

ESTIMATED TIME TABLE
PERTAMINA-UOP-VENDOR ROTATING EQUIPMENT TRAINING, 23-27 JUNE 2003

No	Time	Monday 23 June 2003	Tuesday 24 June 2003	Wednesday 25 June 2003	Thursday 26 June 2003	Friday 27 June 2003
		Pumps	Pumps (continued)	Reciprocating Compressor (Cont'd.)	Centrifugal Compressor	Centrifugal Compressor
1	08. ⁰⁰ – 08. ⁴⁵	Opening Remarks <i>UOP</i>	Pump Wrapup <i>UOP</i>	Trouble Shooting & Repair <i>UOP/DR</i>	Theory Rotodynamics <i>UOP/DR</i>	Remaining life assessment <i>DR</i>
2	08. ⁴⁵ – 09. ³⁰	Pump Curves NPSH <i>UOP</i>	Pump Alignment Demo <i>PT. Putranata</i>	Lube Oil Systems <i>UOP</i>	Control and control systems <i>UOP</i>	Drivers (steam turbine, motor) <i>UOP/DR</i>
	09. ³⁰ – 09. ⁴⁵	COFFEE BREAK				
3	09. ⁴⁵ – 10. ³⁰	BEP Mechanical Seals & Sealing Systems" Reliability and condition monitoring <i>UOP</i>	Pump Alignment Demo <i>PT. Putranata</i>	Preparation of Maintenance and Inspection manual <i>UOP/DR</i>	Surge Control Dry Gas Seals <i>UOP</i>	Turn around maintenance <i>DR</i>
4	10. ³⁰ – 11. ¹⁵	System Hydraulics Pump Control Fan Laws <i>UOP</i>	Pump Alignment Demo <i>PT. Putranata</i>	Turnaround Maintenance <i>DR</i>	Bearings and lube oil consoles Condition Monitoring <i>UOP</i>	Engineering software for design <i>DR</i>
5	11. ¹⁵ – 12. ⁰⁰	Bearings & Lubrication Systems Oil Mist <i>UOP</i>	Condition Monitoring Demo <i>PT. Putranata</i>	Case Studies <i>UOP/DR</i>	Review Vendor Bid Proposals Trouble shooting and repair <i>UOP/DR</i>	Maintenance & Instruction Manuals <i>DR</i>
	12. ⁰⁰ – 13. ⁰⁰	LUNCH				

			Reciprocating Compressor			
6	13. ⁰⁰ – 13. ⁴⁵	Multi-stage Double Suction & Sundyne Pumps UOP	Theory UOP	Case Studies UOP/DR	Anti Surge Control Demo CCC	Case Studies UOP/DR
7	13. ⁴⁵ – 14. ³⁰	Trouble Shooting & Repair Pump Vendor Selection & Procurement UOP	Capacity Control Design limitations UOP	Revamp considerations DR	Anti Surge Control Demo CCC	Case Studies UOP/DR
	14. ³⁰ – 14. ⁴⁵	COFFEE BREAK				
8	14. ⁴⁵ – 15. ³⁰	Standards Proportioning pump Slurry Pumps UOP	Valves UOP	Reciprocating Comp. Monitoring Demo & Discussion DR	Anti Surge Control Demo CCC	Wrap-up UOP/DR
9	15. ³⁰ – 16. ¹⁵	Revamp Considerations UOP	Maintenance and Reliability UOP	Reciprocating Comp. Monitoring Demo & Discussion DR	Recip vs Centrifugal case study UOP	Closing & Certificate

Note: The timeframes associated with each topic need to remain flexible as depending on the questions of the participants more/less time may be needed to cover certain areas.



Rotating Equipment Training

PUMPS

Presentation Material

Pertamina

June 23-27, 2003

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Training Services

Pumps

EDSPO-1

Pumps

- Introduction

- Pump Curves

- Head versus Capacity
- NPSH
- Efficiency

EDSPD-2

Pumps ***(continued)***

■ **Single-Stage Centrifugal Pump Design**

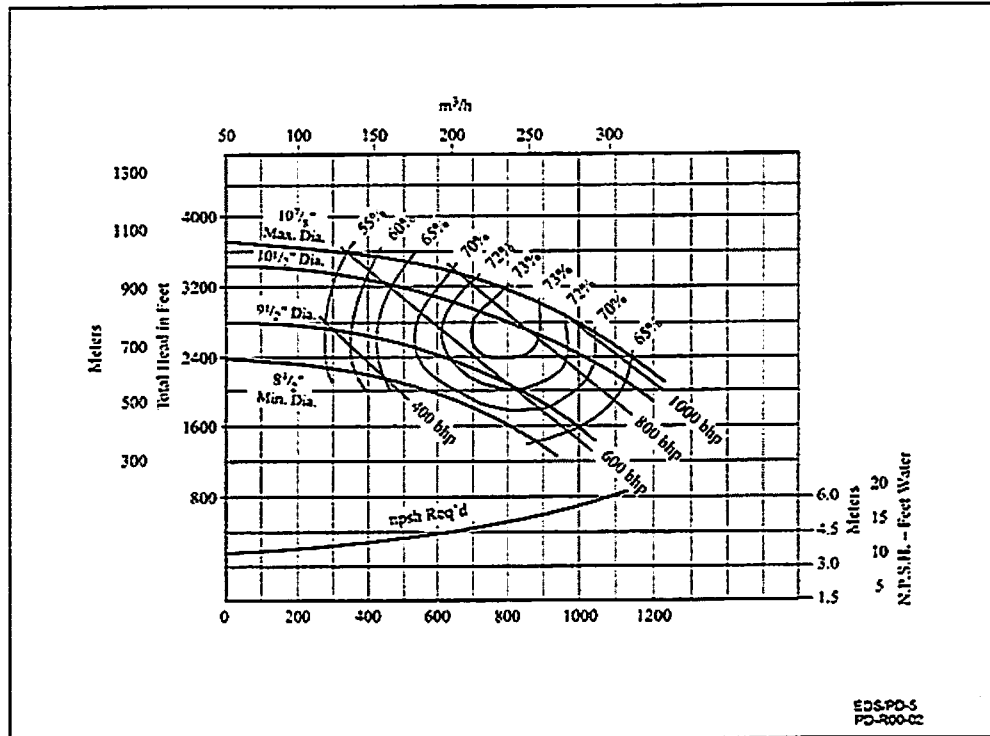
- **Pump Components**
- **Impellers**
- **Axial Thrust**
- **Shaft Sealing**
- **Cooling and Flushing**
- **Bearings**
- **Couplings**

EDSPO-3

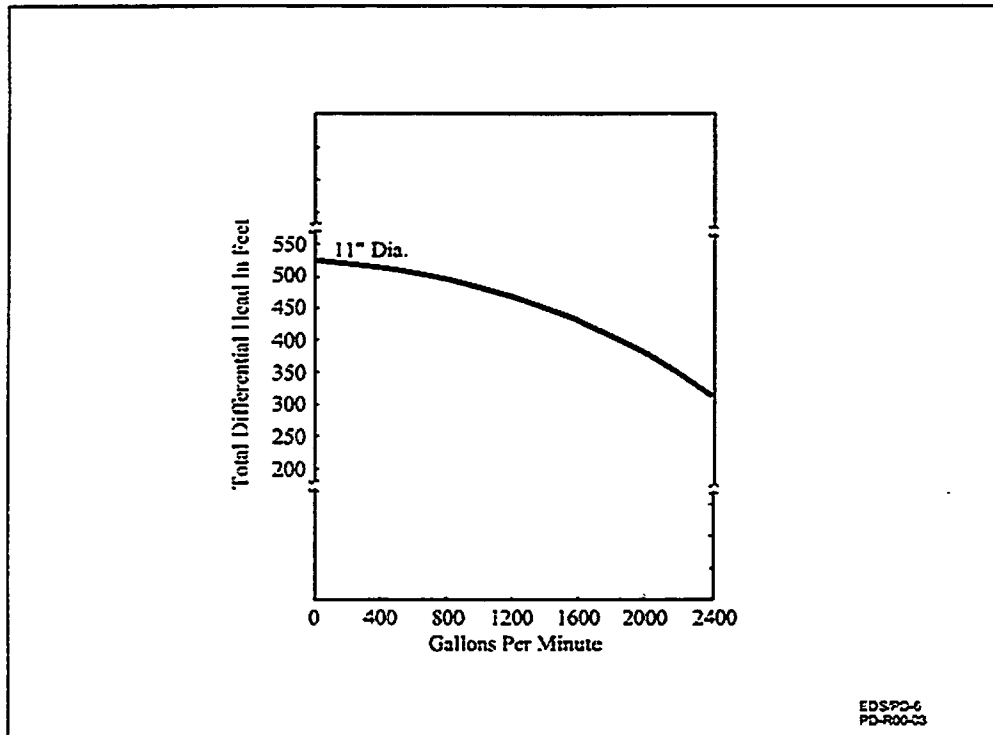
Pumps ***(continued)***

- ▣ **Sealless Pumps**
- ▣ **Pump Selection and Performance**
- ▣ **Double Suction, Multi-Stage, and Sundyne Pumps**
- ▣ **Hydraulics**
- ▣ **Fan Laws**
- ▣ **Reliability**
- ▣ **Pump Control**
- ▣ **Standards**
- ▣ **Other Pumps**

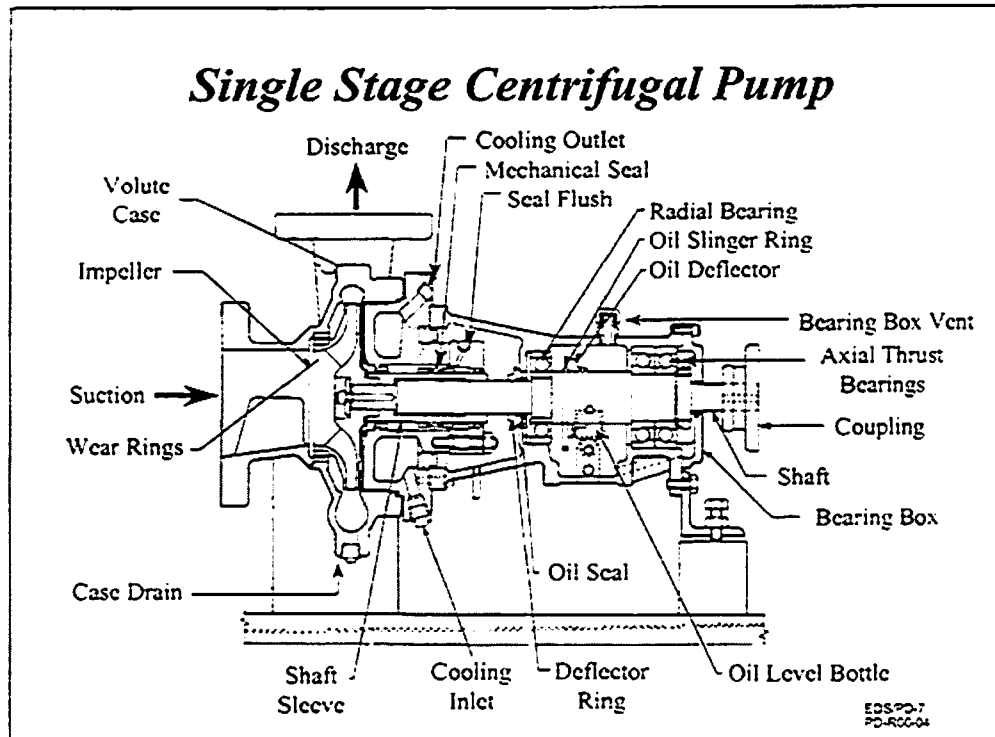
EDS-PO-4



This is a typical pump curve. The pump curve gives information on how the pump will perform, the NPSH required by the pump, and the impeller size range for the casing. All pump manufacturer's curves are similar so, if you can read one manufacturer's curve, you can read anybody's.

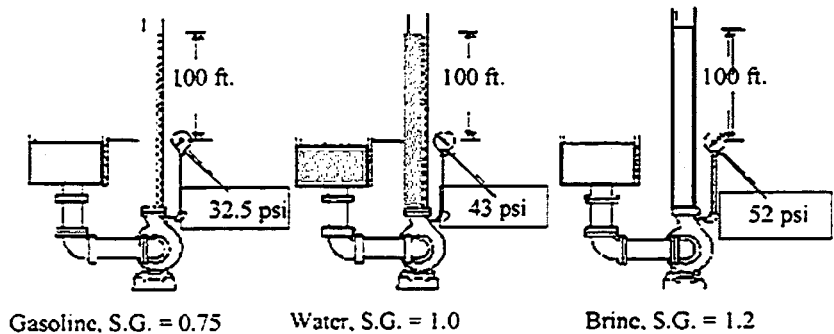


Head-capacity curve. Once this curve is established based on the impeller diameter and speed, the pump will always operate on this curve. Note how the curve rises as the flow goes down. This is a characteristic of all centrifugal pumps.



Single stage centrifugal pump. As the centrifugal force of the impeller throws the fluid out towards the casing, the velocity of the fluid goes up. As the fluid leaves the pump, this velocity energy is changed to pressure energy.

Identical Pumps Handling Liquids of Different Specific Gravity



$$\frac{\text{Disch.}}{\text{Pres.}} = \frac{100' \times 0.75}{2.31} = 32.5 \text{ psi}$$

$$\frac{\text{Disch.}}{\text{Pres.}} = \frac{100' \times 1.0}{2.31} = 43 \text{ psi}$$

$$\frac{\text{Disch.}}{\text{Pres.}} = \frac{100' \times 1.2}{2.31} = 52 \text{ psi}$$

same head.

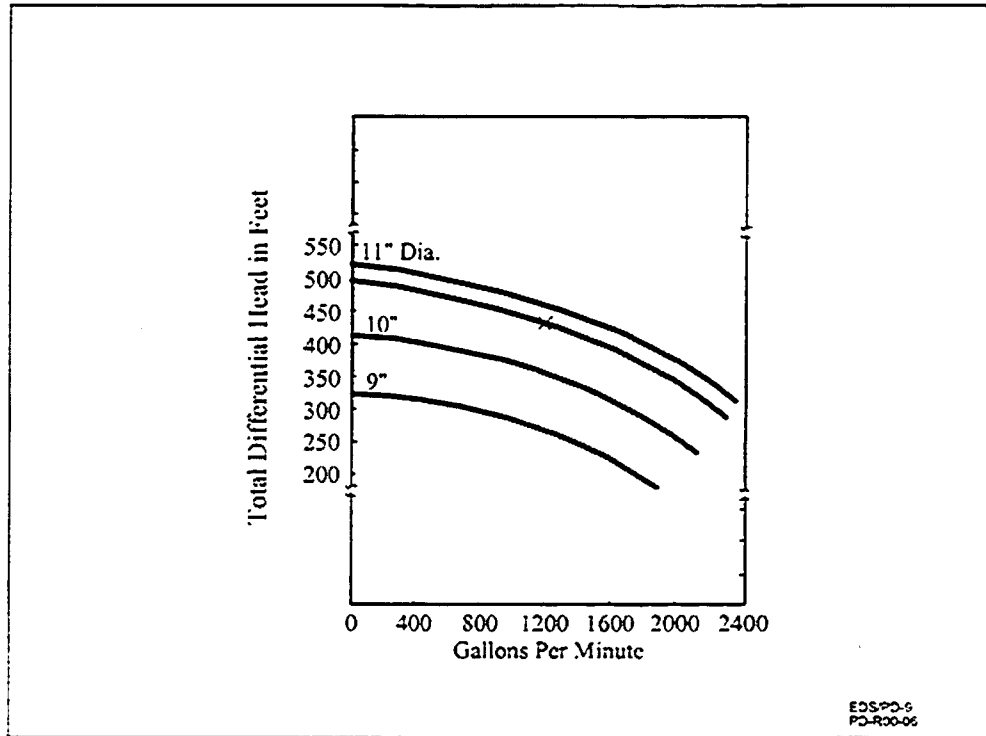
EDSPO-8
PD-R00-05

Pump performance is measured in feet or meters of head. Head is the height of the column that the pump can move the fluid. Pump head is a function of impeller diameter and speed. It is not a function of the density or specific gravity of the pumped fluid. Here are three identical pumps pumping out of three identical tanks. Note that the head or column height is identical even though the specific gravity of the fluid is different.

curve → flow

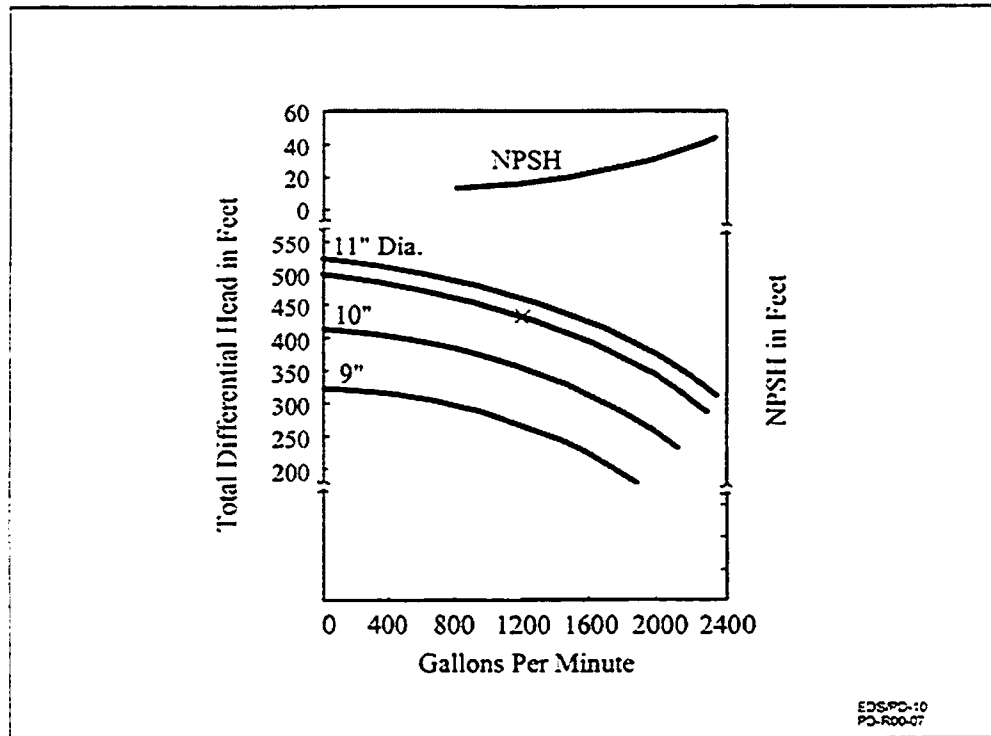
→ pressure

Head = f (flow, speed)

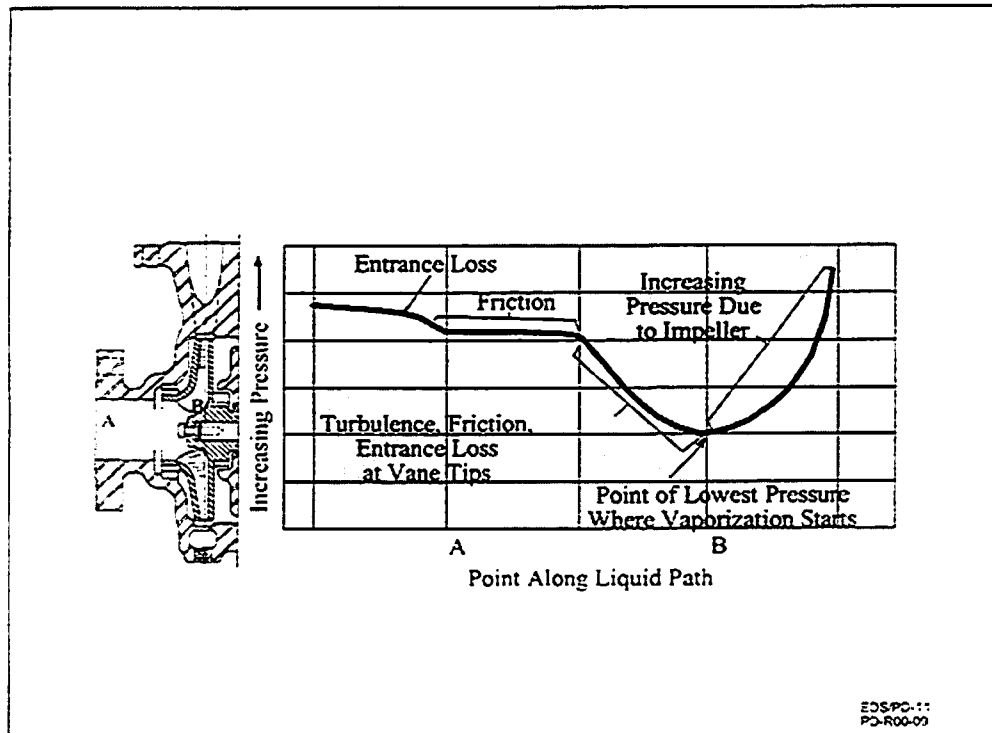


Each pump casing size can handle more than one size impeller. This pump casing can handle impeller diameters between 9 and 11 inches. Also, the impeller can be trimmed to any size between 9 and 11 inches to meet the rated operating point. The impeller diameter does not have to be a whole inch size.

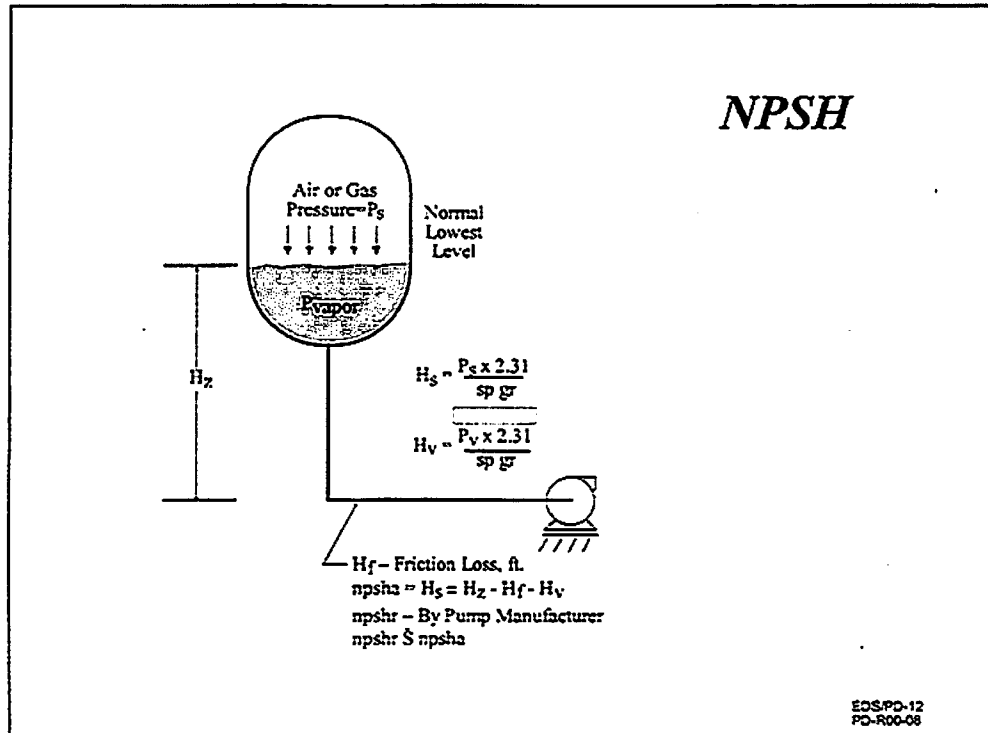
Trimming to get right



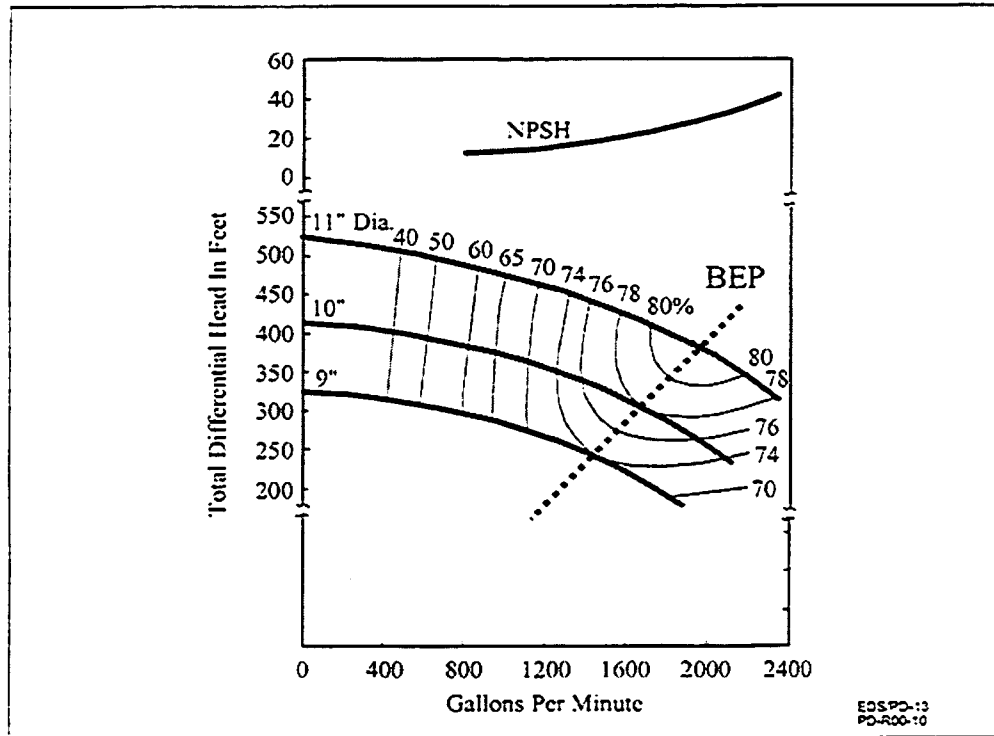
The pump curve also gives the NPSH required by the pump. Note how the NPSH required curve rises with increasing flow.



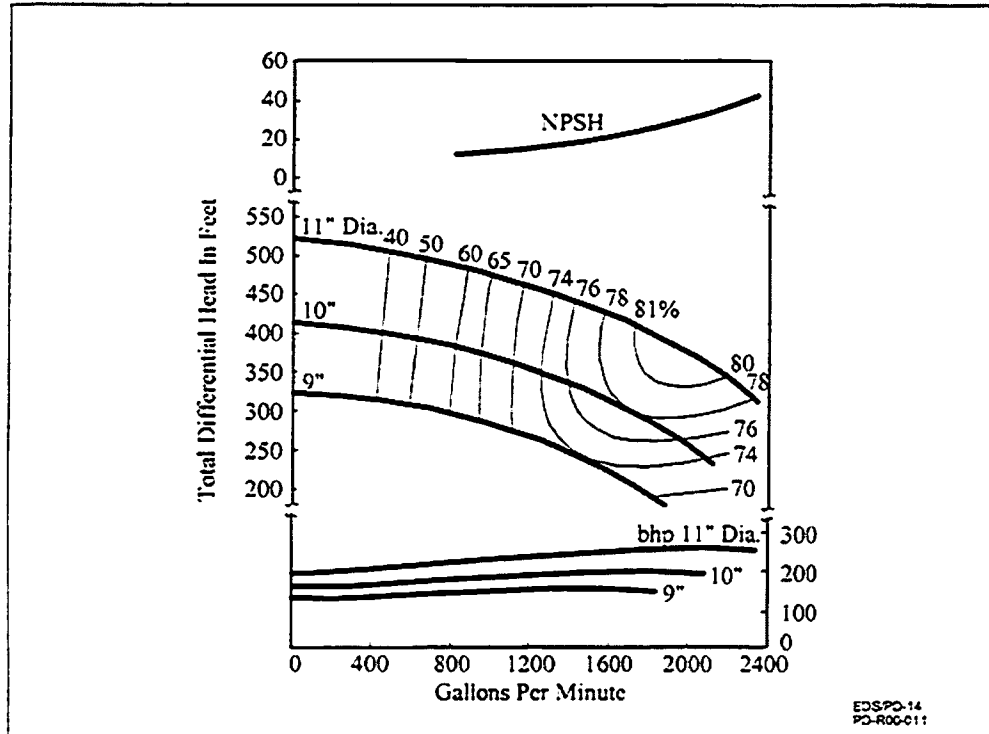
The fluid loses pressure in the pump before the pressure starts to rise. As the fluid enters the pump, there are entrance and friction losses. As the fluid enters the rotating impeller, there are turbulence and friction losses at the vane tips. If this pressure drop is enough to drop the pressure of the fluid below its vapor pressure point, flashing will occur. This phenomena, called cavitation, will quickly destroy an impeller and a pump. The NPSH available must be greater than the NPSH required.



The NPSH available is a function of the pumping system. NPSH available is the pressure at the pump suction minus the fluid vapor pressure. It is the pressure that can be lost in the pump inlet area before flashing or cavitation begins. For a bubble point or vapor pressure point fluid, the NPSH available is gained with vessel elevation.

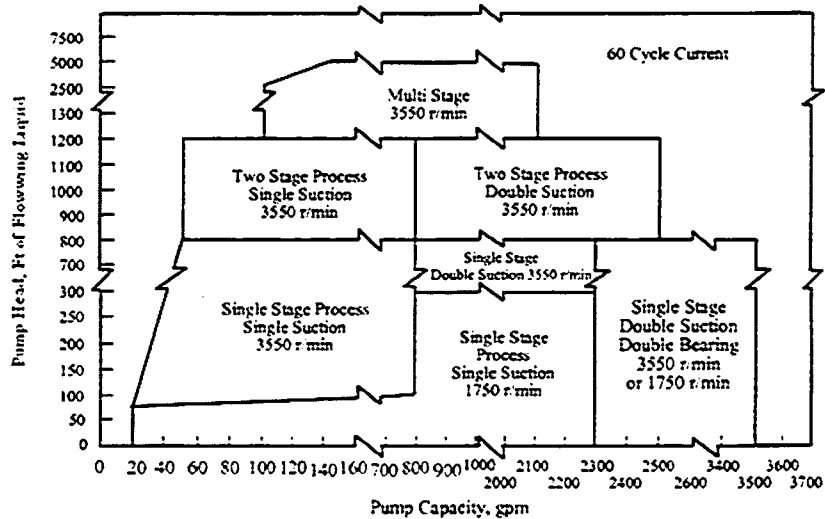


The pump curve shows the efficiency of the pump at any operating point. Note that the efficiencies rise with rising flow to the best efficiency point (BEP), and then quickly drop off. Optimum pump operation is at or near best efficiency point.



Pump curves also show the HP requirement for the pump. Do not use these curves. CALCULATE HP. These HP curves only apply if the specific gravity of the fluid is 1.0. Also, it is difficult to get a good, accurate reading. HP is a simple calculation shown later in this talk.

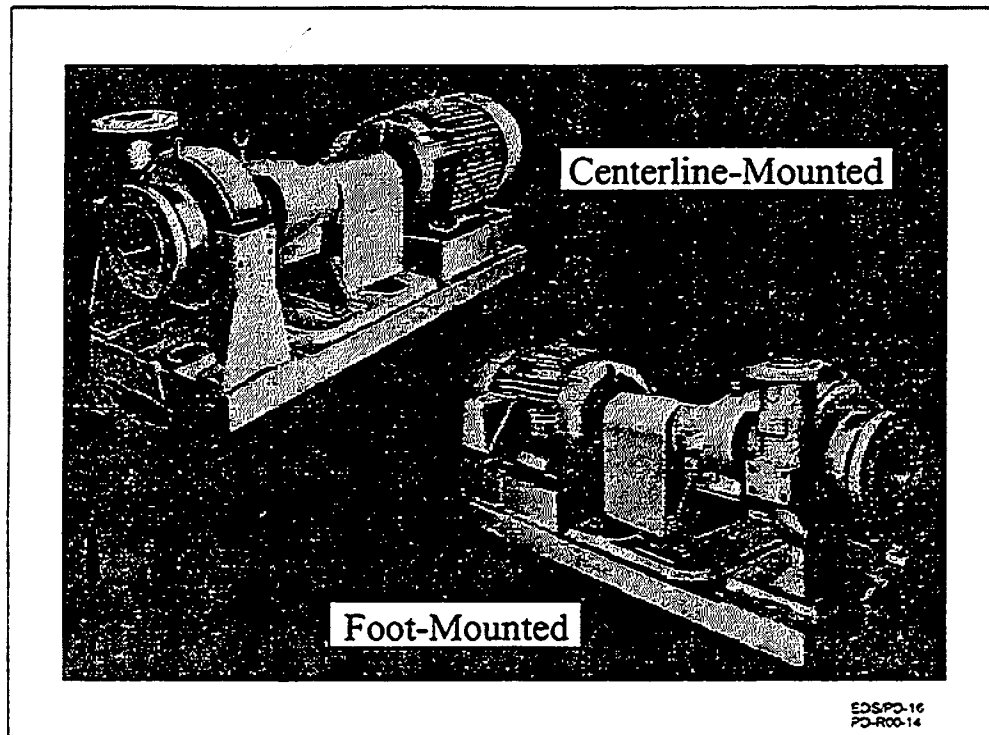
Pump Selection



ESS-PO-15
PD-R00-12

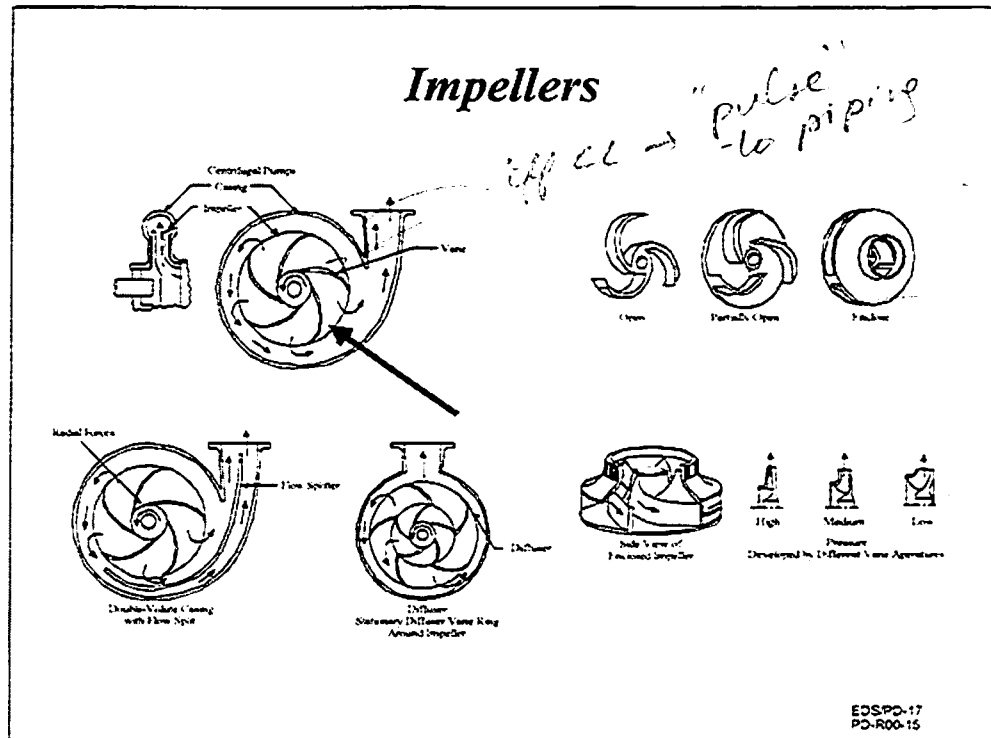
This chart shows the approximate head-capacity ranges of single stage full and half speed pumps, double suction pumps, and two and multi-stage pumps. Low flow, high head applications are Sundynes.

most



All API pumps today are centerline mounted. The centerline mount allows the pump casing to grow both up and down as the casing heats up. This keeps the shaft in the horizontal plane and helps prevent seal leaks and shaft misalignment.

low flow → hi force



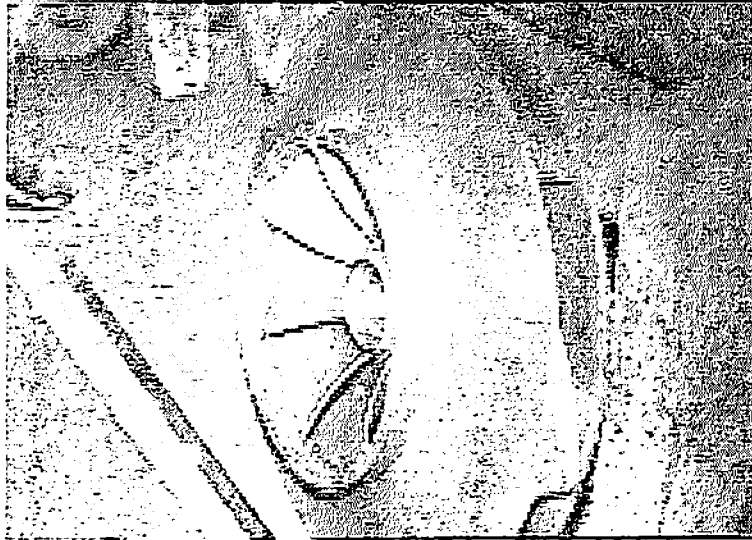
All API pumps today have closed impellers with covers or shrouds on both sides of the vanes. This gives the fluid a more defined path through the pump and raises efficiency. The flow splitter in the outlet or double volute equalizes the radial forces around the impeller and minimizes the load on the radial bearings.

Single Suction Enclosed Impeller

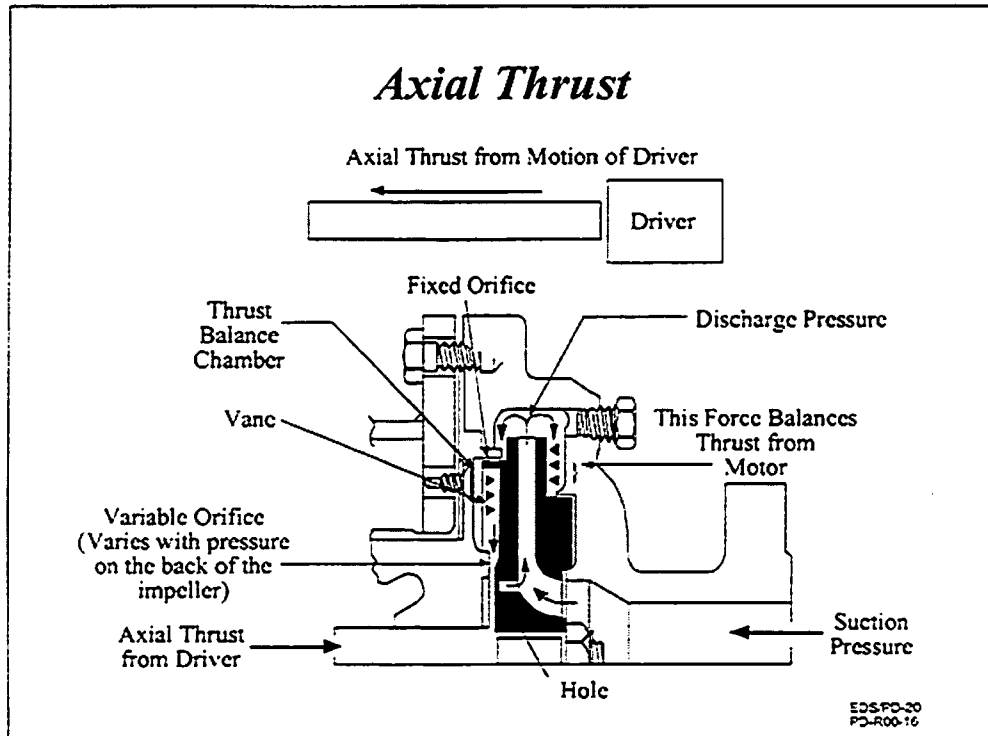


Single suction enclosed impeller.

Single Suction Impeller



Large single suction impeller. Note the impeller vanes at the inlet and outlet. This is a half speed impeller. Full speed impellers are only allowed up to 15 inches in diameter to control tip speeds.



The motor puts an axial thrust on the impeller. This is offset somewhat by the suction pressure acting in the opposite direction. To put the impeller in axial balance, some process fluid is allowed to flow across an orifice from the back side of the impeller. Some small holes are drilled in the back cover of the impeller to equalize the pressure on either side. This minimizes the force on the thrust bearings.

Impeller Specific Speed

$$N_s = \text{rpm (gpm)}^{1/2} / (H)^{3/4}$$

- Below 4200 - centrifugal or radial
- Between 4200 - 9000 mixed flow
- Above 9000 - axial flow

ECSPO-21

Impeller specific speed is an index of hydraulic design defined as the speed at which an impeller, geometrically similar to the one under consideration, would run if it were reduced in size to deliver one gpm at one foot of head. This is a useful tool for the hydraulic designer in designing impellers to meet varying conditions of head, capacity, shape of the curve, suction conditions, and speed. The specific speed helps define the fluid path through the impeller, if there is a sharp right angle, centrifugal, or a very gradual right angle, mixed flow.

Suction Specific Speed

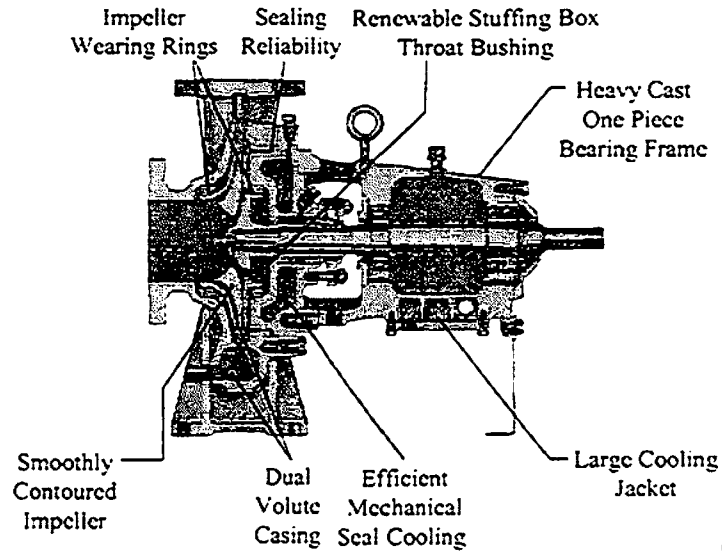
$$S = \text{rpm (gpm)}^{1/2} / (\text{NPSH})^{3/4}$$

- Can range between 3000 - 20000
- UOP limits to 11000

EOS-PO-22

The suction specific speed relates rpm, gpm, and NPSH required. UOP limits the suction specific speed to 11000. If a pump manufacturer wants to reduce the NPSH required of a certain pump, he can increase the impeller eye area to reduce friction drop and reduce NPSH required. This increased eye area increases the internal circulation in the suction area of the pump. This can build up heat which can also flash the fluid and reduce pump reliability. This also reduces the stable operating range of the pump. As the flow is reduced, the pump becomes less efficient and more heat is built up in the pump. At higher suction, specific speeds this can promote cavitation.

***Model 3735 High Temperature/High Pressure Process Pumps
Heavy Duty Design Features to Meet the Total Range
of Process Industries***



EDSPO-22
PO-R00-13

This is a single stage (one impeller), single suction (one entry into the impeller), overhung (impeller is cantilevered on one set of bearings) pump. This is called a Process pump. The metallurgy is as follows:

Casing	Carbon Steel
Impeller	Carbon Steel <500°F, 11-13% Cr >500°F
Shaft	Carbon Steel
Wearing rings	11-13% Cr → 8 plastic (in Comp. v. lve)
Throat bushing	11-13% Cr
Throttle bushing	Bronze or non-sparking material

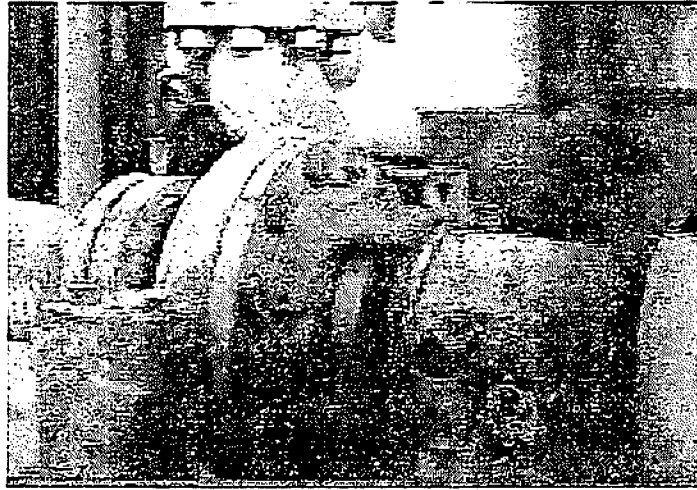
Single Stage Overhung Pump



EDSPO-24

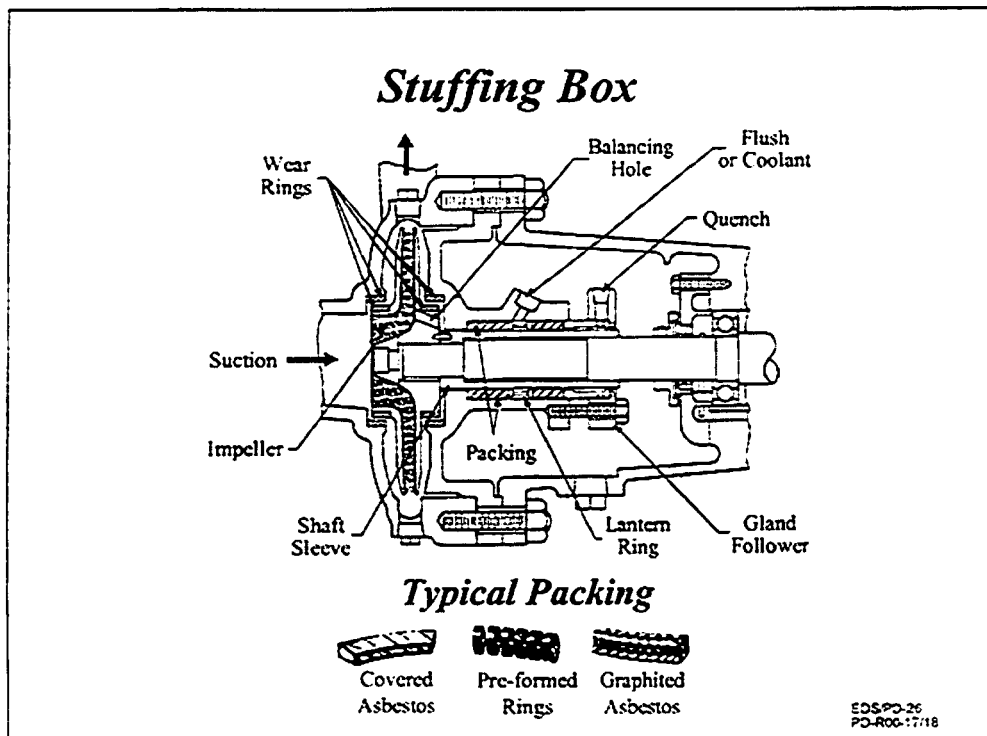
Single stage, single suction, overhung pump. Note the vent connection on the top of the casing.

Single Stage Pump



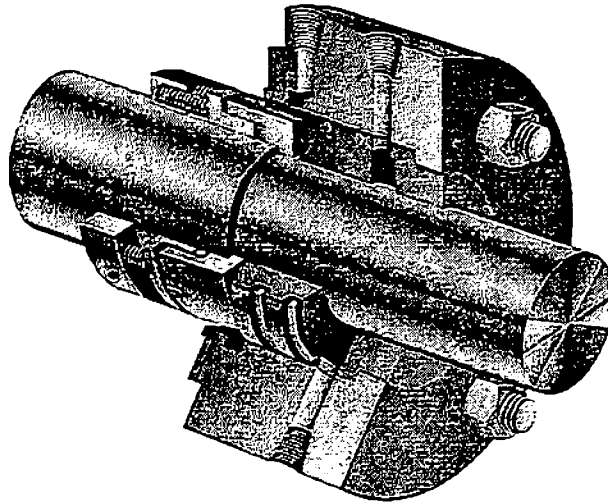
EDS-PQ-25

Single stage, single suction, overhung pump. This pump is self-venting as the discharge is at the high point of the casing. This design is typical.



