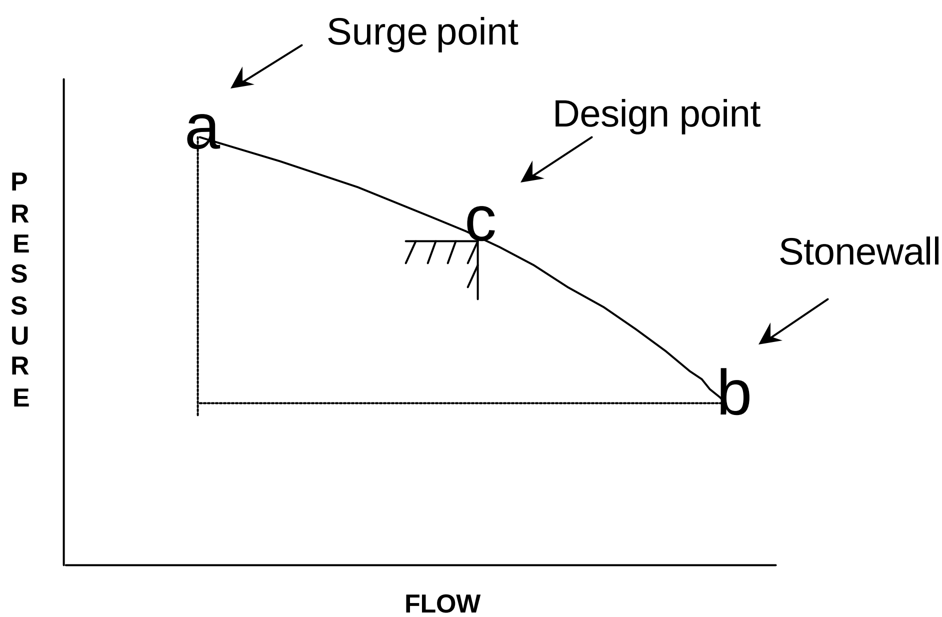
A high compression ratio can result from a high discharge pressure, or low suction pressure, or a combination of the two. For example, at 100% speed, surge will occur at a flow rate less than 58% and a compression ratio slightly over 110%. At these conditions of low flow and high compression ratio, discharge gas will backflow through the compressor. As soon as backflow occurs, the discharge pressure drops, and forward flow starts again. When discharge pressure rises to the surge point, flow reversal occurs again as shown on the following pages.

Surge may occur several times a minute, or several times a second, depending upon a compressor's size, speed, etc. Surge is obvious from a definite sound that may be characterized as a whine, scream, moan, or a sharp audible hammer tapping noise, or thumping sound in large capacity pipeline gas transmission centrifugal compressors.



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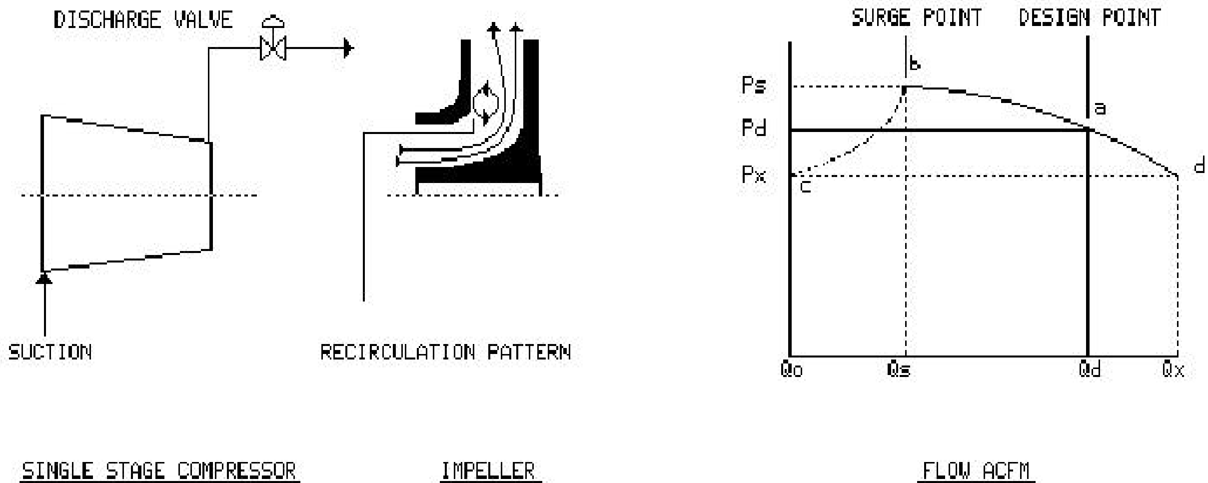

1. As the flow through a centrifugal compressor is progressively reduced by slowly closing the discharge valve in "A" above), the discharge pressure increases.

2. With this reduction in flow, a recirculation pattern develops in the impel as shown in "B".

3. At some minimum flow, the recirculation flow pattern collapses. The impeller can then no longer develop the discharge pressure existing or the system. This flow is identified as Qs in "C".

4. Since the pressure developed is less than that in the downstream syst flow reversal occurs. The delivered flow from the compressor then immediately drops to zero.

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5. When the pressure of the downstream system has dropped to Px, which corresponds to the pressure which can be developed at zero flow, the compressor delivers at rate Qx.

6. If the position of the discharge valve has not been altered, the flow and the discharge pressure change along the compressor characteristic curve until the surge point is again reached.

7. The change from b -> c -> d -> a -> b represents a complete surge cycle.

The relative "violence" of surge is determined principally by:

- The pressure differential across the compressor.
- The volume of the down-stream system.
- The slope of the characteristic curve.
- The gas being handled.

The location of the surge points depends on such factors as the number of impellers, the design ratio, the relative impeller loading, blade angle and the method of control.

It is not desired and not recommended to operate a centrifugal compressor in surging conditions as any time because:

- High rotor vibration levels may occur, with consequent damage to seals.
- High transient thrust loads develop, which may cause thrust bearing failure.
- Excessive heating of the gas which aggravates the surging and affects close tolerance clearances (i.e. bearings, seals etc.)
- Cyclic loading from practically no load to overloaded conditions in short periods of time can be harmful to the driver and the driver controls as well as drive system (couplings, gearing, shafting, impellers, etc.)

In conslusion, the effects of operating a centrifugal compressor in surge may cause severe damage to the compressor and piping. Check valve flappers often break, and internal damage to the compressor can occur. In addition, the back flowing gas tries to reverse the rotation of the machine so added strain is put on couplings between the compressor and the driver.

SURGE PREVENTION

Surge can be avoided by maintaining the flow above the surge point. This is done by competent operation and through the operation of a reliable surge prevention system that will sense that the compressor approaching surge at approximately 15% of the surge line as flow rates fall at a given speed. This will signal opening of the compressor gas recycle valve allowing flow of some discharge gas back to the suction side of the compressor in order to maintain enough flow through the unit so that operation in surge is avoided.

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Generally, a valve called an anti-surge control valve is installed in the discharge from the compressor system. The valve is regulated with a flow controller that senses the flow rate of gas entering the compressor. The set point on the flow controller is set above the reduced flow rate surge occurs. If the flow to the compressor is less than the set point in the controller, the controller signals the anti-surge control valve to open and allow enough recycle gas to flow into the suction line so that the total flow to the compressor is above its surge point.

On variable speed machines, the gas flow rate that results in surging varies with speed. The set point in the surge flow controller has to be reset each time the speed changes. This is done automatically with a speed controller, which resets the set point on the surge flow controller when speed changes. Normally, the surge flow controller set point is reset by change in compression ratio or differential pressure [head] of the compressor. Regardless of the sophistication and complexity of the surge prevention system, the basic control is that of maintaining flow above the surge condition. In any event, recycle control regulates flow [capacity] to maintain a certain compression ratio for the gas being compressed.

The combination of flow and compression ratio that results in surge at each different compressor speed should be determined at the time the unit is commissioned. This information is then used for subsequent control of the surge prevention system to prevent it from occurring during normal operation.

Surge can also occur during start up and shutdown of a centrifugal compressor, so surge protection must be provided during these operations, particularly on large volume flow machines.

SUMMARY OF SURGE PREVENTION SYSTEM:

1) Surge occurs at a low flow rate and high differential pressure in a centrifugal compressor. At these conditions, gas back flows through the compressor unit.

2) The primary means of preventing surge is to prevent flow from dropping to the point that Surge will occur by recycling discharge gas to the suction.

3) Secondary control devices that measure compression ratio in some form are often used to override the primary flow control device and increase flow of recycle gas to prevent the compression ratio from reaching the surge condition.

4) The surge prevention system should be set to avoid surging conditions at the lowest power Consumption of the driver.

CRITICAL SPEED

All solid materials have certain vibration levels called **natural frequencies**. Because of this peculiar things can happen under certain circumstances. A glass, for example, will shatter from sound vibrations at its natural frequency. An automobile part may vibrate at one speed, bot not at speeds above or below the speed inducing the vibration. This is its natural frequency. If a material is subjected to vibration at its natural frequency for any extended period of time, it will usually disintegrate.

The rotating member of the centrifugal compressor, shaft assembly [rotor and impellers], can be affected by vibration. The rotor has a natural frequency of vibration that occurs at a certain speed of rotation called **critical speed**. If the unit operates at its critical speed for an extended period, some form of breakage is almost certain to occur. In almost all cases, the critical speed is below

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the normal operating speed range of a compressor. The minimum operating speed is usually specified by the compressor manufacturer safely above the critical speed. Compressor starting control logic programs include sequences that pass through the 1st critical speed as quickly as possible. Passage through the 1st critical speed is accomplished by bringing the unit up to the minimum operating speed as quickly as possible when the unit is started. Vibration monitors will usually show a flutter on the start up or shutdown when the speed reaches the 1st critical speed point.

Secondary or **harmonic** natural frequencies of lesser magnitude occur a higher speeds than the low speed or 1st critical. The different speeds which result in natural frequency vibration are referred to as the 1st critical [lowest speed] and the 2nd critical speed, etc. Most centrifugal compressors operate at a speed between the 1st and 2nd critical speed.