Before there were mechanical seals, pumps were sealed by “stuffing” an absorbent material called packing around the shaft. Since the process fluid had to lubricate the surface between the stuffing and the shaft, the packing had to leak, typically about 200 cc/hr for a new application. Over time, the packing would become saturated with fluid and the leakage would increase until the pump had to be shut down and the packing replaced. Today, UOP does not specify any pumps with packing.

Single Mechanical Seal

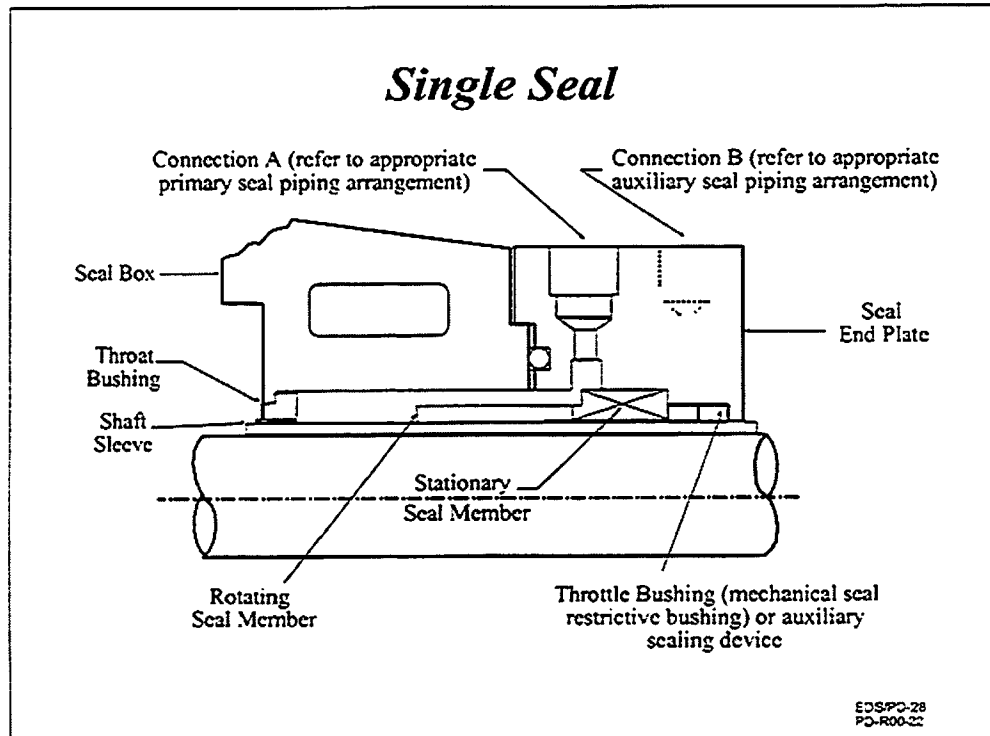


EDSPO-27

Single mechanical seal. Most API pumps today have single mechanical seals. The single mechanical pusher type seal has two members, a rotating member and a stationary member. The main sealing takes place due to the friction between the rotating seal face and the stationary seal face. Since the pumped fluid lubricated this seal face, the single mechanical seal does leak. Typical leak rates are about 2 cc/hr or about 100 ppm of emissions in the air surrounding the seal. As the seal faces wear, springs in the rotating member keep a tight fit between the two seal faces. O-rings prevent leakage between the seal and the shaft and between the seal and the pump casing.

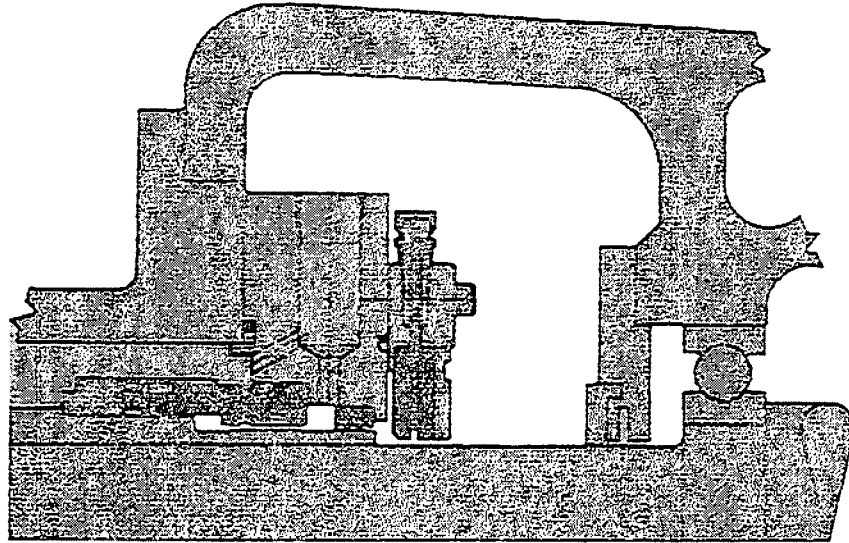
2cc/hr

90° seal not worn out



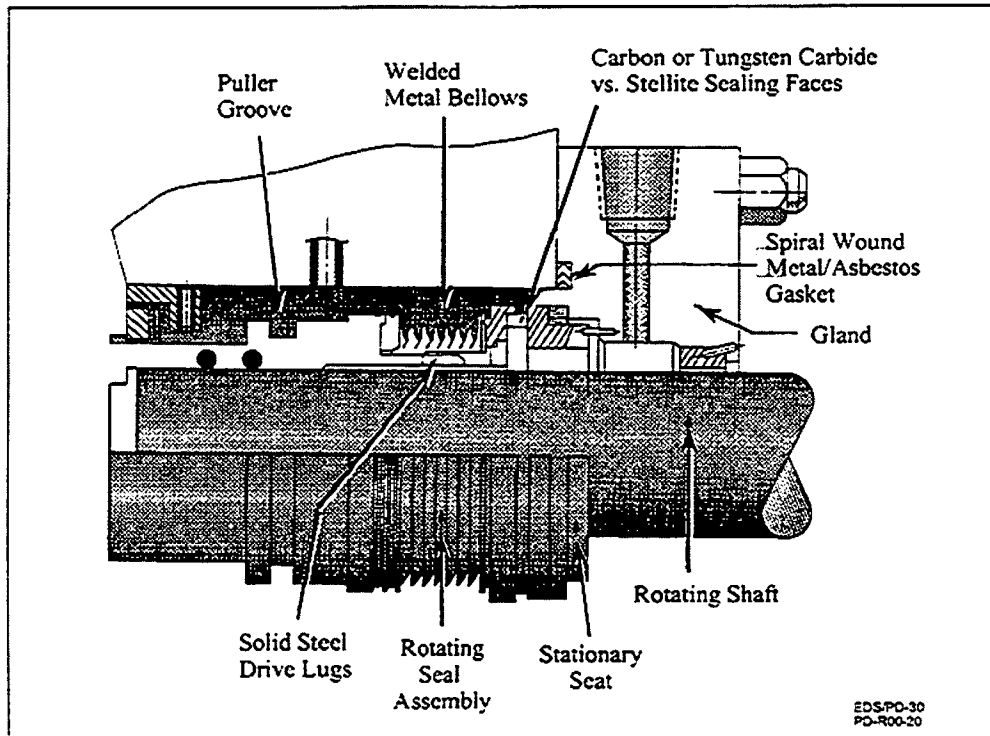
Single mechanical seal. Used for most non-hazardous services.

Single Mechanical Seal

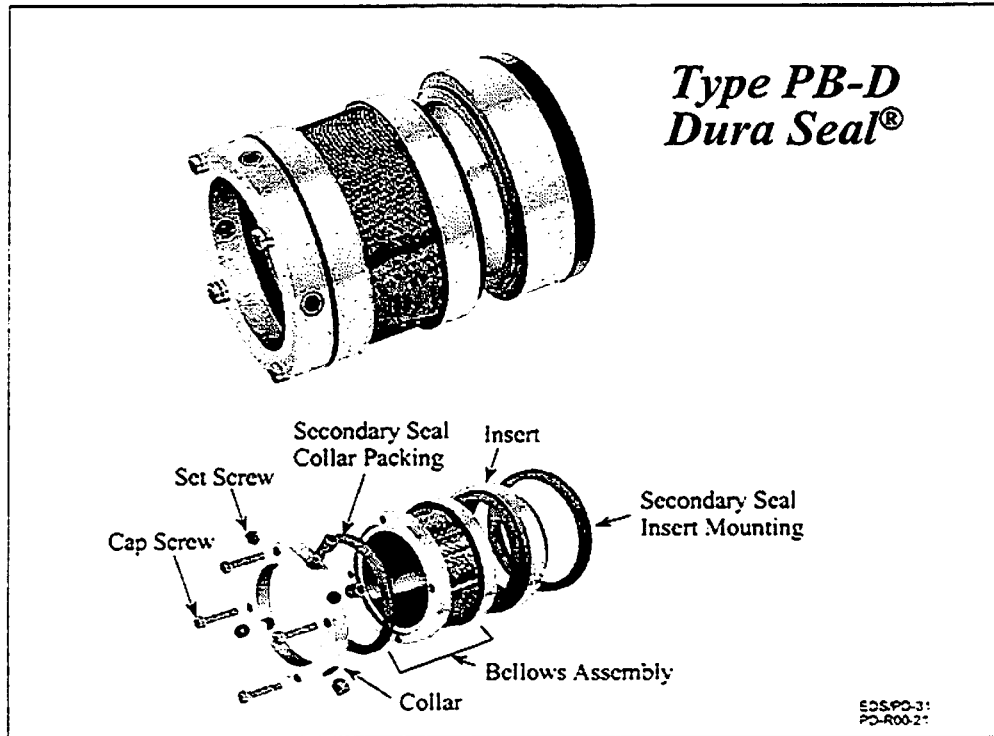


ECS-PO-25
PO-R00-19

Here is another view of the single mechanical seal. Note the yellow process fluid coming from the pump discharge to the process side seal face. The rubbing seal faces generate heat. If the pumped fluid is at vapor pressure or bubble point and heat is added, the fluid could flash around the seal and the seal faces could lose their lubricant. Process fluid flows from the discharge of the pump through an orifice. The pressure is kept high enough around the seal to stay above the vapor pressure point even though with the seal faces are adding heat.



Bellows seals are specified for high temperature applications, above 550°F. Bellows seals have two members, a rotating member and a stationary member, similar to the pusher type seal.



When the seal face wears on a bellows seal, the metal bellows expands like an accordion. The o-rings between the seal and the shaft do not move along the shaft as they do in a pusher type seal. Since the ring material starts to break down at higher temperatures, pusher type seals are temperature limited due to the dynamic o-ring. Since the o-ring on the bellows seal is static, the bellows seal can operate effectively at temperatures up to 800°F.

Single Seals

- <1000 ppm (Most <100 ppm)
- Comply with Regulations in Most Cases
- 3+ Years Life

EDSPD-32

5~7 year MTBR.

Liquid Tandem Seals ***(Unpressurized Dual Seals)***

- <50 ppm (Most <10 ppm)
- Vent to Flare
- Diesel Buffer Liquid
- 3+ Years Life

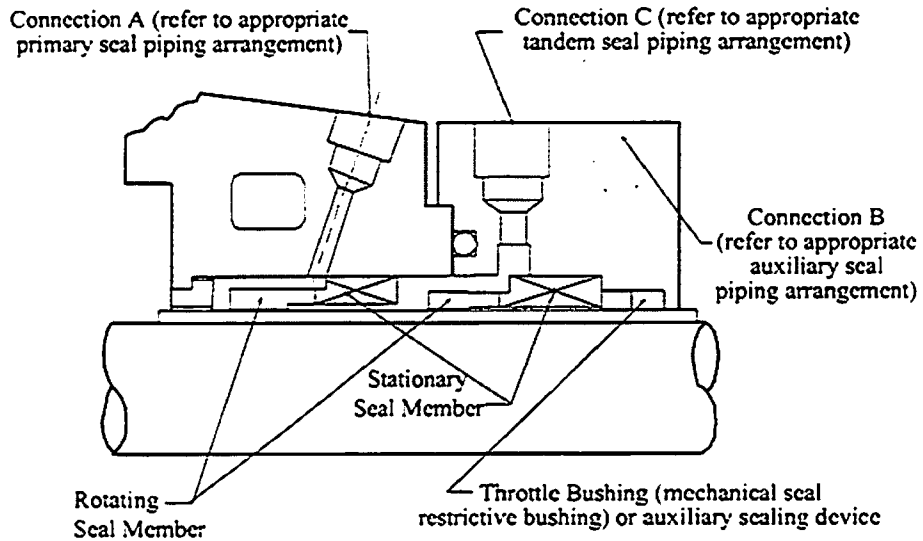
EOS-PO-33

Tandem seals are now referred to as unpressurized dual seals. The buffer between the two seals is vented to flare and is unpressurized. Leakage of process fluid is greatly reduced from the single mechanical seal. Any process fluid that leaks across the inner seal is contained by the outer seal. Release of process fluid to the atmosphere is under 10 ppm.

→ light H-C

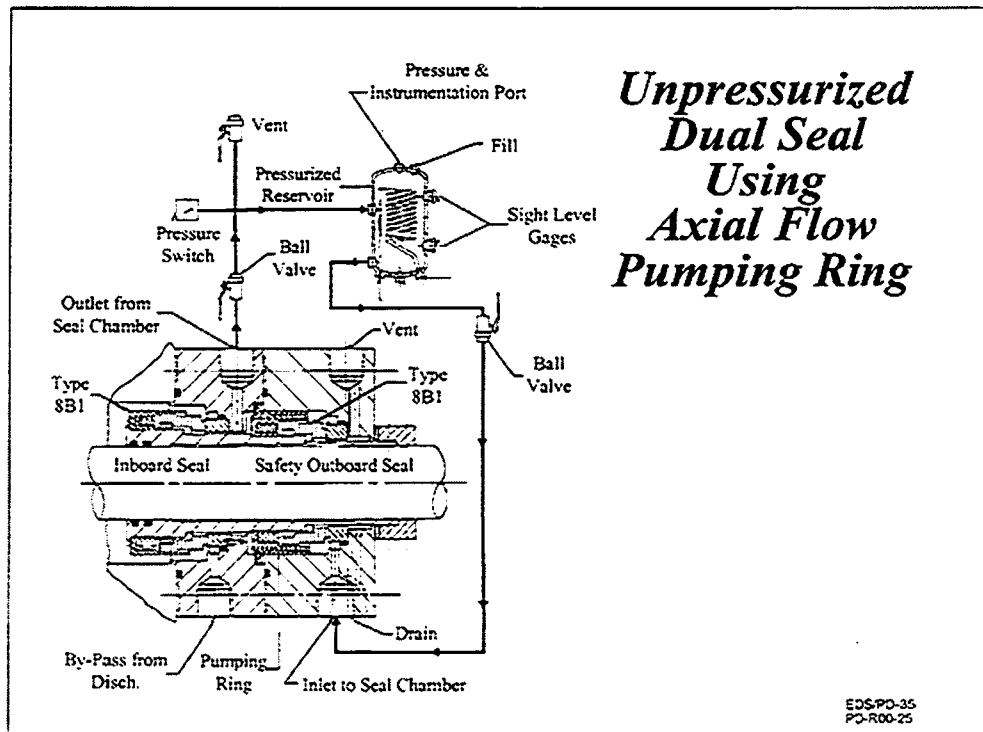
milb air → sampling monthly

Unpressurized Dual Seals

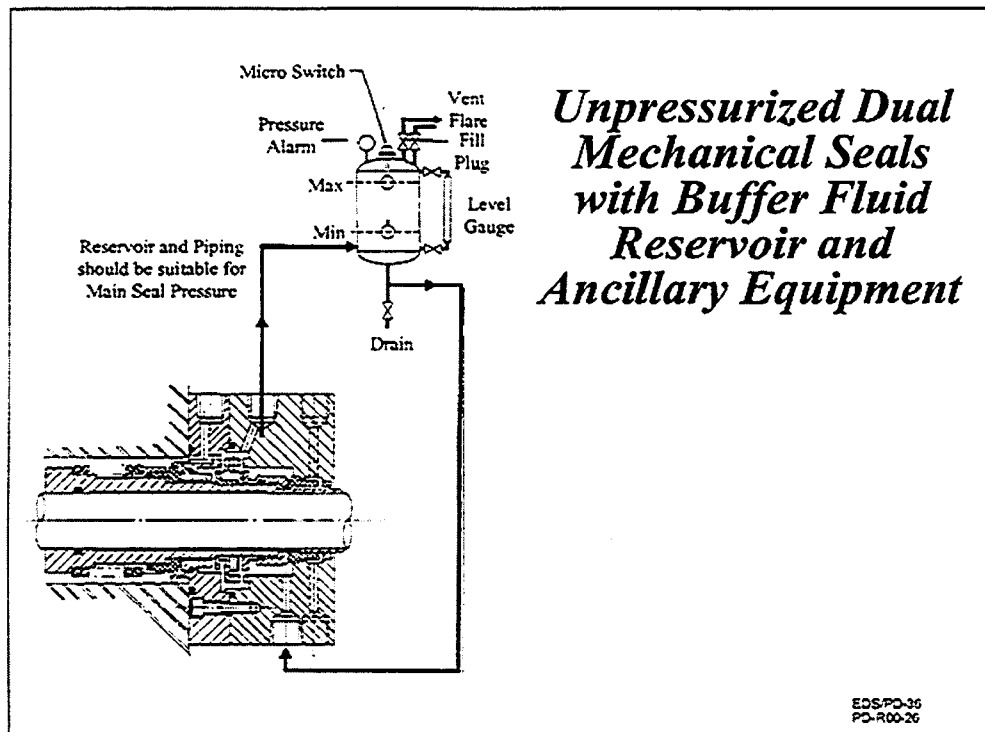


Unpressurized dual mechanical seal. Used for following:

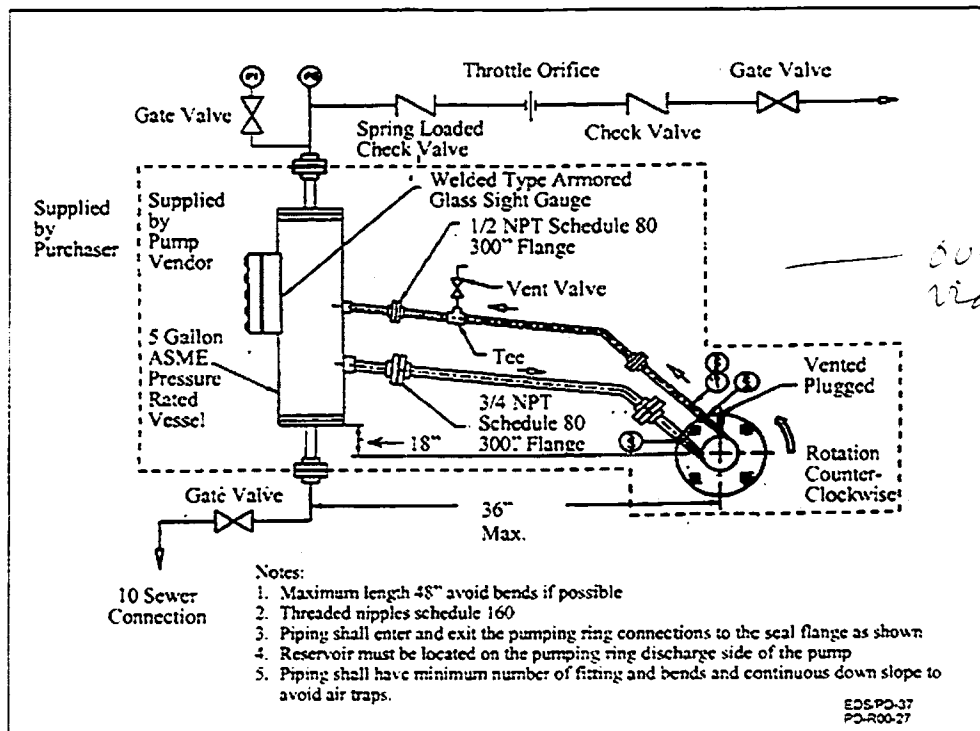
- Light hydrocarbons - Vapor pressure over 30 psig
- 1 wt % Benzene
- 25 wt % C₆-C₉ aromatics
- 5 mol% H₂S
- Other environmentally hazardous materials



Buffer fluid is circulated from the seal pot to the buffer area and back to the seal pot with pumping rings on the shaft. Leakage of process fluid is into the buffer area. The seal pot has a pressure alarm for flashing fluids and a level alarm for non-flashing fluids to warn of an inner seal leak.

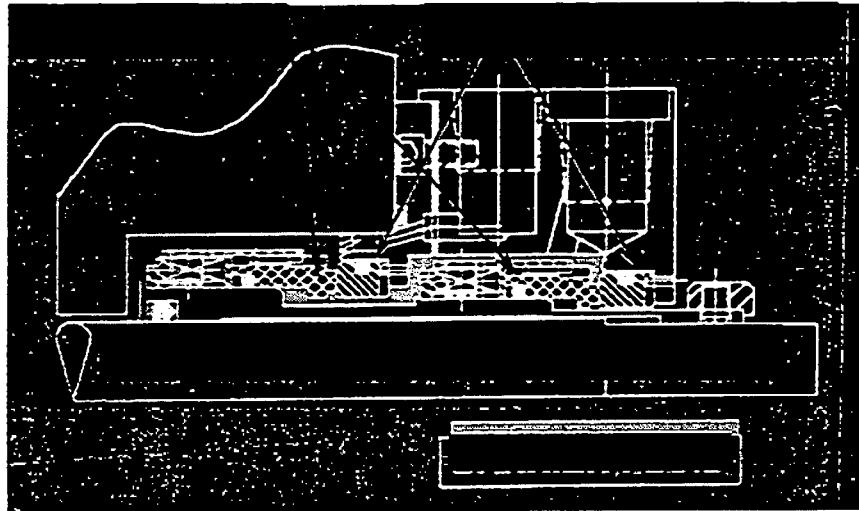


Note that the seal pot is vented to flare. Buffer is at a lower pressure than the process so leakage is into the barrier. No buffer fluid enters the process. If the pump is double bearing or multi-stage with two sets of seals, there are two separate seal pots.



API Piping Plan 52. If the pumped fluid is hazardous and non-flashing, the drain is piped to a closed drain header.

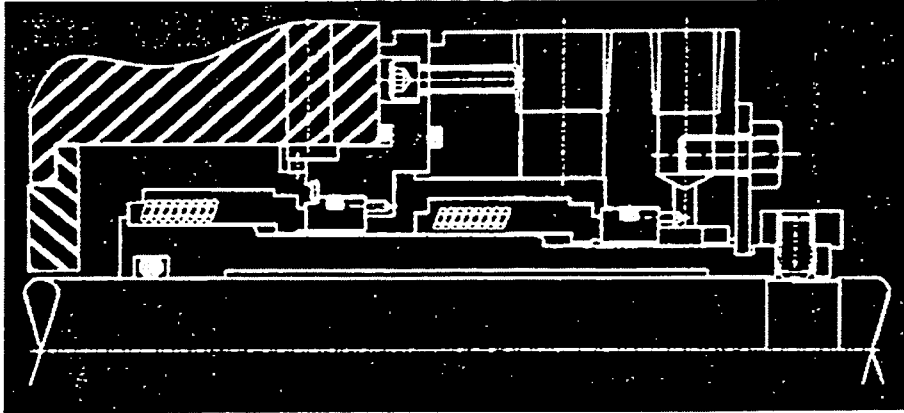
Tandem Mechanical Seal



EDS-PC-38
PO-R00-28

Note the orange buffer fluid between the seal faces. Also note the O-ring (above and to the left of the "P" in Primary to prevent leakage between the shaft and the seal. The O-ring above the "e" in Primary Seal prevent leakage of process fluid between the seal and the pump casing.

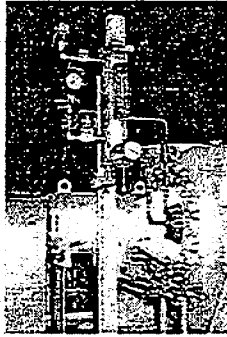
***Arrangement 2: Standard Dual Seal
Buffer Pressure < Process Pressure***



EDSPO-39
PD-R00-29

Unpressurized Dual Seals. Buffer fluid pressure (between the seal faces) is less than process pressure. No buffer fluid will leak into the process.

Series 6000 Slurry Pumps Seals and API Support Piping Plans

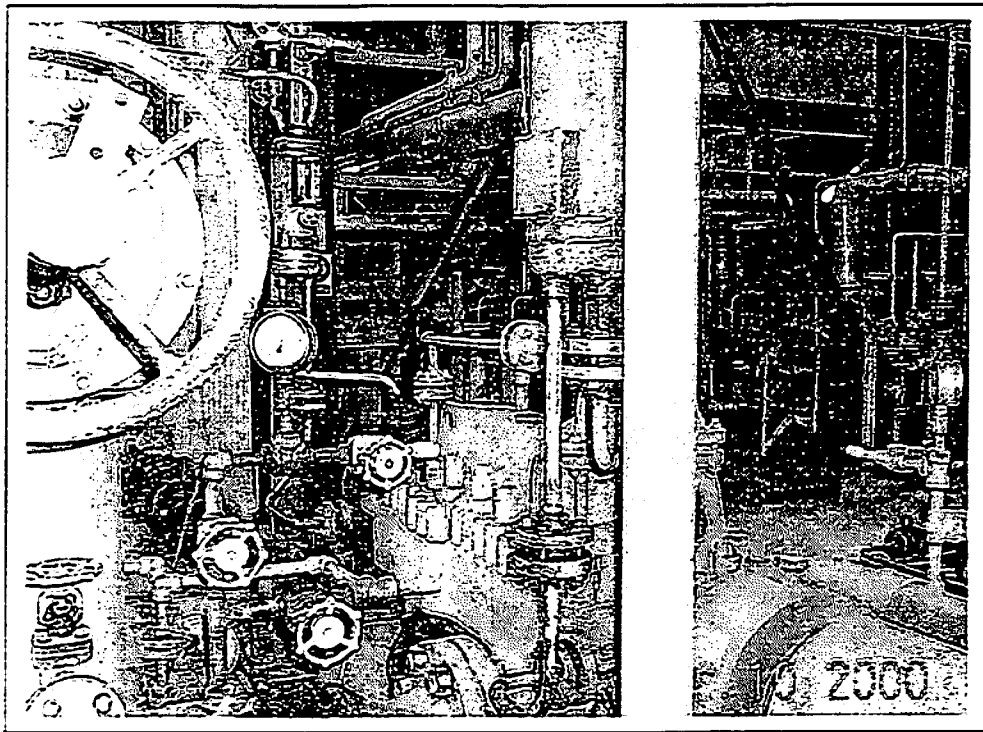


Each seal system is carefully analyzed to ensure long life under both normal and upset operating conditions.

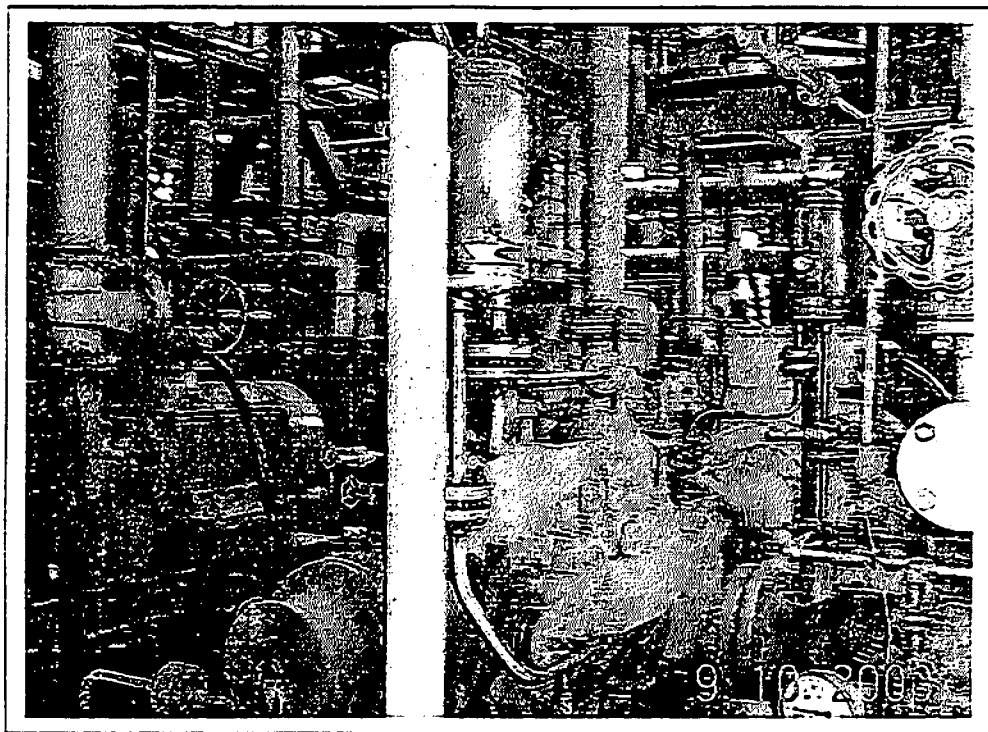
**LAWRENCE
PUMPS INC.**

EDS-PO-40

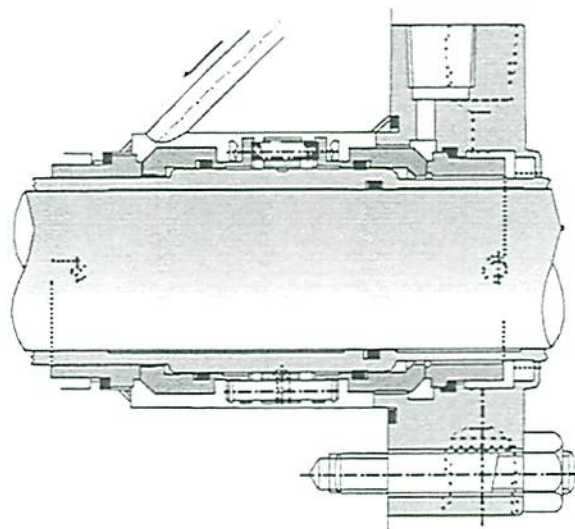
Photograph of actual API Plan 52.



Multi-stage pump with unpressurized dual seals. Note the two seal pots the the API Plan 52.



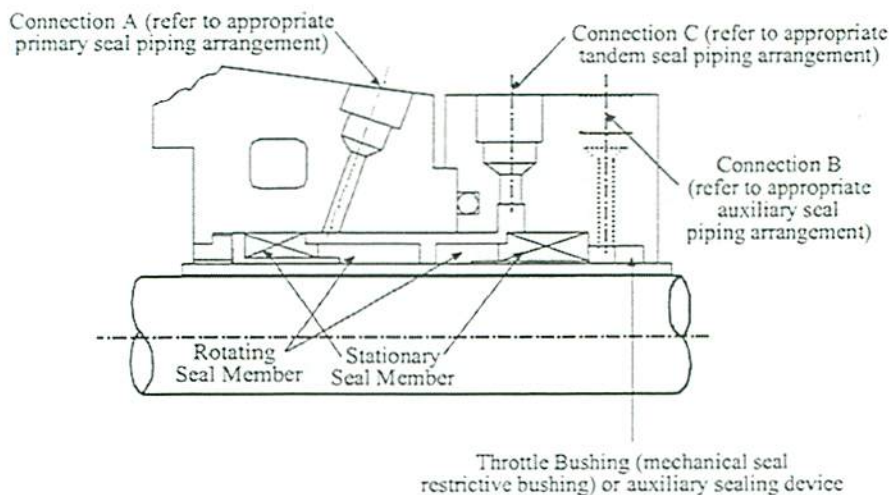
Pressurized Dual Seal



EDS-PO-43
PO-R00-30

Pressurized Dual Seals. Barrier fluid is pressurized to a higher pressure than the process and leakage is into the process. Barrier fluid must be compatible with the process. Most UOP processes do not allow the addition of an "alien" fluid. Therefore, pressurized dual seals are used only in special cases.

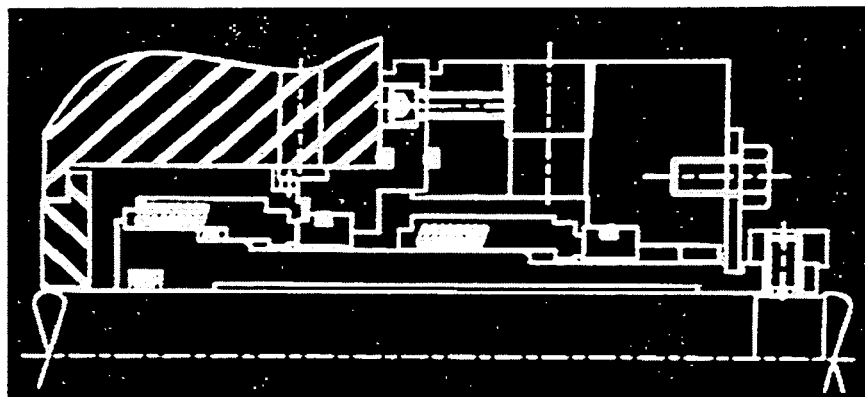
Pressurized Dual Seals



EDS/PO-44
PO-R00-24

Pressurized dual seals. The barrier between the seals is pressurized from a header or a seal pot pressurized with **nitrogen**. Barrier fluid lubricates both sets of seal faces and leaks into the process. Therefore, the barrier fluid must be compatible with the process. Pressurized dual seals are used when pumping very corrosive fluids such as HF acid and when pumping from a vacuum.

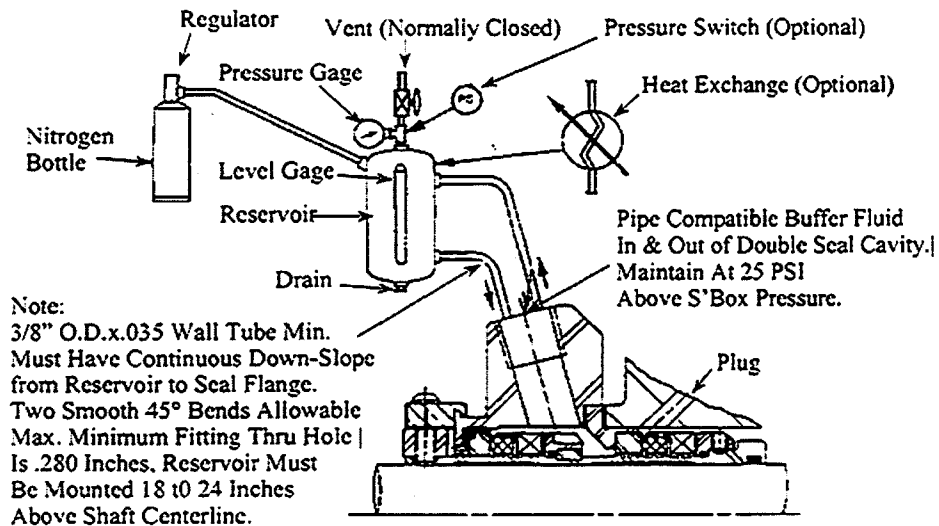
*Arrangement 3: Standard Dual Seal
Barrier $P > \text{Process } P$*



EDS/PO-45
PO-R00-31

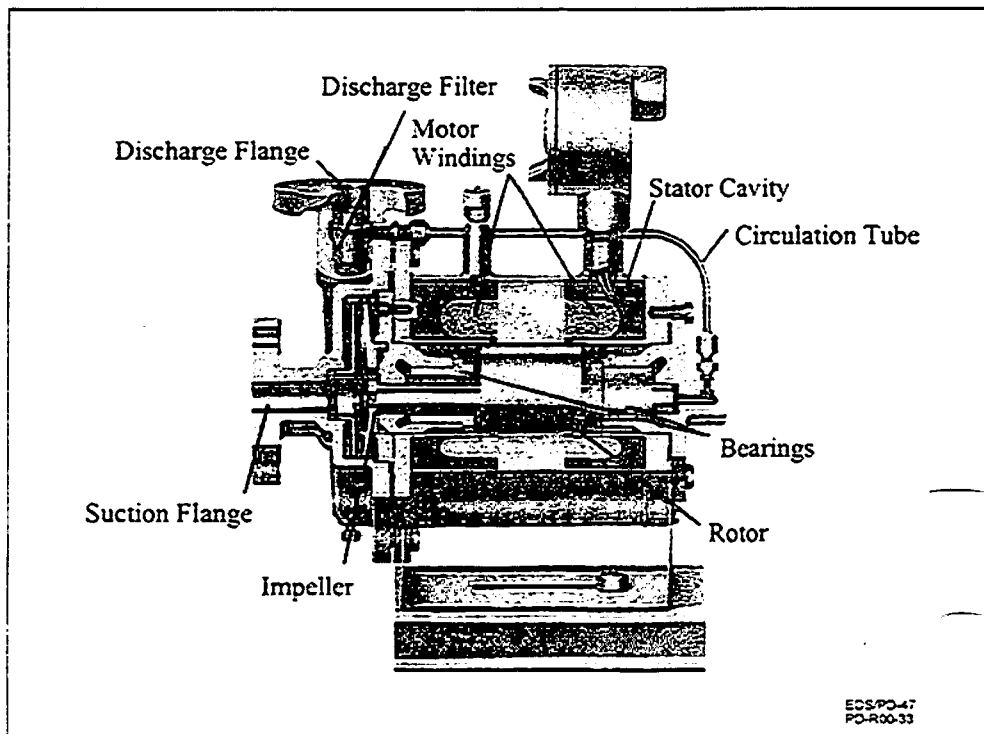
Pressurized dual seals. Barrier fluid is at a higher pressure than the process.
This is a zero emissions seal.

Double Seal Piping



EDS/PO-46
 PO-R00-32

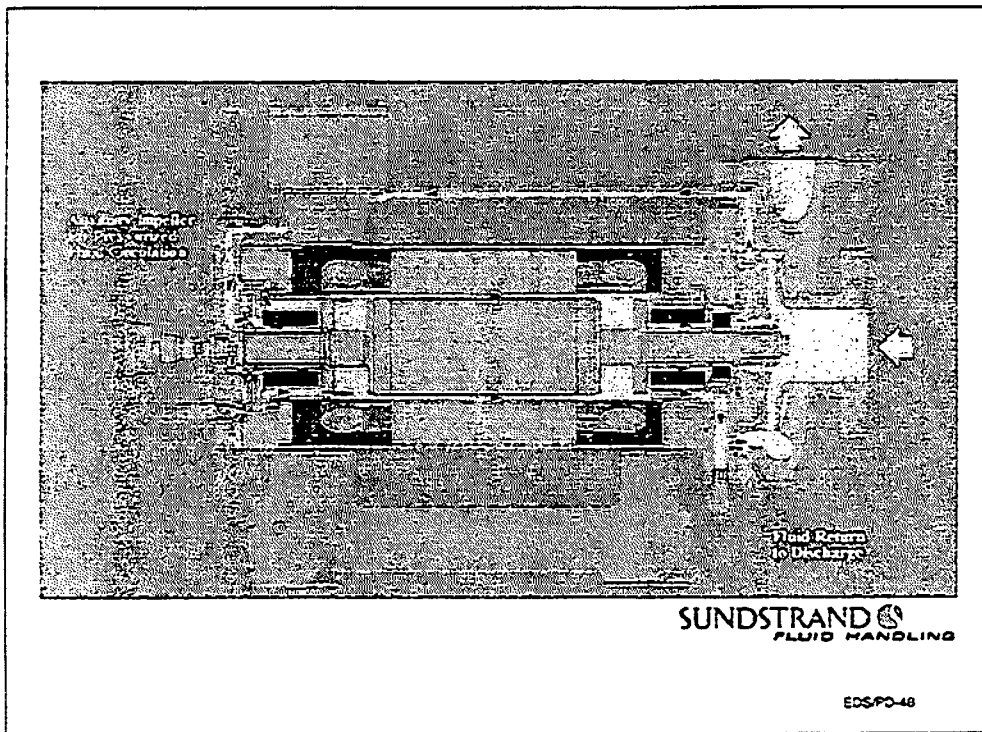
API Plan 53. Similar to an API Plan 52. The seal pot is pressurized with nitrogen to pressurize the barrier between the seals. A pressure switch on the seal pot will warn of loss of nitrogen and a seal leak.



— many problems
— very sensitive

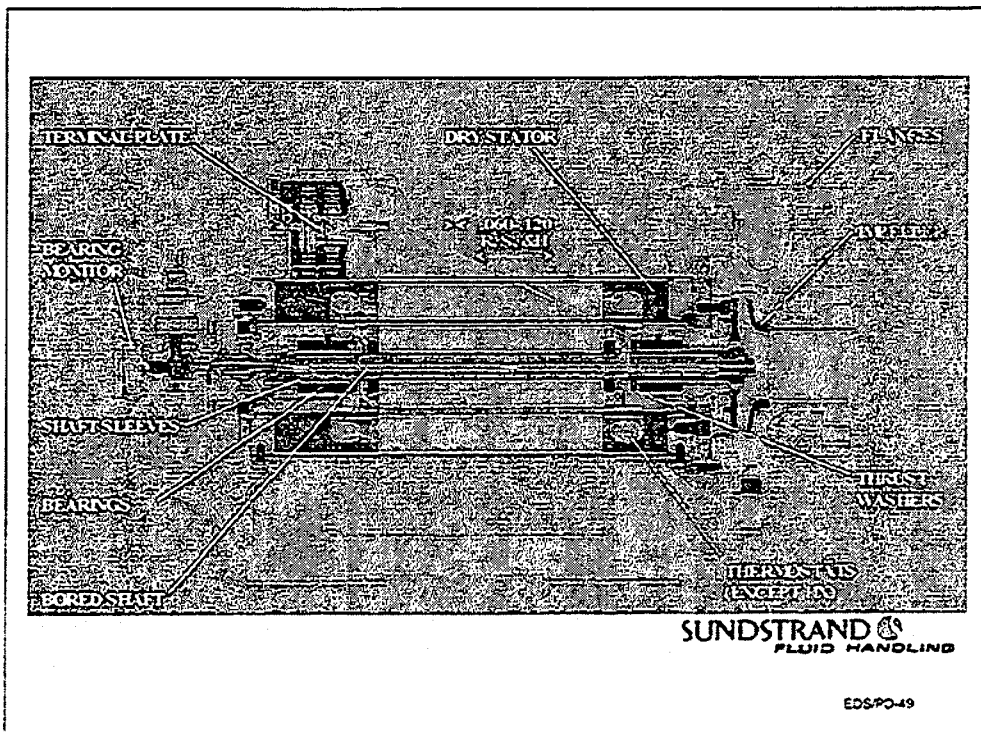
Sealless canned motor pump. Zero fugitive emissions. The motor windings turn a magnet on the pump shaft across a containment barrier. The process fluid lubricate the bearings on the pump shaft and remove heat from the motor windings. Therefore, the pump cannot be run dry (bearings will not be lubricated) or be run blocked in (heat will not be removed from the motor). UOP specifies instrumentation (alarms and shutdowns) to prevent pump damage in the event of mis-operation.

Not UOP std
Many in LNG critical → No seal or
@t high temps

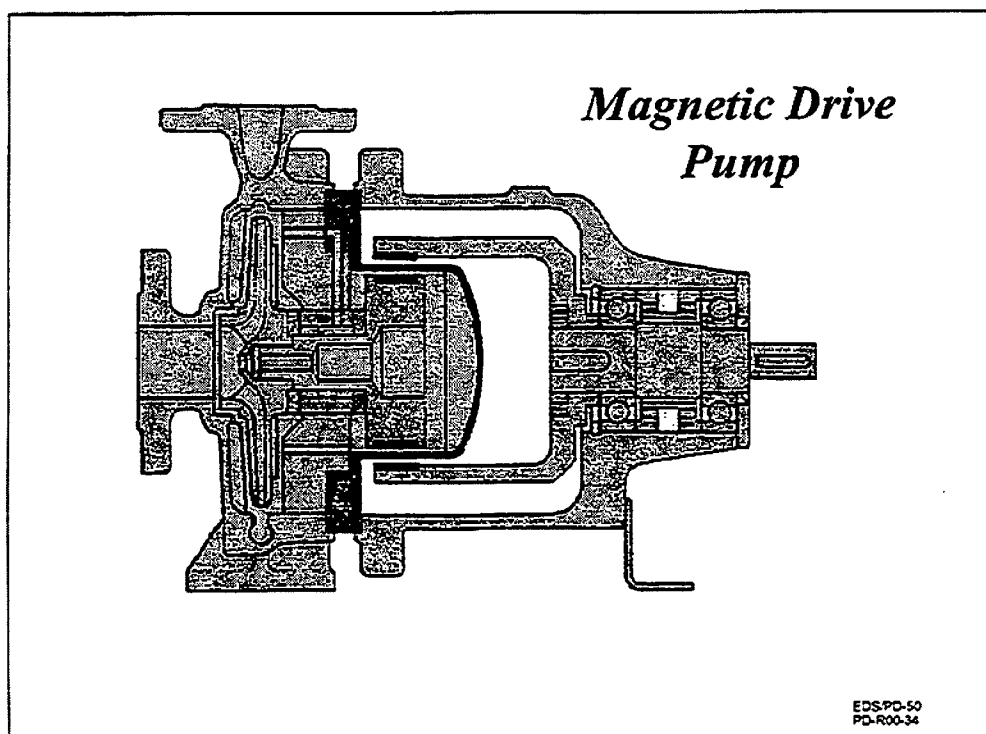


Sealless canned pump. Note the process fluid circulating from the pump discharge to the back end of the pump. The fluid then travels through the pump, lubricating the shaft bearings and removing heat from the motor windings.

If the process fluid is corrosive, (HF acid) the bearing fluid could be from an external source.



Sealless canned pump. For this type, the process fluid is circulating back through the shaft instead of externally.



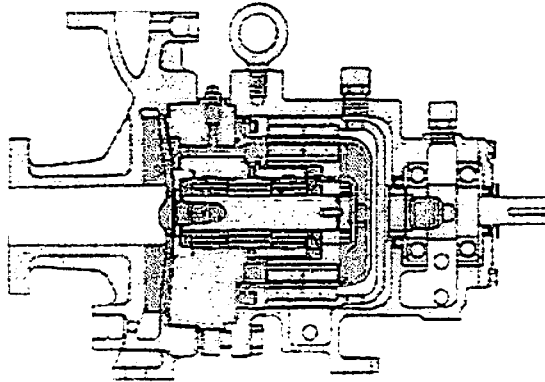
Sealless magnetic drive pump. Magnets on the motor shaft turn magnets on the pump shaft across a containment barrier. This is an alternate design to the canned motor pump. Process fluid still lubricates the pump shaft bearings. Zero fugitive emissions.

motor - separated

Magnoseal™

Standard Features

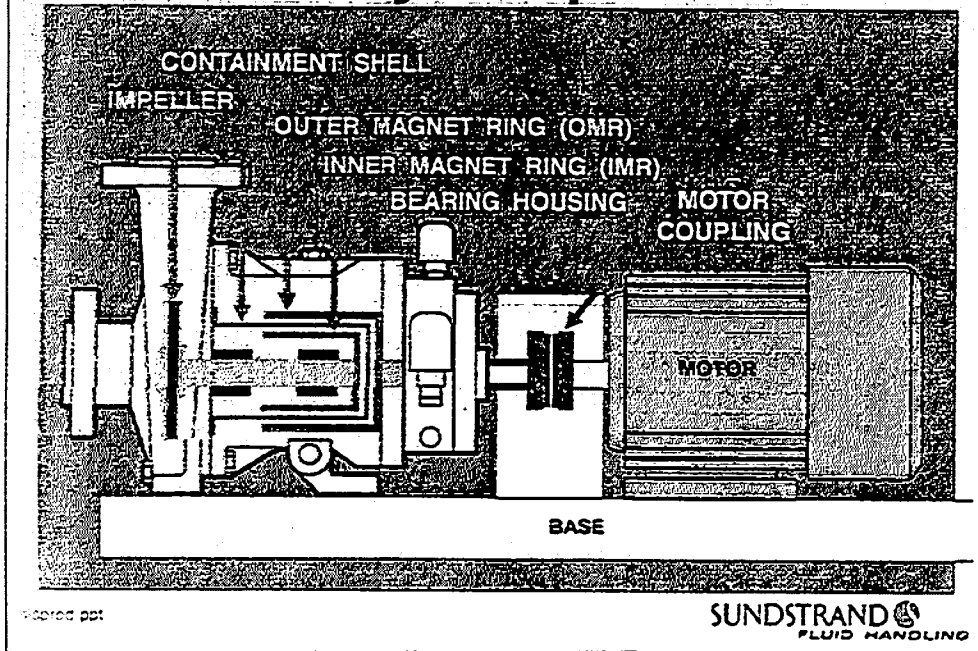
- ASME/ANSI Dimensions
- Magnetic Couplings to 100 HP
- Engineered Composite and Metal Containment Shells
- Precision Cast Semi-Open Impeller
- Wear Resistant Silicon Carbide Bearing System



EDS-PD-51
PD-R00-35

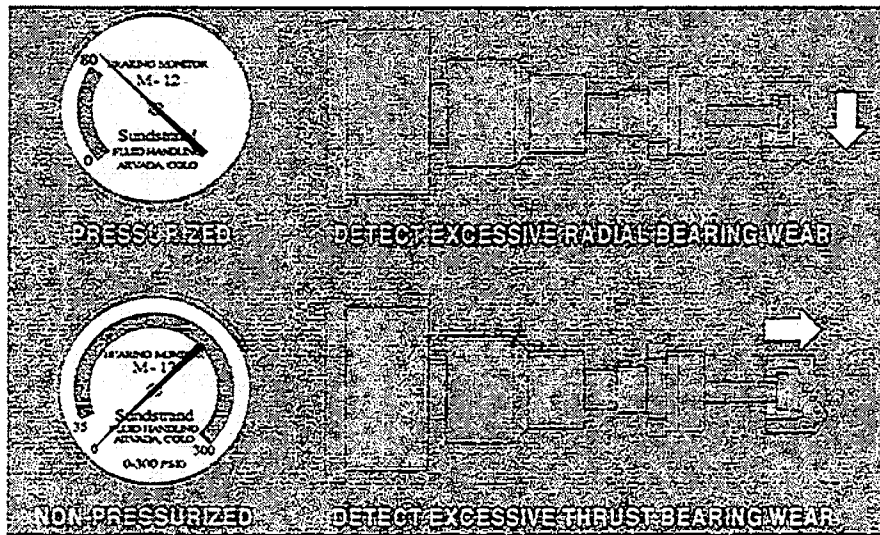
Sealless magnetic (Mag) drive pump. Note that the magnetic couplings are good only to 100 HP.