COMPRESSOR MAINTENANCE

RECOMMENDATIONS AND BEST PRACTICES

General

The best maintenance program begins with proper centrifugal compressor installation and includes correct start-up, shutdown, and operating procedures. A conscientiously followed program of preventive maintenance, a part of which is the gathering and interpretation of accurate operational data and periodic inspection records, must be followed. These records should be reviewed on a weekly basis to determine any trends, which would indicate the extent and type of maintenance that is or soon, may be required. Maintaining, analyzing, and acting upon accurate shift and daily records as well as P.M. inspection results is the key to long compressor life.

When taking shift or daily records, the operator should perform a walk around visual inspection for external evidence of required maintenance. This inspection should include close examination of all external piping, tubes, fittings, hardware, lockwire, cover plates, connectors and sight glasses, as well as looking for evidence of oil and gas leaks. In addition, all casings, housings, and tanks should be inspected to determine that all attachments and fasteners are in proper order and secure.

Regular general inspections should be performed periodically; time intervals between these major inspections should be determined by local operation conditions and experience. During the regular inspection, thoroughly clean the aerodynamics, impellers, rotor, casing and coupling; check for worn and damaged parts. If it becomes necessary to replace a part, it is of extreme importance that the cause of the failure be fully determined and that any corrective action that is required be taken before the unit is restarted.

Maintenance should be done only by trained, qualified personnel who are familiar with centrifugal compressors. Prior to disassembling any internal components, a review of the manufacturer's maintenance manual is recommended to avoid any unnecessary disassembly or damage due to incorrect disassembly. Servicing the compressor without a working knowledge of its internal parts could cause damage, expense, and unnecessary time. Care must be taken during disassembly to avoid damaging parts; when reassembling, work procedures must insure that no dirt enters the compressor.

Dirt is one of the biggest enemies of any piece of machinery. All the safety and control devices provided on a compressor can do little to save the machinery when dirt has been introduced into the system. Only the personnel doing the work have control over this area. Special care must be taken when removing and reconnecting the oil inlet and outlet piping to keep dirt from entering these openings.

If a prolonged shutdown is anticipated, it is necessary to protect the bearings, rotor shaft, seals and all finished surfaces of the compressor against corrosion. The procedure to be followed depends on the length of time the unit is to be shut down and on local ambient conditions. Various compounds for corrosion protection are available, and each manufacturer has recommendations for their use. The lubricating oil should be drained, the lubricating system flushed and refilled to the proper level with fresh, clean oil.

SAFETY PRECAUTIONS

Safety precautions must be carried out when performing maintenance work on any piece of equipment.

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Tag the control panel. This tag should be put in place by the person doing the work and removed only by him. Switches, which allow startup of any system being serviced, should be tagged; this tag should state that maintenance personnel are working on the system.

Disable the valves so that they cannot accidentally be operated, thereby causing serious injury or possible death. Relieve all pressure and insure that pressures are safe before breaking any high-pressure flanges. Do not remove pressure gauges or pressure switches until the lines n which they are mounted are relieved of all pressure. As some compressor lubricating oil can be toxic through skin absorption if allowed to touch the skin for long periods, any necessary precautions to avoid extended contact should be taken.

On re-assembly of any compressor, there are several standard rules which should be followed. All internal threaded fasteners should be installed with loctite or lockwire. Tightening of threaded fasteners, both inside and outside the compressor, should be done according to manufacturers torque data.

GENERAL INFORMATION:

Loctite – an aerobic compound which incorporates a single component polyester-type, liquid resin which hardens when confined between close fitting metal parts to provide a seal and/or locks parts in place.

Lockwiring – a technique used to prevent a fastener from disengaging and dropping free, should it for some reason become loose. If installed correctly, lockwiring will also aid other locking devices in keeping the fastener tight.

Tightening requirements – screws, bolts, studs, and nuts are to be tightened to specified torque values or otherwise tightened per the manufacturer's tightening instructions found in the maintenance manual.

O-RINGS

O-rings – anytime a housing or assembly which uses o-ring seals is removed from the compressor, the o-ring seals should be replaced prior to re-assembly. Use only correct material o-rings. To assure maximum sealing and to reduce the risk of assembly damage, lubricate the o-ring with petrolatum (petroleum jelly). Do not twist the o-ring when installing it in the groove.

COMPRESSOR TOOLS

Compressor tools – Despite the high reliability inherent in today's centrifugal compressor designs, major maintenance work is still periodically required. For that reason, the manufacturers have designed a series of special tools, which greatly facilitate such maintenance work as compressor rotor removal. Other special tooling (e.g., sleeves to protect bearing and seal surfaces of the shaft, hydraulic tooling for assembly and removal of such shrunk-on components as impellers, couplings, etc., lifting brackets, pullers) is specialized for each individual compressor.

COUPLINGS

Couplings – Shaft couplings must permit considerable axial shaft displacement and minor angular misalignment. Misalignment at high speeds must be minimized by a “hot alignment” check after the unit is warmed up.

There are several coupling manufacturers making many different coupling designs. The gear-type coupling is probably the most common; lubrication for a gear-type coupling is provided by a grease filling or an oil circulating system. Diaphragm couplings are preferably used today because of high speed and torque capabilities as well as the maintenance benefits possible because this coupling requires no lubrication.

Performing maintenance on the drive end of the compressor requires the removal of the coupling guard and spacer. Once the coupling guard is removed, determine if the coupling spacer, hubs, gear sleeves and shafts are match-marked. If not, match-mark them before removal to insure reinstallation in their original position. Each bolt and locknut is normally weigh-balanced as a unit. After removing each bolt, re-assemble the corresponding locknut to assure that they are maintained as a unit. The coupling has been dynamically balanced as a unit; therefore, bolts and nuts should not be intermixed with other couplings.

Once the spacer is removed, clean and inspect all coupling parts for cracks, chips, or other damage. Dye check parts if cracks are suspected. If a hub or spacer is defective, the coupling must be replaced.

Depending on the particular application, the coupling hubs could be fit to the shaft by splines, keyed to a tapered shaft, or assembled with a hydraulic fit. In any of these cases, the hub removal/installation tools supplied with the compressor should be used according to the manufacturer’s instructions for disassembly and re-assembly.

JOURNAL BEARINGS

Journal bearings – Since a great variety of different types of compressors built by various manufacturers are in service today, a piece-by-piece description of all disassembly and re-assembly procedures will not be provided. We will discuss the disassembly and inspection procedures used with a tilting pad, self-aligning type journal bearing. In our example, each journal bearing contains five babbitt-faced bearing pads.

When removing or installing any bearing assembly from the compressor, be sure to install the assembly sleeves of guide sleeves on the shaft. These sleeves are supplied with your special tools and are used to protect both the bearings and the shaft. A special puller will also be required.

The impeller shaft should be centered to relieve the bearings of radial load, and the shaft should be held in this position while the bearing is removed. Whenever a bearing is removed, the rotor shaft must be supported, centered, and maintained in that position to prevent damage to the rotor shaft seals.

Once the bearing housing is removed, all components should be disassembled and cleaned in a solvent suitable for the lubricant being used. Air and oil passages should be blown out with compressed air. Carefully examine the bearing pads to ensure that none of the following conditions exist:

1. Significant wear over more than 50% of the pad surface.
2. Any wiping of babbitt that has smeared over the surface of the pad.
3. Babbitt that has been peened so that it overhangs the steel back at the babbitt-steel interface.
4. Surface cracks in the babbitt. Use of a 5-power glass is recommended for this inspection.
5. Evidence of the steel back having rocked, resulting in witness marks more that 3/8” in circumferential width.

Pads showing any of the above signs must be replaced.

To check and determine the bearing clearance on a tilting-pad bearing, the following procedure should be used:

1. Accurately measure the bearing housing inside diameter, the shaft outside diameter, and the average thickness of the pads. Pad thickness must be measured at the thickest radial section of each pad.

2. Clearance is equal to the housing inside diameter minus twice the average shoe thickness minus the shaft outside diameter.

3. If this value exceeds the maximum allowable bearing clearance listed in the manufacturers’ maintenance manual, new pads must be installed to provide correct clearance. It is desirable, in terms of bearing pad life, to use an unworn pad in the location that shows the greatest wear.

On re-assembly, coat all fastener threads with Loctite #271 or a similar product. When installing the pads in the bearing housing, make sure that the capscrews do not prevent the pads from seating in the bearing housing at any position. These bolts only serve to prevent pad rotation and must not bottom in the bearing pad. Re-machine these bolts to provide clearance if necessary.

OIL FILM SEALS

Oil film seals – Seal assemblies are removed and disassembled from the inlet or discharge end of the compressor in the same manner as the journal bearing (except in the case of the overhung pipeliner). When removing the seals, be certain that the shaft is supported. Extreme care should be taken to avoid damaging the seal rings.
The use of special tools such as seal guide sleeves and the correct pullers will be necessary for removal of the seal assemblies.

1. Disassemble and clean all parts; blow out all passages.
2. Inspect labyrinth seals for nicks, cracks, burrs, or evidence that rotor contact has been made.
3. Inspect the seal housing, retainer, and the seal rings for cracks, nicks, distortion, scoring, and evidence of insufficient lubrication. Replace all defective parts.
4. Check the running clearance between the inner and outer seal rings and impeller shaft. If the actual clearance exceeds the maximum allowable clearance listed in the manufacturer’s maintenance manual, replace the seals.
5. Inspect the lapped fit of the seal rings to the seal housings. If necessary, relap these parts. If lapping fails to make these parts usable, they must be replaced.

For detailed instructions, refer to the manufacturer’s maintenance manual.

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DRY GAS SEALS

Dry gas seals are a complete assembly inserted into the compressor from both or one end depending on the compressor type and model. Special removal tools are employed to remove the seal assembly. Entering sleeves employed to protect the shaft during removal of the seal assembly must have the correct OD [outside diameter] prior to removal of the seal from the compressor. Special tools are used to retract the seal where it can be taken to the bench for further disassembly. Disassembly of the seal should be done cautiously, by experienced, highly qualified personnel practiced or certified to conduct disassembly and inspection of the seal. If in doubt about the procedures, contact the seal supplier and seek help from a qualified service representative to protect and preserve the carbon ring which is expensive to replace. Special schools are available to gain the knowledge and skill required to conduct break down maintenance on the dry gas seal assembly. It is recommended anyone conducting maintenance become qualified prior to implementing maintenance procedures on dry gas seals.