Theory of Operation



Magnetic drive pump. Note the inner and outer magnet rings. Alignment is critical between the two rings to avoid slippage.

*alignment sensitive
(none in canned pump)*

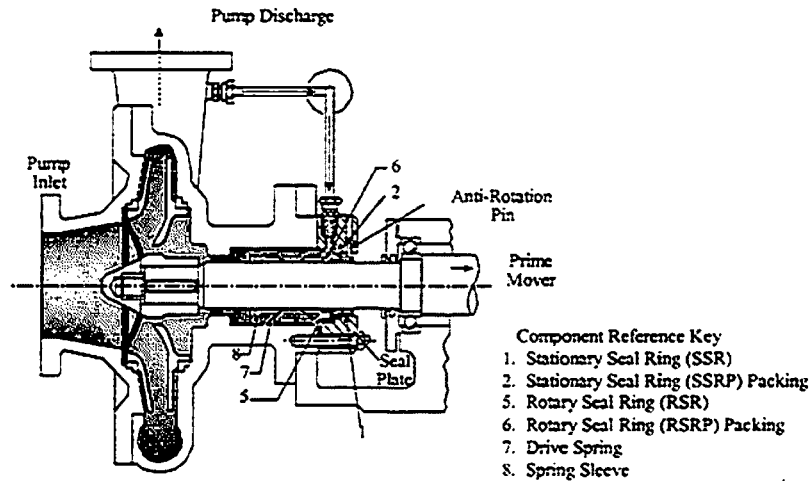


SUNDSTRAND
FLUID HANDLING

ECS-PO-53

Bearing wear monitors.

A Simple Flexibox Seal (Type R) in a Typical Process Pump Application



EDSPO-54
PD-R00-35

API Piping Plan 11. Process fluid at discharge pressure piped to the process side of the seal face. Pressure in this area is maintained with a throat bushing. The friction of the seal faces create 50-100°F of heat. The pressure must be raised around the seal face to prevent the bubble point fluid from vaporizing. At a pumping temperature of 300°F, a cooler is added to this line so the seal is flushed with pressurized, cooled fluid.

API Cooling Water Piping Plan K

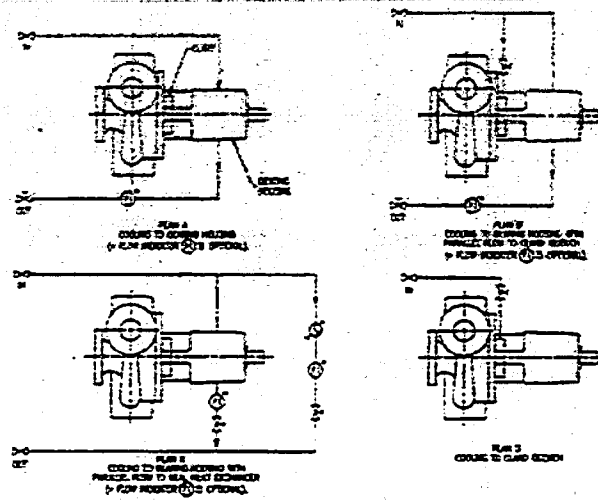
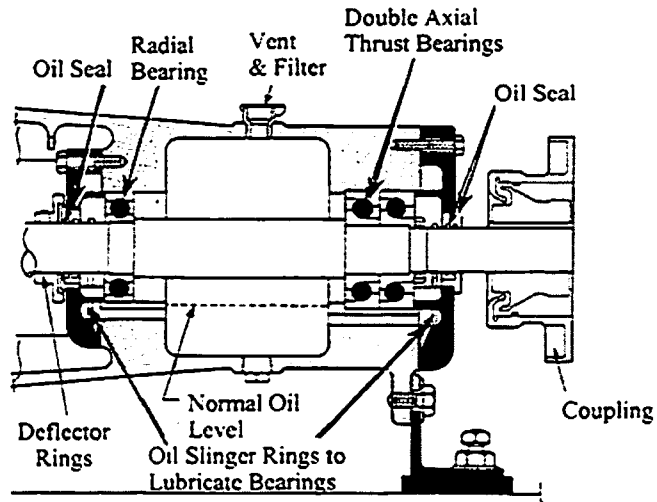


Figure D-44—Cooling Water Piping for Overhung Turbines—Plans A, B, C, and D

EDS/PO-59

API Cooling Water Piping Plan K. This plan provides cooling water for the seal flush cooler and the bearing box.

Bearing Box

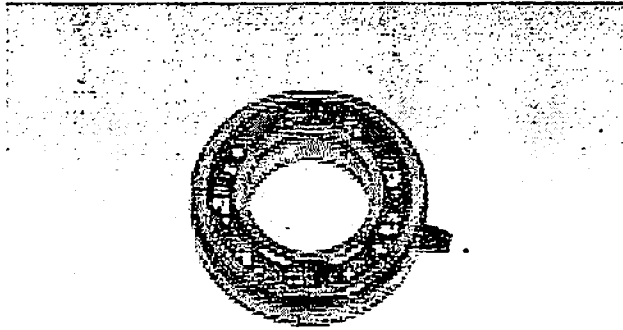


EDS/PD-60
PO-R00-37

Bearings locate the shaft. They also are designed to absorb any extraneous forces. Most pumps have ball type bearings as shown. The bearing box has an oil level. The oil is distributed to the bearing with a ring around the shaft. The ring, sitting loose on the shaft, rotates with the shaft and carries oil up to the bearings. Larger multi-stage pumps have sleeve type bearings with an external, pressurized oil system.

External oil mist lubrication systems are becoming popular. In a oil mist system, a mist of about 1000 parts water to 1 part oil is pumped through the bearing box and exhausted. The bearing box is then always in a clean, fresh environment. The operators do not have to worry about changing or topping off the oil.

Anti-Friction Ball Bearings



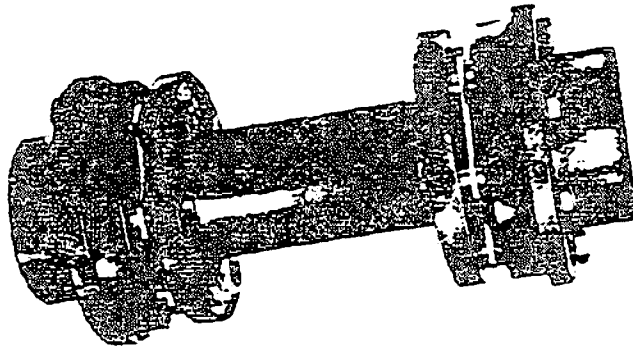
EDS-PD-61

This diagram illustrates the exploded view of a mechanical assembly, likely a pump or motor component. The main assembly is shown in a perspective view, with various parts labeled with callouts. The components include:

- Top Cover:** The uppermost part of the assembly, shown in an exploded position.
- Impeller:** The central rotating component, shown in an exploded position.
- Motor Housing:** The main body of the motor, shown in an exploded position.
- Base:** The bottom support structure, shown in an exploded position.
- Motor Mounting Bracket:** The bracket that holds the motor, shown in an exploded position.
- Motor Mounting Bolts:** The bolts used to secure the motor mounting bracket.
- Motor Mounting Nuts:** The nuts used to secure the motor mounting bracket.
- Motor Mounting Washers:** The washers used to secure the motor mounting bracket.
- Motor Mounting Spacers:** The spacers used to secure the motor mounting bracket.
- Motor Mounting Screws:** The screws used to secure the motor mounting bracket.
- Motor Mounting Screws (Top):** The screws used to secure the top cover.
- Motor Mounting Screws (Bottom):** The screws used to secure the base.
- Motor Mounting Screws (Side):** The screws used to secure the side cover.
- Motor Mounting Screws (Front):** The screws used to secure the front cover.
- Motor Mounting Screws (Back):** The screws used to secure the back cover.
- Motor Mounting Screws (Internal):** The screws used to secure internal components.
- Motor Mounting Screws (External):** The screws used to secure external components.
- Motor Mounting Screws (Internal/External):** The screws used to secure both internal and external components.
- Motor Mounting Screws (Internal/External/Top):** The screws used to secure internal, external, and top components.
- Motor Mounting Screws (Internal/External/Bottom):** The screws used to secure internal, external, and bottom components.
- Motor Mounting Screws (Internal/External/Side):** The screws used to secure internal, external, and side components.
- Motor Mounting Screws (Internal/External/Front):** The screws used to secure internal, external, and front components.
- Motor Mounting Screws (Internal/External/Back):** The screws used to secure internal, external, and back components.

Single stage, overhung process pump. Note the finned bearing box for air cooling.

Flexible Couplings *T Series-Type TSC*

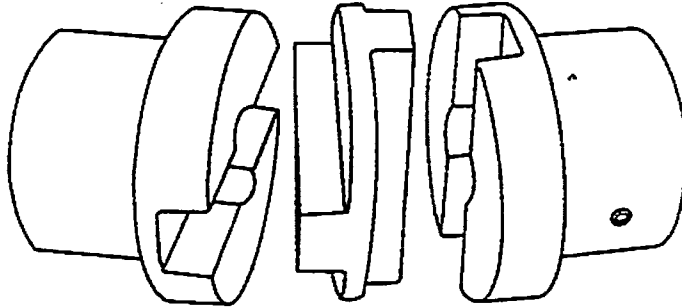


EDS/PD-63
PD-R00-38

Flexible disc type coupling. The coupling is flexible to absorb any misalignment between the pump and driver shafts or vibration from the driver.

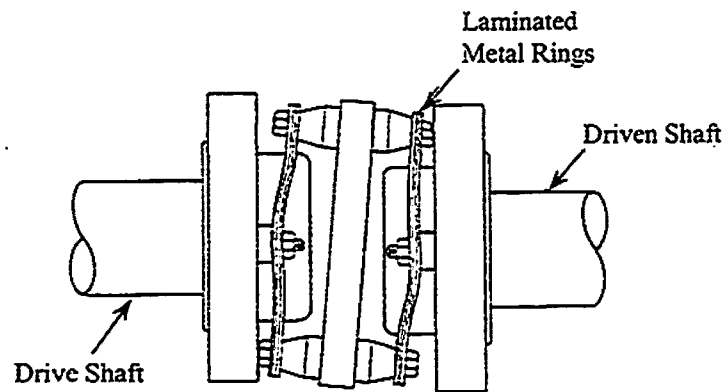
Not to pick-up misalignment
or to transmit power from driver to ...

***Cushioned Single-Engagement
Flexible Couplings***



EDS/PQ-64
PD-R00-39

Laminated Metal Ring Couplings



EDS/PD-65
PD-R00-40

The rings must take up any vibration or misalignment between the driver and pump.

Reliability and Maintenance

What is Reliability?

The main objective of reliability is to achieve the highest plant availability at the lowest possible cost in order to maximize profit.

The goal is to achieve the World Class target of 95+% plant availability!

EDS-PD-66

Reliability and Maintenance

typically unsparred

- Critical Equipment - Centrifugal Compressors, Some pumps - Unsparred
 - Continuous Monitoring System
- Pumps, Reciprocating Compressors - Spared
 - Periodic Monitoring of Vibration Data - Collect and Analyze
 - Root Cause Analysis

EDS-PO-67

Reliability and Maintenance

Equipment Specs and Standards

Vendor Selection

Design and Testing

Process Considerations

Operations and Monitoring

EDSPD-68

Reliability and Maintenance

95% on-line availability for pumps, 5 year MTBR

	MODE		RELIABILITY	MAINT. S
1	Reactive		Low	High
2	Preventative		Moderate	High
3	Proactive		High	Low

- 1. Reactive – Run to failure**
- 2. Preventative – Time-based maintenance**
- 3. Proactive – Condition-based maintenance**

EDSPD-69

Reliability and Maintenance

Reactive - Run to Failure

- Process Interruption ✓
- No Opportunity for Diagnosis ✓
- Frequent Failures ✓
- Other Parts are Effected
- Replace Good Parts

EDS/PQ-70

Reliability and Maintenance

Preventative - Time-based Maintenance

- No Opportunity for Diagnosis
- Replace [✓] Good Parts

EDS/PD-71

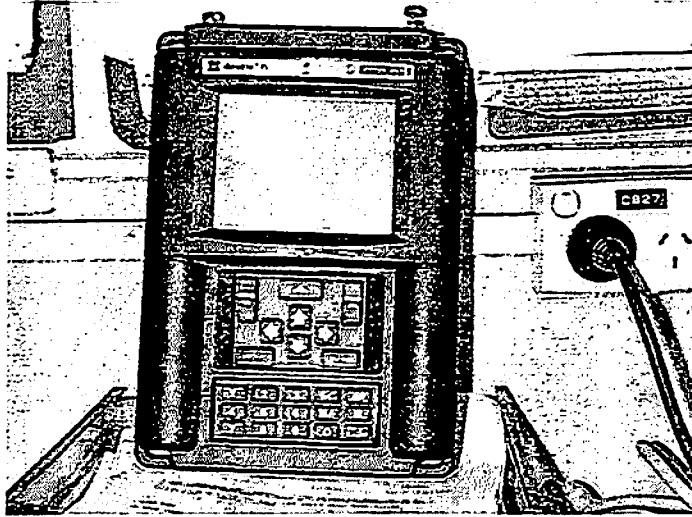
Reliability and Maintenance

Proactive - Condition-based Maintenance

- Repair Before Pump Fails
- Replace Only Bad Parts
- Diagnostics and Root Cause Analysis

EDS-PD-72

Portable Vibration Monitor



EDS-PD-73



EDS-PO-74

Reliability and Maintenance

<u>Cause</u>	<u>Frequency</u>	<u>Amplitude</u>
Unbalance	RPM x 1	Steady
Bent Shaft	RPM x 1 or 2,	Axial high
Cavitation	Random	Fluctuating
Misalignment	RPM x 1 and 2	
Parallel		Radial
Angular		High Axial
Foundation, Unsteady RPM,		

ECS-PD-77

***Reliability and Maintenance
Procurement***

EPC during vendor/contractor proposal review

"I am concerned with 3 things:

Price, price and price"

EOSPO-78

Reliability and Maintenance Procurement

1. Price

2. Functionality

3. Utilities

4. Reliability

EDS/PD-79

***Reliability and Maintenance
Procurement***

Main Air Blower quits

Cost up to \$500,000/day
in lost production

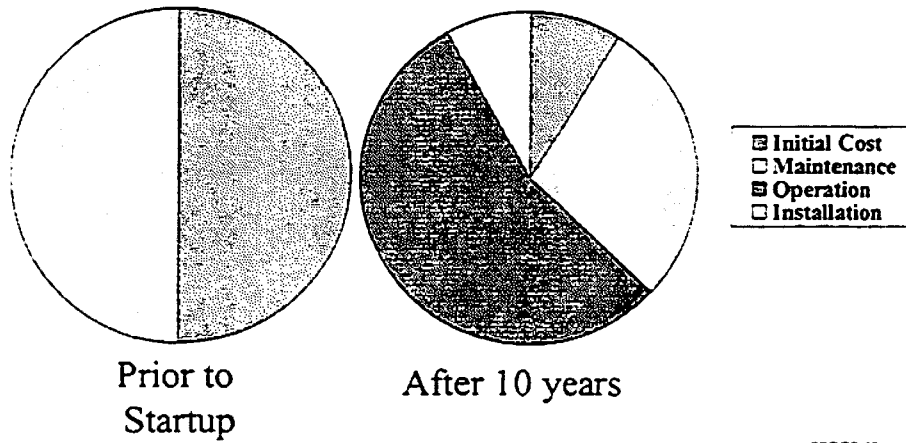
EDSPD-80

Procurement

1. Functionality
2. Reliability
3. Utilities
4. Price

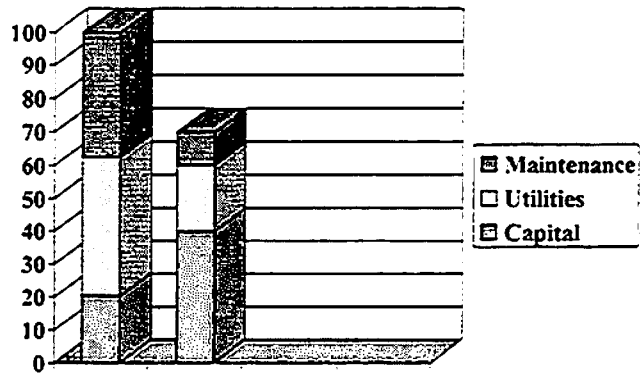
EDSPD-01

Reliability and Maintenance Procurement



EDS-PD-82

Reliability and Maintenance Life Cycle Costs



ECS/PO-83

Reliability and Maintenance

Best Practices - Pump and System Design

- Suction Specific Speed < 11000
- $L^3/D^4 < 60$ (inches)
- Design system for operation at or near BEP
- 5 foot NPSH margin
- Minimum 5 pipe diameters straight pipe on suction
- Install API Flush Plan 23 if pumping temperature > 300°F (149°C)

EOS-PO-84

Reliability and Maintenance

Best Practices - Pump Operation

- Do not start and stop often
- Do not run pump dry
- Operate at or near BEP
- Visually inspect pump often (once per shift)
- Measure and record vibrations
- Check cooling water and seal flush temps
- Inspect and change bearing oil (3-6 months)
- Do not "hose down" pumps
- Report problems immediately
- Training

EOS/PD-85

Bruce:
switch 3 months

Reliability and Maintenance

Best Practices - Pump Reliability

- Alignment
- Balance
- Bearings
- Lube oil
- Monitoring - trend analysis
- Repeat

EDS/PD-86

Reliability and Maintenance

Reliable Reciprocating Compressor Design

- Limit Piston Speed
- Limit Piston RPM
- Limit Cylinder Size
- Limit Discharge Temp (250 °F)
- Lubricate Cylinders
- Coat Piston Rod
- Vibration and Temperature Monitor

ECS-PD-87

Reliability and Maintenance

Reliable Centrifugal Compressor Design

- **Limit Maximum Impeller Yield Strength**
- **Dry Gas Seals**
- **Report all Operating Cases**
- **Voting Type Shutdown**
- **Anti-surge controls (where required)**

EDS/PO-88

Affinity Relationships

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

Q = Capacity, gpm

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

N = Rotative Speed, rpm

H = Head, feet

HP = Horsepower

$$HP_2 = HP_1 \left(\frac{N_2}{N_1} \right)^3 \left(\frac{D_2}{D_1} \right)^3$$

D = Impeller Diameter

ECS-PD-89

Affinity relationships or fan laws.

The flow varies proportional to the speed variation and the head varies proportional to the square of the speed. These laws explain why high flows and low heads are achieved with low speed pumps and low flows and high heads are achieved with high speed (Sundyne) pumps.

Processing

↓
Multi - Sundyne (Cheaper than ...)

Horsepower

$$BHP = \frac{GPM \times Head \times SP. GR.}{3960 \times Eff.} = \frac{lb/min \times Head}{33,000 \times Eff.} = \frac{GPM \times PSI}{1714 \times Eff.}$$

60 - 50 Cycle

$$Q_{60} = 1.2 Q_{50}$$

$$H_{60} = 1.44 H_{50}$$

$$NPSH_{60} = \frac{NPSH_{50}}{0.76}$$

EOS/PS-00

When using pump curves for 60 cycle and the pumps will be in a county with 50 cycle power, the flow, head, and NPSH required must be corrected before a pump can be selected.

Horsepower in Field

Measure amp draw of motor

Watts = Amps x Volts

$$\text{BHP} = \frac{1.73 \times \text{Amps} \times \text{Volts} \times \text{motor eff} \times \text{motor power factor}}{746}$$

Motor eff = 0.95 (Approx)

Motor power factor = 0.90 (Approx)

For single phase motor don't multiply by 1.73 (3^{1/2})

ECS-90-91

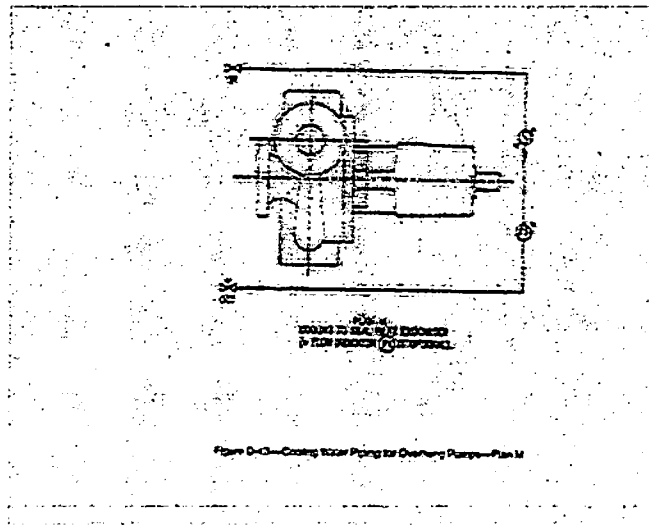
Horsepower in Field Power Factor

Power factor is the ratio between the KW and the KVA drawn by an electrical load where the KW is the actual load power and the KVA is the apparent load power.

It is a measure of how effectively the current is being converted into useful work output and more particularly is a good indicator of the effect of the load current on the efficiency of the supply system.

EDS-PD-92

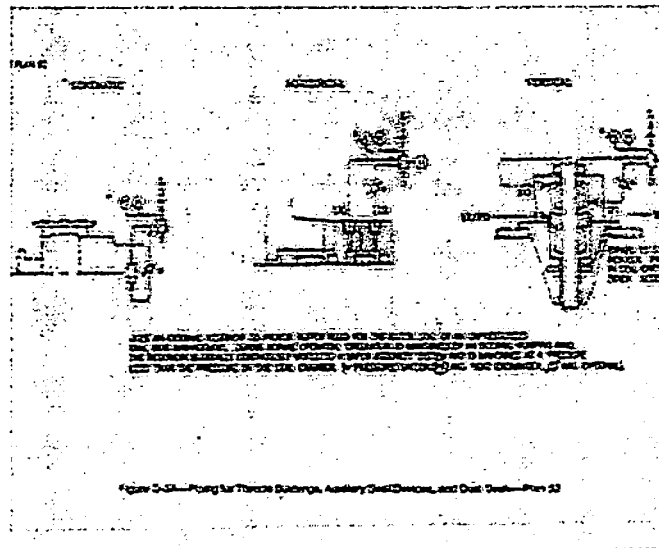
API Cooling Water Piping Plan M



EDS/PO-58

API Cooling Water Piping Plan M. This plan is used when the pumping temperature is between 350°F (177°C) and 500°F (260°C). The cooler is the same cooler used for Piping Plan 23.

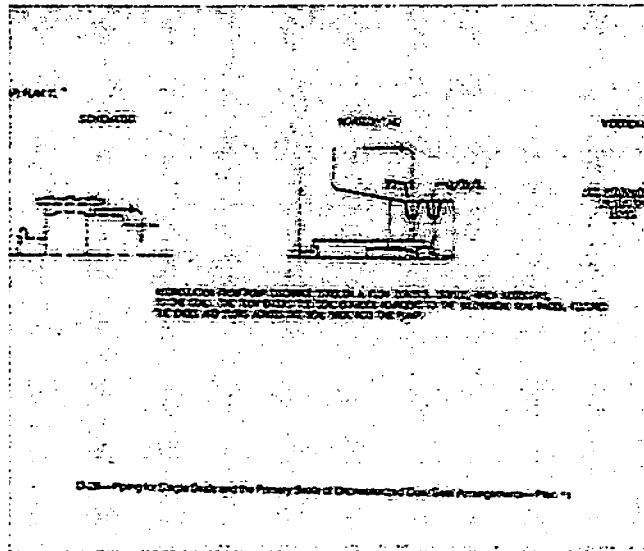
API Piping Plan 52



EDS-PD-57

Schematic of API Piping Plan 52.

API Piping Plan 11



Schematic of API Piping Plan 11.

Horsepower in Field

Example
Amps = 30
Volts = 360

$$\text{BHP} = 1.73 (30) (360) (0.95) (0.90) / 746$$

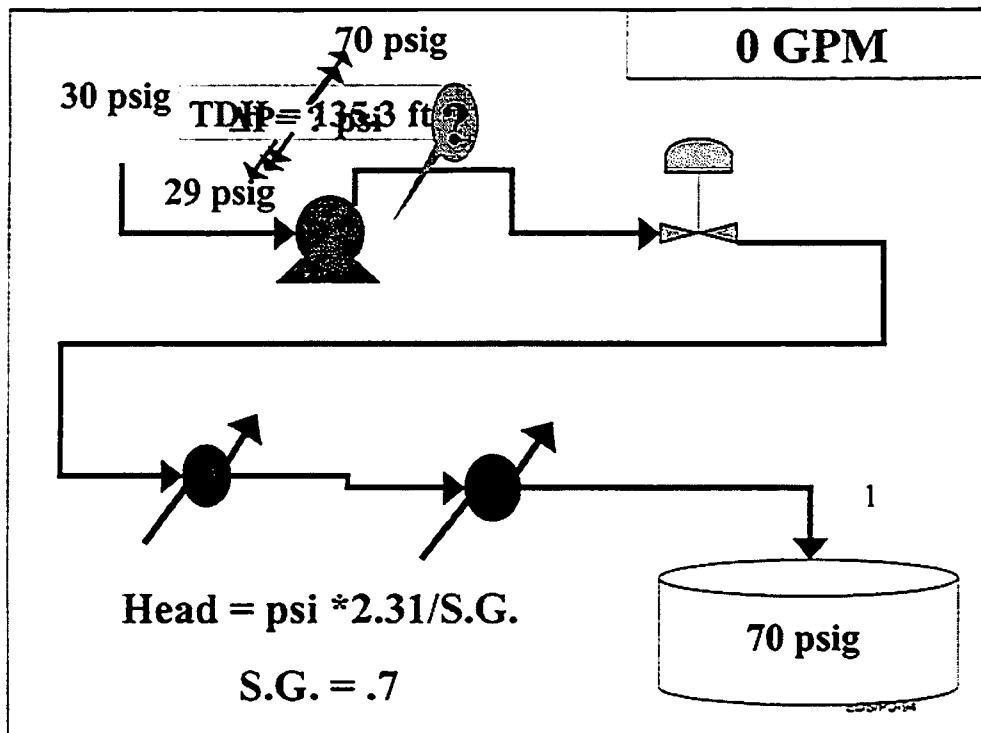
$$\text{BHP} = 21.4$$

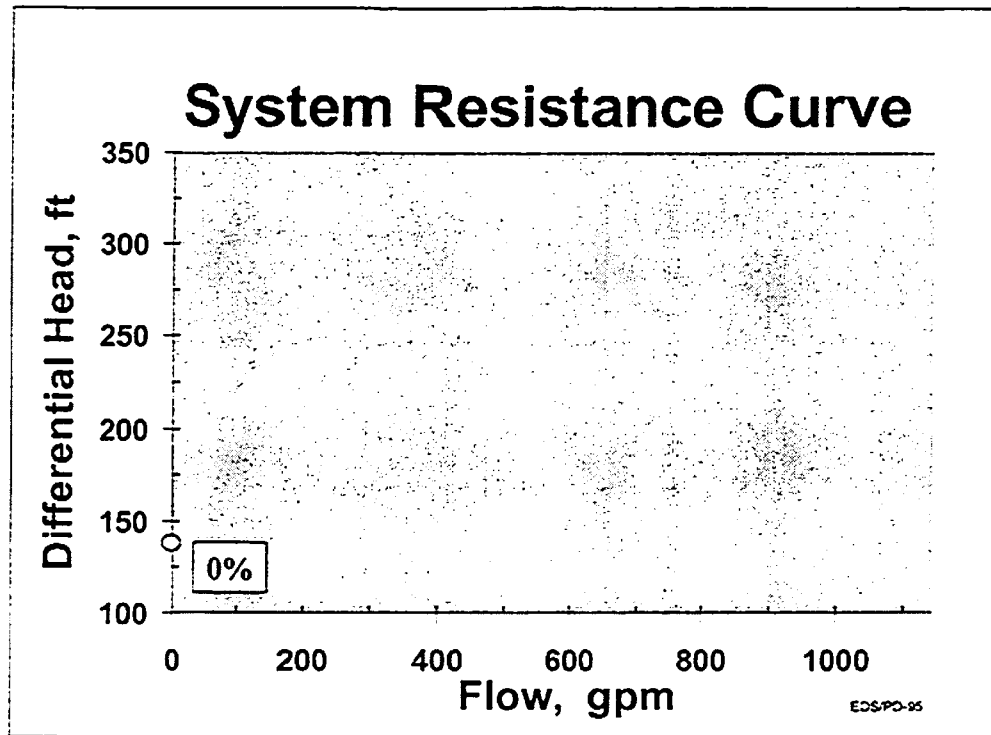
GPM = 300
P2 = 170
P1 = 95

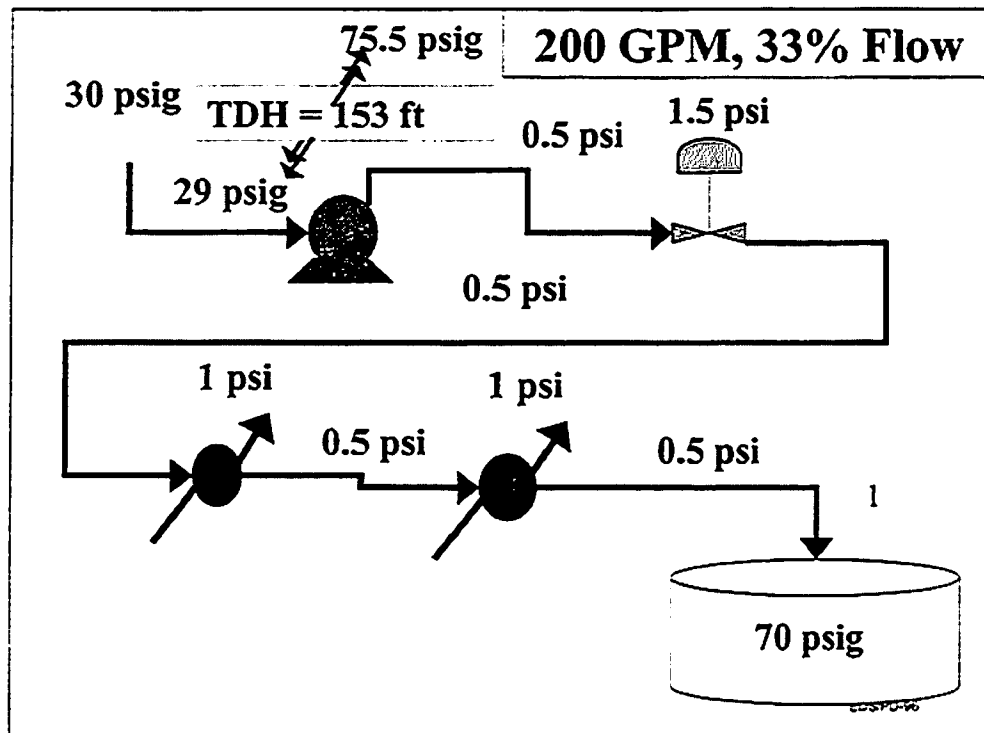
$$\text{Eff} = (300) (170 - 95) / (1714) (21.4)$$

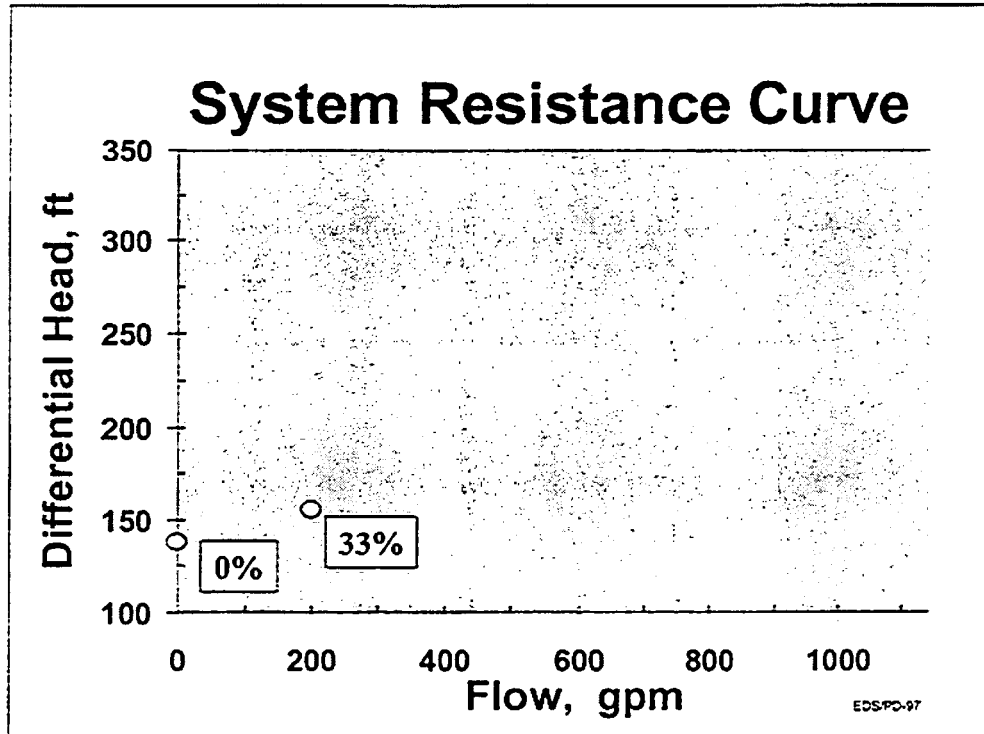
$$\text{Eff} = 61\%$$

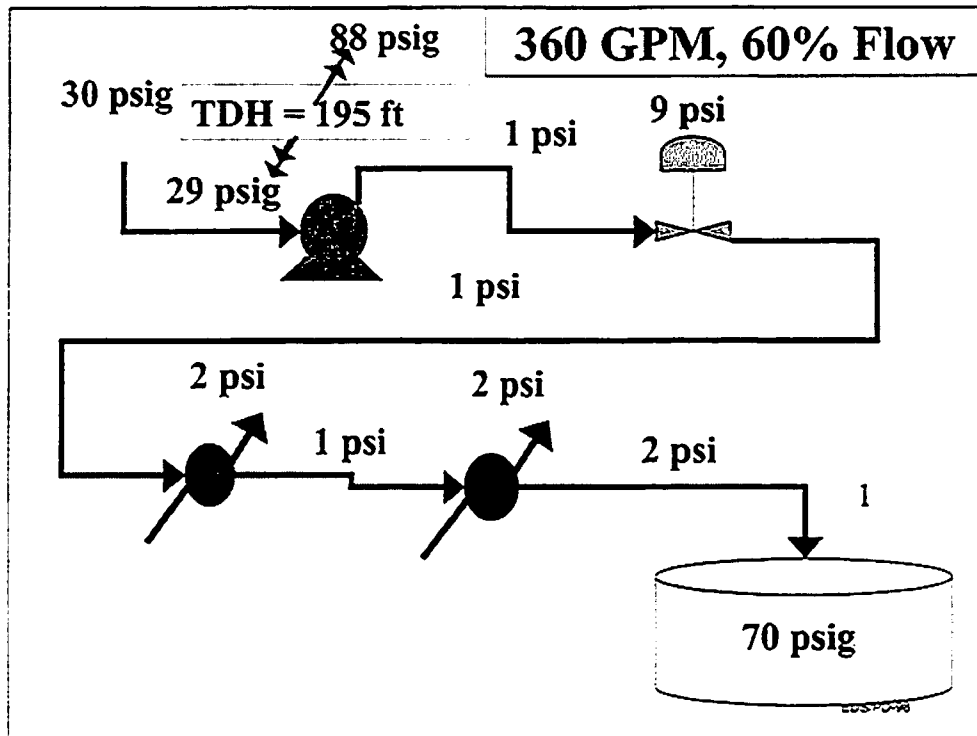
EDS/PO-03

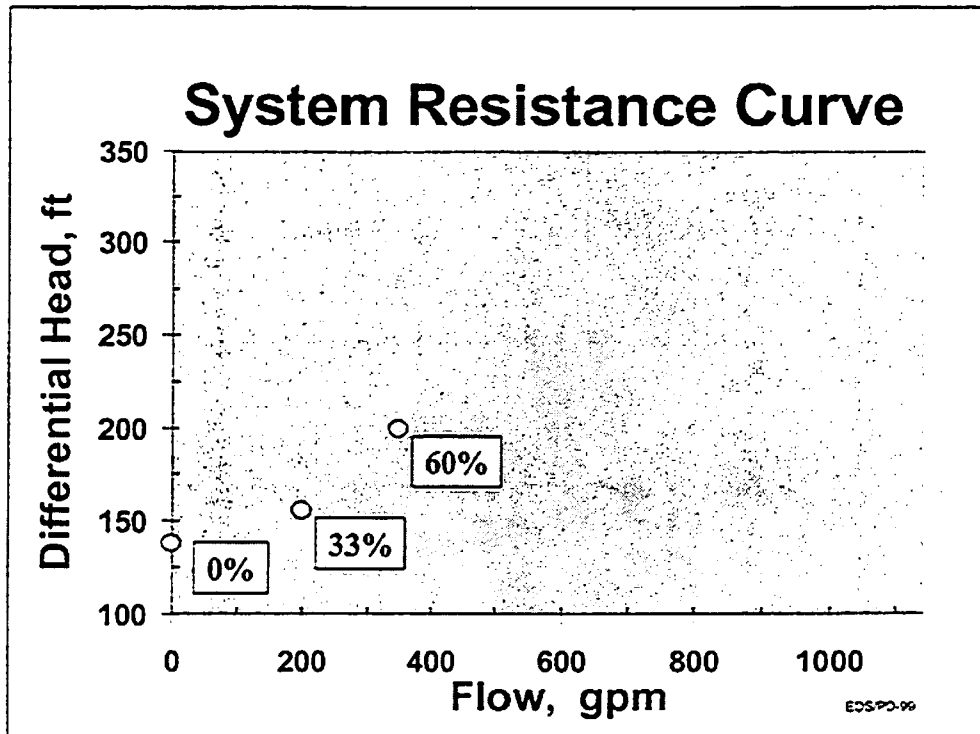


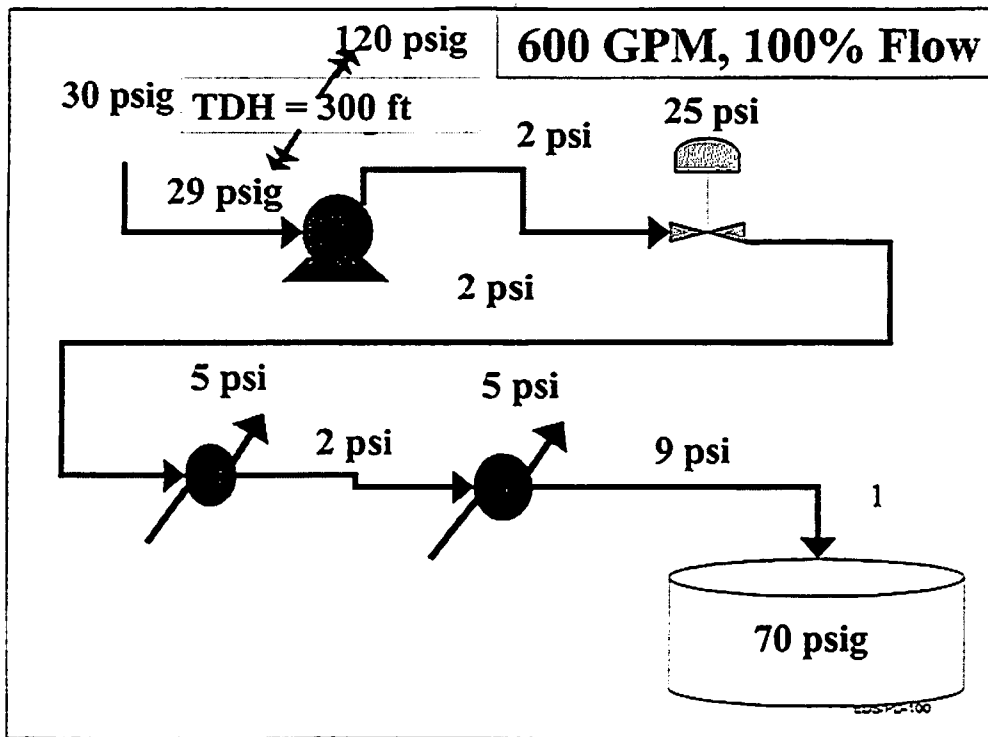




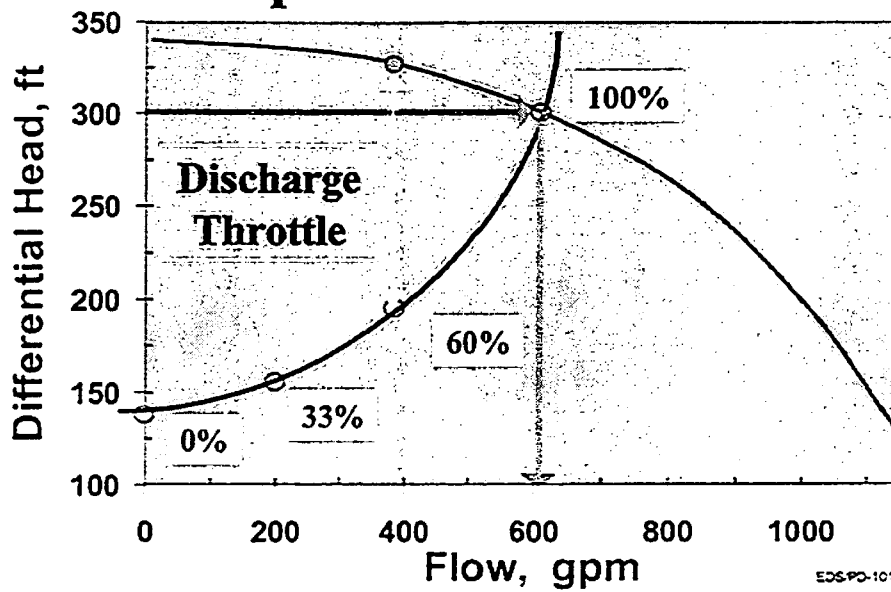








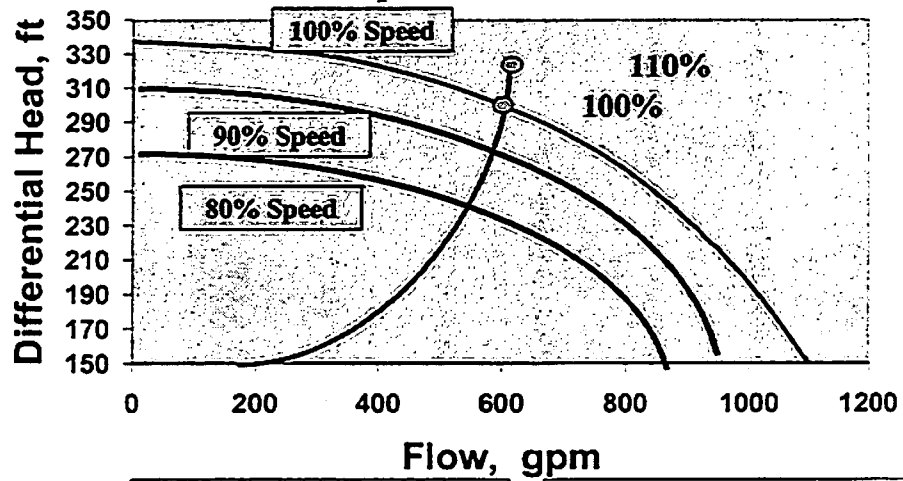
Pump Performance Curve



EDS-PO-101

Pump Performance Curve

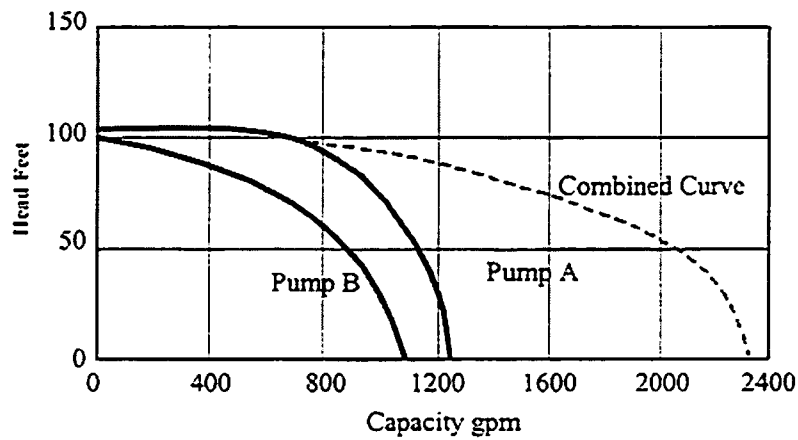
Variable Speed Driver



No discharge throttling

No wasted utilities

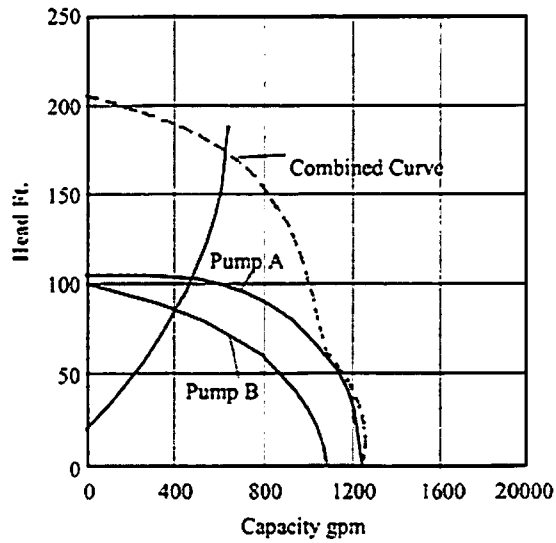
Two Centrifugal Pumps in Parallel



EDS-PO-103
PO-R00-53

When pumps are operated in parallel, the combined performance curve is obtained by adding horizontally the capacities of the same heads. It is preferred that the head-capacity curves rise to shutoff. If the curves droop and if the second pump comes on-line at low flow, the pump could “hunt” between two flows at the same head.