During operation the mating ring rotates providing a pressure “dam” of cooling, sealing gas sufficient to separate the primary [carbon ring] from the mating ring. A relatively small amount of seal gas flows to vent.

Most problems are caused when liquids enter the system and/or the primary ring is “cocked” in its orientation to the mating ring, especially, during start up phases. Forces normally applied to lift the primary ring from the mating ring when improperly seated can produce potentially high loads on the primary ring and damage the seal.

Recommended Practices:

1. Maintain the integrity of the dry gas seal system at all times through good preventive maintenance of the filters and flow controllers and make certain there is no ingress of oil from the bearing area and proper operation of a barrier seal, if installed, to restrict liquids form entering the seal and adversely affecting operation of the carbon elements of the seal.

2. Make sure that vents are free flowing and flowing the correct amount of gas.

3. Make sure the special tool entering sleeves [if provided], are the correct OD prior to removal of the seal assembly. Raising a burr from an oversized entering sleeve can damage the interior mating surfaces of the dry gas seal. MEASURE AND VERIFY THE DIAMETER OF THE SHAFT PROTECTION SLEEVE TOOL IS CORRECT BEFORE REMOVAL OF THE SEAL ASSEMBLY.

4. Dry gas seals must be re-installed to rotate in the right direction. Seal assemblies are usually marked with a directional rotation arrow. Install to ensure proper direction or rotation. Failure to install the seal and ensure proper rotation can result in ineffective sealing and destruction of the seal during start up and/or operation.

5. Dry gas seals require correct axial position in the compressor. Failure to ensure correct axial position can, and, most likely, will result in destruction of the seal.

THRUST BEARING

The thrust bearing covered in our example is the tilting pad, self-leveling bearing, separated by a steel, precision-lapped thrust collar. Because the thrust collar is mounted with an interference fit, the use of a special puller will be required for its removal.

Completely disassemble, clean, and inspect all parts of the thrust bearing for cracks, distortion, wear or other damage. Inspect the thrust bearing pads for scoring, pitting, and excessive wear. Pads are serviceable so long as the bearing surface is not worn through and is free of defects. Replace any defective parts.

When re-assembling the leveling links, be certain that all lower links are assembled with the rounded surface down and all upper links are assembled with the flat surface up. The flats of the upper links must be in line with the support disc radii on the pads. In installations where bearing pad thermocouples are used, make sure that the pads drilled for the thermocouples are installed in the same position from which they were removed.

Install the inner thrust bearing in the bearing housing, making sure the thrust bearing locking dowel engages in the bearing housing. Next, install the thrust collar spacer on the impeller shaft. If this spacer must be replaced, some grinding may be necessary. This thrust collar spacer is one of the most vital parts of the compressor as it centers the rotor assembly in the stationary aerodynamic assembly.

Carefully heat the thrust collar to 125^{o}F above the temperature of the rotor shaft and install the thrust collar, spacer, and locknut on the impeller shaft. Tighten the locknut to push thrust collar against spacer. After the thrust collar and shaft have cooled, remove the locknut. With the outboard thrust bearing assembled in the thrust housing, and installing the shim between thrust housing and the bearing housing, secure these components with socket head screws. Install the lockwasher and locknut on the impeller shaft. Tighten the locknut against the spacer, and back off at least on locking tab, and lock with tab. <u>DO NOT</u> overtighten this thrust collar locknut or a bent shaft may result.

Checking and/or Adjusting Thrust Bearing Clearance is accomplished as follows:

1. Force the rotor shaft as far toward the inlet end of the compressor as possible, position a dial indicator against the rigid coupling hub on the rotor shaft, and set the indicator on zero.

2. Force the shaft back toward the discharge end of the compressor as far as possible and read the dial indicator. Clearance should typically be .011 to .017 inch. Check with the manufacturer’s maintenance manual for each compressor.

3. If the dial indicator reading shows excessive end play, the thrust pads are worn. The desired clearance may be obtained by reducing the shim thickness.

4. If the indicated clearance is less than .011 to .017 inch (or the clearance specified in the manufacturer’s manual). Shims may be added to increase the end play. However, if an excessive amount of shimming is required, the pads in the thrust bearing may be cocked or out of place.

Setting Rotor Shaft for Proper Running Clearance:

If a new compressor rotor or thrust collar spacer is installed without centering the rotor, serious damage will result to the compressor. The procedure used for locating the rotor assembly is as follows:

1. The rotor assembly must be installed in the stationary aerodynamic assembly with the seals and journal bearings installed and the thrust bearing, collar and spacer removed in order to measure the overall travel of the rotor.

2. Move the rotor shaft as far as possible toward the “cover” end of the compressor. Position a dial indicator against the end of the rotor shaft at the coupling end and set the indicator at zero.

3. Move the rotor shaft toward the drive end (coupling end) as far as possible and record the reading of the dial indicator.

4. Install thrust collar spacer, inboard thrust bearing and thrust collar.

5. Recheck the rotor movement as outlined in steps 2 and 3. The reading Should now be approximately ½ of the recorded reading taken in Step 3.

6. If the rotor movement obtained in step 5 is greater than ½ the total movement recorded in step 3, the thrust collar spacer must be ground to obtain the correct position.

7. If the rotor movement obtained in step 5 is less than ½ the total movement recorded in step 3, the thrust collar spacer must be replaced with one of greater thickness to obtain the correct rotor running position.

COMPRESSOR IMPELLERS

The compressor impellers used on the overhung-shaft type Pipeliners are removed and installed by hydraulically expanding the impellers. A hydraulic pump and hose capable of supplying 45 thousand PSI hydraulic pressure is required.

When removing the impeller, match-mark the impeller and shaft to facilitate re-assembly, and install the special impeller removal tool on the impeller shaft. Make sure the tool is seated against the impeller, and then back off ¼ inch on the capscrews. Before installing the hydraulic pump, remove all air from the pump and line by operating the pump handle until oil (free of air bubbles) flows from the line. Build up the hydraulic pressure until the impeller “pops” free and rests against the impeller tool. Approximately 25 to 30 thousand PSI should free the impeller. Repeat this procedure until the impeller is removed.

Do not attempt to remove the impeller hydraulically without the impeller removal tool installed, or damage to the unit and injury to personnel may result. DO NOT exceed a hydraulic pressure of 45 thousand PSI. If a new impeller is used, the impeller and rotor shaft must be dynamically balanced.

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On re-assembly, make sure the impeller and impeller shaft match-marks are aligned. Position the impeller tool and tighten the capscrews evenly against the impeller tool to a torque of 75 ft. lbs. Remove all air from the hydraulic pump and install hydraulic tubing. Apply hydraulic pressure and at same time maintain the 75 ft. lb. torque on the capscrews. Approximately 30 to 40 thousand PSI will be necessary to expand the impeller. When the impeller is fully seated, the hydraulic pressure will continue to increase and the torque on the bolts will increase sharply. DO NOT exceed 45 thousand PSI hydraulic pressure. After the impeller is seated in place, release the hydraulic pressure. Do not remove the impeller tool until the impeller has remained in place for at least 15 minutes with the hydraulic pressure released.

On vertically split barrel casings and beam style Pipeliners, the rotor assembly (including impellers) is removed as a unit with the aerodynamics. This combination is commonly referred to as the bundle; the bundle is removed from the casing by adding support brackets on the inlet end of the compressor and then using a bundle puller. The bottom half of the aerodynamics is equipped with four small rollers, which ease this procedure.

Another method involves the installation of rails on the cover end of the compressor; using a saddle and cap (special tool) to support the rotor, the bundle assembly is rolled out of the compressor case by connecting a hydraulic pump to an actuator on the opposite end (or drive end) of the compressor shaft.

With the pump hose connection on the hydraulic actuator loose, operate the pump handle until fluid free of air bubbles is observed at the actuator hose connections. Tighten the hose connection fitting and slowly build up pressure until the actuator extends and pushes the rotor shaft toward the cover end of casing. During removal and assembly, the drive end of the rotor is supported by special tools such as insert tool sleeves. Refer to the compressor manufacturer’s maintenance manual for detailed instructions.

Once the bundle is removed from the casing, the aerodynamics must be “split”. The aerodynamics are bolted together along the horizontal centerline of the bundle. Once the capscrews that secure the two halves are removed, use a suitable lifting device to carefully lift the top half of the aerodynamics off and set this top half aside.

Be extremely careful when removing and handling the rotor and aerodynamic assemblies in order to prevent damage to the impellers and labyrinths. Do not remove an impeller or balance piston unless it is damaged. If a new impeller or balance piston is installed, the rotor assembly must be dynamically balanced. Inspect the impellers for deposits, corrosion, cracks, breaks, or excessive wear. Inspect the rotor shaft for nicks, scoring, or excessive wear. Inspect all labyrinth seals and balance piston seal rings closely for the slightest damage; replace as necessary. The cause of any excessive damage to labyrinths must be thoroughly investigated.

On Pipeliners, other parts of the aerodynamics such as the diaphragm, discharge scroll, and the seal holder are removed with the aid of the specialized assembly tool. This tool is provided with interchangeable adapter plates and adjustable balancing weights.

When re-assembling the compressor, remember how important cleanliness is to the proper operation and life of the unit. The following procedure should be followed:

1. Clean all parts in a suitable solvent and dry thoroughly.
2. Blow out all passages with compressed air.
3. Inspect and carefully check every item.
4. Use only new o-rings and gaskets where needed.

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5. Loctite or lockwire all internal threaded fasteners.

6. Tighten all threaded fasteners to the correct torque value.

Disassembly and re-assembly of the compressor should be done with the utmost care. Maintenance should be performed only by properly trained, qualified personnel who are familiar with centrifugal compressors. Prior to disassembling the internal components, a review of the manufacturer's maintenance manual is recommended to avoid any unnecessary disassembly. Care must be taken to avoid damaging parts, and no dirt should enter the compressor upon re-assembly.

Compressor Instrumentation
What is its purpose?

Purpose:

To provide data on an operational compressor necessary to assess:

1) Mechanical performance of various components or systems at a specific point in tin
   Examples: Vibration and bearing temperature

2) Aerodynamic performance of the compression process at a specific point in time.
   Examples: Inlet and discharge gas pressure and temperature

3) Trend of any critical parameter over time.
   Examples: Vibration, bearing temperature

Compressor Instrumentation/Mechanical Performance
What is Measured?