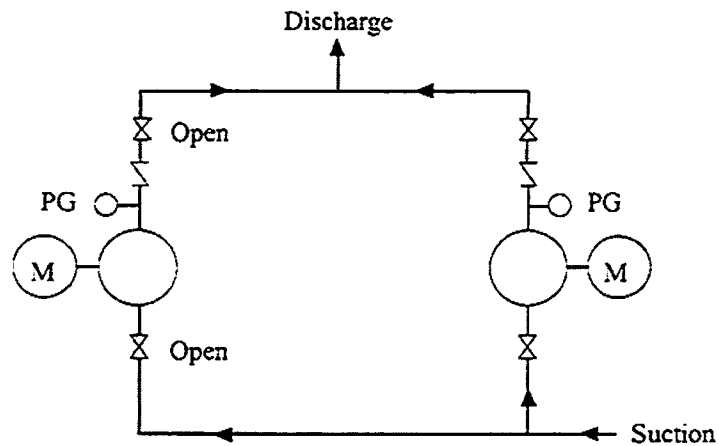
Two Centrifugal Pumps in Series



EDS-PD-104
PD-R00-54

For series operation, the combined performance curve is obtained by adding vertically the heads at the same capacities. Note that the maximum suction pressure of the downstream pump is the shutoff pressure of the upstream pump.

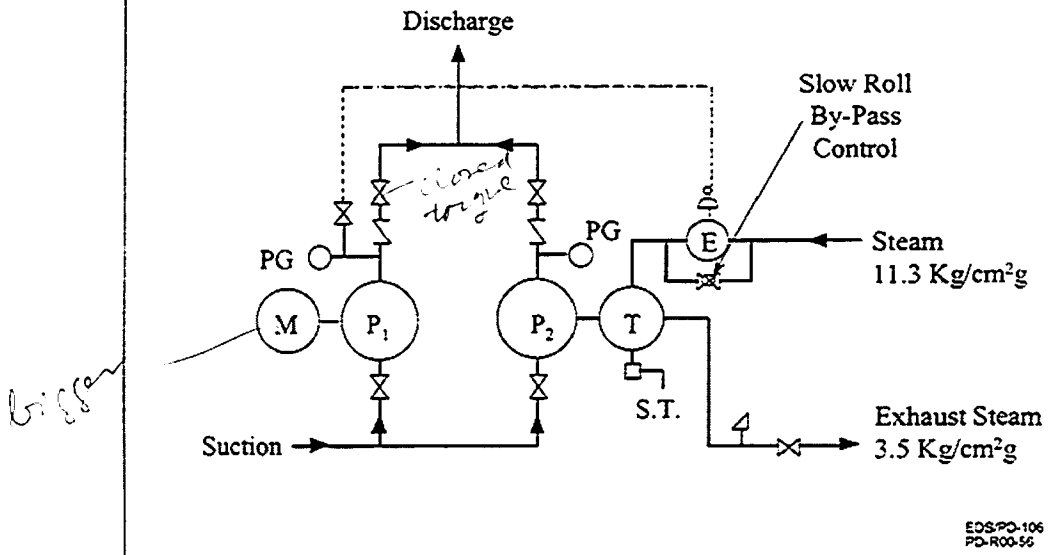
Typical Motor/Motor Spare Pump Arrangement



ECS/PO-105
PO-R00-55

Typically, there are two pumps installed, one operating and one spare. If a pump goes out of service, an operator has to come out and start up the spare pump. Pumps are typically started with the discharge valve closed or pinched open. The least amount of starting torque required by the motor to start the pump is with the discharge valve closed.

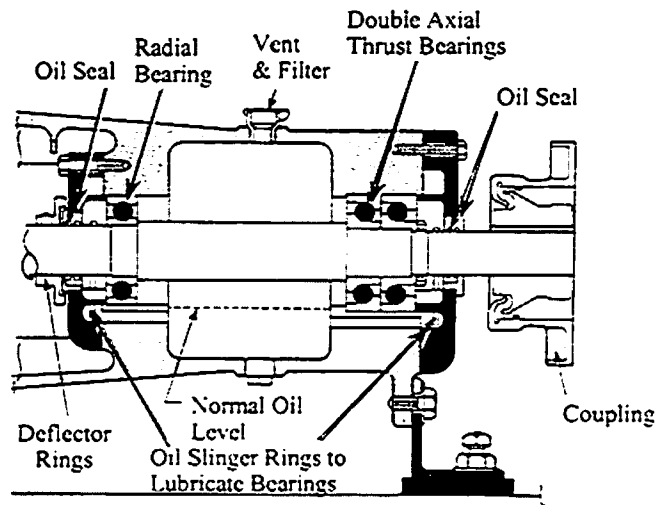
Typical Motor-Auto Cut-In Turbine Spare Pump Arrangement



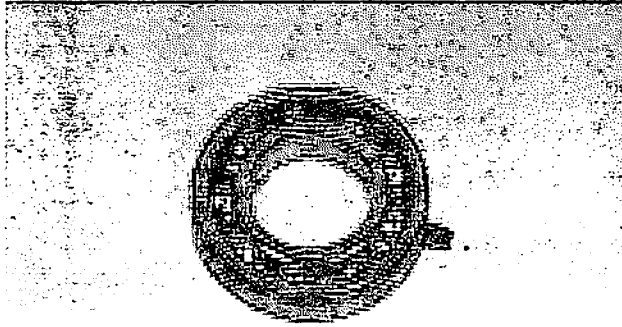
Critical service pumps are on automatic start. Examples of critical service pumps are Boiler Feedwater, Surface Condenser Condensate, Compressor Lube Oil, and HF Acid pumps. If a critical service pump goes out of service, equipment, personnel, or catalyst could be damaged before an operator could get the spare pump in operation. Therefore, the spare pump must come on-line automatically when the operating pump goes down.

*Lead to 1000 cc. Acid
20 p.p.m. c.*

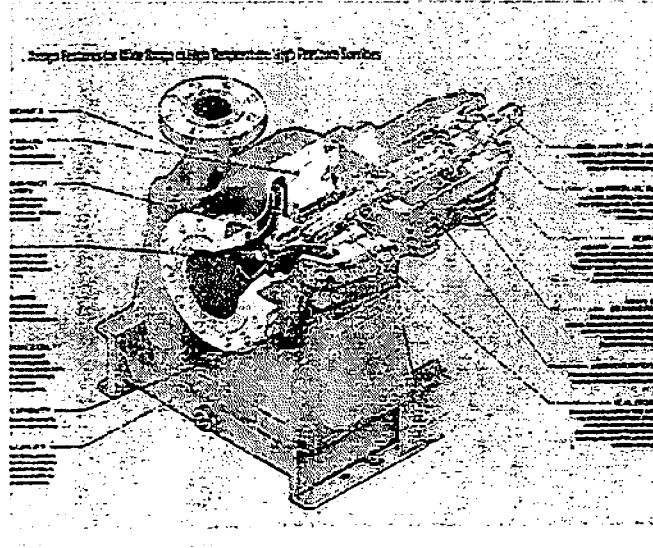
Rolling Element (Ball) Bearings



Rolling Element Bearings



Ring Oil Bearing Lubrication



Rolling Element Bearings

- Per API 610 Minimum Requirements
- 25000 hours (3 yrs) at rated capacity
- 16000 hours (2 yrs) at max radial & axial loads

Bearings Enemies

- Wrong oil level
- Water
- Solids

Bearings Oil Level

- Just Right - Half way up bottom bearing
- Too Low - Inadequate Lubrication
- Too High - Excessive Heat
- Per SKF, oil has useful life of 30 yrs @ 30 C (80 F) Cut in half for every 10 C (18F) rise ✓
- At 100 C (212 F) useful life is 90 days ✓

Bearings Water Where Does It Come From?

- House Cleaning
- Seal Gland Quench
- Aspiration
- Open Oil Cans
- Other

Bearings Water What Are Problems?

- Pitting and Corrosion increase fatigue
- Free atomic H_2 causes hydrogen embrittlement accelerating fatigue
- Water/oil emulsion is poor lubricant
- 0.002% H_2O can reduce bearing life 48%
- 6% H_2O can reduce bearing life 83% ✓

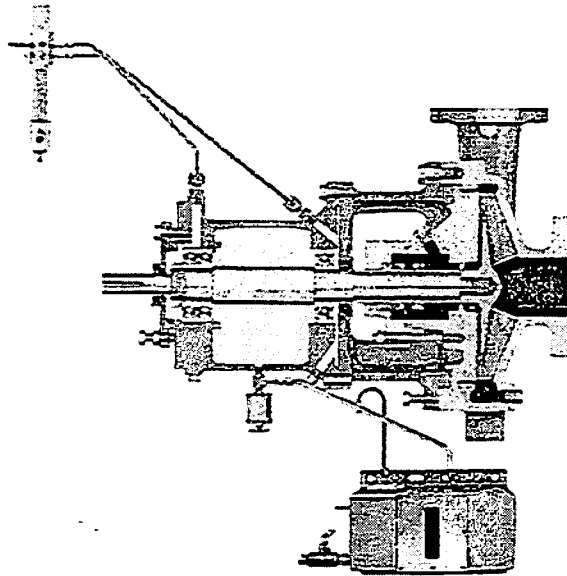
Bearings Solids Where Do They Come From?

- Seal Cage and Bearing Box Seal Wear
- Oil Flinger Ring
- Solids in contaminated oil
- Air borne particles
- Introduced during assembly

Oil Mist

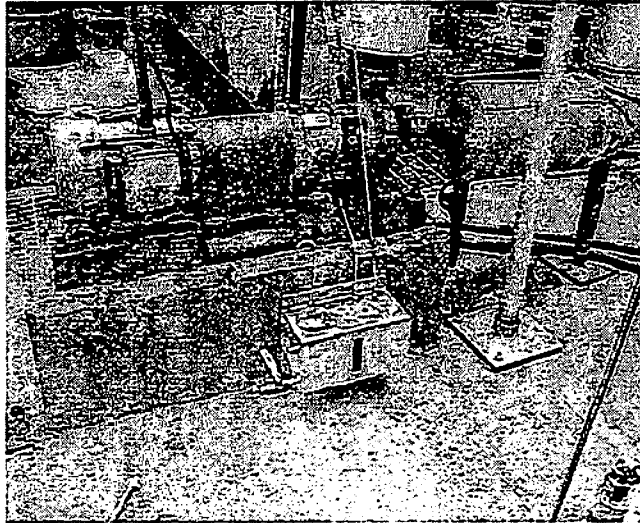
Wrong oil levels, water contamination,
solid abrasion all go away with oil mist
lubrication .

Pure Oil Mist

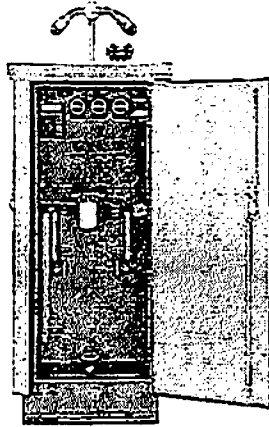


fresh clean supply
by much layer

Pure Oil Mist



Pure Oil Mist



Engineered for large Process units:

- Serve up to 80 Pumps with Drivers
- required Maximum of 30 SCFM of Air
- Consume Less than 2 Gallons of Oil per Day ...justifying use of Synthetic Oils for Maximum Benefits
- 1 part oil per 200,000 parts air

Oil Mist Benefits

- The Proper Amount of Clean Oil is Applied Continuously
 - Clean Once Through Lubrication
 - Bearing Housings are Pressurized Preventing External Contamination
 - Internal Metal Surfaces are Always Coated with oil which Prevents Corrosion (Important For Stand-by Equipment)
 - L_{10} Bearing Life is Extended by a Factor of 6
- Source: Texas A&M University Research

pressurized

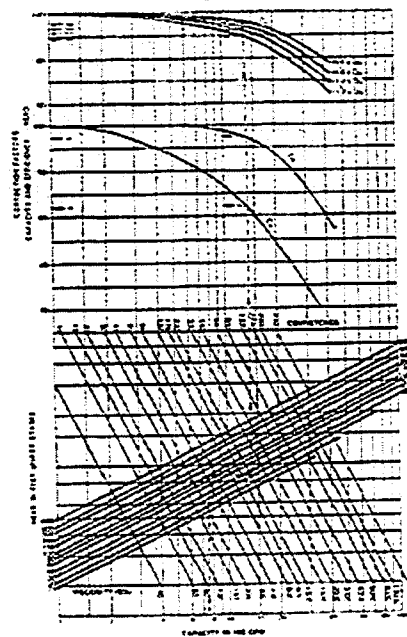
benefits

*8 hrs.
link*

Oil Mist Benefits

- Bearing failures reduced up to 90%
- Dirt particles are not delivered to the bearings
- Dirt particles do not accumulate in the oil sump
- Wear particles are carried away
- Bearings operate 18 to 27°F cooler
- Bearings see only fresh oil
- Seal failures may be reduced up to 30%

Determining Pump Performance When Handling Viscous Fluids



ECS/PO-122

When the fluid is viscous, over 20 centipoise, the pump has to work harder to move the fluid and there is an efficiency correction factor. The factor is in addition to the normal pump efficiency.

Minimum Continuous Flow Depends On:

- Temperature Rise H₂O =

$$\frac{\text{Head at Min Flow}}{778} \left(\frac{1}{\text{Eff}} - 1 \right)$$

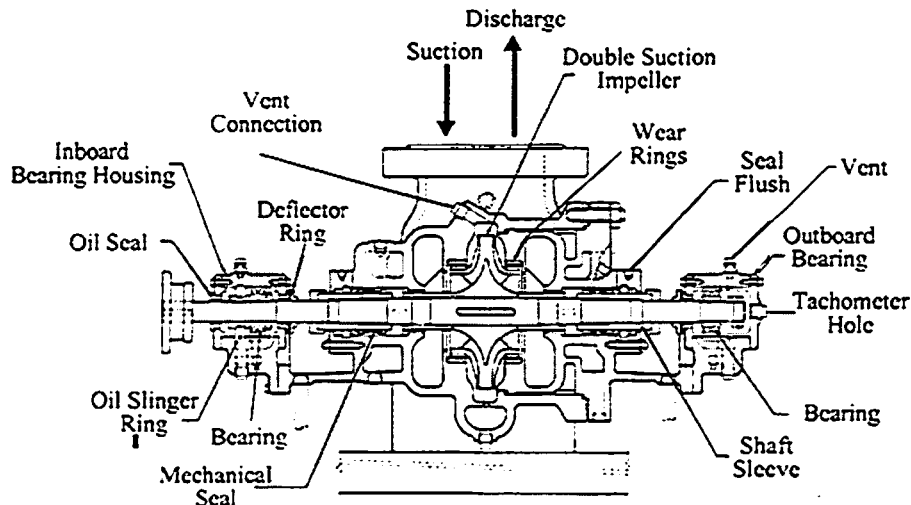
- NPSH

- Radial Reactions On Impeller

EDS/PO-123

Every pump has a minimum continuous flow rate. The pump should not be operated at a lower flow. If a turndown case requires a flow rate lower than the pump's minimum continuous flow, a spillback must be added. Temperature rise across the pump is limited to 15°F. As can be seen from the above formula, as the flow decreases, the temperature rise increases. When looking at catalog pump curves, typically the minimum continuous flow rate is where the NPSH required curves end.

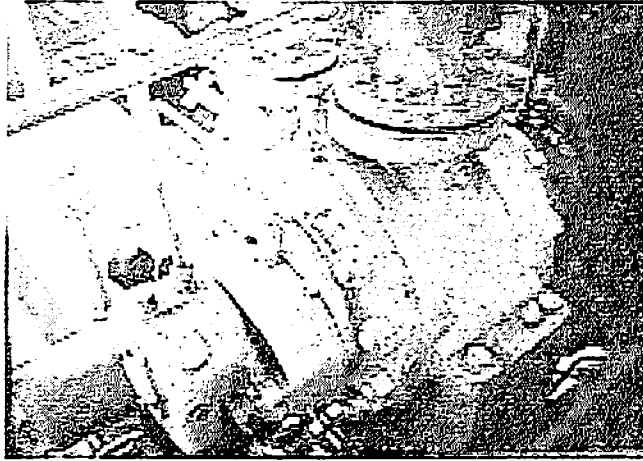
Double Suction Single Stage Pump



EDS-PD-124
PD-R00-43

Double suction, single stage pump. Double suction pumps are used for high flow rates, typically over 1000 gpm. The single impeller has two eyes. Half of the flow is distributed to either side of the impeller. Note that there is one suction nozzle. It is important to have a long straight length of pipe at the suction (at least 6 pipe diameters) and the last elbow should be in the vertical plane perpendicular to the pump shaft to ensure an even distribution of fluid on either side of the impeller. Note the between bearing design. All double suction, two-stage, and multi-stage pumps have the impeller(s) between the bearings and not cantilevered on one set of bearings. This is to prevent the heavy impeller(s) from putting too much of a load on a single set of bearings.

Double Suction



EDS-PD-125

Single stage, double suction between bearing pump.

Heavy Duty Design Features for Refinery, Petrochemical and Power Generating Services.

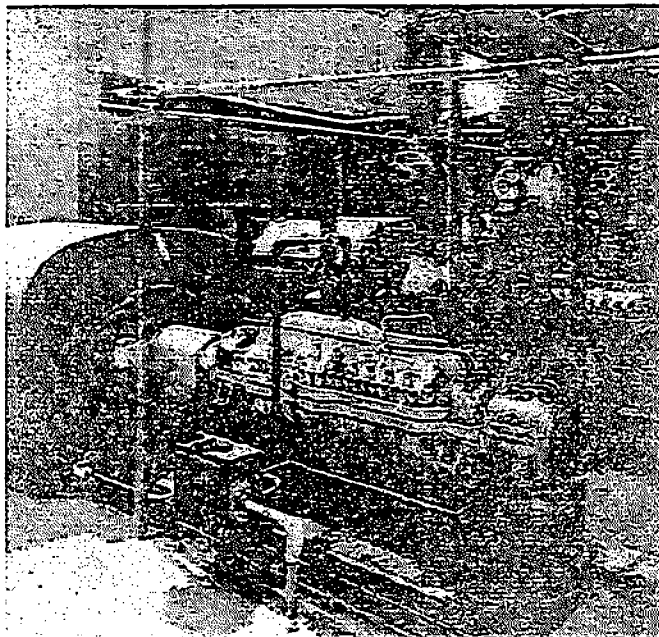


126

Heavy Duty Design Features for Refinery and Petrochemical Services



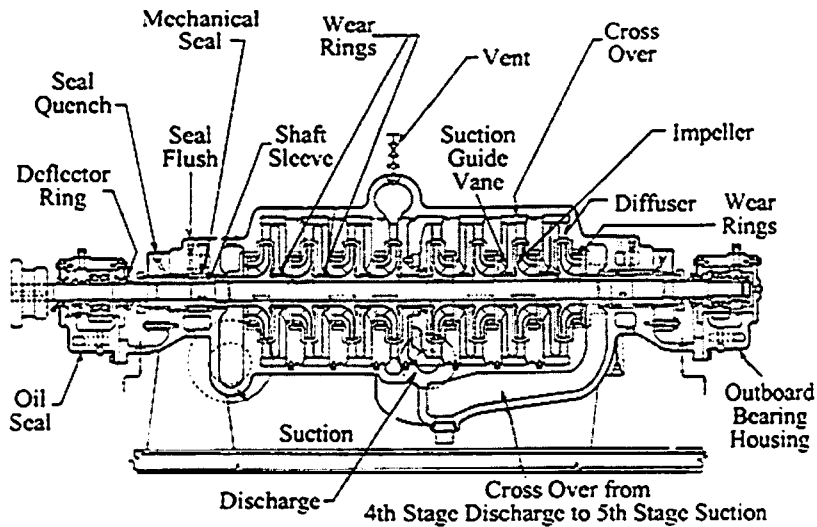
127



EOS-PO-128
PO-R20-44

Horizontally (axially) split multi-stage pump.

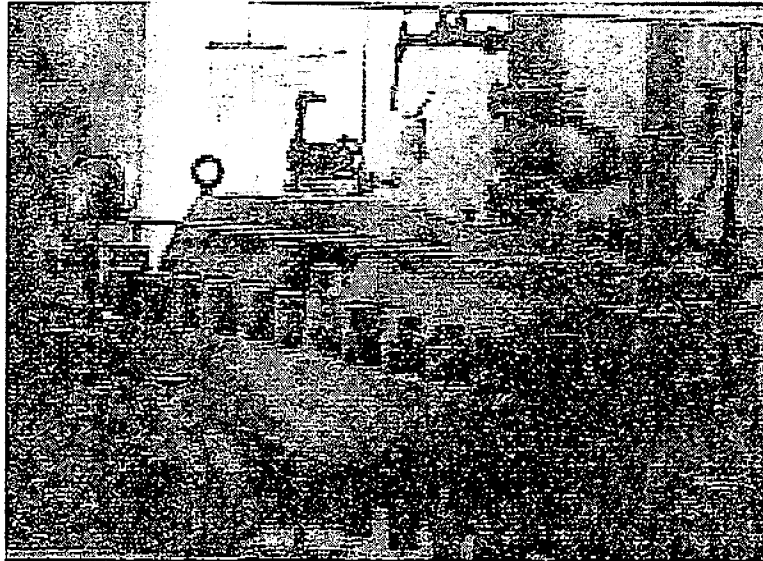
8 Stage Centrifugal Pump Opposed Impellers



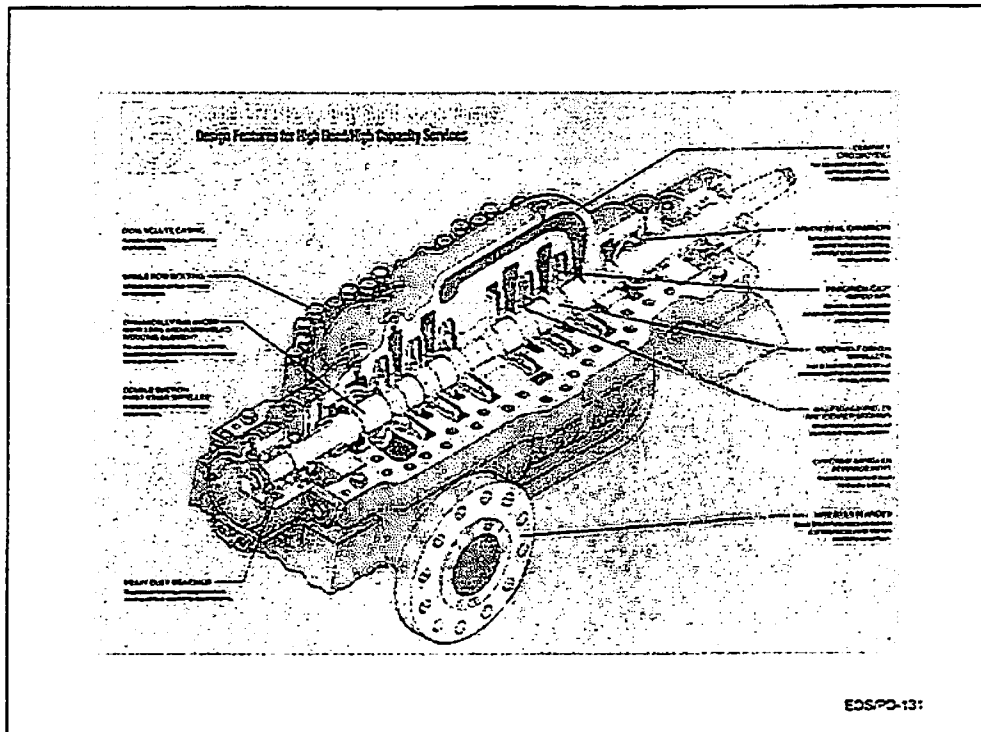
EDS-PQ-129
PQ-R00-45

Inside of horizontally split multi-stage pump. The impellers are opposed to each other. The first stage is on the far left of the pump. The fluid travels to the left for the first four stages. After the fourth stage, the fluid crossed over to the far right and travels to the right for the 5th through 8th stages. This is to balance the axial thrust on the bearings.

Horizontally Split Multi-Stage

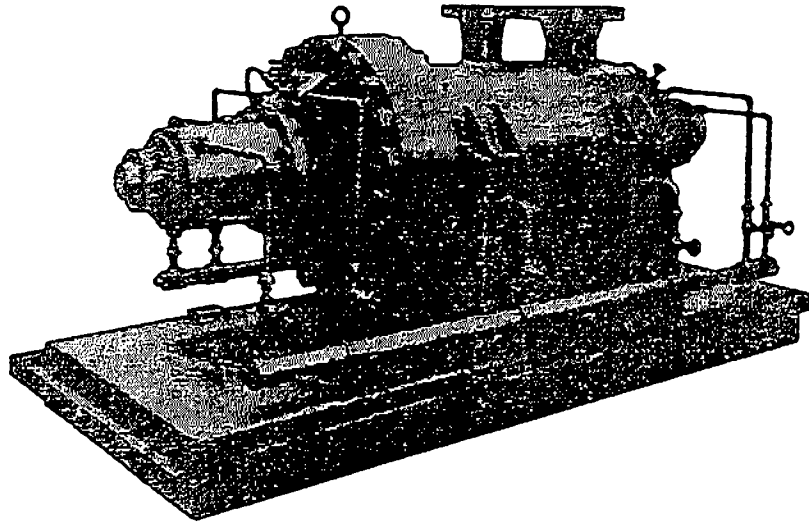


ES-10-130



Six stage, axially split pump. Note the crossover piping internal to the casing.

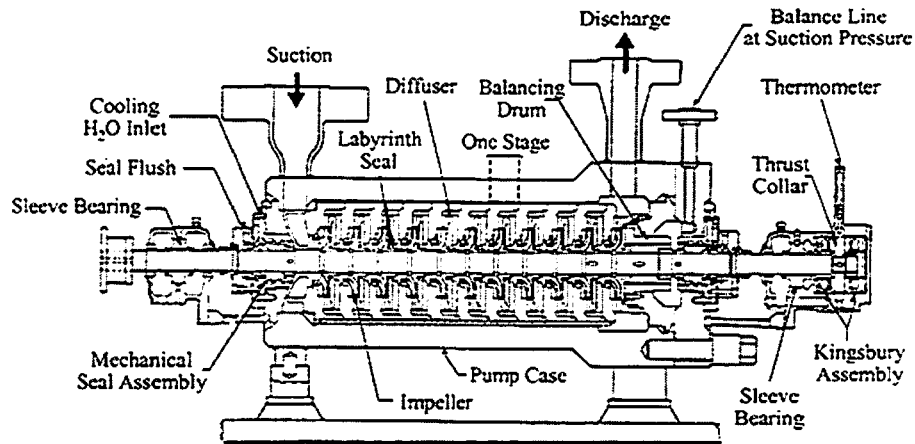
Double Case Centrifugal Pumps



EDS/PD-132
PD-R00-46

Radially split multi-stage pump. Radially split multi-stage pumps are more expensive and take longer to repair than axially split multi-stage pumps. The axially split multi-stage pump has a large casing split. Therefore, to reduce the possibility of process fluid leakage, API 610 does not allow the use of axially split multi-stage pumps if the pumping temperature is over 400°F, the discharge pressure is over 1450 psig or the specific gravity is under 0.7.

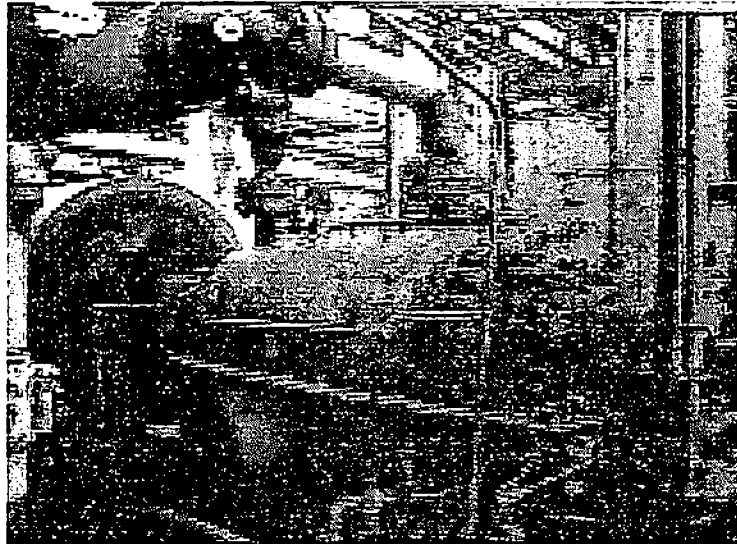
Multi-Stage Pump with Balancing Drum



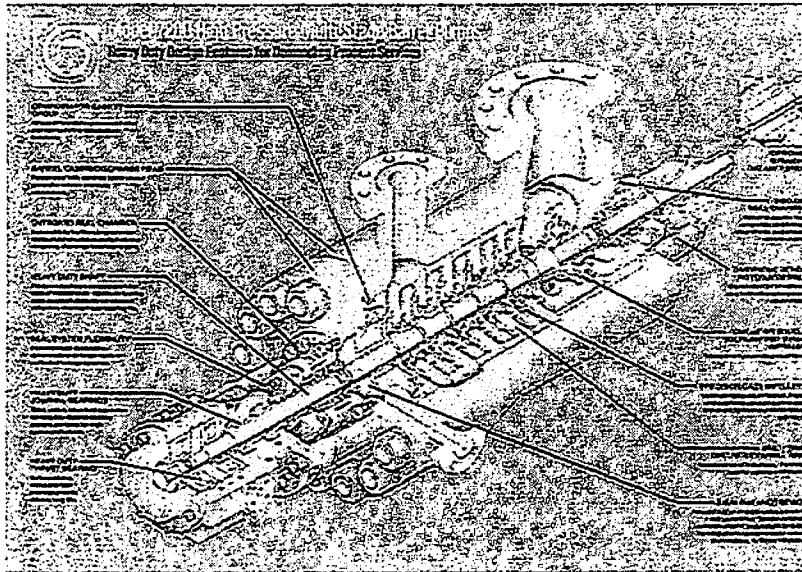
ECS/PO-133
PO-R00-47

Inside a radially split multi-stage pump. Note that the impellers are all facing the same direction. This is because the design of the forged, barrel type casing does not allow for the cross over piping. To balance the axial thrusts, a balance drum attached to a line at suction pressure is installed on the discharge side. This drum absorbs the axial thrust. Also, this enables both seals to seal against suction pressure.

Radially Split Multi-Stage



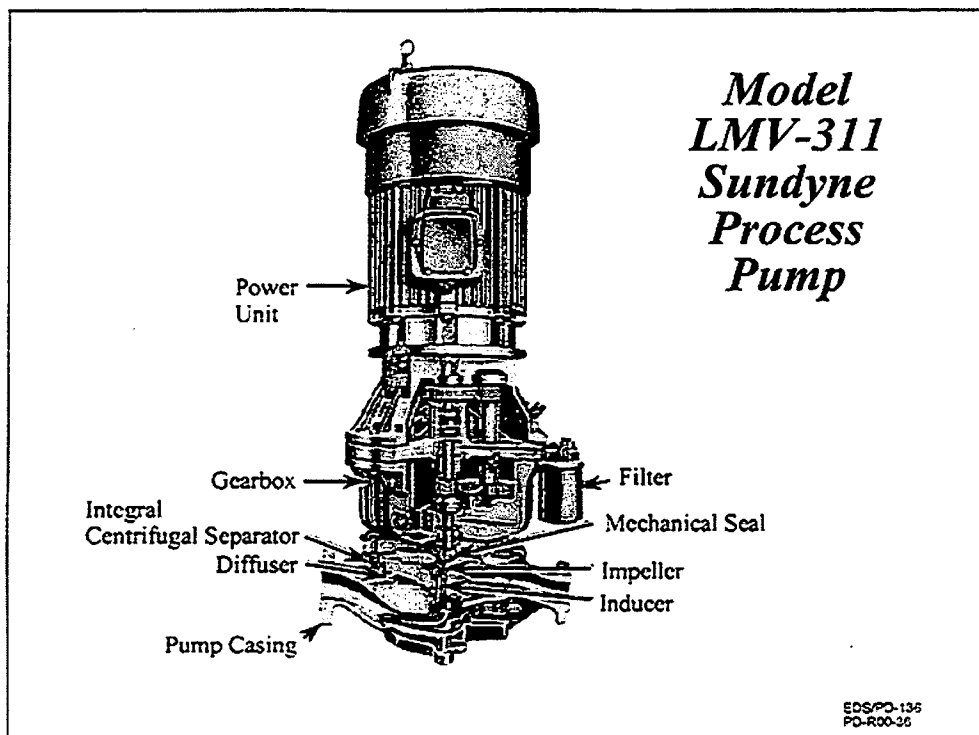
EOS-PO-134



Six stage radially split pump. Note the double suction first stage for reduced NPSH. Also note the balance piston discharge.

Double suction $\downarrow$ NPSH_R

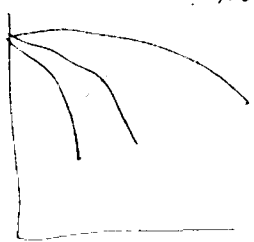
Balance drum vs labyrinth



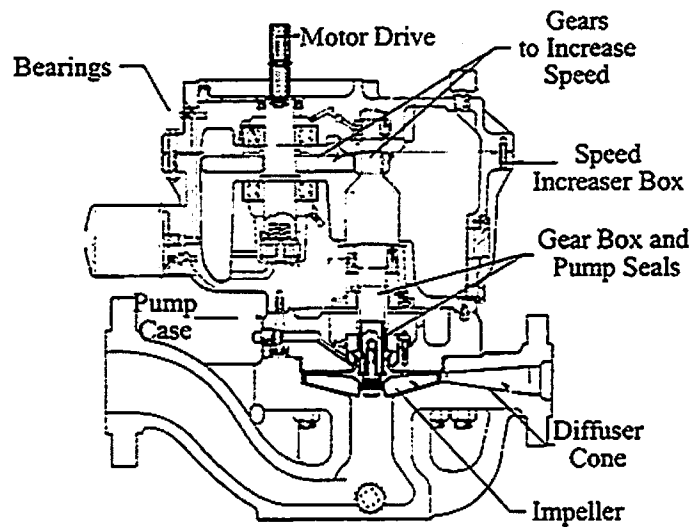
Sundyne pump. This is a high speed, integrally geared pump used for low flow, high head applications. Sundyne is the only manufacturer having good success with this design of pump. This pump is built to API 610 standards. It can achieve high heads using high speed rather than multiple impellers.

No. Sole Source

same wheel, diff. diffuser

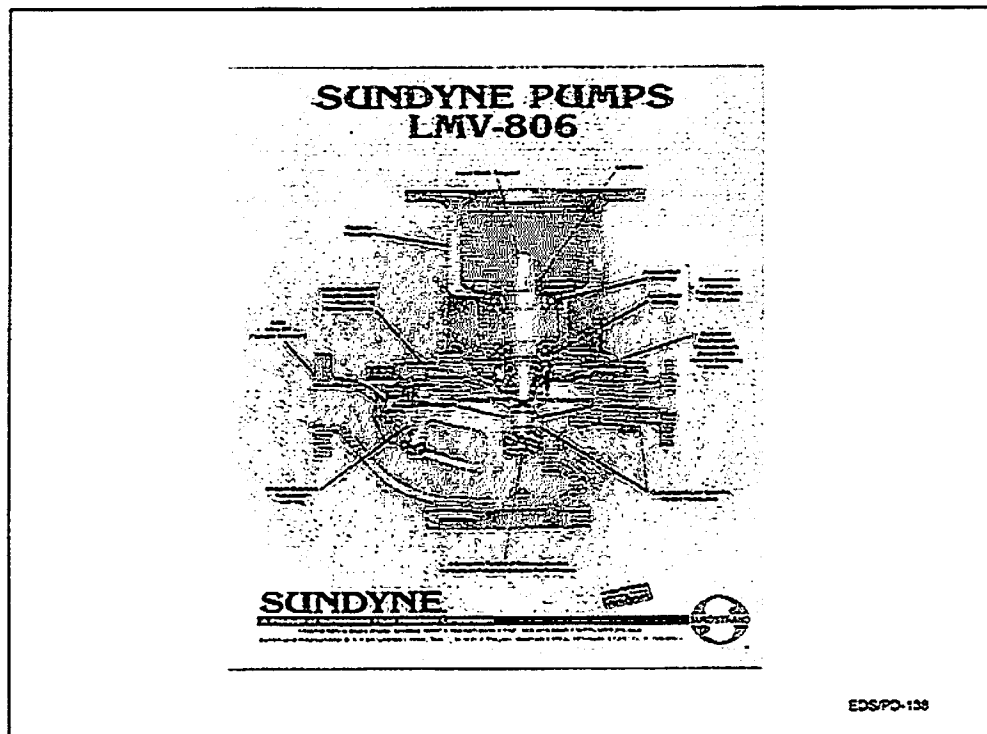


Sundyne Pump

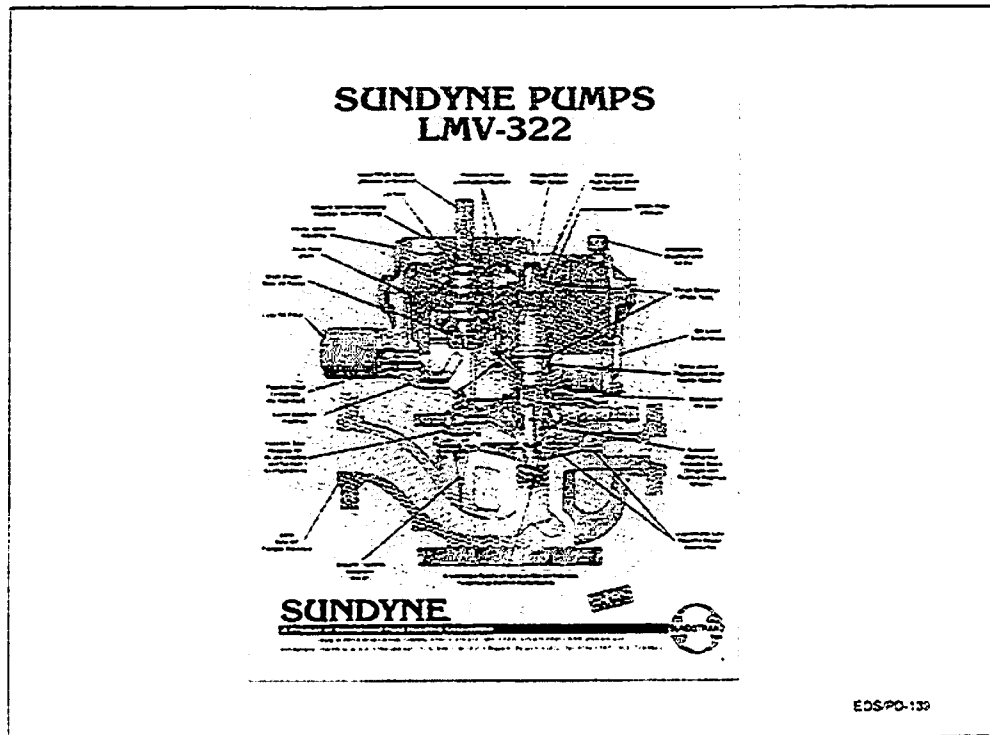


EDS/PC-137
PC-R00-49

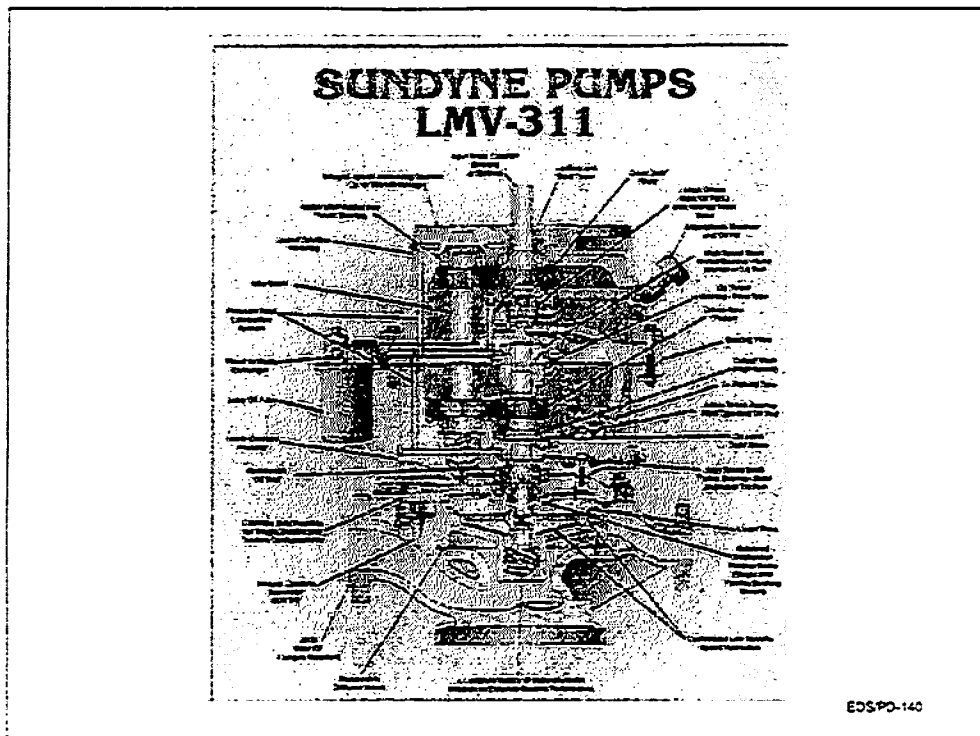
Open view of Sundyne speed increasing gear box.



Simple Sundyne with no gear box.

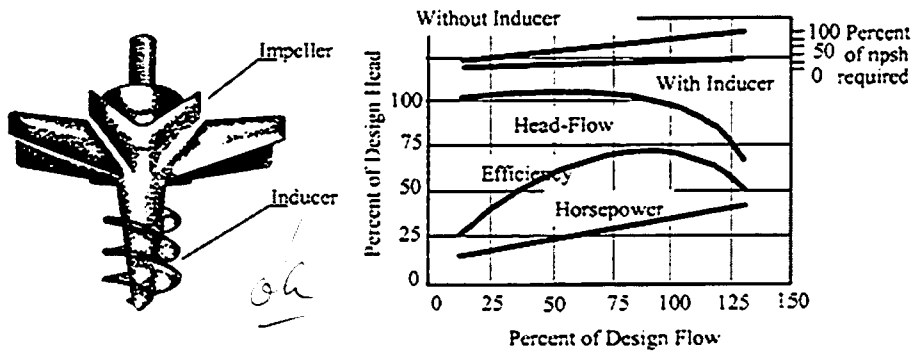


Sundyne with single gear between motor shaft and impeller shaft. This type gear box is good to 50 hp.



Sundyne with idler shaft gear box. This type gear box is good to 200 hp. Note the internal circulating oil system for the bearings and gears. Also, note the internal cyclone separator seal flush system.

Impeller and Inducer



EDS/PO-141
PO-R00-50

Many Sundynes have inducers which lower the NPSH required. The inducer is a spiral mechanism which bolts on to the inlet of the impeller. UOP does not recommend the use of inducers for horizontal pumps. There have been mixed results when inducers are applied to horizontal pumps but they work well on Sundynes.

No wasted utilities

TROUBLESHOOTING GUIDE	
TROUBLE	POSSIBLE CAUSES
Liquid not delivered	Pumps not primed
	Air or vapor pocket in suction line
	Pump not up to rated speed
	Wrong rotation
Failure to deliver rated capacity and pressure	Impeller or passages clogged
	Available npsh not sufficient
	Pump not up to rated speed
	Wrong rotation
	Impeller or passages partially clogged
	Wear rings worn or impeller damaged
	Air or gases in liquid
	Viscosity or specific gravity not as specified
	Air or vapor pocket in suction line
	Air leak in stuffing box
	Total head greater than head for which pump designed
	Injection of low vapor pressure oil in lantern and ring of hot pump

TROUBLESHOOTING GUIDE	
TROUBLE	POSSIBLE CAUSES
Pump loses prime	Air leak in suction line
	Air leak in stuffing box
	Air or gases in liquid
Pump overheats driver	Speed too high
	Specific gravity or viscosity too high
	Packing too tight
	Misalignment
	Total head lower than rated head
	Low voltage or other electrical trouble
	Trouble with engine, turbine, gear or other allied equipment
Pump vibration	Available npsh not sufficient
	Air or gases in liquid
	Misalignment
	Worn bearings
	Damaged rotating element
	Foundation not rigid
	Pump operating below minimum recommended capacity
	Impeller clogged

TROUBLESHOOTING GUIDE	
TROUBLE	POSSIBLE CAUSES
Stuffing box overheats	Packing too tight
	Packing not lubricated
	Incorrect type packing
	Gland cocked
Bearings overheat or wear rapidly	Incorrect oil level
	Misalignment or piping strains
	Insufficient cooling water
	Bearings too tight or preloaded
	Oil rings not functioning
	Suction pressure appreciably different than specified
	Improper lubrication
	Vibration
	Dirt or water in bearings

Purchasing Pumps Technical Evaluation

1. Does it meet flow and head?
2. Check completed API data sheets line by line Does it meet the spec?
3. % of BEP
4. L^3/D^4 → John sheet

ECS-PO-93

D-R
flow
Eigens
Proven Pumps.

— experience
— after sales service

Notes Oper & Maint & chg

Purchasing Pumps Technical Evaluation

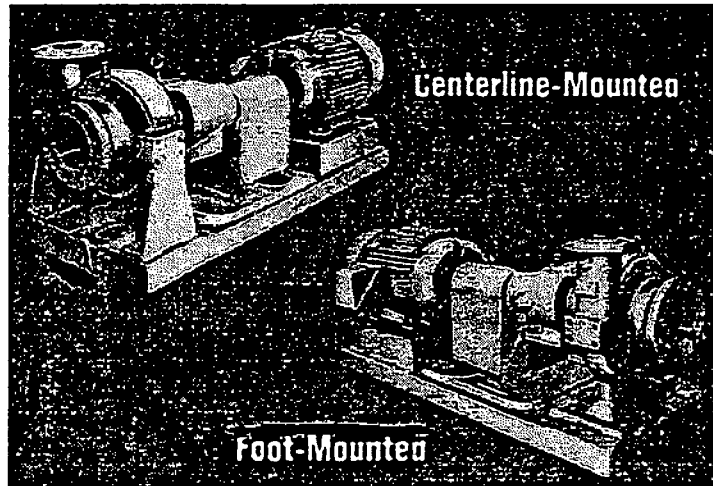
5. **Suction Specific Speed**
6. **Seals**
7. **Materials**
8. **Efficiency**
- 9 **Exceptions to Specs and Standards**
10. **Experience**

EDS/PO-64

ASME vs. API

	ASME	API
Pump Type	End Suction Single Stage Only	End or Top Suction Single or Multi-Stage
Dimensions	Standardized	Non-Standardized
Impeller	Open or Closed	Closed
Std. Materials	Cast Iron or 316SS	Steel or 11-13% CR
Stuffing Box	Packed or Mechanical Seal	Mechanical Seal
Support	Foot	Centerline
Minimum Bearing Life	17,500 HR	25,000 HR
Wearing Rings	Optional	Required
Flanges	CL. 125 or CL. 150 Min.	CL. 300 Min.
Running Clearances	None Specified	Specified
Shaft Runout	0.05 MM	0.025 MM
Temperature Limits	500°F 300°F for Cast Iron	800°F
Throat Bushing	Optional	Required
Cost	30 to 50% of API	

ASME vs. API



ASME vs. API

- API is engineered product
- ASME is off-the-shelf ✓
- 100 GPM, 200 FT
 - API \$14,000, ASME \$6000
- 600 GPM, 300 Ft
 - API \$30,500, ASME \$14,500
- Affect ~20% of pumps

ASME vs API

- API 8th Edition
- Nonflammable, nonhazardous
- Max discharge pressure 275 psig
- Max suction pressure 75 psig
- Max temp 300 F
- Max speed 3600 rpm
- Max head 400 ft
- Max impeller diameter 13 in

ASME vs API

- API 9th Edition (Released this Spring)
- Same limitations
- Removed nonflammable, nonhazardous

ASME vs API

- Allowed (endorsed) by API
- Major pump vendor says no difference in reliability in allowable range
- Much UOP experience (non-hydrocarbon)
- Cheaper

API 610

5-11-5

ASME B 73.1

5-21-1

ECS/PD-143

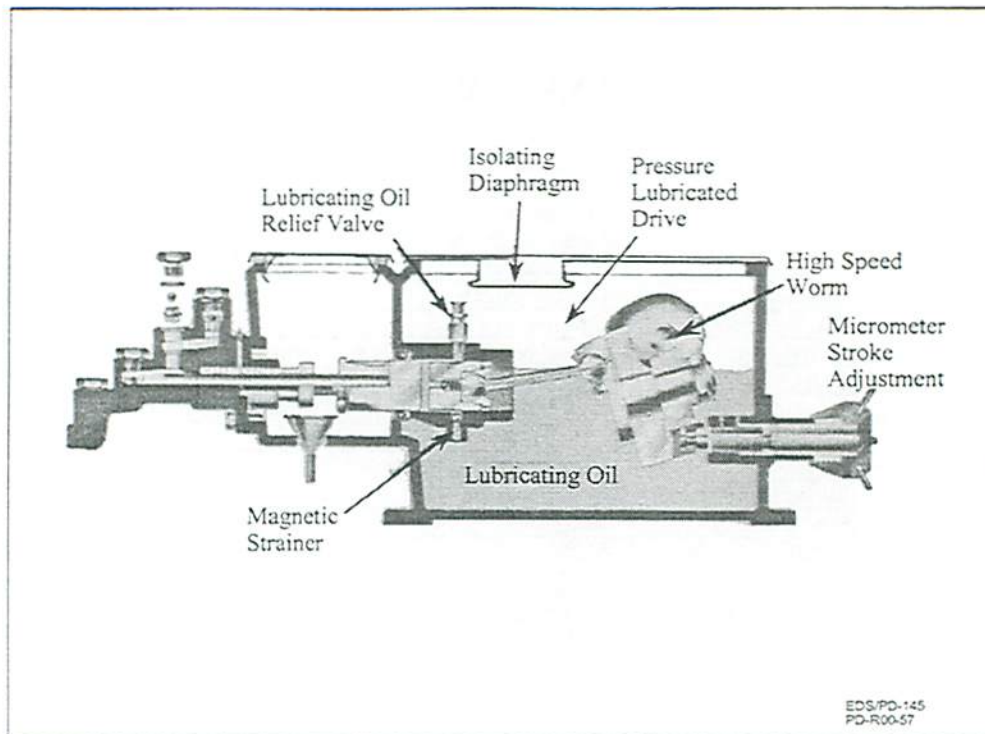
Hydrocarbon pumps are designed to API 610. Non-flammable pumps (water, caustic) are typically designed to ASME B73.1.

ASME vs. API

	ASME	API
Pump Type	End Suction Single Stage Only	End or Top Suction Single or Multi-Stage
Dimensions	Standardized	Non-Standardized
Impeller	Open or Closed	Closed
Std. Materials	Cast Iron or 316SS	Steel or 11-13% CR
Stuffing Box	Packed or Mechanical Seal	Mechanical Seal
Support	Foot	Centerline
Minimum Bearing Life	17,500 HR	25,000 HR
Wearing Rings	Optional	Required
Flanges	CL. 125 or CL. 150 Min.	CL. 300 Min.
Running Clearances	None Specified	Specified
Shaft Runout	0.05 MM	0.025 MM
Temperature Limits	500°F 300°F for Cast Iron	800°F
Throat Bushing	Optional	Required
Cost	30 to 50% of API	

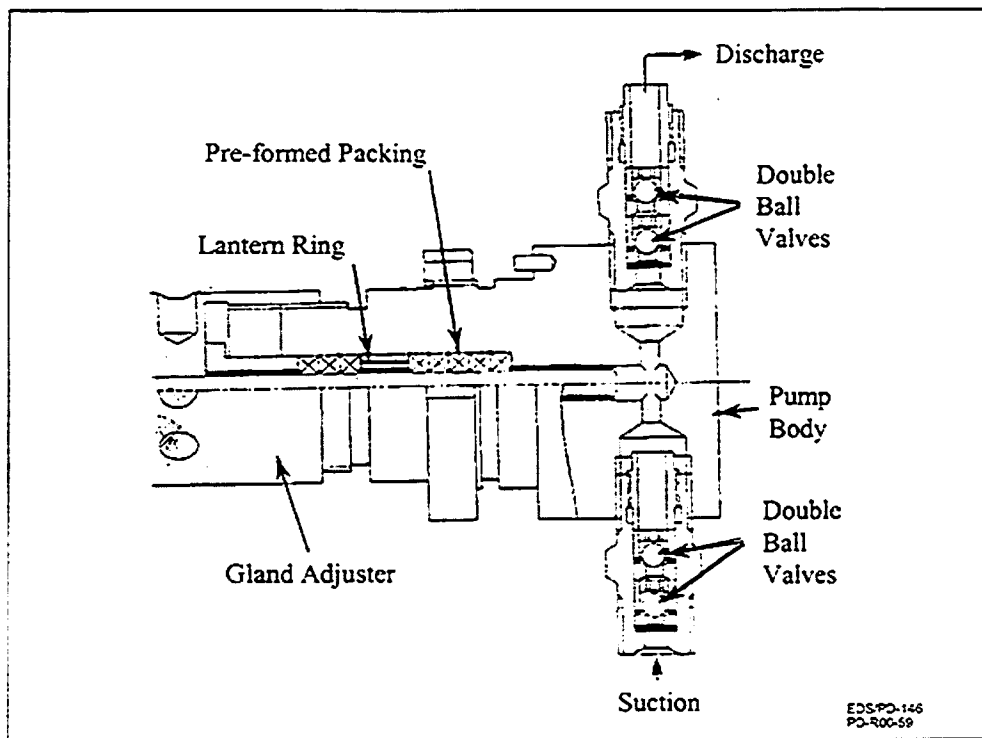
EDS/PO-144

The ASME pump is a light duty pump used for non-hazardous, non-flammable services. It is not a custom engineered pump. UOP accepts ASME pumps for low flow, low pressure water, and caustic services. They cost about 30% of API pumps. Also, dimensional equivalence among manufacturers is obtained as the ASME standard lists dimensions of major parts for the various model sizes. This can reduce spare parts.



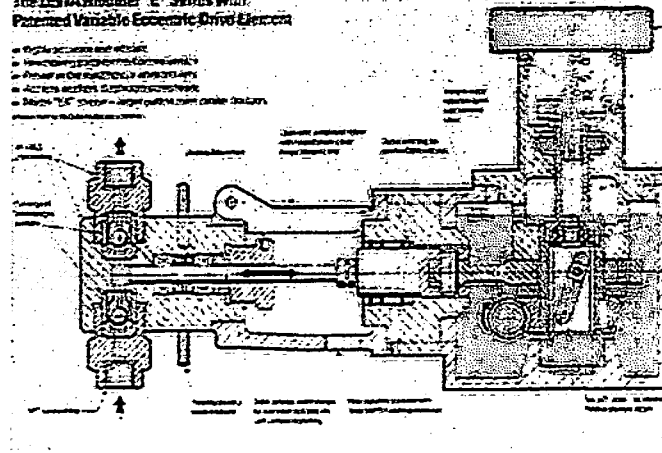
Proportioning pump. Proportioning pumps are used for low flow injection type services. They are positive displacement. There is no pump curve. Stroke speed is about 100 strokes/min. The flow rate can be adjusted by changing the stroke length. This can be done manually or automatically. Proportioning pumps can have rates down to 20cc/hr. Relief valves are required because the pump will always try to overcome the discharge pressure. If blocked in, the pump could be damaged.

constant flow for changing masses

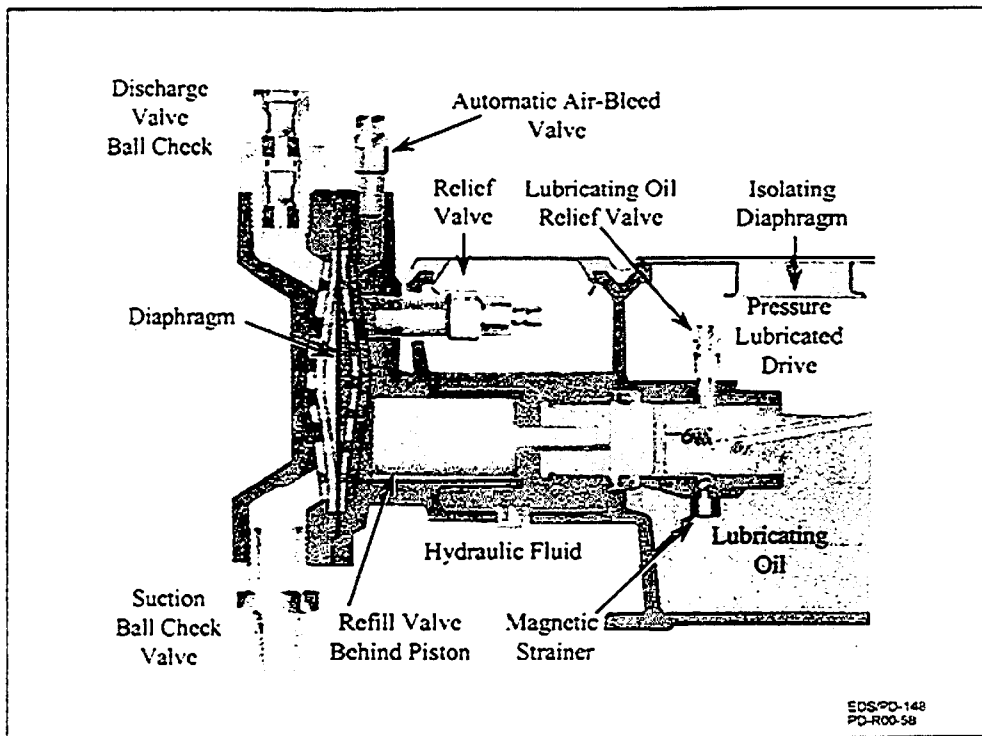


Liquid end of a plunger type proportioning pump. Ball valves are free to open and close with differential pressure. Process suction pressure forces the suction valves open and the cavity fills with liquid. The plunger squeezes the liquid to a higher pressure. When the pressure inside the cavity is greater than the discharge pressure, the discharge valves open, discharging the fluid. Most proportioning pumps require pulsation bottles in the discharge piping. The bottles dampen the pulsations caused by the pump.

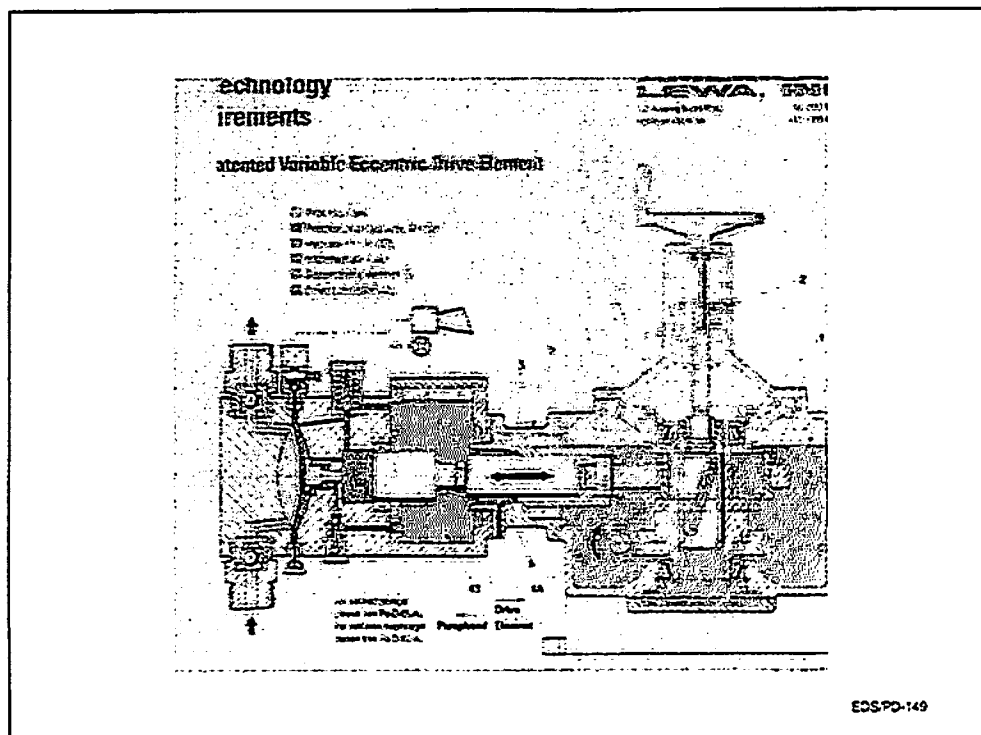
- Highly accurate test of choice
- Increasingly popular for the following reasons
- Powerful in the hands of the clinician
- Accurate results
- More "user friendly" - larger quality control panels



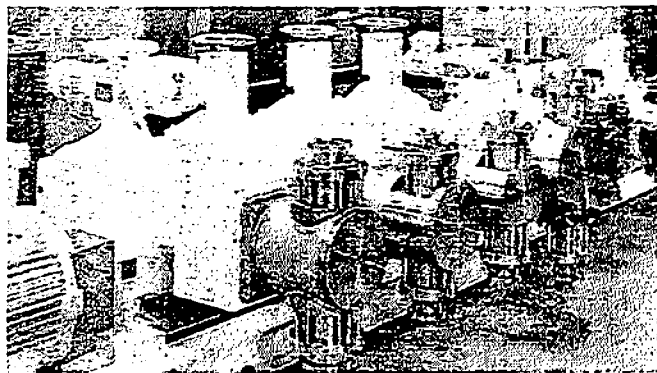
Plunger type proportioning pump.



This is a diaphragm type proportioning pump. The plunger-type (previous 2 slides) is sealed with packing, which leaks. The diaphragm type has no process fluid leakage.



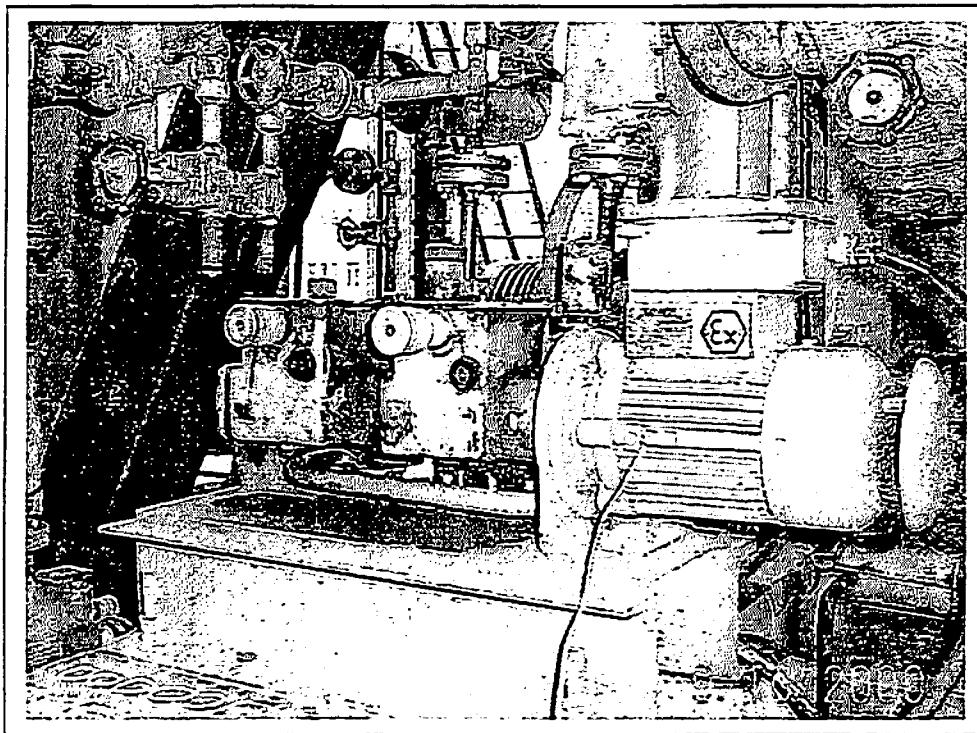
Diaphragm type proportioning pump.



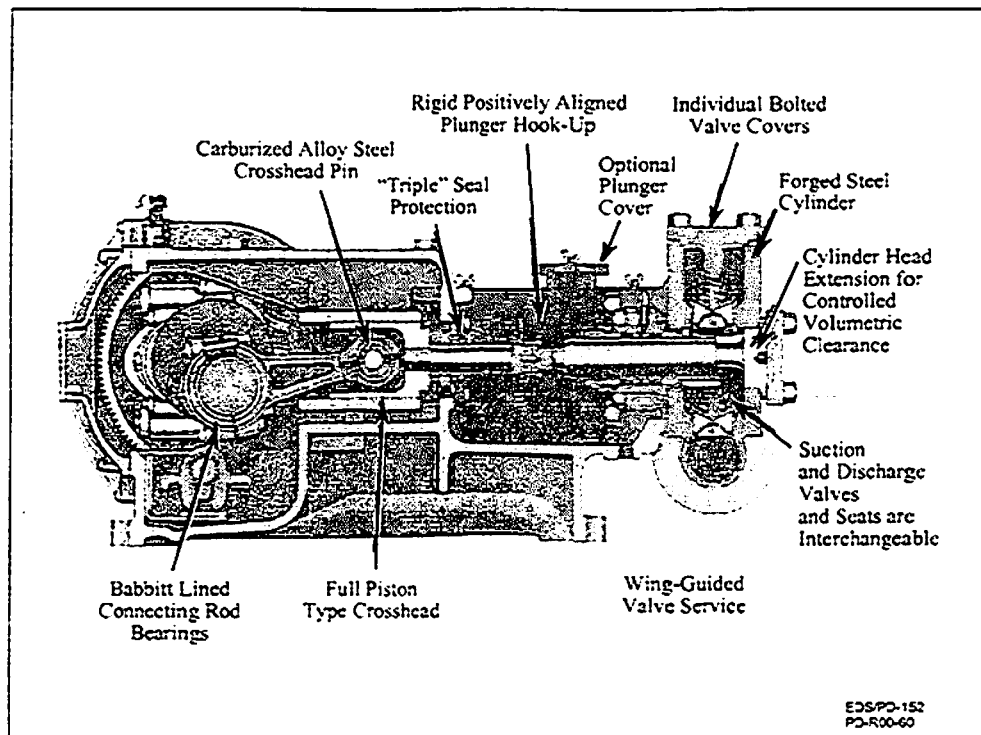
Four proportioning pumps on a single driver.

EDSPO-150

Four proportioning pumps on a single driver.



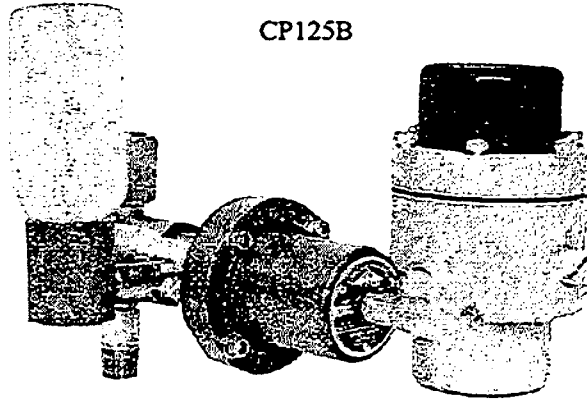
Platforming chlorination/water injection pump. Note the separate stroke adjustments for each pumping head.



Reciprocating pump. Reciprocating pumps are used for services requiring very high pressures such as ram and test pumps or high pressure wash water pumps. They are positive displacement. The only capacity control is with spillback or variable speed.

Oscillamatic™

CP125B



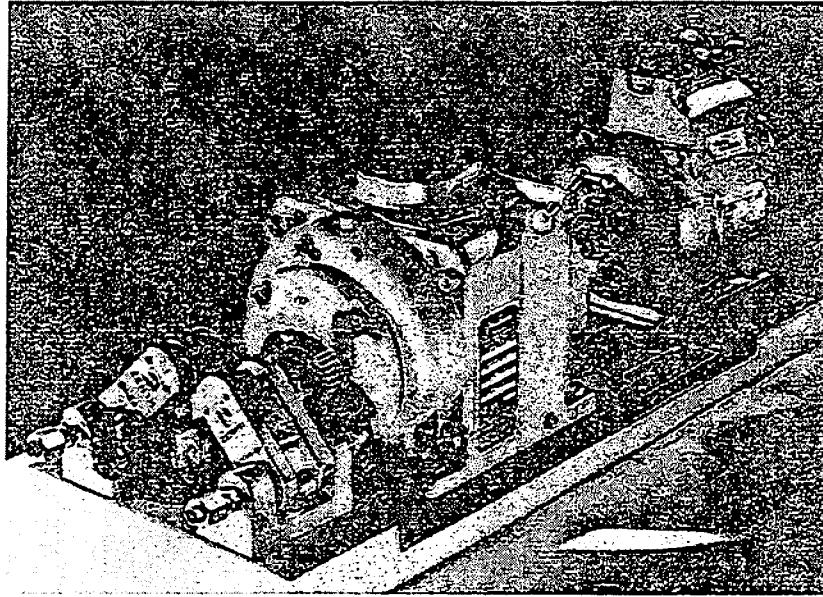
Pneumatic Chemical Injectors
Extremely Low Volume
Highly Accurate

EDS-PQ-153
PQ-R00-61

Pneumatic Chemical Injection pump. These proportioning pumps are air driven. There is no driver. They are used for very low flow services.



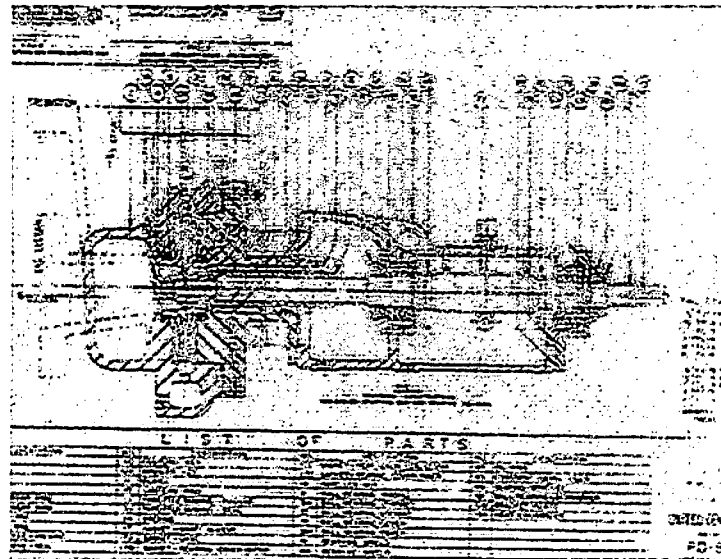
154



EDS/PO-155
PO-R00-63

Variable speed gear type pump. Used for low flow, viscous fluids. The speed can be adjusted in the gear box to adjust the capacity.

Traditional Slurry Pump



Note the wearing ring and throat bushing flushes. The areas of close clearances are flushed with heavy cycle oil to eliminate any catalyst fines.

from 1000
Boiler

MCB/Circulating Pump

■ Slurry Pump

■ HCO flush to wearing rings, throat bushing, seals (LCO backup) 4-6 gpm

■ Diffusion coating wetted parts

■ Two turbine drivers ✓

■ Maximum size impellers

■ Packing/mech. seal - now two mech. seals

■ Last 3 years - Lawrence pump alternate

EDS-PD-157

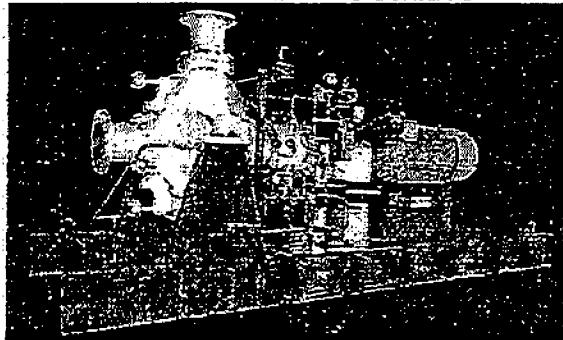
↓ erosion

CB Product Pump

- No flush due to product guarantees*
- Normal process pump with diffusion coating of wetted parts*
- ☒ *Last 3 years Lawrence*
- ☒ *Seals are being flushed (2-3 gpm)*

EDS/PO-158

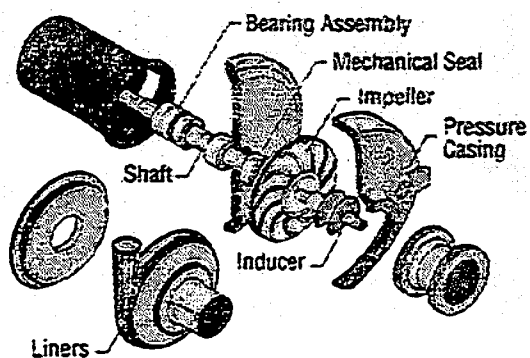
**Hydrocarbon Processing Industry
Slurry Pumps
Series 6000**



**LOWE'S
PUMP**

EDS/PO-159

LPI Fully Lined Pump Technology

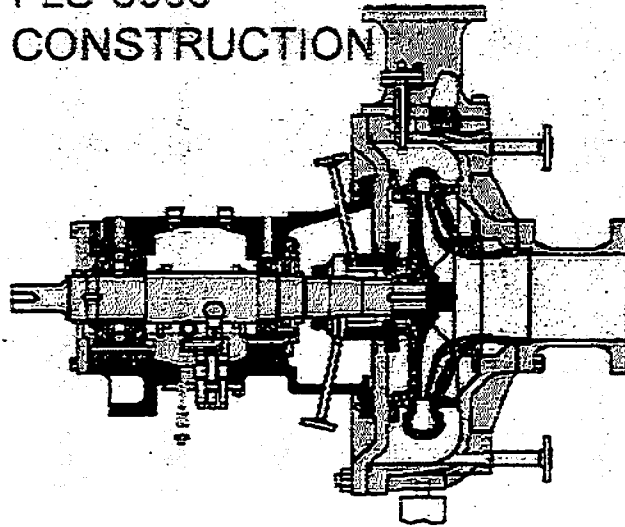


LPI LAWRENCE PUMPS INC.

EOS-PD-160

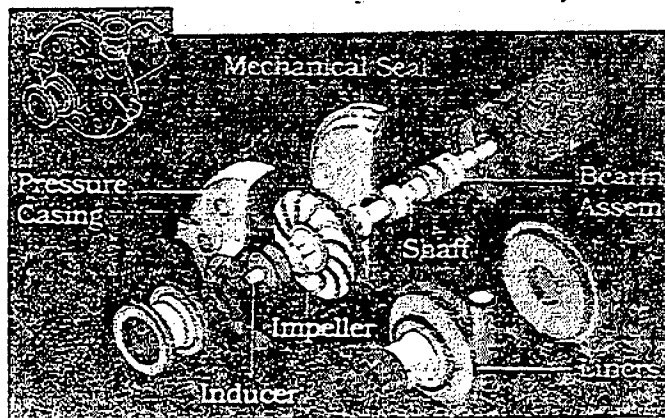
Massachusetts

FLS-5000
CONSTRUCTION



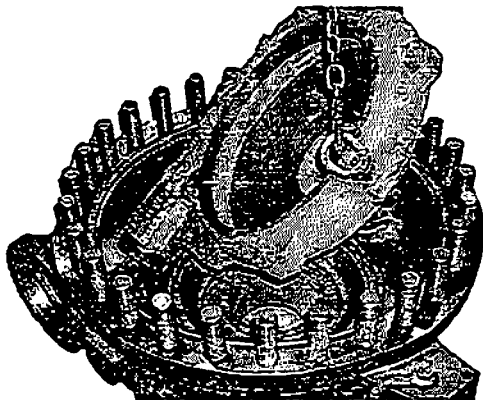
EDS-PO-181

API-610 Fully Lined Pump



EOS-PD-162

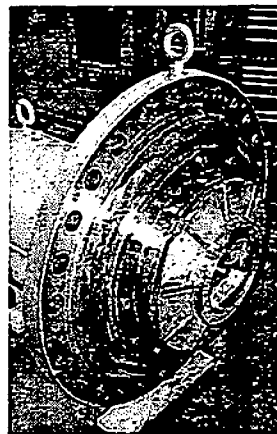
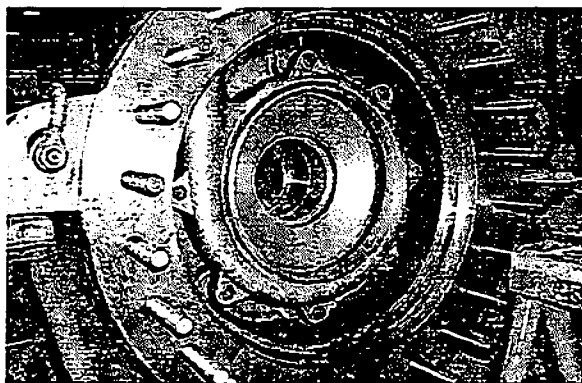
Replaceable Liners



- **Material**
 - 28 % Chrome Iron
- **Tolerances**
 - Each Liner has Machined Fits
- **Installation**
 - Each Liner is Bolted into Place

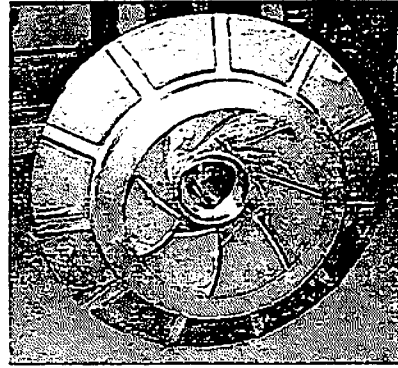
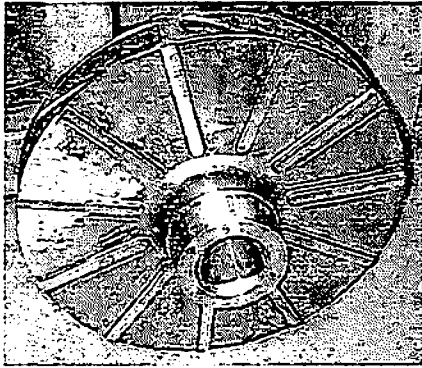
EOS-PO-163

Liners in Pressure Casing



EDS-PO-164

Slurry Impeller Design



EDSPD-165

**LAWRENCE
POMPS INC.**

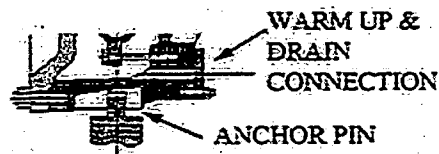
CASING LINER MATERIAL

- Casing Liner Material is an abrasion resistant iron-nickel-chromium-molybdenum alloy which has been specifically developed for erosive slurry applications
- Liner Hardness can be increased from 350 BHN to over 600 BHN by using a special heat treatment process. (Casings on unlined pumps cannot be hardened to adjust for wear)
- Surface Finish of the Liners is controlled by machining. Finished machined surfaces of the liners opposite the impeller helps to reduce the gouging action caused by the vortices which form in the spiraling inflow from the volute.

EDS/PO-168

Casing Features

- Casing drain is located outside of high velocity area of the volute so that it is protected from erosive wear.
- A warm up connection is located at the bottom of the casing to ensure even heating of the pump during start up.
- A 2" anchor pin is provided at the bottom of the casing to maintain pump alignment under full API nozzle loads & process temperature changes.



ECS-PD-167

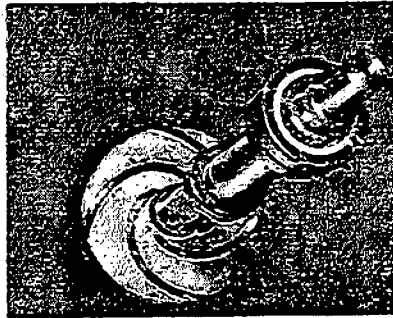
Series 6000 Slurry Pumps Rotating Element

Each shaft is engineered to provide the necessary space for single, double or tandem seal arrangements and a conservative L/D³ deflection index.

Impeller Arrangements

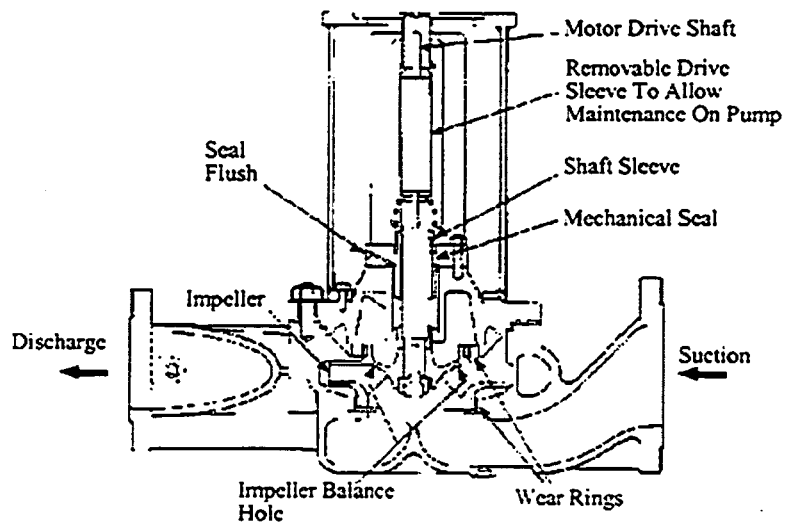
The standard LPI Slurry Pump is built with an enclosed impeller with front and back repeating vanes. Optional casing and impeller wear rings are available.

For unique applications involving highly viscous fluids, large diameter solids or fibrous material, LPI offers Open, Semi-Open and Recessed impeller designs.



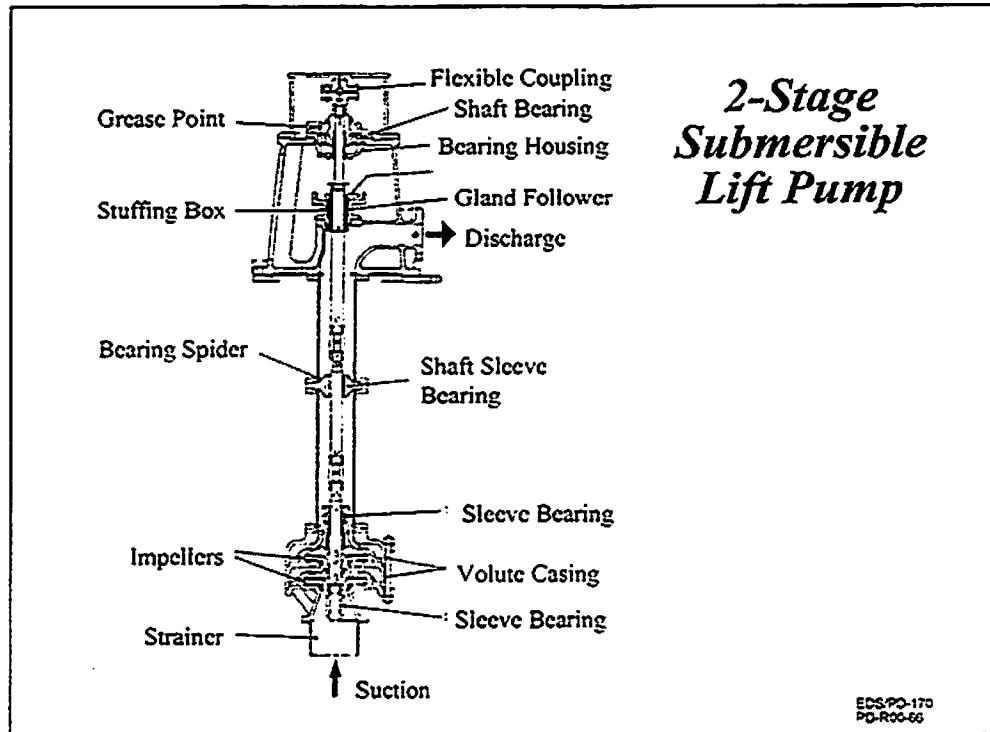
EDS/PO-168

Vertical Inline Pump



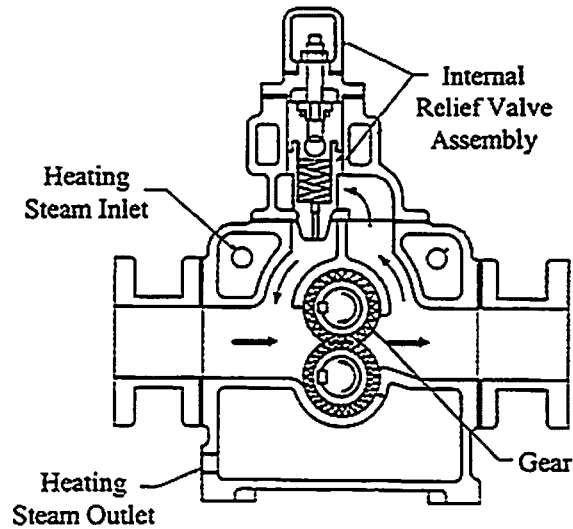
EDS/PO-169
PD-R30-65

Vertical Inline pump. This pump is available in API or ASME models. They are perceived to be more difficult to maintain than a horizontal pump. This is because the motor must be removed to remove the impeller. They take up much less space than a horizontal pump so, if plot space is a concern, vertical pumps could be considered.



Sump Pump. Sump pumps are on-off services. When the sump level is high, the pump evacuates the sump down to the minimum level and the pump automatically shuts down.

Gear Pump



EDS/PD-171
PD-R00-67

Gear pump. Used for high viscosity fluids such as lube oil. This is a positive displacement pump. As in the screw pump, the process fluid provides lubrication for the rotors.

↑
dan



UOP LLC • 25 East Algonquin Road • Des Plaines, Illinois 60017-5017 USA

CENTRIFUGAL PUMPS AND DRIVERS

PROJECT SPECIFICATION

#REF! - 501

SHEET 5

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV

APPLICABLE TO: ☐ PROPOSALS ☐ PURCHASE ☒ AS BUILT

FOR _____ UNIT _____
SITE _____ SERVICE _____
NO. REQ _____ PUMP SIZE _____ TYPE _____ ☒ NO. STAGES _____
MANUFACTURER _____ MODEL _____ SERIAL NO. _____

NOTES: INFORMATION BELOW TO BE COMPLETED: ☐ BY PURCHASER ☐ BY MANUFACTURER ☒ BY MANUFACTURER OR PURCHASER

GENERAL (3.1.1)

PUMPS TO OPERATE IN (PARALLEL) _____ NO. MOTOR DRIVEN _____ NO. TURBINE DRIVEN _____
(SERIES) WITH _____ PUMP ITEM NO. _____ PUMP ITEM NO. _____
GEAR ITEM NO. _____ MOTOR ITEM NO. _____ TURBINE ITEM NO. _____
GEAR PROVIDED BY _____ MOTOR PROVIDED BY _____ TURBINE PROVIDED BY _____
GEAR MOUNTED BY _____ MOTOR MOUNTED BY _____ TURBINE MOUNTED BY _____
GEAR DATA SHT. NO. _____ MOTOR DATA SHT. NO. _____ TURBINE DATA SHT. NO. _____

OPERATING CONDITIONS

☐ CAPACITY, NORMAL _____ RATED _____
OTHER _____
☐ SUCT PRESS MAX./RATED _____ / _____

☐ DISCHARGE PRESSURE _____
☐ DIFFERENTIAL PRESSURE _____
☐ DIFF. HEAD _____ NPSHA _____
☐ PROCESS VARIATIONS _____ (3.1.2)
☐ STARTING CONDITIONS _____ (3.1.3)
SERVICE: ☐ CONT. ☐ INTERMITTENT (STARTS/DAY) _____
☐ PARALLEL OPERATION REQ'D (2.1.11)

SITE AND UTILITY DATA

LOCATION: (2.1.29)
☐ INDOOR ☐ HEATED ☐ UNDER ROOF
☐ OUTDOOR ☐ UNHEATED ☐ PARTIAL SIDES
☐ GRADE ☐ MEZZANINE _____
ELECTRIC AREA CLASSIFICATION (2.1.22 / 3.1.5)
CL _____ GR _____ DIV _____
☐ WINTERIZATION REQ'D ☐ TROPICALIZATION REQ'D

SITE DATA (2.1.29)

ALTITUDE _____ BAROMETER _____ 0
RANGE OF AMBIENT TEMPS: MIN/MAX _____ / _____
RELATIVE HUMIDITY: MIN / MAX _____ / _____ %
UNUSUAL CONDITIONS: (2.1.23) ☐ DUST ☐ FUMES
OTHER _____
UTILITY CONDITIONS:

STEAM: DRIVERS HEATING
MIN _____ 0 _____ 0 _____ 0
MAX _____ 0 _____ 0 _____ 0
ELECTRICITY DRIVERS HEATING CONTROL SHUTDOWN
VOLTAGE _____
HERTZ _____
PHASE _____

COOLING WATER: (2.1.17)

TEMP. INLET _____ MAX. RETURN _____
PRESS. NORM _____ DESIGN _____
MIN RETURN _____ MAX ALLOW DP _____ 0
FOULING FACTOR _____

SITE AND UTILITY DATA (CONT'D)

WATER SOURCE _____
CHLORIDE CONCENTRATION (PPM) _____ (3.5.2.6)
INSTRUMENT AIR: MAX/MIN PRESS _____ / _____

LIQUID

☐ TYPE OR NAME OF LIQUID _____
☐ PUMPING TEMPERATURE _____
NORMAL _____ MAX. _____ MIN. _____
☐ VAPOR PRESSURE _____ @ _____
☐ RELATIVE DENSITY (SPECIFIC GRAVITY):
NORMAL _____ MAX _____ MIN _____
☐ SPECIFIC HEAT _____
☐ VISCOSITY _____ @ _____
☐ MAX. VISCOSITY _____ 0
☐ CORROSIVE/EROSIVE AGENT _____ (2.11.1.8)
☐ CHLORIDE CONCENTRATION (PPM) _____ (3.5.2.6)
☐ H₂S CONCENTRATION (PPM) _____ (2.11.1.11)
LIQUID (2.1.3) ☐ HAZARDOUS ☐ FLAMMABLE
☐ OTHER _____

PERFORMANCE

PROPOSAL CURVE NO. _____ RPM _____
☐ IMPELLER DIA. RATED _____ MAX. _____ MIN. _____ mm
☐ RATED POWER _____ EFFICIENCY _____ %
☐ MINIMUM CONTINUOUS FLOW:
THERMAL _____ 0 STABLE _____ 0
☐ PREFERRED OPERATING REGION _____ TO _____ 0
☐ ALLOWABLE OPERATING REGION _____ TO _____ 0
☐ MAX HEAD @ RATED IMPELLER _____ 0
☐ MAX POWER @ RATED IMPELLER _____ 0
☐ NPSHR AT RATED CAPACITY _____ 0 (2.1.8)
☒ SUCTION SPECIFIC SPEED _____ (2.1.9)
☐ MAX. SOUND PRESS. LEVEL REQ'D _____ (dBA) (2.1.14)
☐ EST MAX SOUND PRESS. LEVEL _____ (dBA) (2.1.14)

REMARKS: _____
ESTIMATED SHUTOFF PRESSURE _____ 0

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CENTRIFUGAL PUMPS AND DRIVERS

PROJECT SPECIFICATION

#REF! - 501

SHEET 6

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV

CONSTRUCTION

APPLICABLE STANDARD:

- ☐ API 610 8TH EDITION
☐ OTHER _____ (SEE REMARKS)

PUMP TYPE: (1.1.2)

- | | | | |
|------------------------------|------------------------------|------------------------------|--------------------------------|
| ☐ OH2 | ☐ BB1 | ☐ VS1 | ☐ VS6 |
| ☐ OH3 | ☐ BB2 | ☐ VS2 | ☐ VS7 |
| ☐ OH6 | ☐ BB3 | ☐ VS3 | ☐ OTHER |
| | ☐ BB4 | ☐ VS4 | |
| | ☐ BB5 | ☐ VS5 | |

☐ NOZZLE CONDITIONS: (2.3.2)

	SIZE (IN)	FLANGE RATING (LBS)	FACG	POSITION
SUCTION				
DISCHARGE				
BALANCE DRUM				

PRESSURE CASING CONNECTIONS: (2.3.3)

	NO.	SIZE (NPS)	TYPE
☐ DRAIN			
☐ VENT			
☐ PRESSURE GAUGE			
☐ TEMP GAUGE			
☐ WARM-UP			
☐ BALANCE / LEAK-OFF			

☐ CYLINDRICAL THREADS REQUIRED (2.3.3)

CASING MOUNTING: (SEE SEPARATE SHEET FOR VERTICALS)

- | | |
|-------------------------------------|--|
| ☐ CENTERLINE | ☐ NEAR CENTERLINE |
| ☐ FOOT | ☐ SEPARATE MOUNTING PLATE |
| ☐ IN-LINE | ☐ SUMP |

CASING SPLIT:

- ☐ AXIAL ☐ RADIAL

CASING TYPE:

- | | | |
|--|---|-----------------------------------|
| ☐ SINGLE VOLUTE | ☐ MULTIPLE VOLUTE | ☐ DIFFUSER |
| ☐ OVERHUNG | ☐ BETWEEN BEARINGS | ☐ BARREL |

CASE PRESSURE RATING:

- ☐ MAX ALLOWABLE WORKING PRESSURE _____ 0

@ _____ 0

- ☐ HYDROTEST PRESSURE _____ 0

- ☐ SUCTION PRESS REGIONS MUST BE DESIGNED FOR MAWP (2.2.4)

ROTATION: (VIEWED FROM COUPLING END)

- ☐ CW ☐ CCW

- ☐ IMPELLERS INDIVIDUALLY SECURED (5.2.2.2)

REMARKS:

- ☐ BOLT OH3 PUMP TO PAD/FOUNDATION (5.1.2.4)

SHAFT:

- ☐ SHAFT DIAMETER AT COUPLING _____ mm

REMARKS:

SUCTION TYPE

IMPELLER TYPE

CONSTRUCTION (CONT)

- | | |
|--|----------|
| ☐ SHAFT DIAMETER BETWEEN BEARINGS | _____ mm |
| ☐ SPAN BETWEEN BEARING CENTERS | _____ mm |
| ☐ SPAN BETWEEN BEARING & IMPELLER | _____ mm |

REMARKS

COUPLINGS: (3.2.2)

DRIVER - PUMP

- ☐ MAKE _____
- ☒ MODEL _____
- ☐ RATING KW/100RPM _____
- ☐ LUBRICATION _____
- ☒ LIMITED END FLOAT REQUIRED _____
- ☒ SPACER LENGTH _____ mm
- ☒ SERVICE FACTOR _____

DRIVER HALF COUPLING MOUNTED BY: (3.2.11)

- ☐ PUMP MFR ☐ DRIVER MFR ☐ PURCHASER
- ☐ COUPLING PER API 671 (5.2.7)

BASEPLATES:

API BASEPLATE NUMBER _____ (APPENDIX M)

- ☐ NON-GROUT CONSTRUCTION (3.3.13/5.3.7.3.5)

REMARKS:

MATERIAL

- ☐ APPENDIX H CLASS _____ (2.11.1.1)
- ☐ MIN DESIGN METAL TEMP (2.11.4.5) _____
- ☐ BARREL/CASE _____ IMPELLER _____
- ☐ CASE/IMPELLER WEAR RINGS _____
- ☐ SHAFT _____
- ☐ DIFFUSERS _____
- ☐ COUPLING SPACER/HUBS _____
- ☐ COUPLING DIAPHRAGMS (DISKS) _____

REMARKS:

BEARINGS AND LUBRICATION

BEARING (TYPE/NUMBER):

- ☐ RADIAL _____ / _____
- ☐ THRUST _____ / _____

- ☐ REVIEW AND APPROVE THRUST BEARING SIZE (5.2.5.2.4)

LUBRICATION: (2.10)

- | | | |
|---|---|--|
| ☒ GREASE | ☒ FLOOD | ☒ RING OIL |
| ☒ FLINGER | ☐ PURGE OIL MIST | ☐ PURE OIL MIST |
| ☐ CONSTANT LEVEL OILER PREFERENCE (SEE REMARKS) (2.9.2.2) | | |
| ☐ PRESSURE LUBE SYS (5.2.6) | ☐ API-610 | ☐ API-614 |
| ☒ OIL VISC. ISO GRADE (5.2.6.5) _____ | | |



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CENTRIFUGAL PUMPS AND DRIVERS

PROJECT SPECIFICATION

#REF! - 501

SHEET 7

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV

BEARINGS AND LUBRICATION (CONT)

- ☒ OIL HEATER REQ'D ☐ ELECTRIC ☐ STEAM (2.9.2.9/5.2.6.3)
☐ OIL PRESS TO BE GREATER THAN COOLANT PRESS (5.2.6.2.b)

REMARKS

MECHANICAL SEAL OR PACKING

SEAL DATA: (2.7.2)

- ☐ SEE ATTACHED API-682 DATA SHEET (2.7.2)
☐ NON-API 682 SEAL (2.7.2)
☐ APPENDIX H SEAL CODE _____ (2.11.1.1)
☒ SEAL MANUFACTURER _____
☐ SIZE AND TYPE _____ / _____
☒ MANUFACTURER CODE _____

SEAL CHAMBER DATA: (2.1.6/2.1.7)

- ☒ TEMPERATURE _____
☒ PRESSURE _____ 0
☒ FLOW _____ 0
☐ SEAL CHAMBER SIZE (TABLE 2.3)
☐ TOTAL LENGTH _____ mm ☐ CLEAR LENGTH _____ mm

SEAL CONSTRUCTION:

- ☐ SLEEVE MATERIAL _____
☐ GLAND MATERIAL _____
☐ AUX SEAL DEVICE (2.7.3.20) _____
☒ JACKET REQUIRED (2.7.3.17)

GLAND TAPS: (2.7.3.14)

- ☐ FLUSH (F) ☐ DRAIN (D) ☐
☒ QUENCH (Q) ☒ COOLING (C) ☒ LUBRICATION (G)
☒ HEATING (H) ☒ LEAKAGE ☒ PUMPED FLUID (P)
☒ BAL'NCE FLUID (E) ☒ EXTERNAL FLUID INJECTION (X)

SEAL FLUIDS REQUIREMT AND AVAILABLE FLUSH LIQUID:

NOTE: IF FLUSH LIQUID IS PUMPAGE LIQUID (AS IN FLUSH PIPING PLANS 11 TO 41), FOLLOWING FLUSH LIQUID DATA IS NOT REQ'D.