Parameter	Where Measured	How Measured	Output Received
Rotor Shaft Vibration	At each journal bearing	2 Proximity probes @ 90° apart	Mils – 0.001 inch Microns – 0.001 mm Total or filtered
Key Phasor	At free shaft end on non-thru drive units only	1 Proxiimity Probe	Angular, position of phase marker
Rotor Axial Position	Usually at non-drive shaft end	1 or 2 Proximity Probes	Position – as above Axial Vibration – as above Total or Filtered
Journal Bearing Pad Temperature	At each journal bearing	2 Simplex RTD's Each mounted within a separate pad	Deg. C Deg. F
Thrust Bearing Pad Temperature	On both active and inactive sides of the thrust bearing	2 Simplex RTD's each mounted within a separate pad (Total 4 pads)	Deg. C Deg. F
Oil Drain Temperature	At each oil drain pipe	Local – Gauge Remote – RTD	Deg. C Deg. F
Oil Drain Assessment	At each oil drain pipe	Sight flow indicator	Visual assessment Foaming, Level

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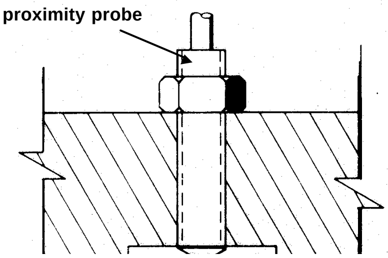
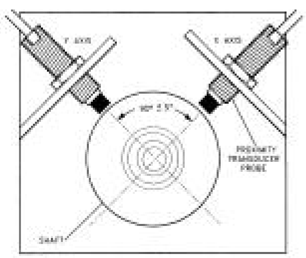
Compressor Instrumentation
How a Proximity Probe Works

1) Probe emits a low power radio frequency
2) If no conductive material is within range, virtually all power released is returned to the probe
3) If conductive material is within range, generated eddy currents cause signal loss
4) Signal loss is utilized to generate an output voltage
5)

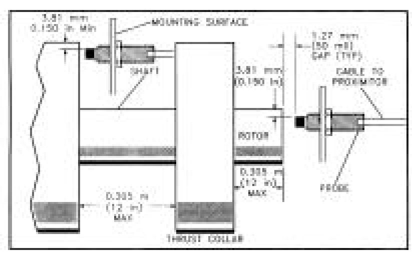
Axial or radial proximity probe



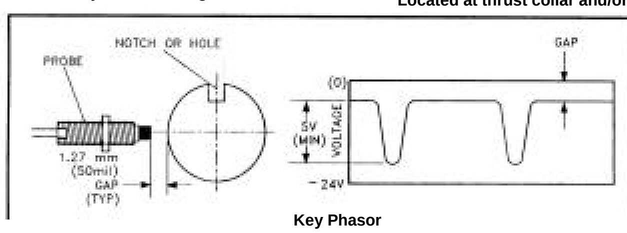
Compressor Vibration/Position Probe Location



Radial Vibration
Located at each journal bearing

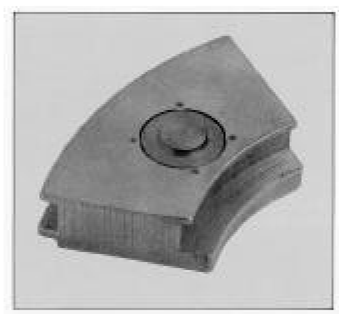


Axial Position
Located at thrust collar and/or shaft end

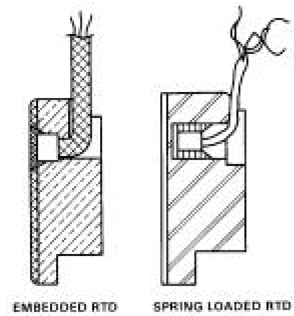


Key Phasor
Located at free shaft end
non-thru drive, beam style units only

Compressor Instrumentation
How Bearing Pads are Instrumented



Typical Thrust Pad



Types of RTD Attachments

RTD: Resistance Temperature Detector
A device which is utilized to measure temperature change as a direct function of sensor resistance change

Compressor Instrumentation / AerodynamicPerformance

What is Measured

Parameter	Where Measured	How Measured	Output Received
Inlet Gas Temperature	Inlet Compressor Flange	Local – Gauge Remote – Transmitter	Deg. C Deg. F
Inlet Gas Pressure	Inlet Compressor Flange	Local – Gauge Remote – Transmitter	PSI KG/CM2 KPA BAR
Discharge Gas Temperature	Discharge Compressor Flange	Local – Gauge Remote – Transmitter	Deg. C Deg. F
Discharge Gas Pressure	Discharge Compressor Flange	Local – Gauge Remote – Transmitter	PSI KG/CM2 KPA Bar
Gas Flow	Customer Piping	Remote – Flow Meter, Orifice, Venturi, Nozzle	SCFM ACFM Lbm/min.
Gas Characteristics	Customer Piping	Remote – Gas Chromatograph, Gas Spectrography	% Gas Components • Mol Weight • Enthalpy • Entropy • Density
Compressor Speed	Driver Output	Remote – Magnetic Sensor	RPM