- ☐ SUPPLY TEMPERATURE MAX/MIN _____ / _____
☐ RELATIVE DENSITY (SPECIFIC GRAVITY) _____ @ _____

- ☐ NAME OF FLUID _____
☐ SPECIFIC HEAT, Cp _____ 0
☐ VAPOR PRESSURE _____ 0 @ _____
☐ HAZARDOUS ☐ FLAMMABLE ☐ OTHER _____
☐ FLOW RATE MAX/MIN _____ / _____ 0
☐ PRESSURE REQUIRED MAX/MIN _____ / _____ 0
☐ TEMPERATURE REQUIRED MAX/MIN _____ / _____

BARRIER FLUID (2.7.3.21):

- ☐ SUPPLY TEMPERATURE MAX/MIN _____ / _____
☐ RELATIVE DENSITY (SPECIFIC GRAVITY) _____ @ _____
☐ NAME OF FLUID _____

ADDITIONAL REMARKS:

MECHANICAL SEAL OR PACKING (CONT)

- ☐ VAPOR PRESSURE _____ 0 @ _____ 0
☐ HAZARDOUS ☐ FLAMMABLE ☐ OTHER _____
☐ FLOW RATE MAX/MIN _____ / _____ 0
☐ PRESSURE REQUIRED MAX/MIN _____ / _____ 0
☒ TEMPERATURE REQUIRED MAX/MIN _____ / _____ 0

QUENCH FLUID:

- ☐ NAME OF FLUID _____
☐ FLOW RATE _____ 0

SEAL FLUSH PIPING: (2.7.3.19 AND APPENDIX D)

- ☐ SEAL FLUSH PIPING PLAN _____
☒ TUBING ☐ CARBON STEEL
☐ PIPE ☒ STAINLESS STEEL
☐ AUXILIARY FLUSH PLAN _____
☒ TUBING ☐ CARBON STEEL
☐ PIPE ☒ STAINLESS STEEL
☐ PIPING ASSEMBLY: (3.5.2.10.1)
☒ THREADED ☒ UNIONS ☐ SOCKET WELDED
☒ FLANGED ☒ TUBE TYPE FITTINGS
☐ _____
☐ PRESSURE GAUGE (PLAN 52/53)
☐ _____
☐ LEVEL GAUGE (PLAN 52/53)
☐ TEMP INDICATOR (PLANS 21, 22, 23, 32, 41)
☐ HEAT EXCHANGER (PLAN 52/53)

REMARKS

INSTRUMENTATION

VIBRATION:

- NONCONTACTING (API 670) _____ TRANSDUCER _____
☐ PROVISION FOR MOUNTING ONLY (2.9.2.11)
☒ FLAT SURFACE REQ'D (2.9.2.12)
☐ SEE ATTACHED API 670 DATA SHEET
☐ MONITORS AND CABLES (3.4.3.3)

REMARKS

STEAM AND COOLING WATER PIPING

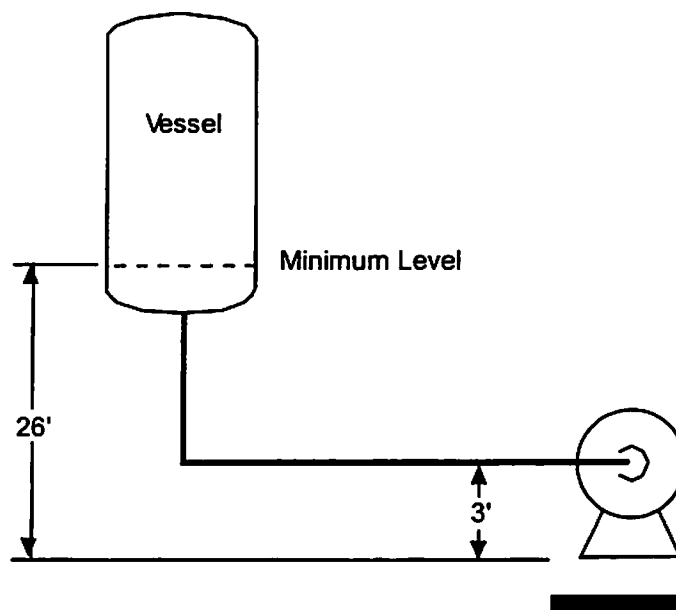
- ☒ COOLING WATER PIPING PLAN _____ (3.5.4.1)
☐ COOLING WATER REQUIREMENTS
SEAL JACKET/BRG HSG _____ 0 @ _____ 0
SEAL HEAT EXCHANGER _____ 0 @ _____ 0
QUENCH _____ 0 @ _____ 0
TOTAL COOLING WATER _____ 0
☐ STEAM PIPING: ☐ TUBING ☐ PIPE

REMARKS

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PROBLEMS

Problem I



Liquid is at bubble point. Friction loss is 2 psig. (6 feet)

$$npsha = (26 - 3) - 6 = 17 \text{ feet}$$

Two existing pumps (operating and spare) Byron Jackson $4 \times 6 \times 13 \frac{1}{4}$ L (curve attached)

Present pump duty	Flow	600	gpm
	Head	500	ft
	npshr	14	ft
	Sp Gr	0.80	
	Temperature	150°F	
	60 Cycle		

New conditions require flow to be increased to 780 gpm.

Now need 18 ft npsh. What can be done?

Problem II

Pump is specified for 1200 gpm, 450 feet of head, 60 cycle.

Quotations are received from three vendors as follows:

Byron Jackson	$4 \times 6 \times 13 \frac{1}{4}$ H	Single Suction, 3550 rpm
United	H x 6 x 13	Double Suction, 3550 rpm
Ingersoll-Dresser	6x26J3B	Double Suction, 1750 rpm

Pump curves are attached.

What are positive and negative qualities of each pump?

Problem III

Required Pump Duty	159 m ³ /h	Rated Capacity
	145 m ³ /h	Normal Capacity
	7.03 kg/cm ² (g)	Suction Pressure
	49.22 kg/cm ² (g)	Discharge Pressure
	0.95	Specific Gravity
	149°C	Suction Temperature
	Water	Fluid Handled
	4.0 Meters	npsh Available
	50 Cycle	

1. Make the best pump selection from the attached curves.
2. How many stages?
3. What is the efficiency?
4. What is the horsepower (kw)?
5. What is the required npsh?



Rotating Equipment Training

COMPRESSORS

Presentation Material

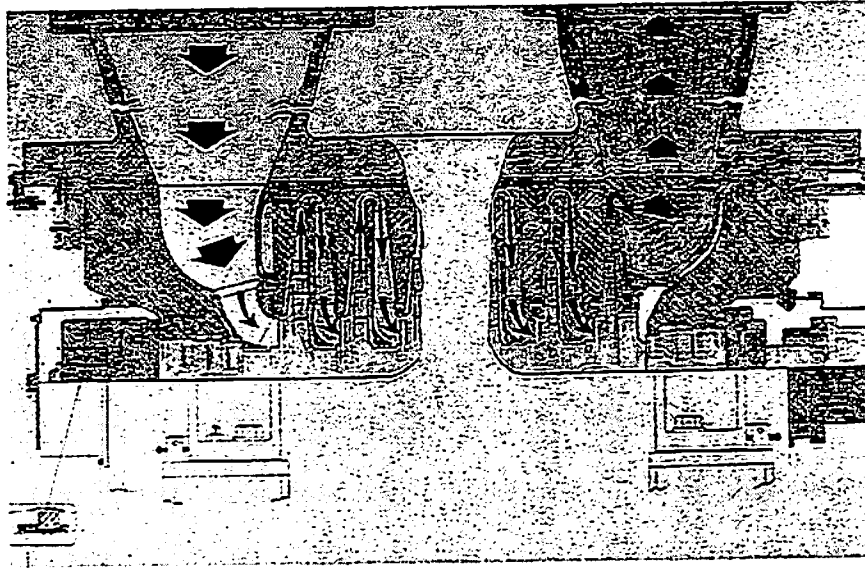
Pertamina

June 23-27, 2003

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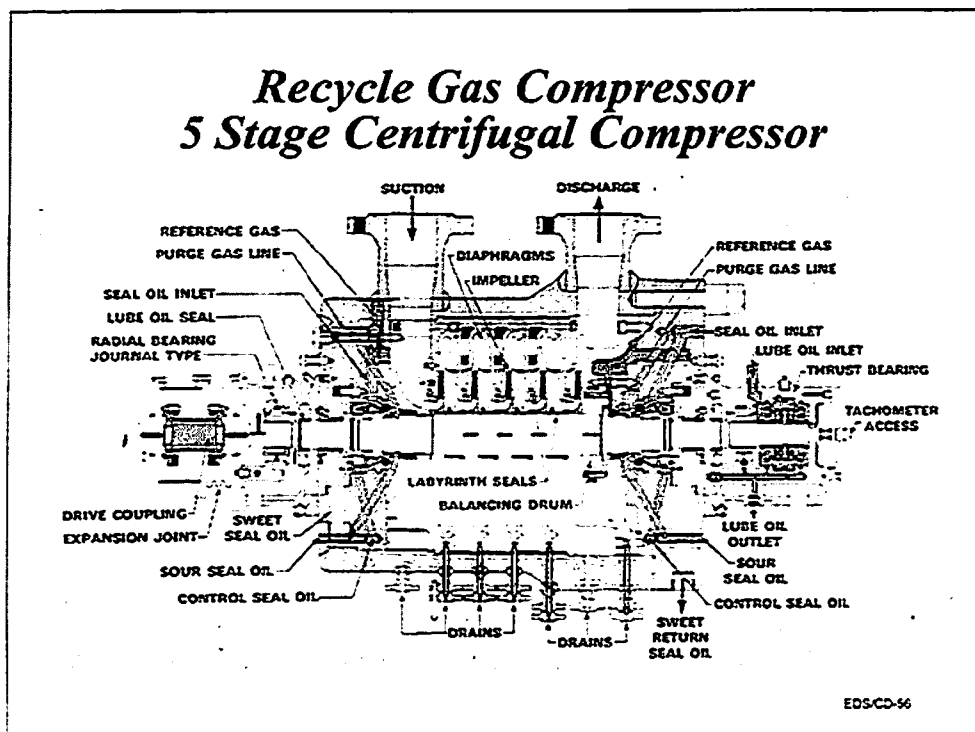
Compressor Assembly



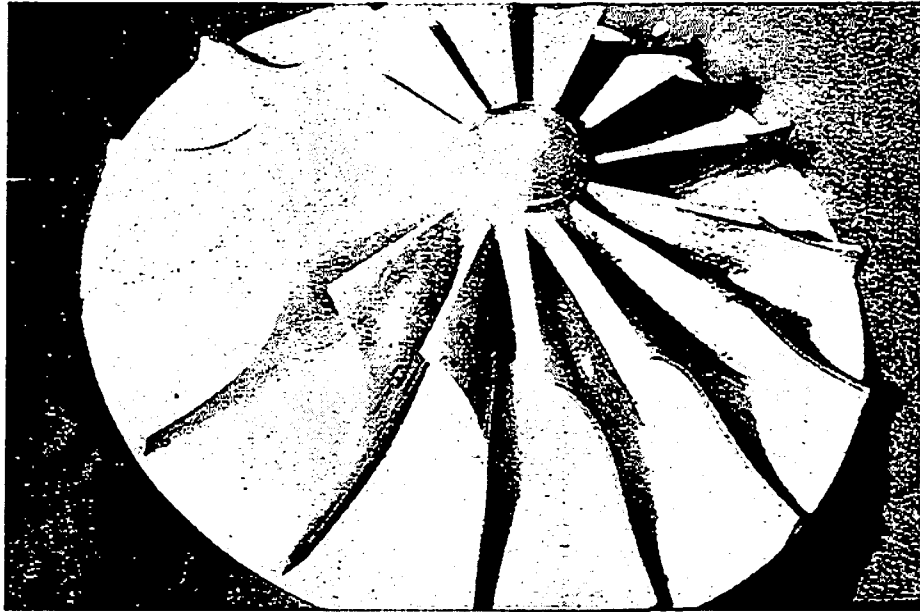
EDS-CO-55

Diaphragms guide the gas from the discharge of one impeller to the suction of the next impeller.

Recycle Gas Compressor 5 Stage Centrifugal Compressor



Five-stage centrifugal compressor. Note the seal and lube oil piping connections. The labyrinth seals minimize the flow of gas back to a lower pressure impeller.



EDSCO-57

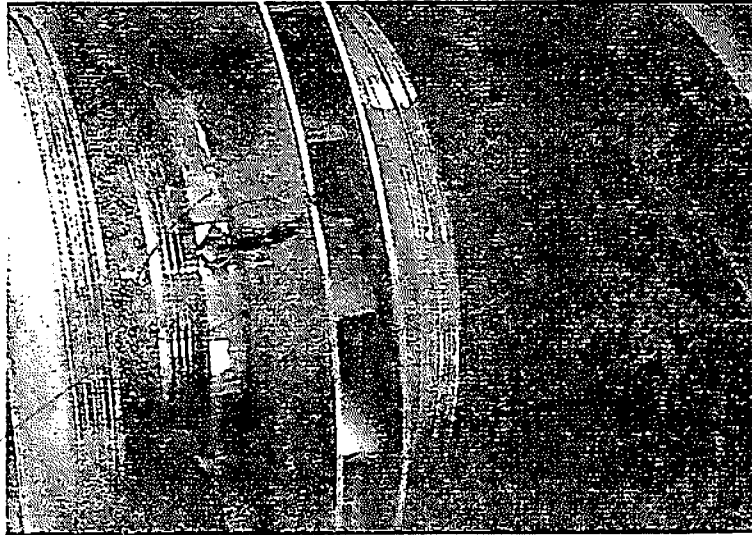
Semi-open centrifugal compressor wheel. Note the curved shape of the blades. This curvature helps define the performance curve shape.



EDS/CD-58

Compressor impeller.

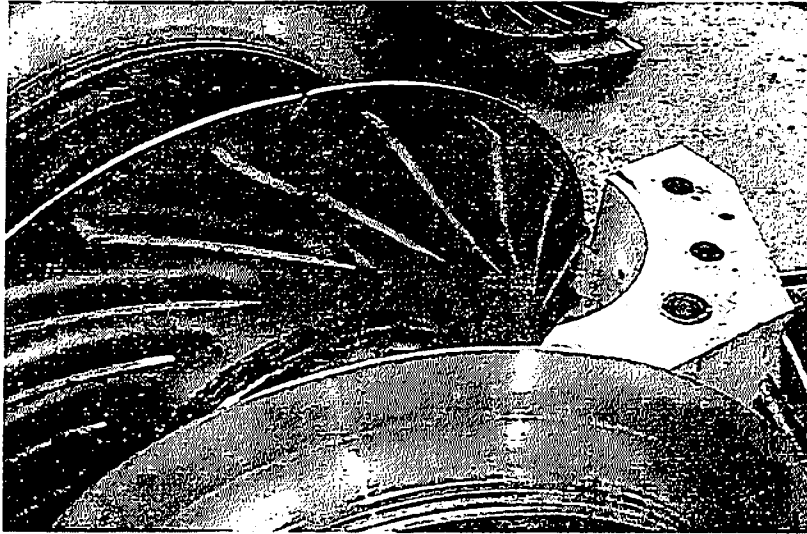
Stress Corrosion Crack



EDS-CD-56

Riveted impeller. This is an old manufacturing technique, now obsolete. Today, most impellers are machined. The cover is then welded in place. The stronger wheels result in higher achievable performance per stage.

Stress Corrosion Crack



EDS-CD-60

Riveted impeller. This is an old manufacturing technique, now obsolete. Today, most impellers are machined. The cover is then welded in place. The stronger wheels result in higher achievable performance per stage.

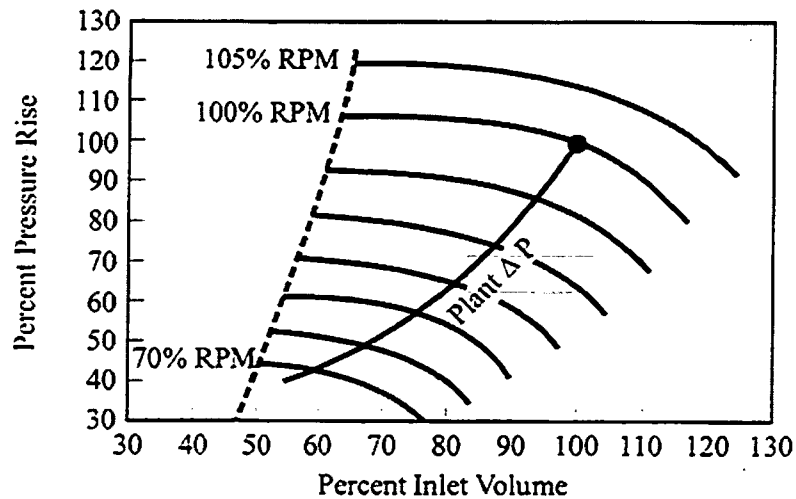
Centrifugal Compressor Control

- Variable Speed
- Suction Throttle Valve

EDSCO-61

Centrifugal compressors are controlled with variable speed or for a single speed driver, suction throttling.

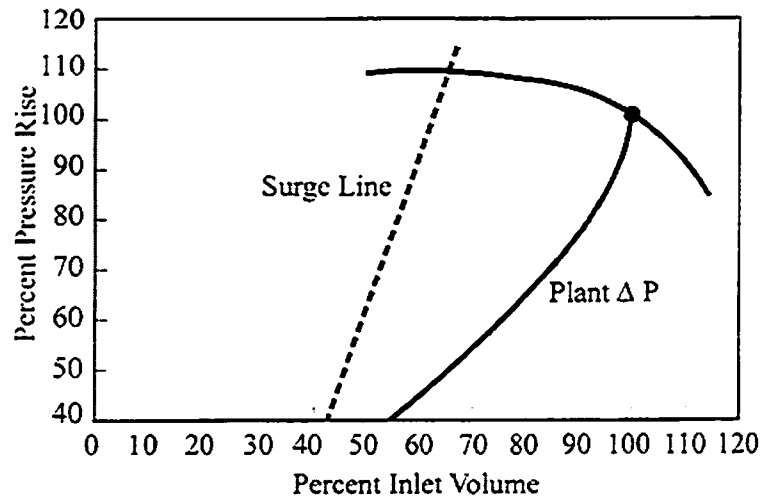
Centrifugal Compressor Typical Variable Speed Performance Curves



EDS/CO-62
CO-R20-12

For a variable speed driver, steam turbine, or variable speed motor, the compressor speed can be varied to meet the head requirements along the system resistance curve. Most variable speed compression trains can operate between 70% and 105% of the design speed.

Centrifugal Compressor Typical Constant Speed Performance Curve

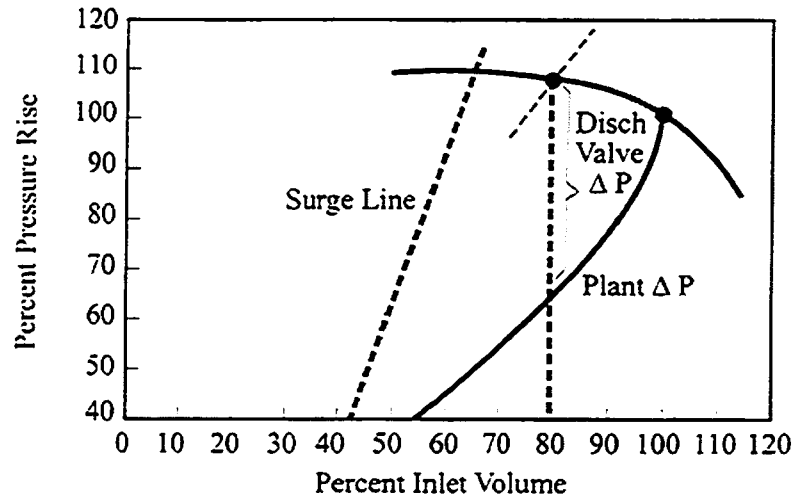


EDS-CO-63
CO-200-13

When a compressor is motor driven, there is only one head-capacity curve. When operating at off-design cases, pressure must be throttled over a control valve.

Variable speed motor > better

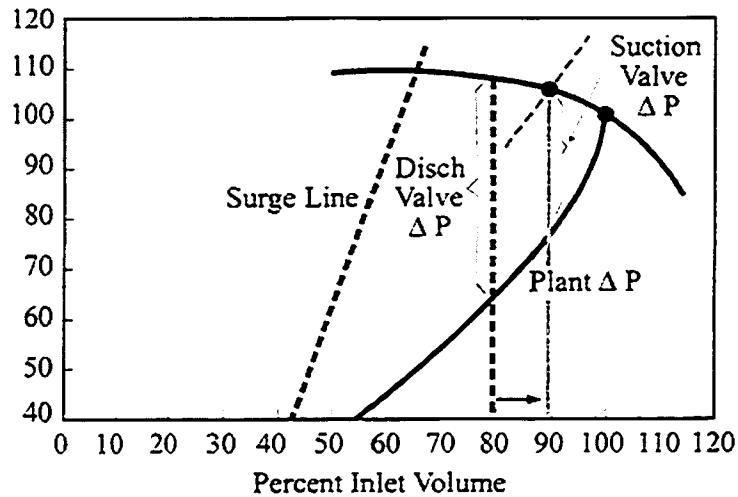
Centrifugal Compressor Typical Constant Speed Performance Curve



EDS/CD-64
CD-R00-14

The throttle valve can go upstream or downstream of the compressor. If the valve goes on the discharge, the volumetric flow rate, ACFM, is directly proportional to the mass flow, lbs/hr. 80% lbs/hr will be achieved at 80% ACFM. Note that at 80% ACFM, almost half the head produced by the compressor is throttled across the control valve. Since HP is a function of mass flow time head, almost half the HP requirement of the compressor is wasted across the control valve.

Centrifugal Compressor Typical Constant Speed Performance Curve



EDS/CD-65
CD-R00-15

If the throttle valve is on the compressor suction, not only does the mass flow decrease as the valve is throttled, but the suction pressure also decreases. ACFM is inversely proportional to suction pressure. With the valve on the suction, 80% mass flow can be achieved at about 90% ACFM. Compared to discharge throttling, less pressure is taken across the control valve and less HP is required for the same mass flow. Therefore, most centrifugal compressors with single speed drivers have suction throttle valves.

Valve → Down
Upstream

$$\frac{1 \text{ kg}}{\text{hr}} \times H$$

$$\sim \frac{\text{NM}^3}{\text{hr}} \times \text{MW}$$

$$\frac{\Delta P^2}{\text{hr}} \sim \frac{\text{NM}^3}{\text{hr}}$$

Total Flow

Suction Throttling

	Rated	TD Turb	TD Disch	TD Suct
SCFD	60 MM	48	48	48
FT	85300	68560	92130	89570
ACFM	22200	17750	17750	19750
BHP	2764	1785	2399	2331

EDS/CD-66

DRIVERS

ELECTRIC MOTORS

(SSM) (VSM)

fixed
speed

Variable
speed

Gas turbines, Gas engines

STEAM TURBINES

Cond Back Pressure

W/C A/C

Extraction, Induction

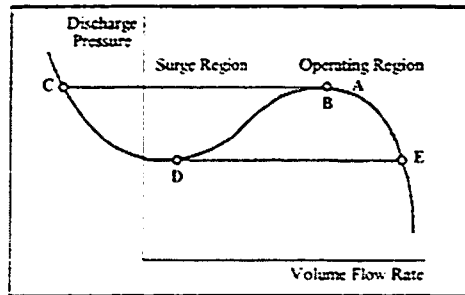
EDS-CO-67

5/72
hoggy ejector → pull vacuum
↓
- then small ejector to maintain

A Model of Surge

■ Surge Definition

Surge is self-oscillations of pressure and flow, including a flow reversal. The surge flow reversal is the only point of the curve when pressure and flow drop simultaneously.



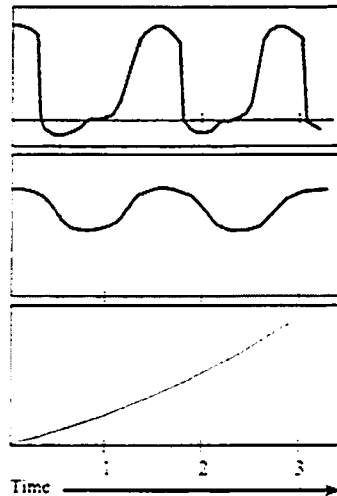
Characteristic Curve of a Typical Centrifugal Compressor

- B to C in 20 to 50 mSec
- Cycle B to B at 0.33 to 3 Hertz

EDS-CD-66
CD-870-20

All centrifugal compressors can and surge. If the compressor rides up its curve due to rising discharge pressure or decreasing gas molecular weight, the compressor will physically not be able to overcome the downstream pressure requirements. Since gas is compressible, pressure and energy will build up downstream. When the compressor curve reaches point B, the compressor will no longer be able to push the gas out the discharge. The gas will then reverse flow through the compressor. Flow at point C is negative. Now that the downstream pressure has been relieved through the compressor, the compressor can start pushing gas out the discharge again. When the pressure builds up and the compressor can no longer keep up with the demand, the gas once again goes back through the compressor to the suction. This can happen multiple times per second.

The Surge Phenomena



- Rapid flow oscillations
- Thrust reversals
- Potential damage
- Rapid pressure oscillations with process instability
- Rising temperatures inside compressor

EDS/CD-69
CD-R00-33

The consequences of surge are severe. The thrust reversals on the shaft will damage seals, bearings, and open up critical internal clearances. Since the same gas is passing through the compressor multiple times a second, the temperature in the compressor rises rapidly.

Surge Description

- Flow reverses in 20 to 50 milliseconds
- Surge cycles at a rate of 1/3 to 3 hertz
- Compressor vibrates
- Temperature rises
- "Whooshing" noise
- Trips may occur
- Conventional instruments and human operators may fail to recognize surge

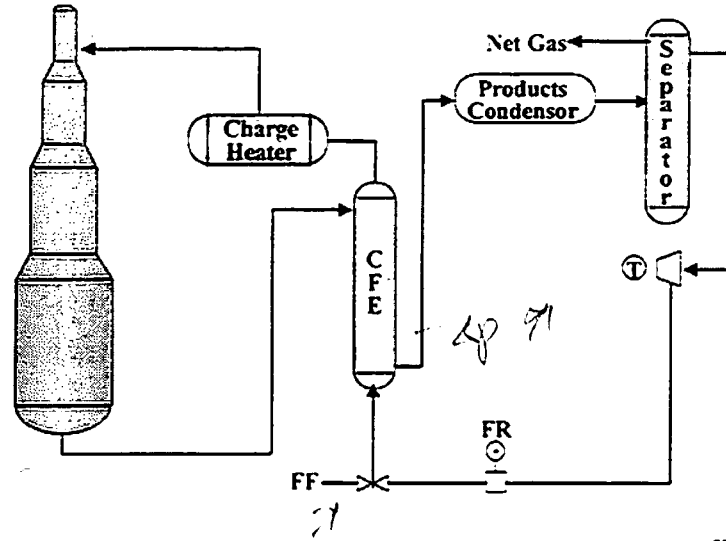
EOS-CD-70

Some Surge Consequences

- **Unstable flow and pressure**
- **Damage in sequence with increasing severity to seals, bearings, impellers, shaft**
- **Increased seal clearances and leakage**
- **Lower energy efficiency**
- **Reduced compressor life**

EDS-71

Platforming Recycle Gas Circuit

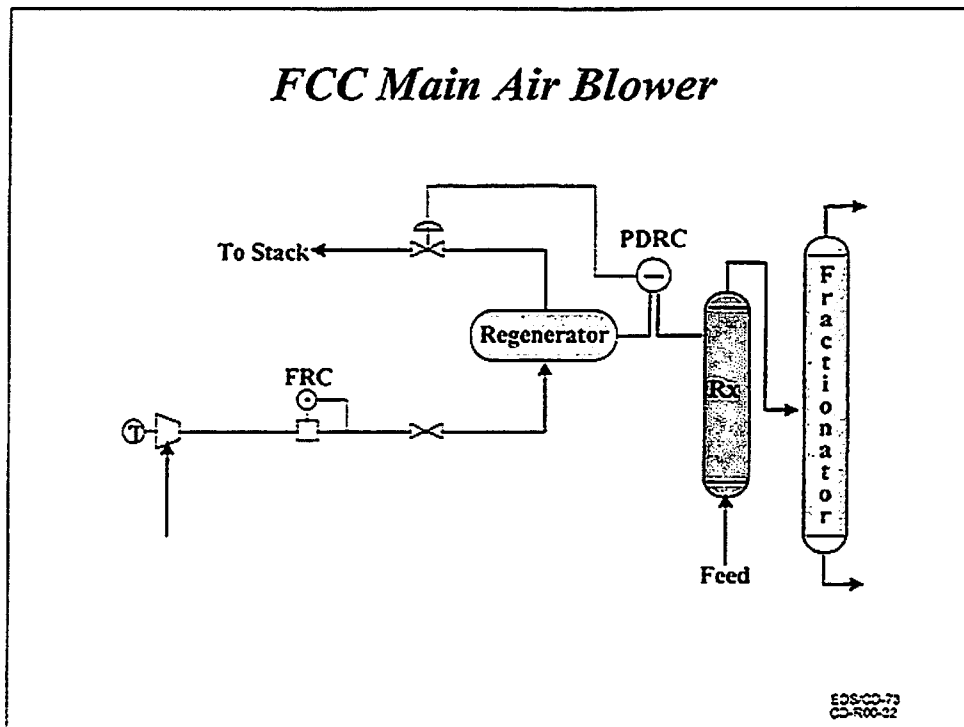


This is a UOP Platforming recycle gas circuit. Note that this is a circulating loop circuit. The compressor discharges into the compressor suction. There are no automatic controls on compressor speed and no automatic control valves. Assuming no blockages occur in the exchangers or reactor, which will only happen slowly over time, there is nothing in this circuit that can put the compressor into surge. Therefore, UOP recycle gas circuits do not require anti-surge control.

surge not necessary

low flow to the reactor

*105
106 ft*

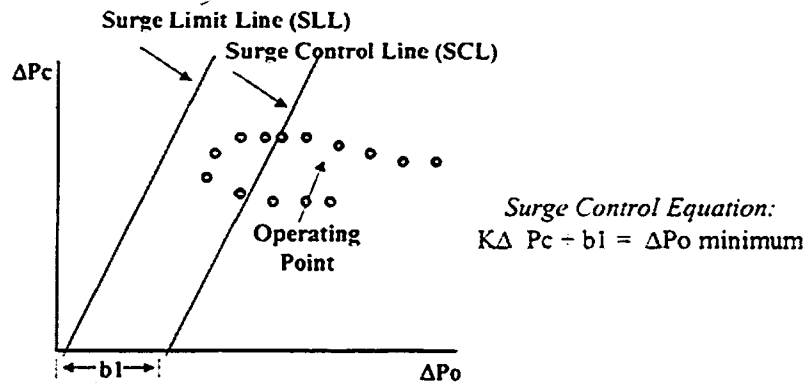


This is a FCC Main Air Blower. There are automatic valves which if not operating properly can put the blower into surge. Therefore, anti-surge equipment is specified.

WGC → surk

Net Gas @ Platform - can surge

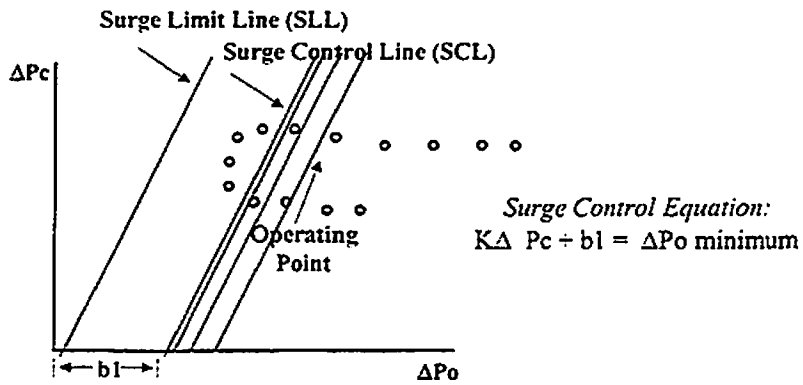
Antisurge Controller Operation



EDS-CD-74
CD-R00-33

The anti-surge controller measures gas pressure, temperature, and flow a minimum of 40 times per second. If the operating point hits the surge control line, the anti-surge controller sends a signal to open the spillback valve. This allows the gas downstream of the compressor an alternate path around the compressor instead of back through the compressor. A margin, $b1$, is left between the surge control line and the actual surge line. This ensures that the spillback valve will open in time to prevent the compressor from surging.

Antisurge Controller Operation



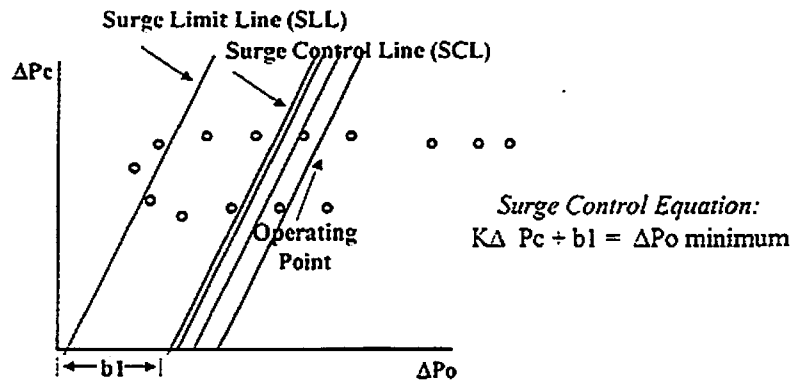
EDS-75
CD-R00-33

very quick. grill bad. open.

heat was just y's y's

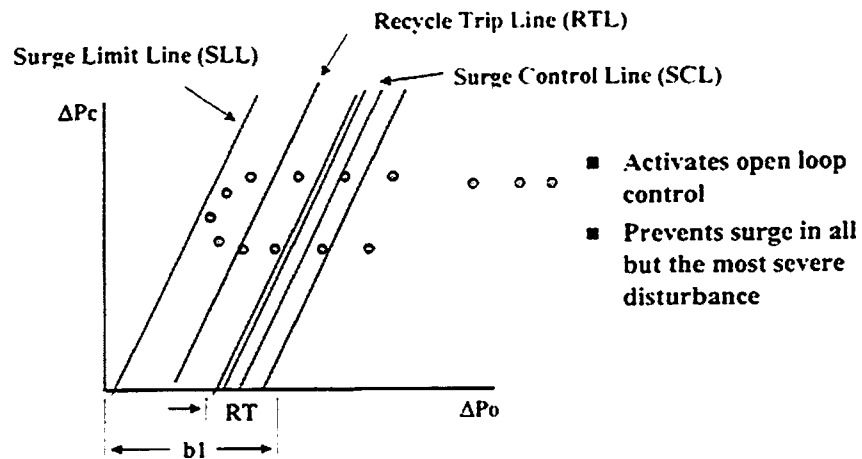
degrees and that up — and to ship

Antisurge Controller Operation



EDS/CD-76
CD-R00-33

Antisurge Controller Recycle Trip Circuit Operation

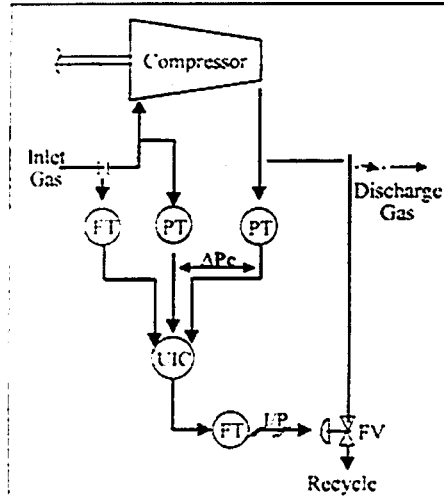


ESS-CD-77
CD-R00-33

The surge controller is also programmed with a recycle trip line. If the operating point hits the surge control line, the spillback valve opens slowly. It is not desirable for the spillback valve to go wide open because catalyst in the circuit could be starved of hydrogen.

If the operating point hits the recycle trip line, now the compressor is in danger of surging. The spillback valve will step open quickly to avoid surge.

Simple Antisurge System

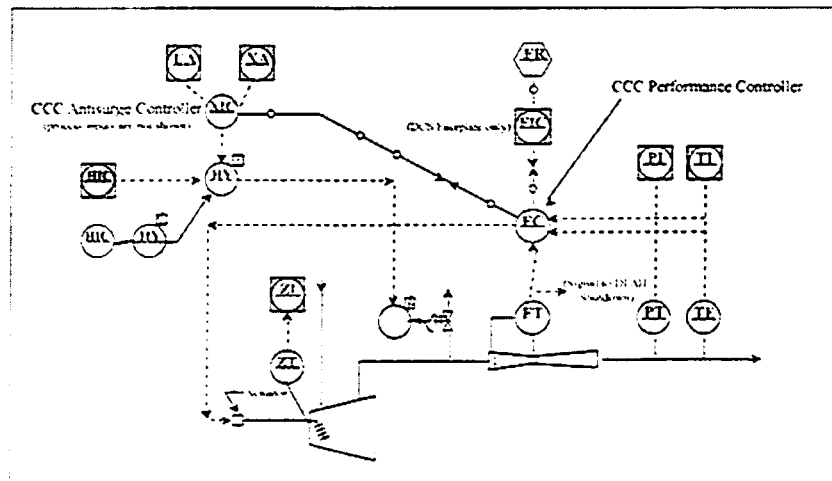


- Flow measured in suction (ΔP_o)
- Suction and discharge pressure transmitters for pressure differential calculation
- Control strategy is proportional plus integral control to manipulate recycle valve
- Increase flow through compressor and reduce discharge resistance when required to prevent surge

EDS-CD-73
CD-R00-25

In a simple antisurge system, flow and pressure measurements on the suction and pressure measurements on the discharge are sent to the antisurge controller, UIC. The UIC controls the recycle valve which is normally closed. If surge is approached, the UIC opens the recycle valve.

FCC Axial Main Air Blower with CCC Performance Controller

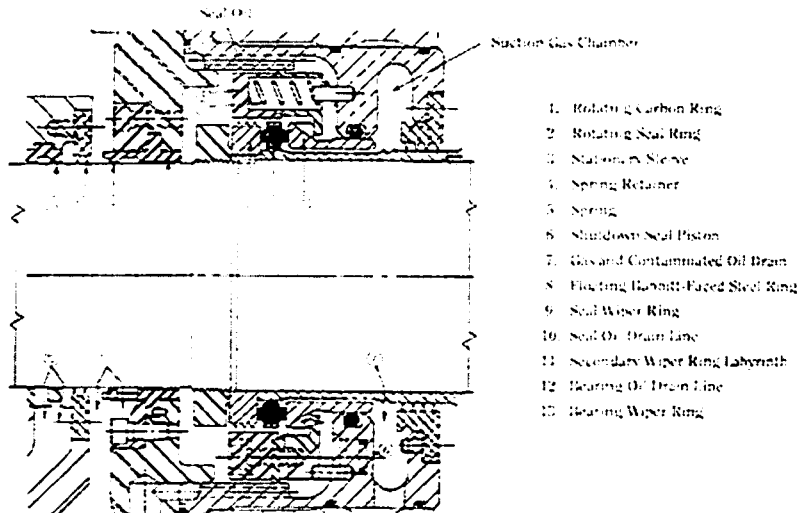


EDS-CD-76
CD-R00-34

This is an actual UOP FCC Main Air Blower P & ID. Note the flow, pressure, and temperature measurements transmitted to the performance controller which is controlling the flow of air to the regenerator. The performance controller is adjusting the axial air blower guide vanes or the turbine driver speed. The performance controller is also sending data to the antisurge controller, which is controlling the normally closed snort valve. If the surge line is approached, the antisurge controller will open the snort valve and uncouple the performance controller from the loop.

Occasionally, in a surge event, the performance controller could actually push the compressor further towards surge by speeding up or slowing down the compressor. With the antisurge and performance controllers "talking" to each other, this is prevented from happening.

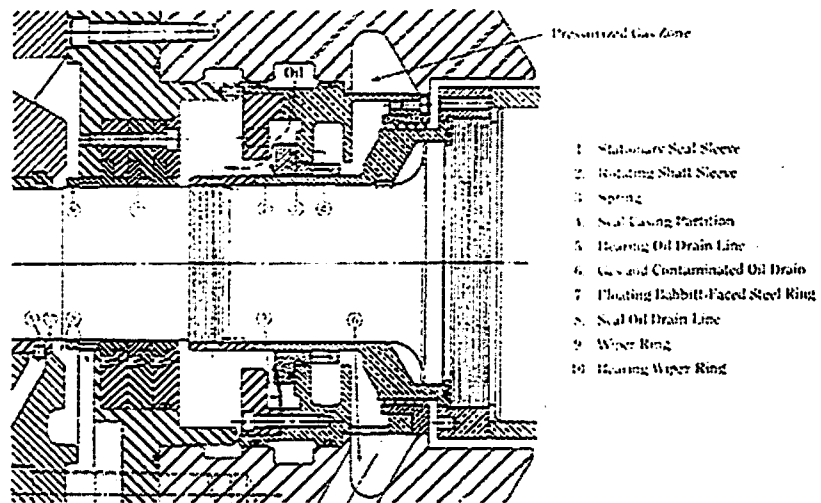
Mechanical Shaft Seal (TSO-Carbon Seal)



EDS-CD-80

Like the pump seal, the contact type compressor seal has a rotating member and a stationary member. A floating carbon bushing is between the two members. Process gas leaks across the labyrinth on the right side of the seal and fills a gas chamber. This gas is at a lower pressure than the process because of the pressure drop across the labyrinth. Seal oil is introduced on the bearing side of the seal faces. Most of the oil goes around the seal back to the reservoir. The oil pressure is maintained higher than the gas pressure in the gas chamber. The oil lubricates the seal and about 10 liters per day of oil leaks into the gas chamber. This sour oil is drained. Note that most of the UOP processes contain some H_2S or HCl . If the sour oil is put back in the oil reservoir, the lube oil can become contaminated at bearing or seal problems can result.

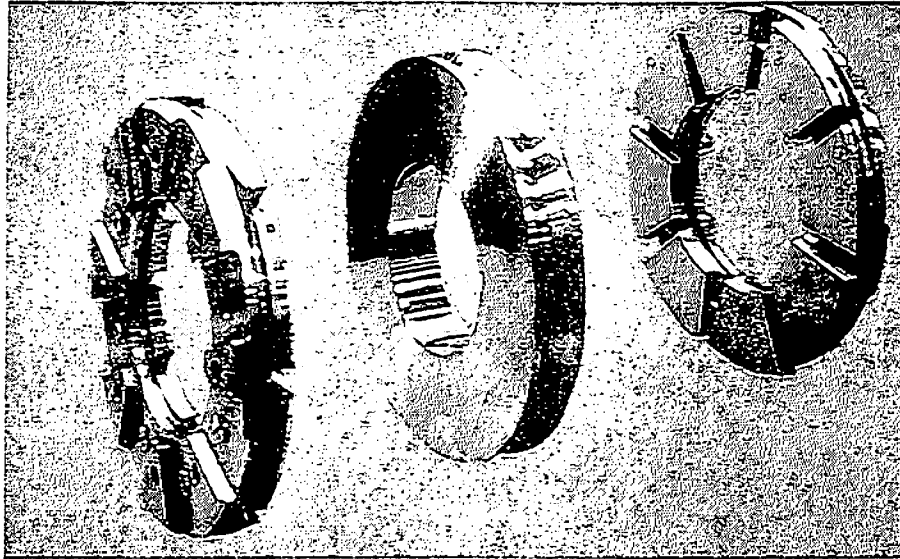
Liquid Film Sleeve Seal (TSO-Sleeve Seal)



ECS-CD-81

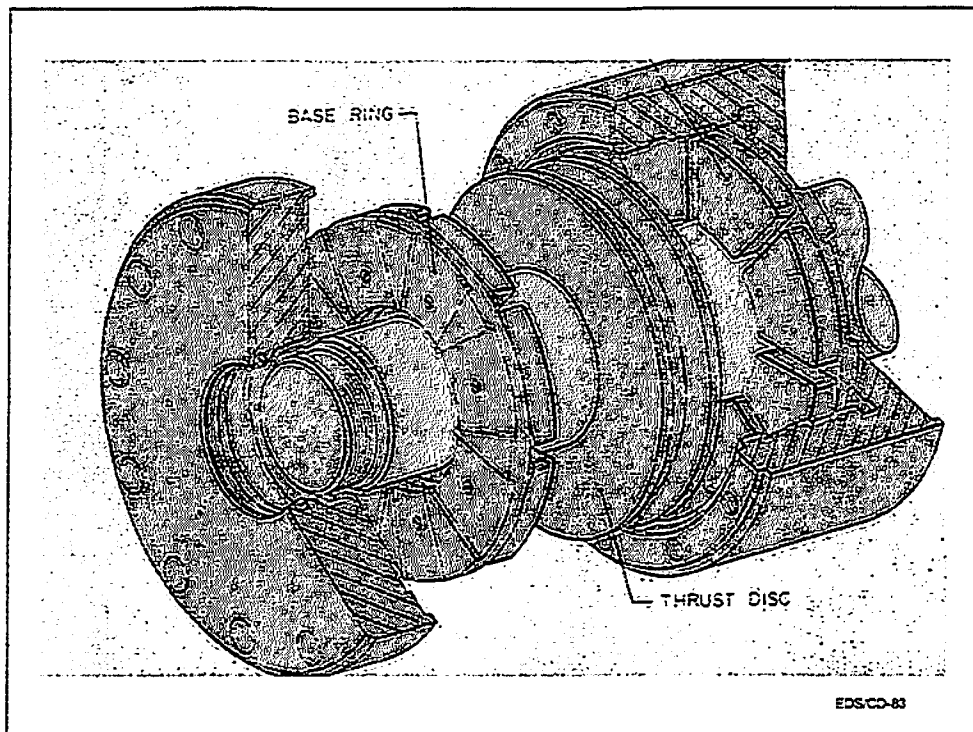
Bushing type seal. Oil leakage is greater than contact type but there is no normal seal wear.

Thrust Bearing



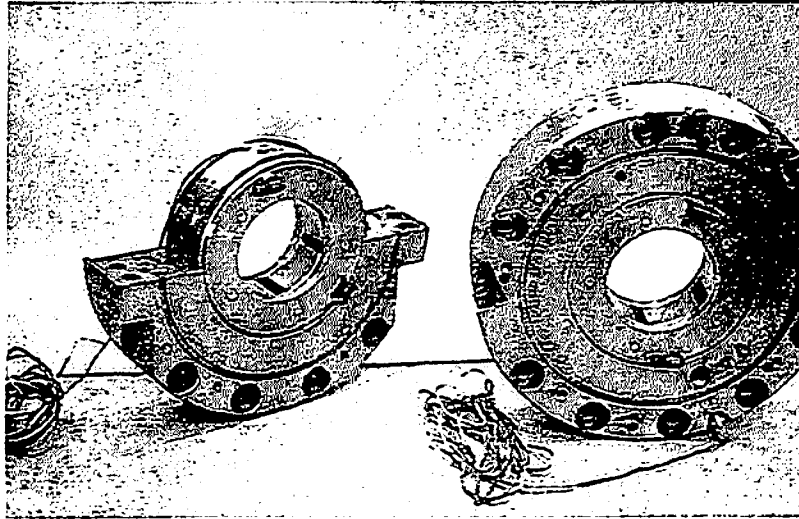
EDS-00-42

Thrust bearings are designed to withstand the axial forces on the shaft. The thrust pads are designed to pivot to the contour of the shaft.