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Compressor Instrumentation

Rotor Shaft

- Two probes @ 90° apart at each journal bearing
- One key phasor probe (only on non-thru drive units when required)

Rotor Shaft Axial

- One or two probes usually at the non-drive shaft end

Bearing Temperature

- One single (simplex) RTD in two drive-end journal bearing
- One single (simplex) RTD in two face-end journal bearing
- One single (simplex) in two thrust pads on both sides of the thrust bearing

TROUBLESHOOTING

Troubleshooting an operating problem is one of trial and error in which the various causes of the problem are investigated until the root cause is found. Causes which can be quickly checked by observing instruments and displays of real time criterion should be check first along with trend logs and report, as they can be done quickly and eliminated as a source of the problem.

The most common operating problems on a centrifugal compressor are:

1. Low compressor capacity
2. High compressor discharge temperature
3. High discharge pressure
4. Problems in the lube oil system
5. Problems in the seal system

TROUBLESHOOTING LOW COMPRESSOR CAPACITY

Cause	Corrective Action
1. Low speed	a. Check speed and increase to that required.
2. Too much recycle gas	a. Check surge or recycle controller and reset set point above surge point.
3. High compression ratio. Suction pressure is low and/ or discharge pressure is up.	a. Calculate compression ration to ensure it is up. b. Determine cause of low suction pressure or high discharge pressure and correct it.
4. Fouled impellers or diffusers, this condition usually builds slowly over a period of time.	a. Conduct an internal inspection, verify and repair.

TROUBLESHOOTING HIGH DISCHARGE TEMPERATURE

Cause	Corrective Action
1. Suction temperature has risen.	a. Check temperature of inlet gas, and correct if it is up.
2. Compression ratio has risen.	a. Calculate compression ratio. Determine cause of increase – suction pressure is down and/or discharge pressure is up – and correct.
3. Fouled impellers or other internals.	a. Inspect and repair.

TROUBLESHOOTING LUBE OIL SYSTEM

Cause	Corrective Action
1. Low oil pressure	a. Low level in oil reservoir. Check level and add oil. b. Oil pressure regulator set too low. Raise pressure setting. c. Worn bearings. Unit vibrates. Drain oil from bottom of tank and check for bearing cuttings. Conduct oil analysis. Inspect bearings and repair. d. Oil pump failure. Start stand-by pump. Check automatic start up. e. High oil temperature. See below. f. Filter is plugged. High pressure drop indicated. Switch and change filter elements in first filter.
2. High oil temperature	a. Heater in oil tank is on. Turn it off. g. Cooling fluid flow to oil cooler is low. Fan problem. Temperature Control Valve [TCV] problem. Check inlet and outlet temperature of oil and cooling fluids, and check pressure drop of fluids. Fouling is in fluid with high pressure drop. Clean cooler. Test TCV valve operation per vendor instructions. Repair or replace. Determine status of fan operation. Check vibration and lubrication. Repair or replace as required – check cooler louver control operation.

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TROUBLESHOOTING SEAL OIL SYSTEM

Cause	**Corrective Action**
1. Low level in head tank.	a. Set point on level controller on head is set too low. Raise set point. b. Level control valve stuck open. Stroke valve. h. Seal oil pump failure. Start stand by pump. i. Excessive seal leakage. Flow out dump line from one or both drain pots is high – traps are cycling time frequent – possible seal ring wear - Inspect seal and repair. j. Filter is plugged. High pressure drop indicated. Switch and change filter elements in first filter.
2. High level in head tank	a. Set point on level controller is set too high. Lower set point. b. Level control valve is stuck closed or plugged. Stroke valve. Repair Valve if it will not stroke.
3. High seal oil temperature	a. Heater in oil tank is on. Turn it off. [mainly for separately seal oil systems] b. Oil cooler is fouled. Check inlet and outlet temperature of oil and cooling Fluid, and check presure drop of the fluids [tube type cooler]. Fouling is in The fluid with high pressure drop. Clean cooler. c. Cooler fans problem. Check operation. Check vibration. Lubricate or repair / replace as needed.

DRY GAS SEALS

Most problems are caused when liquids enter the system and/or the primary ring is “cocked” in its orientation to the mating ring, especially, during start up phases. Forces normally applied to lift the primary ring from the mating ring when improperly seated can produce potentially high loads on the primary ring and damage the seal.

Recommended Practices:

1. Maintain the integrity of the dry gas seal system at all times through good preventive maintenance of the filters and flow controllers and make certain there is no ingress of oil from the bearing area and proper operation of a barrier seal, if installed, to restrict liquids form entering the seal and adversely affecting operation of the carbon elements of the seal.

2. Make sure that vents are free flowing and flowing the correct amount of gas.

3. Dry gas seals must be re-installed to rotate in the right direction. Seal assemblies are usually marked with a directional rotation arrow. Install to ensure proper direction or rotation. Failure to install the seal and ensure proper rotation can result in ineffective sealing and destruction of the seal during start up and/or operation.

4. Dry gas seals require correct axial position in the compressor. Failure to ensure correct axial position can, and, most likely, will result in destruction of the seal.

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