Thrust bearing. The bearing pads ride against the thrust disc and absorb force in either axial direction.

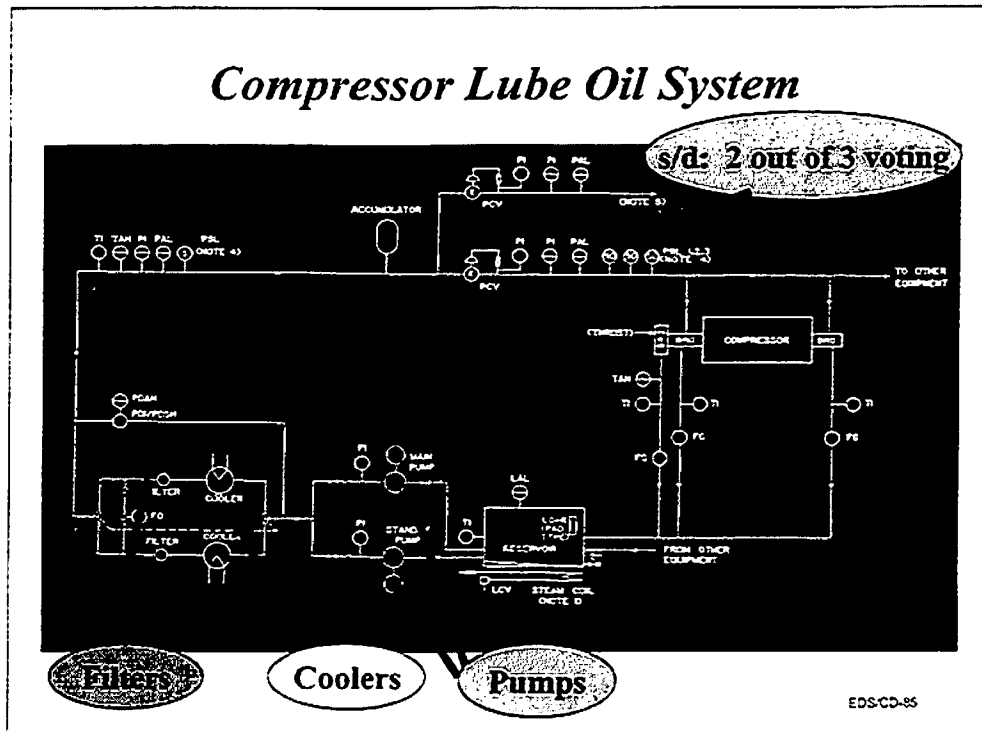
Journal Bearing



EDS-00-84

The journal bearings are designed to withstand the radial forces on the shaft. The bearing pads can tilt to fit the contour of the shaft. Note the temperature monitors on the back side of the pads.

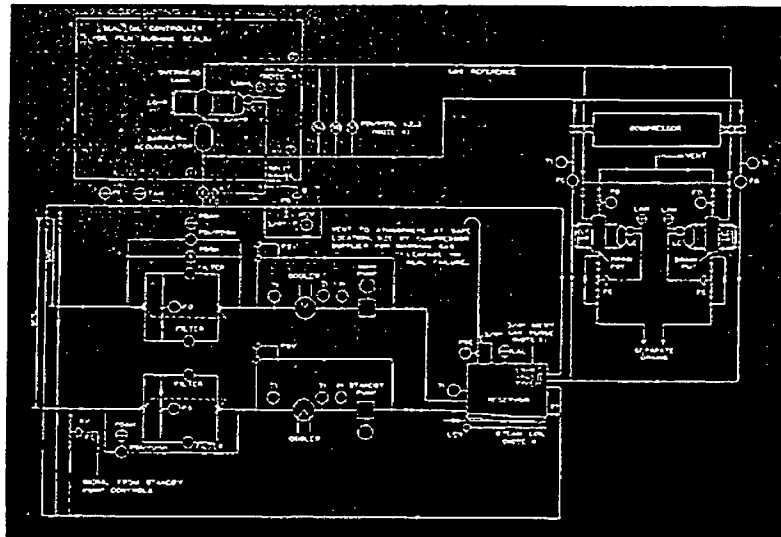
Compressor Lube Oil System



Centrifugal compressor bearings are typically lubricated by an external pressurized oil system. Oil is stored in a large reservoir. There are two pumps, one operating and one spare. On low oil pressure, the spare pump comes on-line automatically. The two pumps should be driven by different power supplies, typically electric motor and steam turbine, to insure a supply of oil to the compressor. The spare pump must be motor driven. A steam turbine will take too long to come on-line. Note the twin coolers and filters. Also note the three separate low oil pressure shutdown switches. The shutdown is two of three voting. Two of the switches must vote to shutdown before the driver is tripped. This avoids spurious shutdowns and insures a shutdown if one is needed.

3. In order to be free from the danger
 of infection.

Compressor Seal Oil System



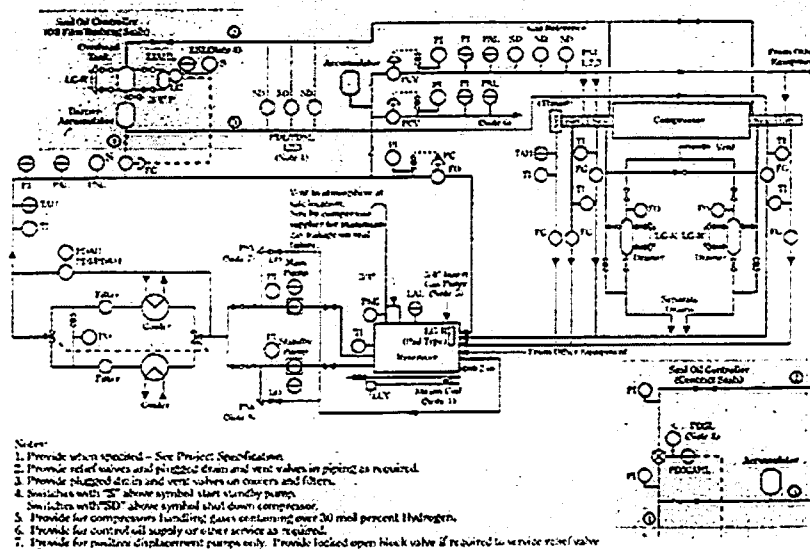
EDS-20

If the compressor has an oil lubricated seal, the seal must be supplied with pressurized oil. The seal oil system is very similar to the lube oil system

Hol vs arak
H.S. - chow

Hol Crack → here we
get gas vs/ some oil

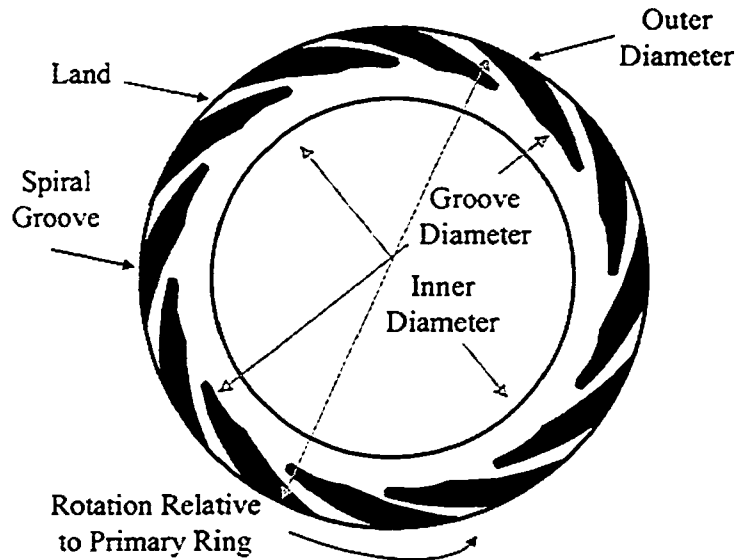
Compressor Lube and Seal Oil System



EDS-CD-87

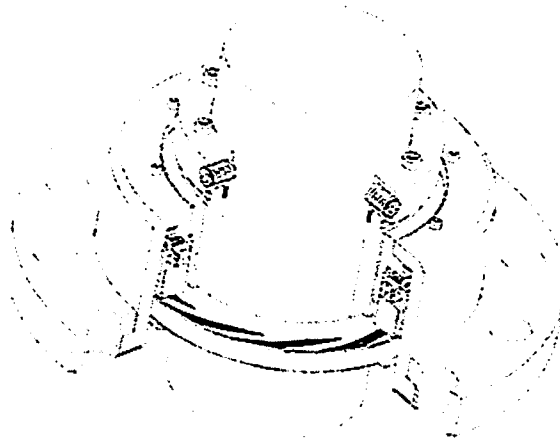
There are many common components between the lube and seal oil systems, such as pumps, coolers, filters, etc. Many compressors have combined lube and seal oil systems. Only one reservoir and one set of pumps, coolers, filters, etc., is required. If the compressor has a high suction pressure or contains a large quantity of H_2S (Hydrocracking Units), the oil systems are kept separate. Otherwise, UOP specified combined lube and seal oil systems. Today, UOP specifies dry gas seals so only the lube oil system is required.

Dry Gas Seal Spiral Groove Arrangement



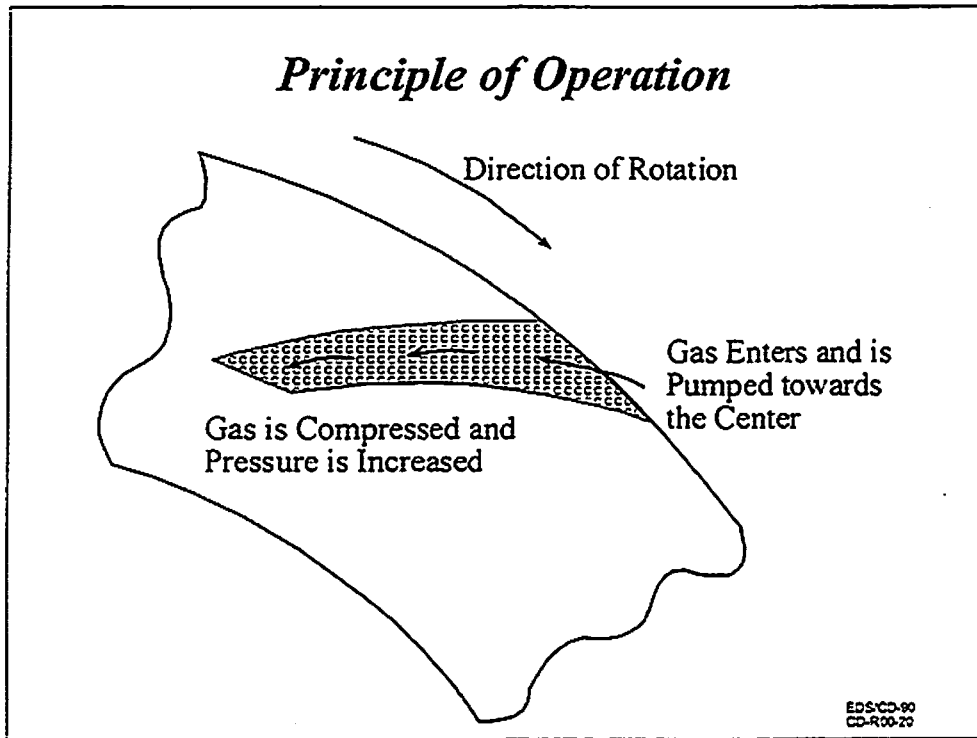
Rotating face of dry gas seal. The sealing face is attached to the compressor shaft and rotates against a solid stationary face. With rotation, gas is pumped from the O.D. inward toward the root of the grooves where a sealing dam is defined. The sealing dam provides resistance to flow, increasing the pressure. The pressure generated separates the carbon ring surface out of contact with the tungsten carbide ring by about 0.0001 to 0.0002 inches. This allows less than 2 cfm of gas to leak across the seal faces.

Single Dry Gas Seal



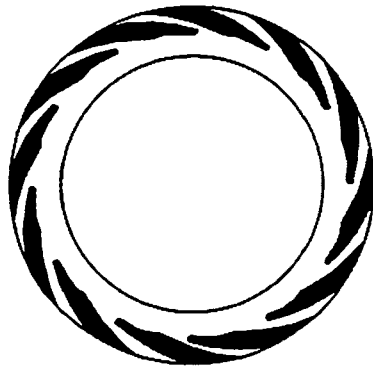
EDS/CD-89
CD-R00-10

Single dry gas seal cartridge.

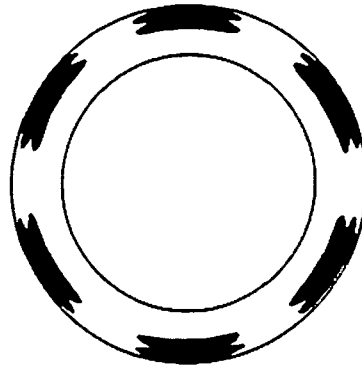


As the grooved face rotates with the shaft, gas is pumped towards the center and, after a few rotations, lift off occurs.

Design Comparisons



**Unidirectional
Spiral Groove**

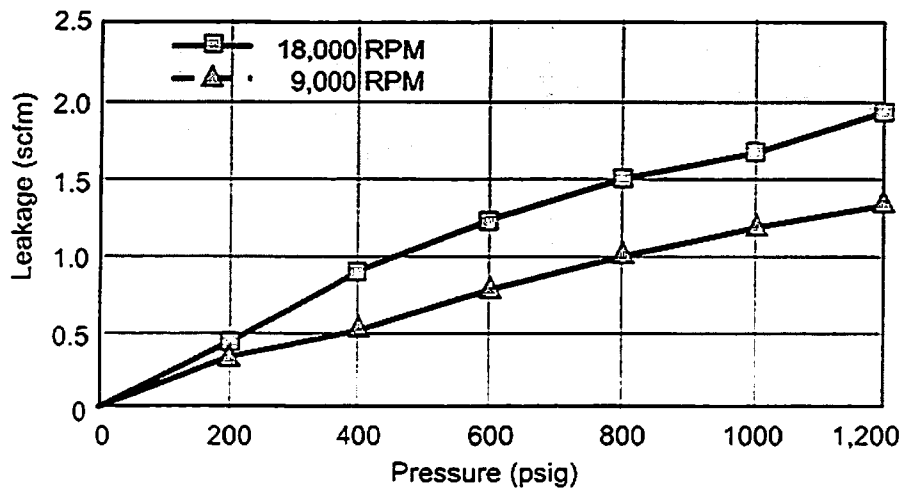


**Bidirectional
Spiral Groove**

EDS-CD-91
CD-R00-21

Some seals are bidirectional. With the spiral groove design, the seals on either side of the compressor shaft are mirror images of each other. Inlet side and outlet side seals are not interchangeable. The bidirectional grooved seal is symmetric so both seals on either end of the shaft are identical.

Dynamic Test Leakage for a 4.937" Diameter Seal



EDS/CD-82
CD-R00-22

Typical seal leakage rates. As can be seen from the tests, leakage is primarily a function of speed and pressure. Molecular weight and temperature can also affect the leakage rate to a lesser degree.

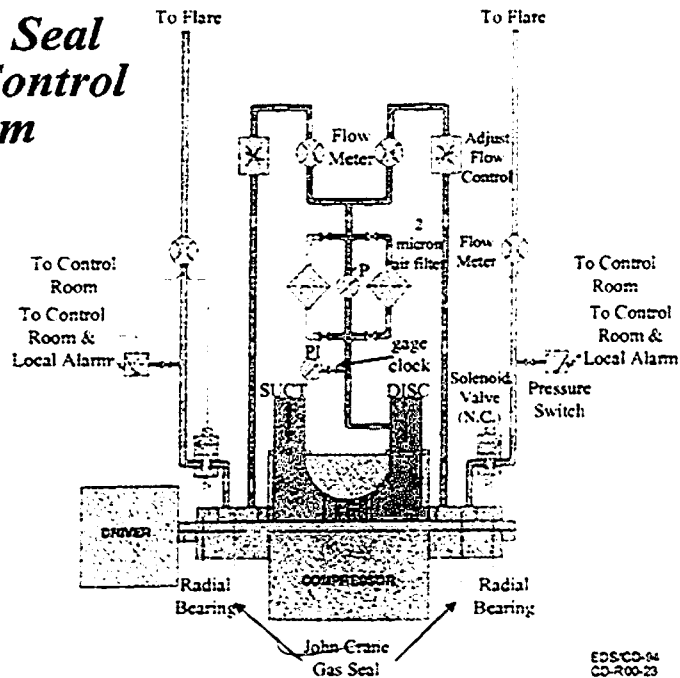
Control System

- Provides three main functions
 - Filtration of buffer gas ✓
 - Regulation of buffer gas ✓
 - Monitoring of seal performance ✓
- Offers local indication of filter and seal performance
- Design is simple to operate and user friendly with minimum maintenance requirements

ECS/CS-92

The control system filters and regulates the gas and monitor seal performance.

Dry Gas Seal Typical Control System

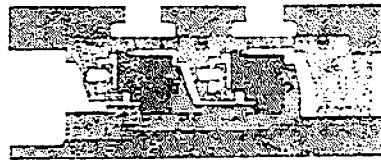


The seal buffer gas is typically process gas from the discharge of the compressor. The process gas is filtered and coalesced, down to one micron liquid and three microns solids. Note that it is critical to supply the seals with a clean, dry buffer gas. The gas is injected on the process side of the seal about 20 psi above the compressor suction pressure. Most the the gas goes across a labyrinth back into the compressor. Under two cfm leaks across the seal. This gas is routed to flare.

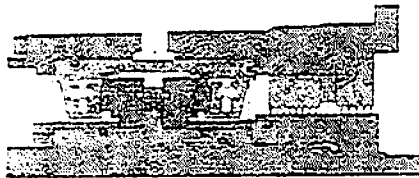
Type 28 Seal Arrangements



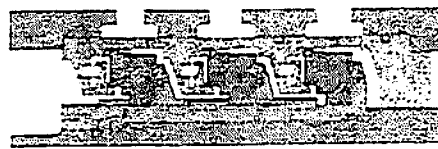
Single Seal



Tandem Seal



Double Opposed Seal

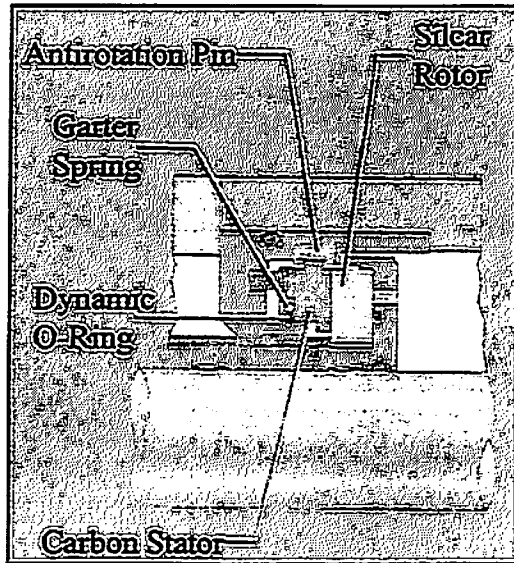


Triple Seal

EDS/CD-35
CD-R00-24

UOP specifies tandem dry gas seals for process gas applications.

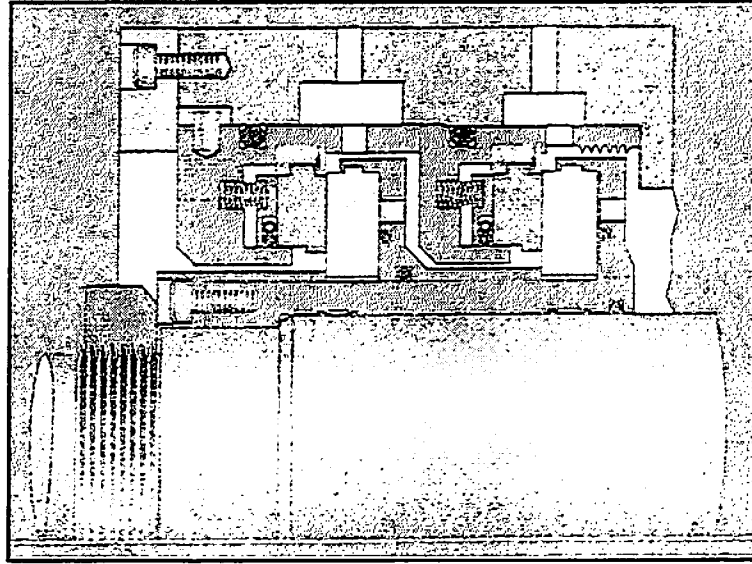
GC-100 Dura Seal®



EDS/CO-86
CO-RSC-25

Single dry gas seal. The grooved silcar rotor is attached to the shaft.

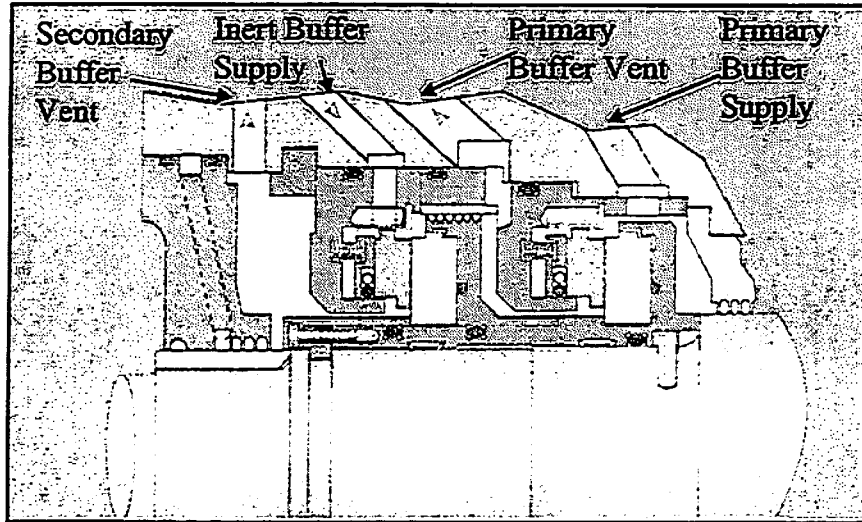
GC-300 Dura Seal®



EDS-CD-67
CD-R06-26

Tandem dry gas seal. Most of the gas which leaks across the primary seal is vented to flare. Under 0.5 cfm leaks across the secondary seal. A nitrogen separation gas prevents the process gas from migrating to the lube oil in the bearing box. The gas that escapes across both seals is routed to atmosphere outside the compressor shelter.

GC-300 Dura Seal® with Interstage Labyrinth



EDS/CD-93
CD-R00-27

If zero fugitive emissions is desired, the nitrogen separation gas can be routed between the two seal faces. This will force all of the gas that leaks across the primary seal out the vent to flare. Only inert nitrogen will leak across the secondary seal to atmosphere.

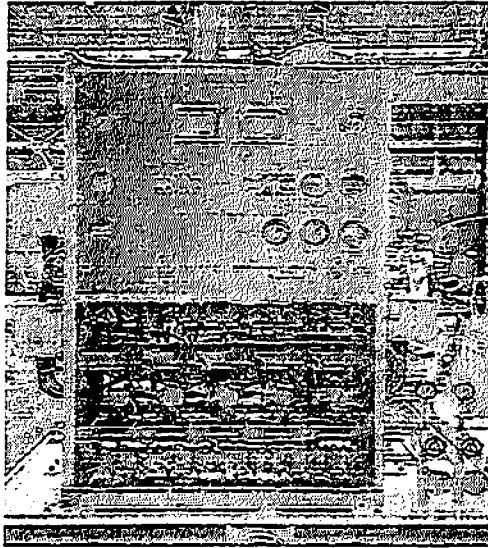
Comparison of Wet Seal vs. Dry Gas Seal

	Wet Oil Seals	Dry Gas Seal
Seal oil support systems costs	Pumps, reservoirs, filters traps, coolers, consoles	None
Seal oil consumption	1-100 gallons/day	No seal oil
Maintenance cost	A major expenditure over equipment life	Negligible
Energy costs	Seal Power loss: 10-30 HP Unit driven pumps: 20-100 HP	1-2 HP
Process gas leakage	Gas Leakage: 25 scfm & higher	Less than 2 scfm
Oil contamination	Of Pipelines: High clean up costs Of Process: Catalyst poisoning	None
Toxic and corrosion applications	Buffer gas consumption (eg N ₂) 40-70 scfm	2-4 scfm
Unscheduled shutdowns	High Downtimes	Very reliable
Aborted startups	Frequent	Rare

EDS-CD-99

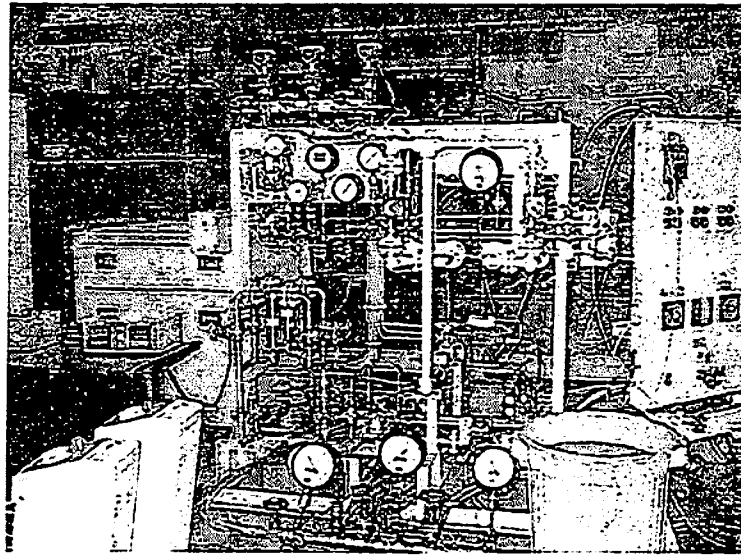
Today almost all process gas compressors are specified with dry gas seals.

Dry Gas Seal Console



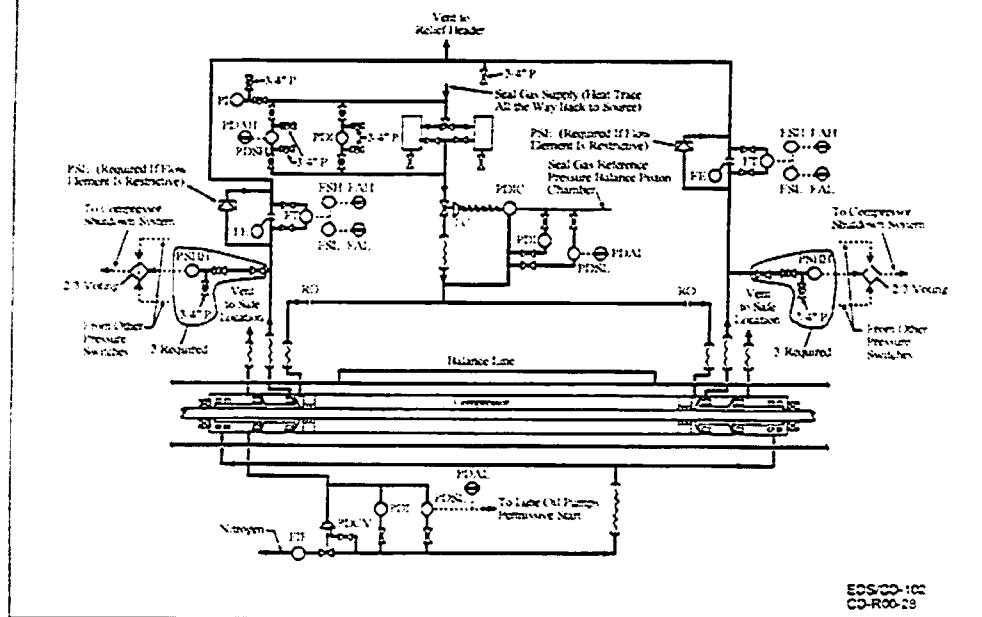
EDS-CD-100

Dry Gas Seal Console



EDS-00-101

Compressor Dry Gas Seal System



UOP dry gas seal schematic. A 2 of 3 voting high-high pressure shutdown is specified on each primary seal vent to shut down in the event of a primary seal leak. A low flow alarm across the primary seal vent warns of a secondary seal leak. If more gas is leaking across the secondary seal, less gas will be traveling out the primary seal vent.

Purchasing Centrifugal Compressors Technical Evaluation

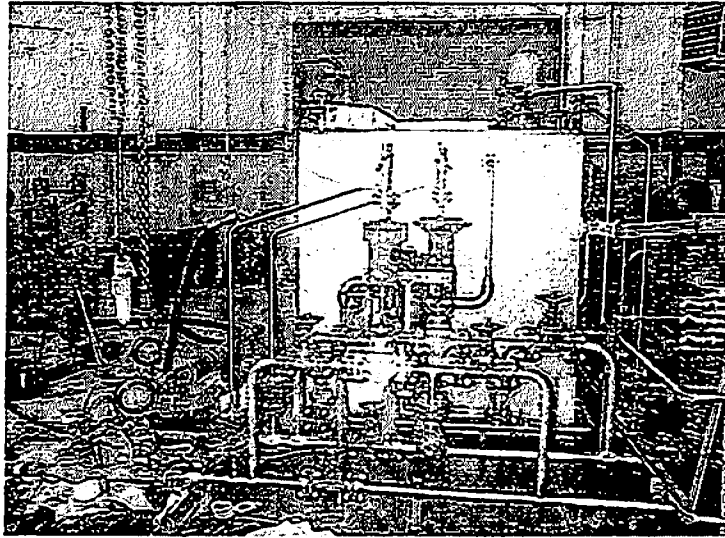
- 1. Scope of Supply**
- 2. Can it meet all operating points (N2 start)**
- 3. Materials (Impeller yield strength)**
- 4. Is proposal complete**
- 5. Does proposal meet API 617**

EDS-PO-03

Purchasing Centrifugal Compressors Technical Evaluation

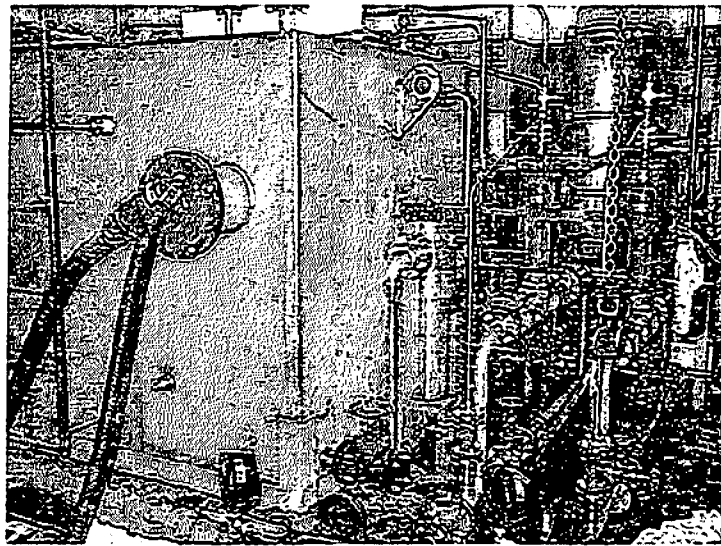
6. Efficiency of various operating points
7. Rotor dynamic stability (How many wheels)
8. Exceptions to specs and standards
9. Efficiency
10. Experience similar casing sizes, rotor sizes, pressures, flows

EDS-PG-24



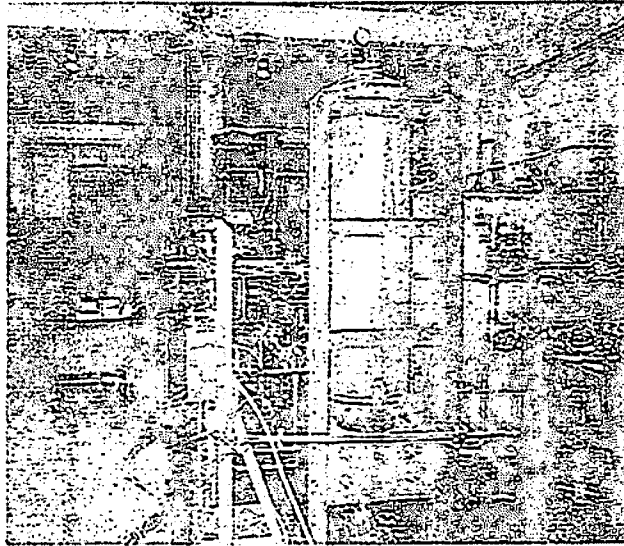
EDS-CO-103

Note the two control valves. One controls the oil pressure for the compressor and turbine bearings; the other controls the oil pressure for the turbine governor control oil.



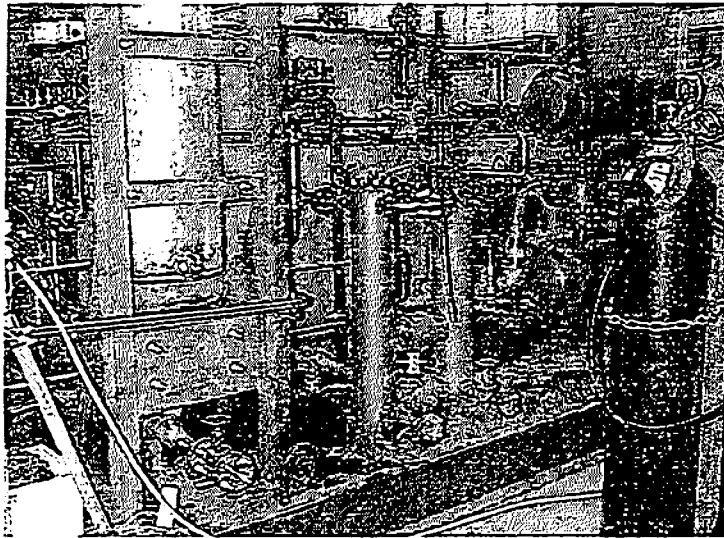
EDS-104

Oil reservoir and oil purifier.



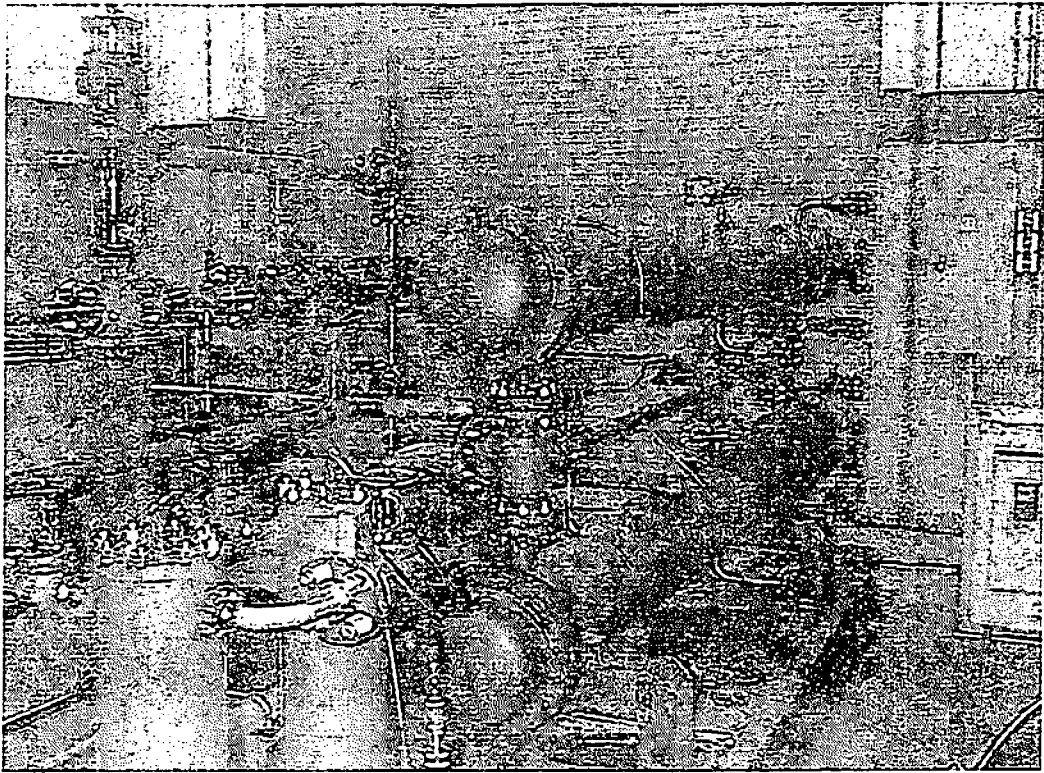
ECS-CD-105

Lube oil accumulator tank. The accumulator holds pressure in the system during an oil pump change over.

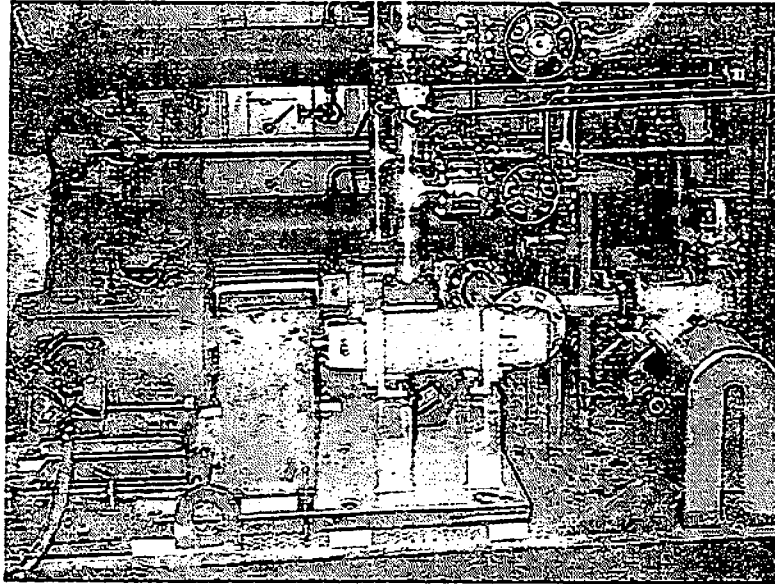


EDS-106

Twin oil filters.

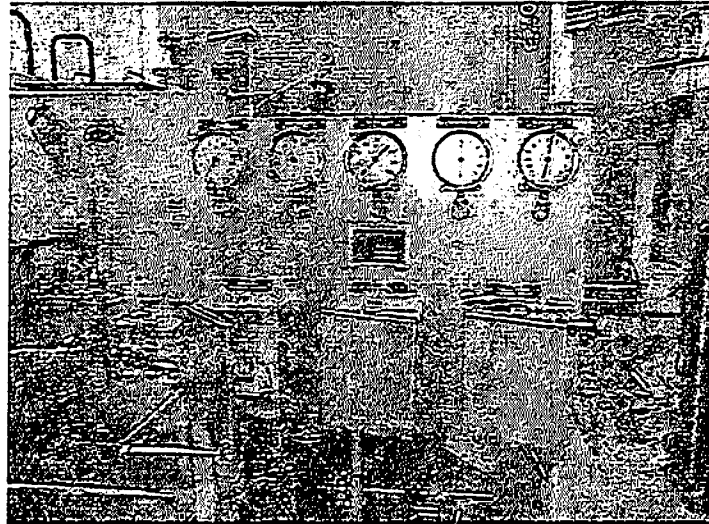


Twin coolers and twin filters.



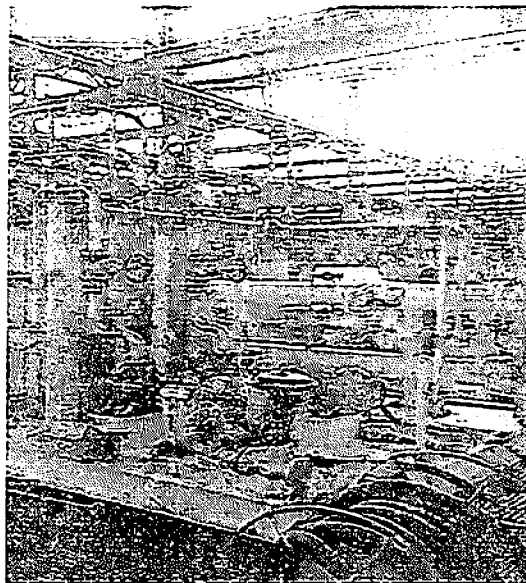
EDS-108

Oil pumps. Typically, one pump is steam turbine driven and one pump is motor driven. The motor should always be the standby pump. The steam turbine will take too long to come up to speed if the motor driven pump fails.

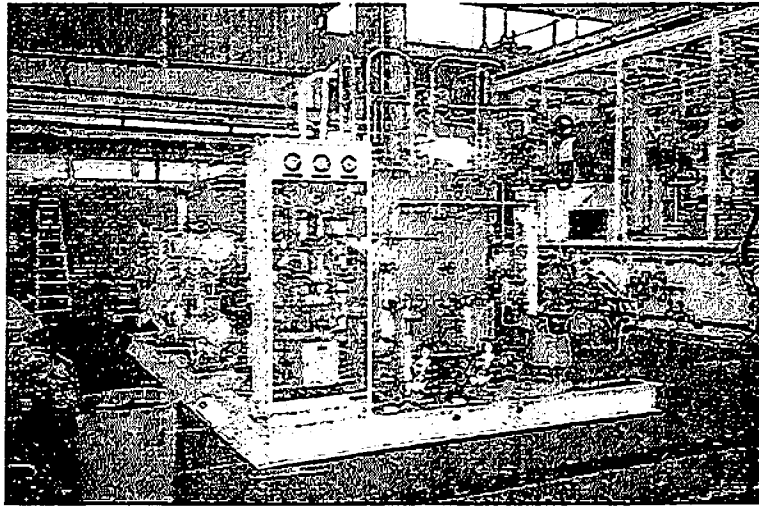


EDS/CO-109

Gauge panel.

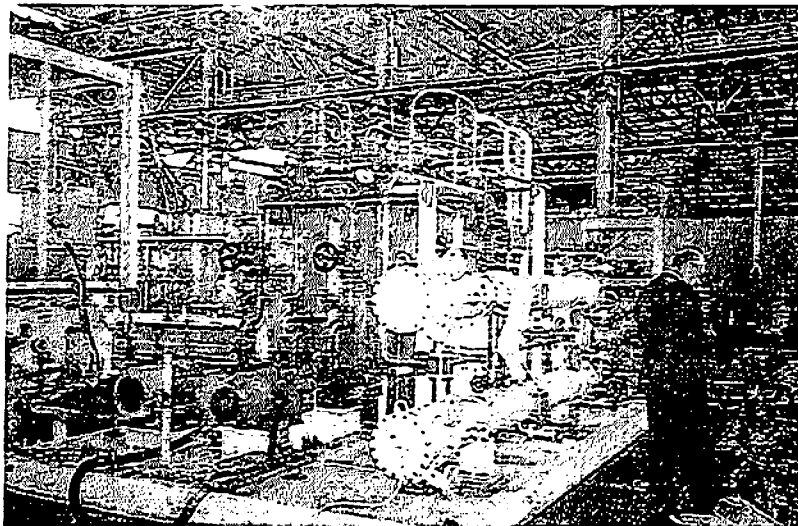


EDS CO-110

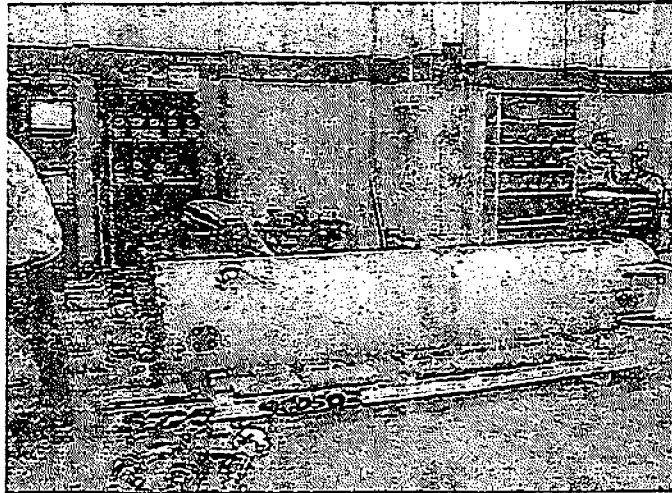


EDS/CD-111

Gauge board, twin filters, twin coolers.

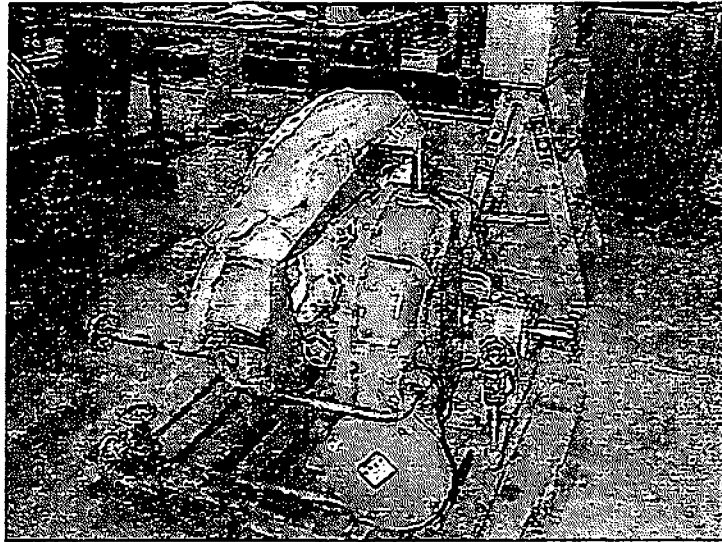


EDS/CO-112



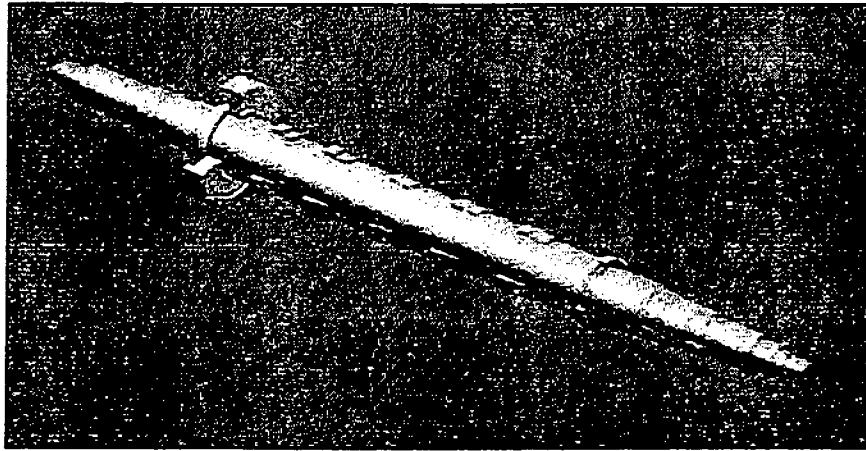
EDS/CO-113

Over head tank laying on its side. If lube oil pressure is lost, the overhead tank supplies oil to the compressor and turbine bearings by gravity feed while the compressor shuts down.



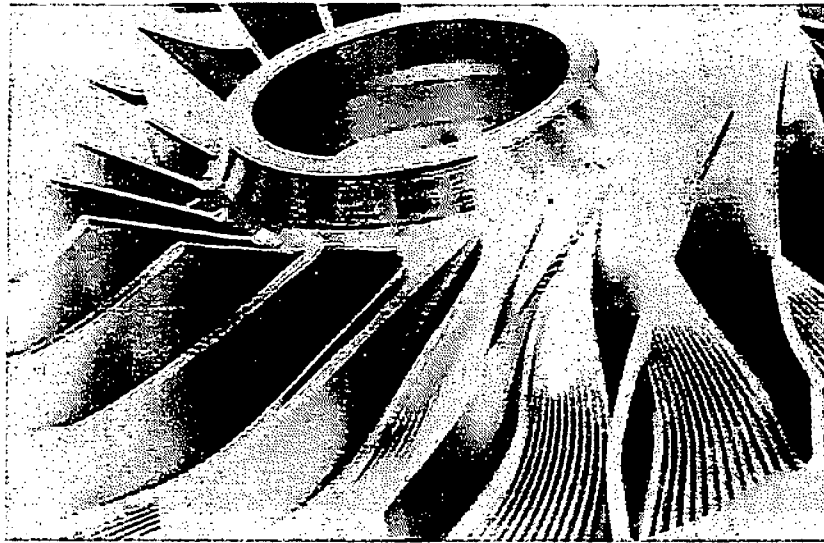
EDSCO-114

Oil pump steam turbine driver.



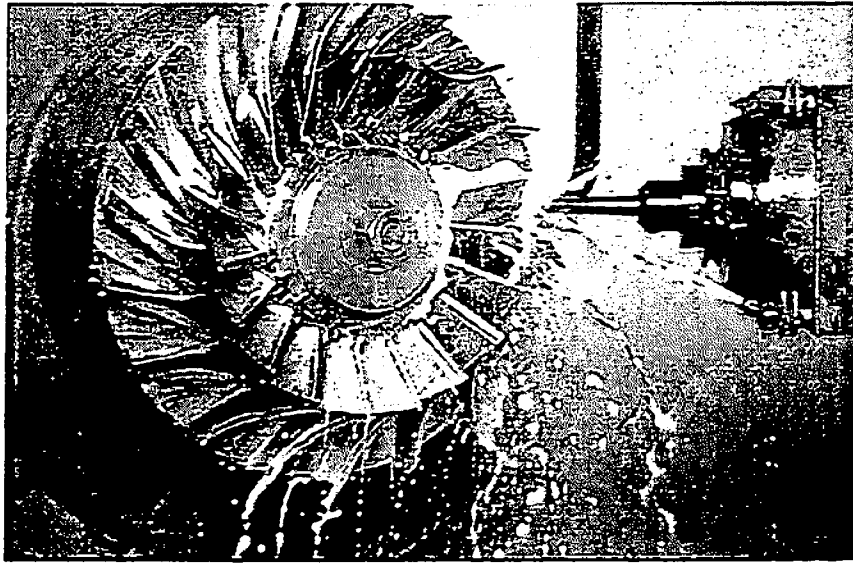
EDS/CD-115

Compressor shaft. Interstage labyrinths prevent gas from traveling back to a previous, lower pressure impeller.



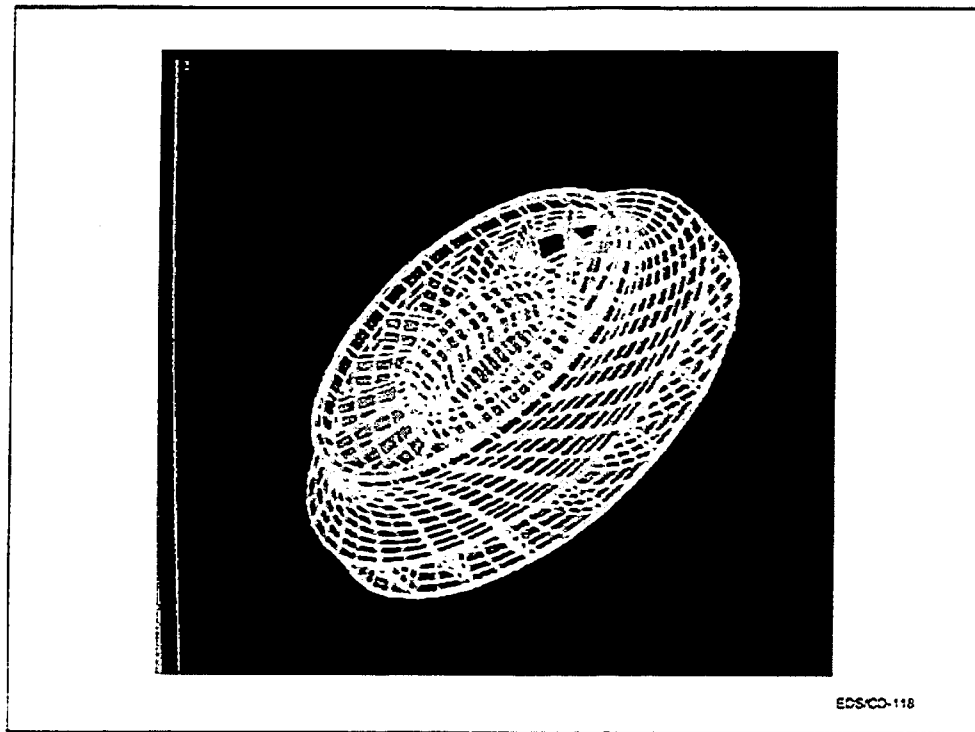
EDS-116

Compressor impeller. Note the contour of the blades which defines the shape of the head curve.

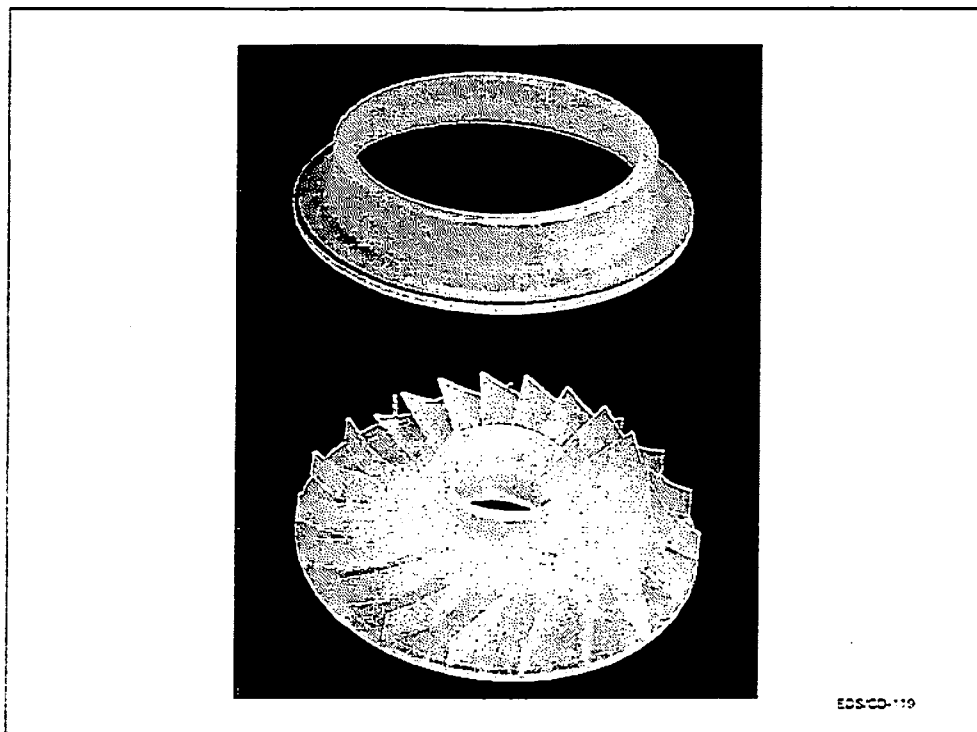


EDS-CD-117

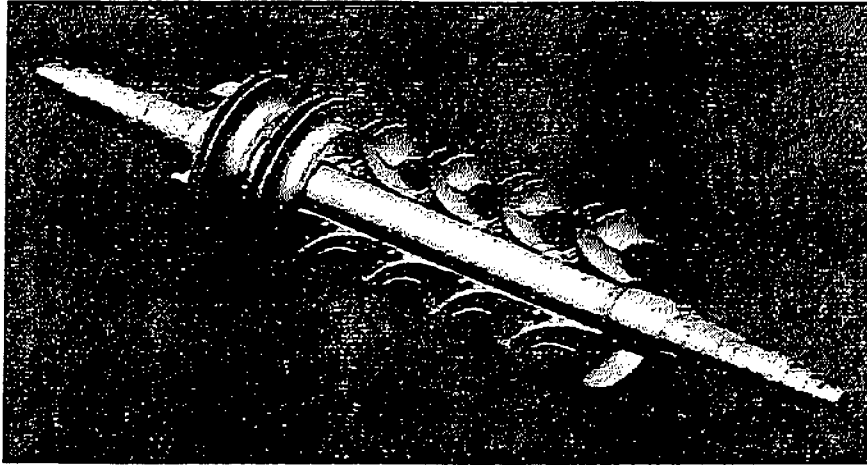
5-axis milling machine.



Finite element analysis of compressor impeller.

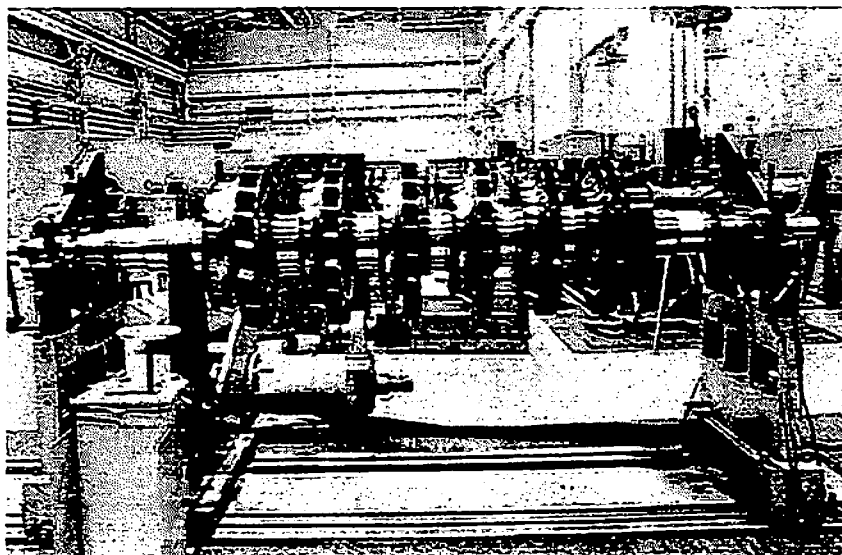


Plastic model of compressor impeller and cover.



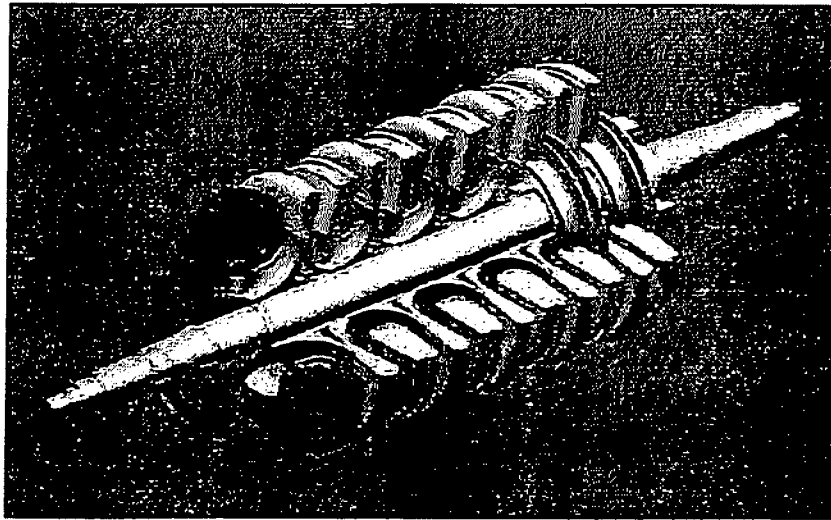
EOS CO-120

Impellers are added to the shaft.



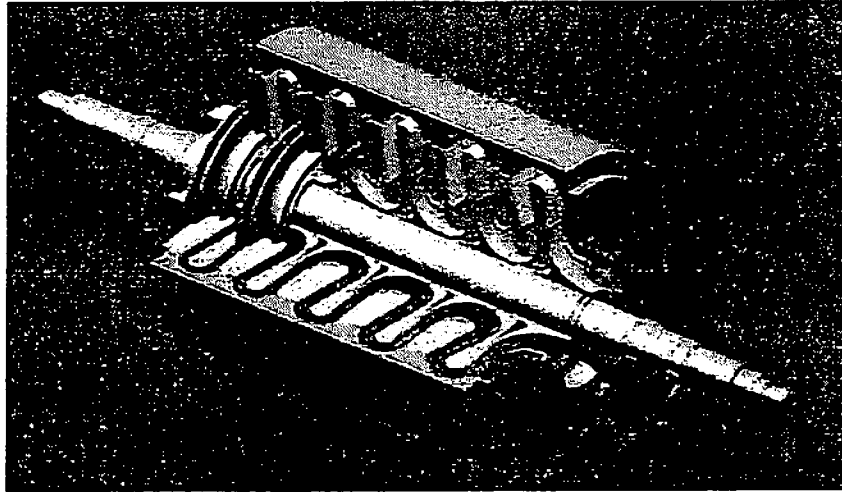
EDS-CD-121

Six stage rotor on balance machine.



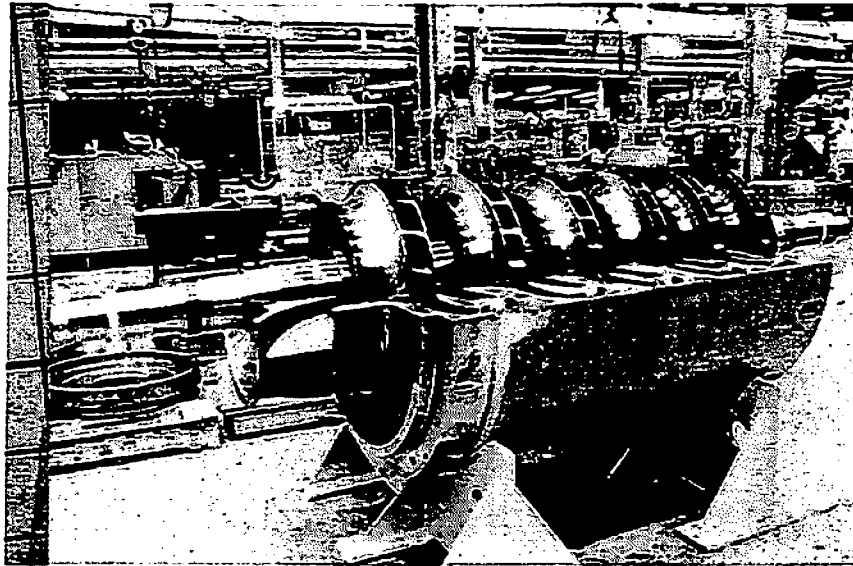
EDS-CO-122

Diaphragms are added which guide the gas from the outlet on one impeller to the inlet of the next impeller.



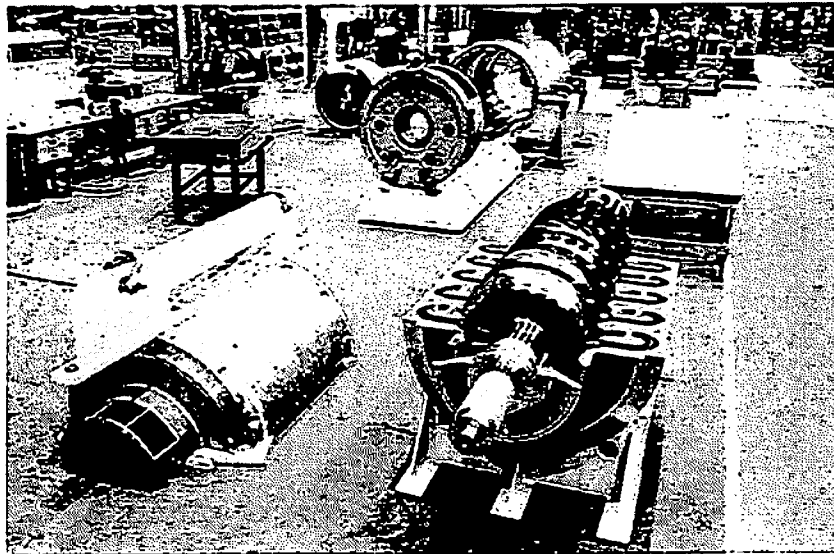
EDS-CD-123

Inner barrel cover and inlet nozzle are added.



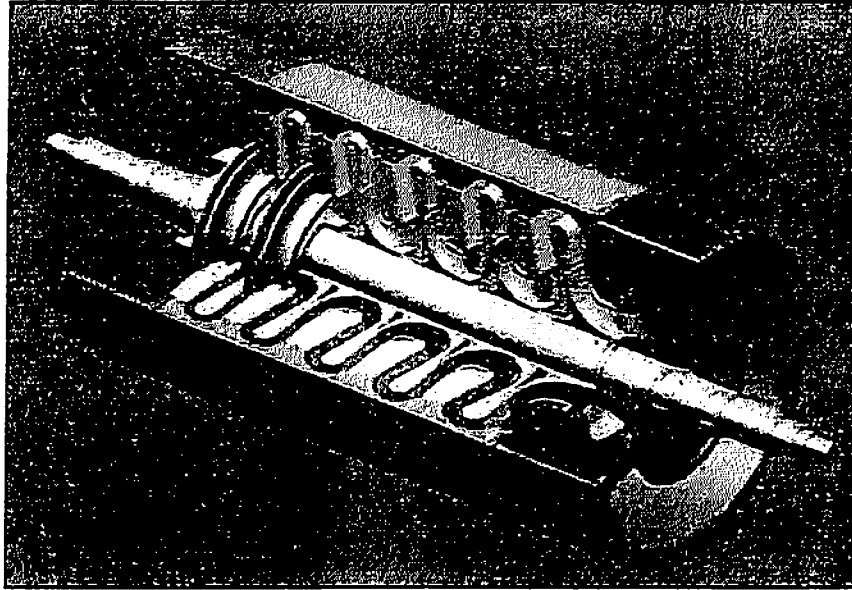
EDS/CO-124

Rotor in bottom half of inner barrel. The inner barrel is not a pressure containing component.



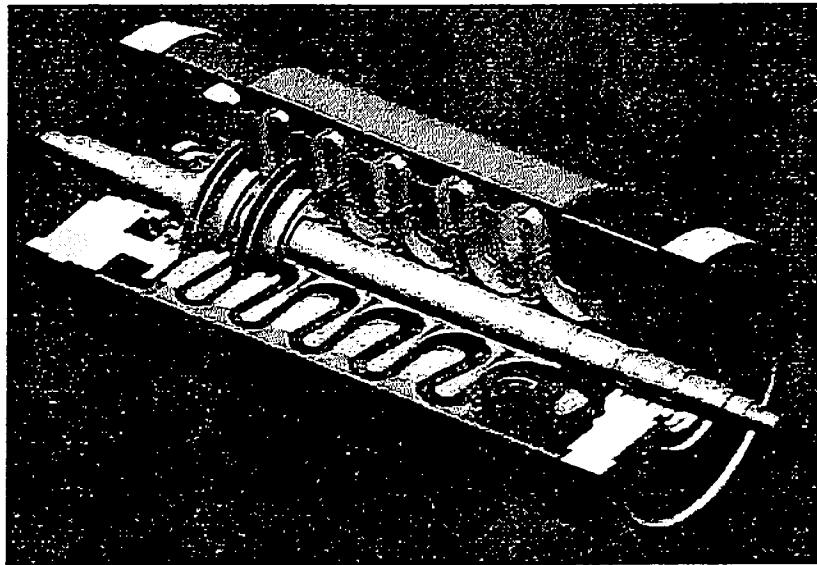
EDS/CO-125

Rotor in bottom half of inner barrel and top half of inner barrel. The end caps are in the background



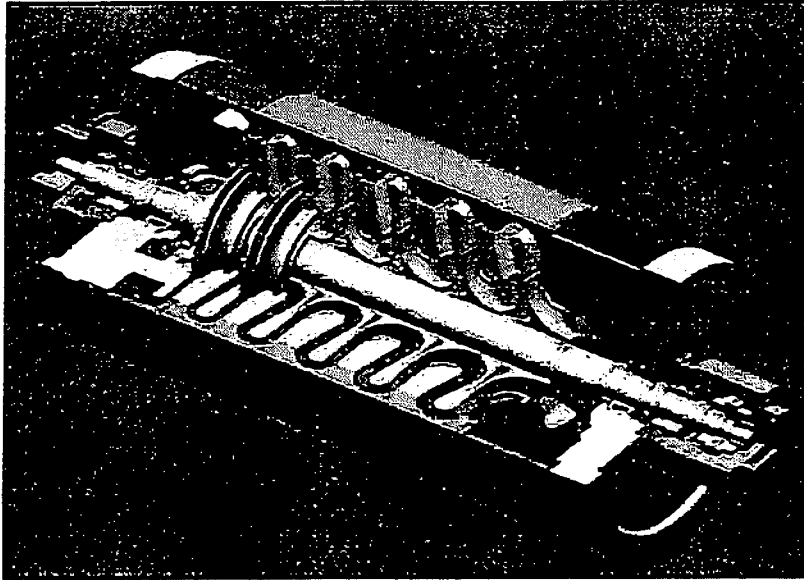
EDS/CO-126

Inner barrel end caps are added.



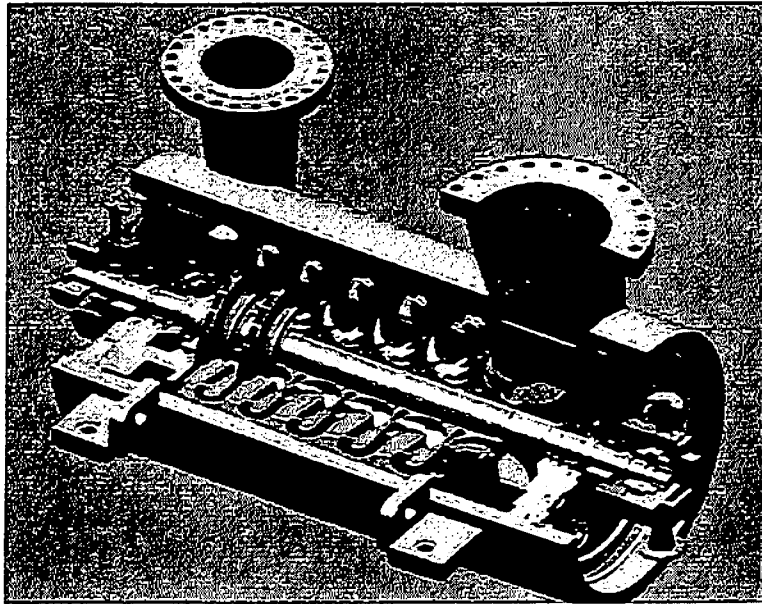
EDS/CO-127

Seals are added.

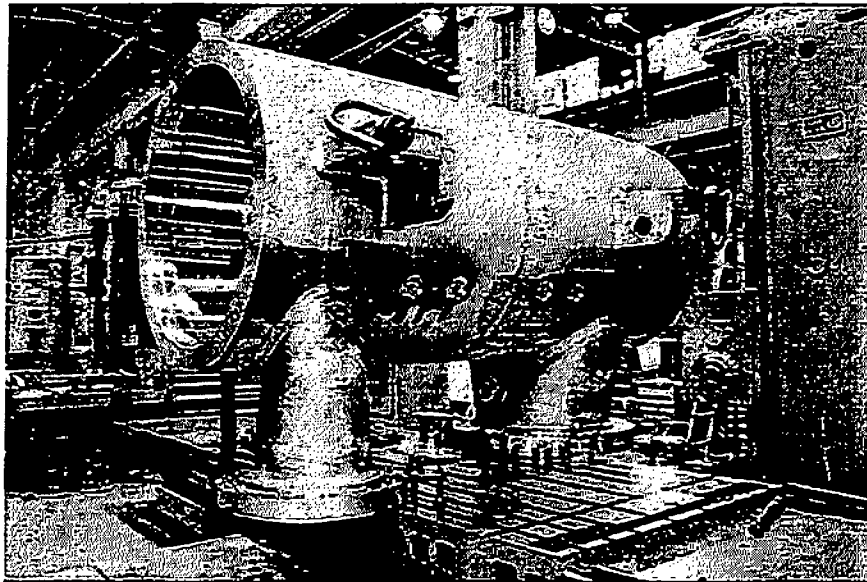


EDS-CO-128

Bearings are added.

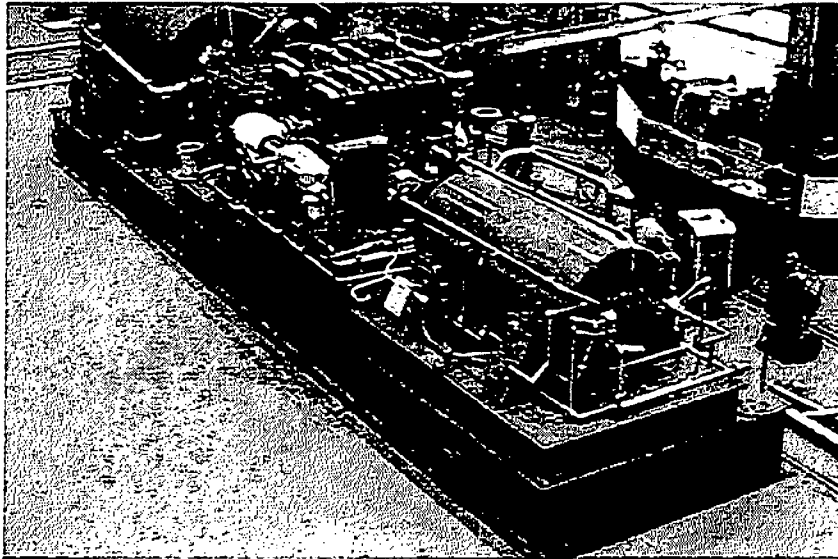


EDSCO-129



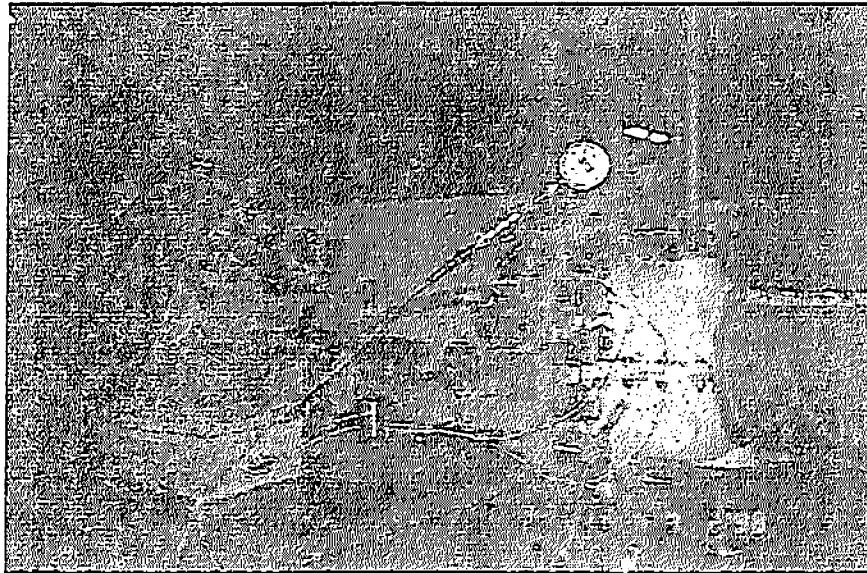
EOS/CO-130

Outer casing. Inlet and outlet nozzles can be on the top or the bottom.



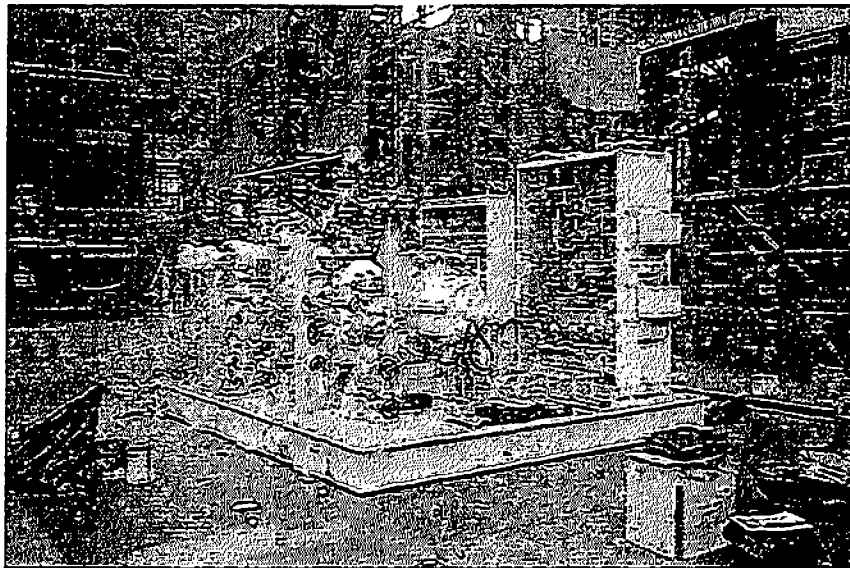
EDS-CC-131

Two-stage compressor. The larger capacity first stage is in the background.



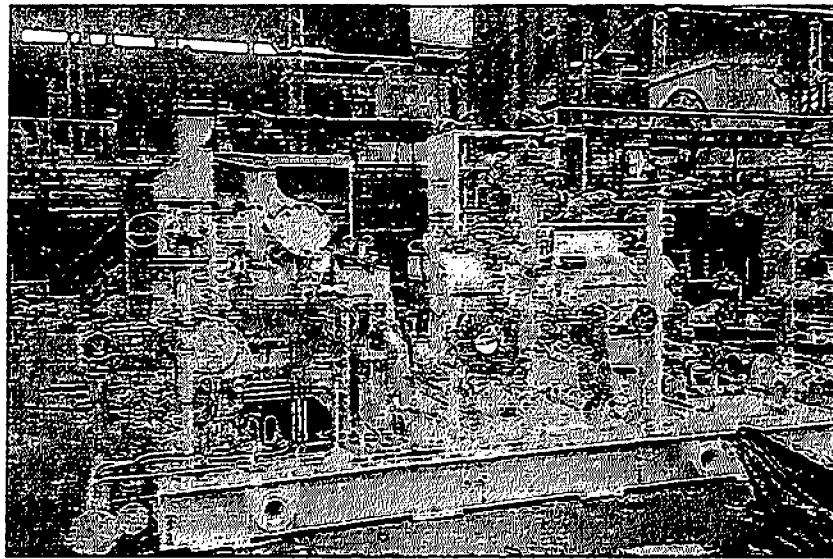
EDS-CO-132

Barrel type compressor during shop performance test.



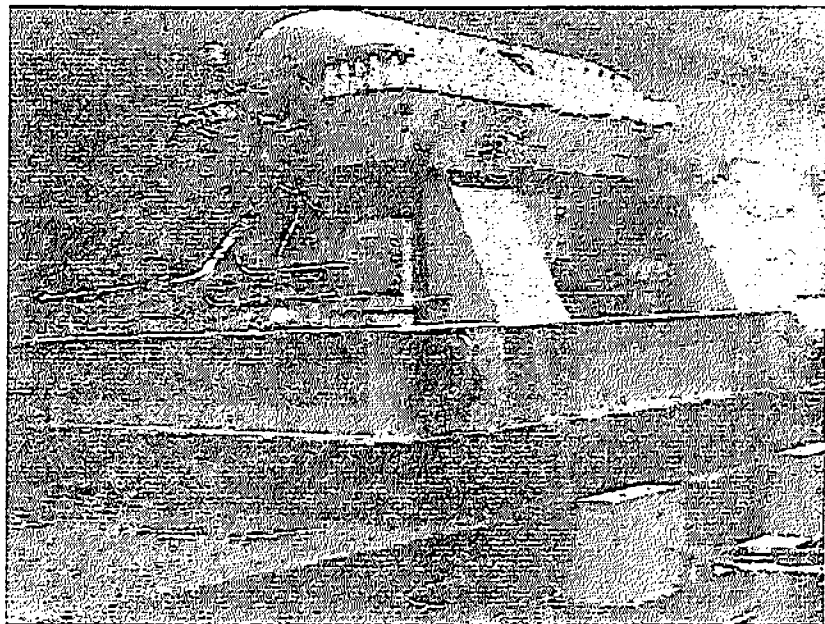
EDS-CO-133

Completed compressor, steam turbine and control panels before boxing.



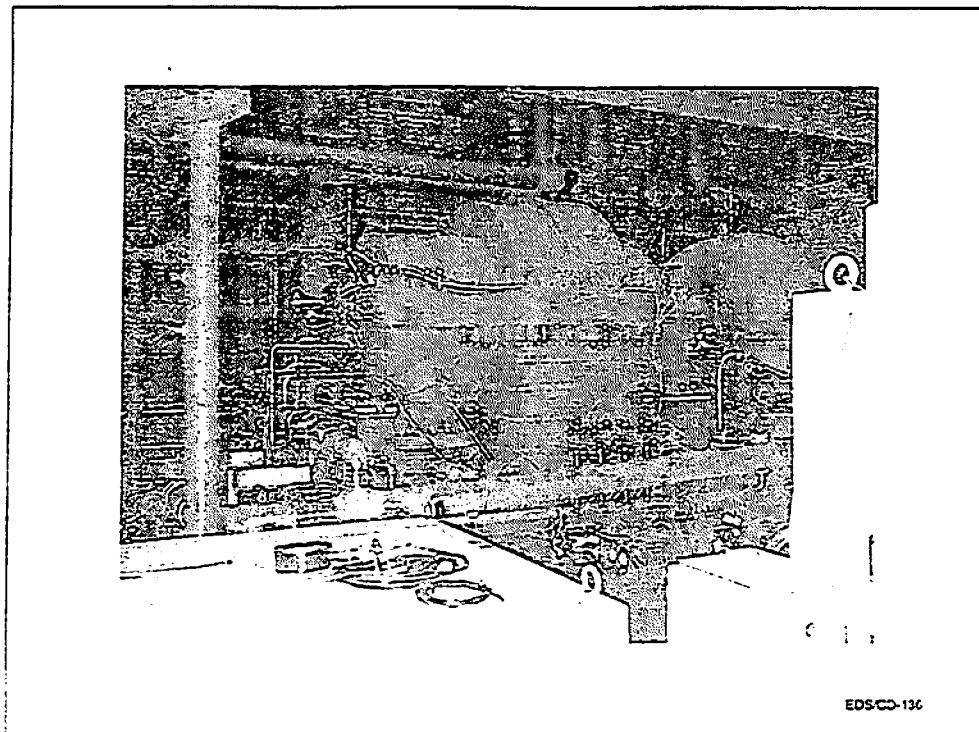
EDS.CO-134

Side view of completed compressor and turbine.

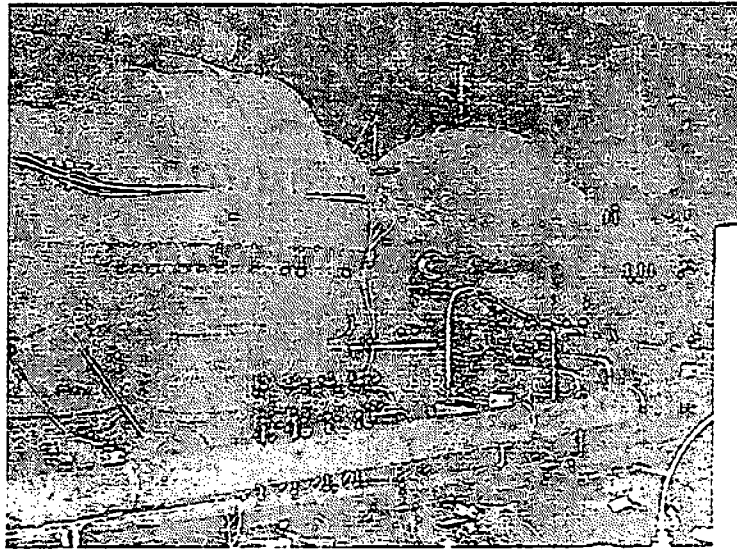


EDS/CO-135

Horizontally split wet gas compressor.

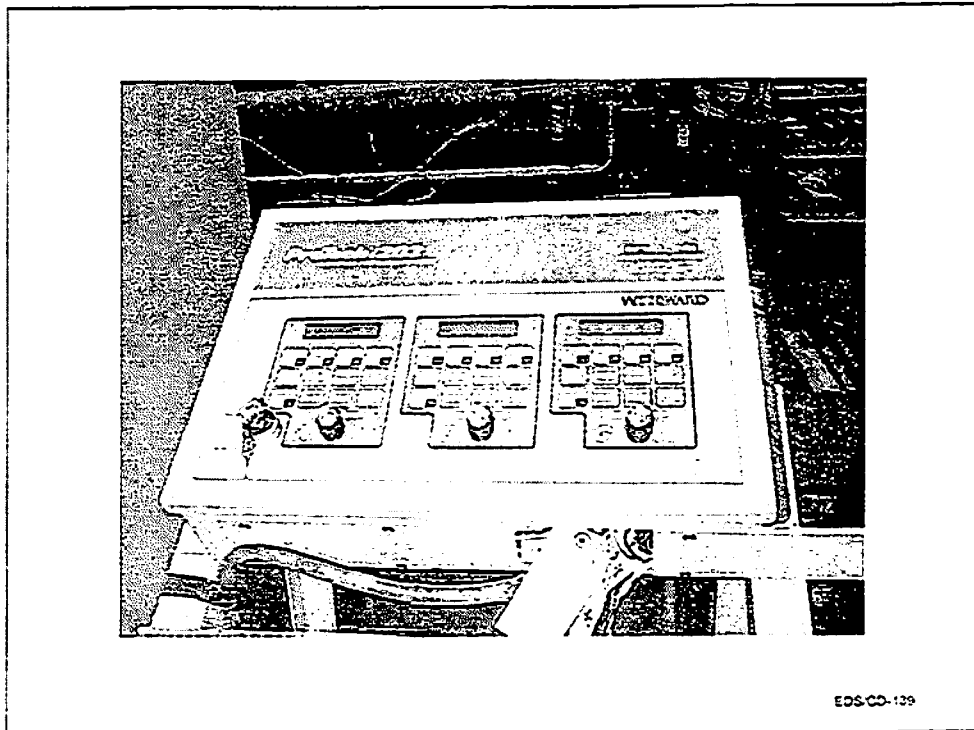


Horizontally split wet gas compressor during shop testing.



EDS/CO-137

Wet gas compressor and steam turbine during shop testing.



Turbine 2 of 3 voting high speed shutdown.

Type of Tests Performed

- **Mechanical Test**
 - Small closed loop on helium
 - Vacuum test
 - Open system on air
- **Seal Check**
- **Performance Test (on Freon)**
- **System Test**
- **Mechanical String**
- **(With or without the contract driver)**
- **Full Load, Full Pressure**

EDS-CD-140

Every centrifugal compressor gets a 4-hour shop mechanical test. This may or may not be done with the job driver. The mechanical test is run at overspeed, 105% of normal operating speed. During the 4-hour vibration, bearing temperature, bearing oil temperature, and seal leakage is checked. Vibration at non-synchronous speed is also checked. Both operating and spare rotors get a mechanical test. The performance test is optional but recommended by UOP. The performance test records five points on the compressor curve and checks the surge point. Head and power requirements at the guarantee must be within API 617 tolerances. A full load, full pressure, test is rare and very expensive and adds to the project schedule.

Critical Speed

A critical speed is where the forcing function or self-exciting frequencies are sympathetic (at synchronous or harmonics) to the resonances of a rotor, bearing or support system.

**Charlie Jackson
The Practical Vibration Primer**

EDS-CD-141

Every centrifugal compressor gets a 4-hour shop mechanical test. This may or may not be done with the job driver. The mechanical test is run at overspeed, 105% of normal operating speed. During the 4-hour vibration, bearing temperature, bearing oil temperature, and seal leakage is checked. Vibration at non-synchronous speed is also checked. Both operating and spare rotors get a mechanical test. The performance test is optional but recommended by UOP. The performance test records five points on the compressor curve and checks the surge point. Head and power requirements at the guarantee must be within API 617 tolerances. A full load, full pressure, test is rare and very expensive and adds to the project schedule.

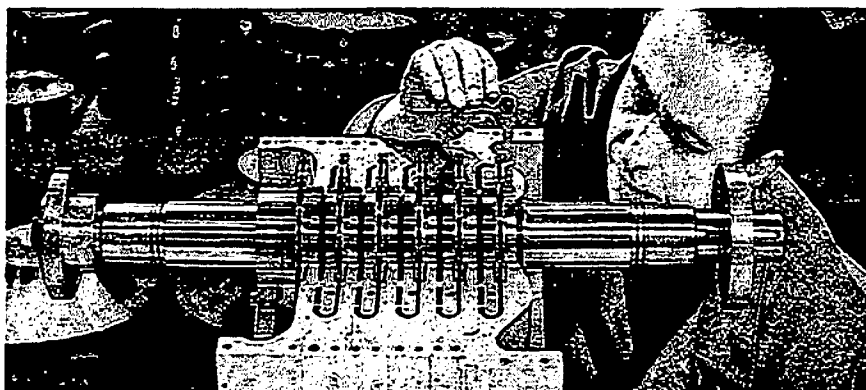
Centrifugal Compressor Disadvantages

- **Defined molecular weight**
- **Small machines inefficient**
- **Limited to approximately 1000 ACFM volume**
- **Surge potential**
- **Low mole weight compression ratio capability**
- **More sophisticated technology**

EDS-142

Centrifugal compressor disadvantages compared to reciprocating compressor:

- Molecular weight of the gas has a big effect on compressor performance as head is inversely proportional to molecular weight.
- Molecular weight has a small effect on reciprocating compressor performance.
- Surge is not a concern with reciprocating compressors.



EDS CO-143

A very small rotor. (Or a very big man)

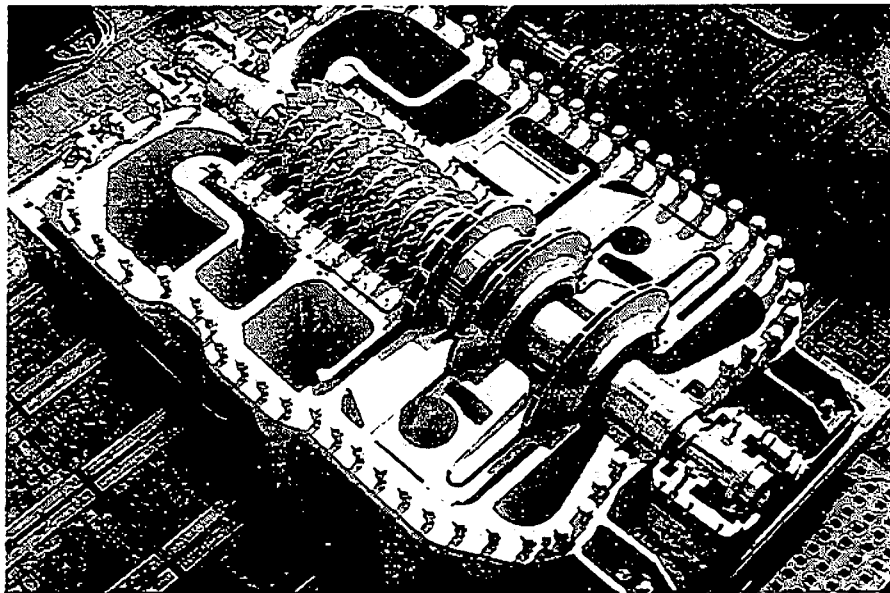
Centrifugal Compressor Advantages

- High reliability–Low maintenance
- Large volumes capacity
- Tolerance of liquids and dirt
- Smooth flow
- Wide capacity range and ease of control
- No process gas contamination problems
- Compatibility with driver

ECS-CD-144

Centrifugal compressor advantages over a reciprocating compressor:

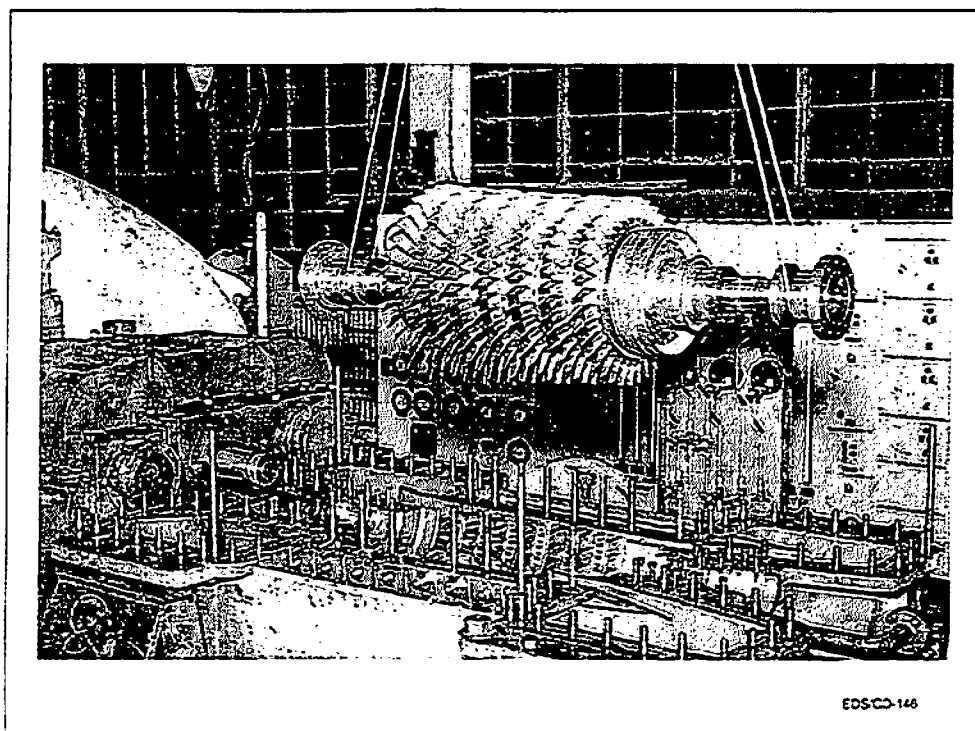
- Reliability is over 99%. No spare is required. There is only one moving part, the rotor.
- Centrifugal compressors have smooth flows, not the pulsating flows of reciprocating compressors.
- Centrifugal compressor is equally compatible with a motor or steam turbine driver.



EDS/CO-145

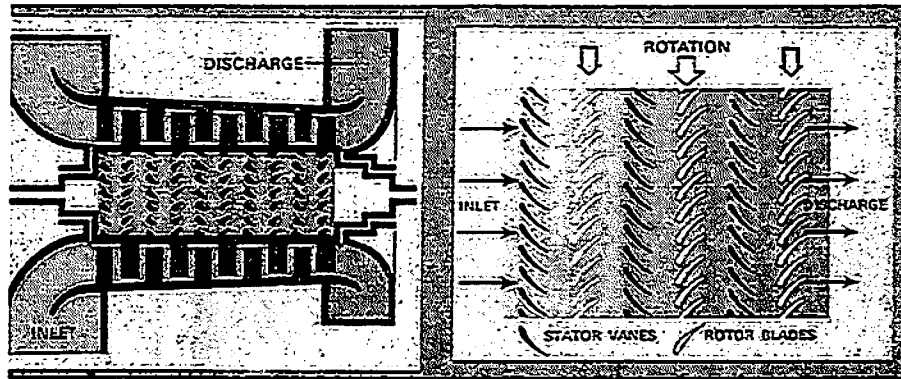
Axial compressor with three centrifugal stages on the discharge side.

Note the axial blades compared to the centrifugal wheels. Note the shape of the blades and the smaller diameter. Many axial compressors have centrifugal wheels on the discharge side which serves to broaden the allowable operating range by flattening out the performance curve and making the compressor more tolerable to choke.



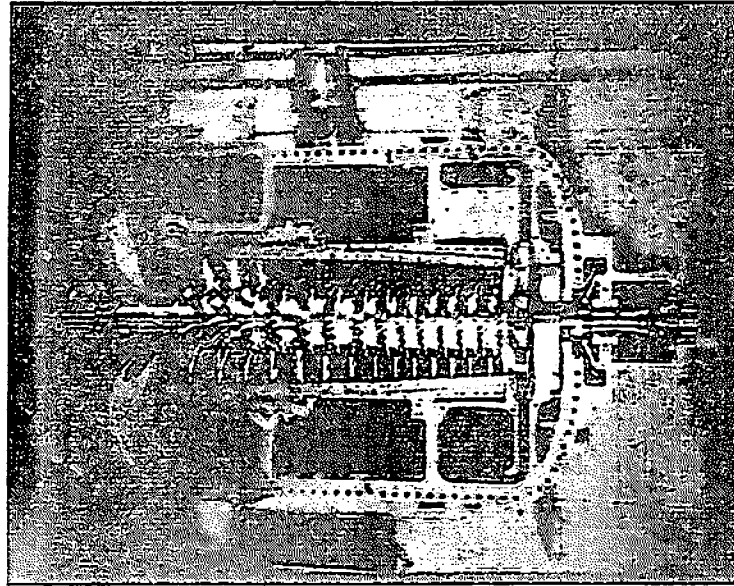
Axial compressor rotor being dropped into bottom half of casing.

Axial Flow Compressors (Dynamic Type)



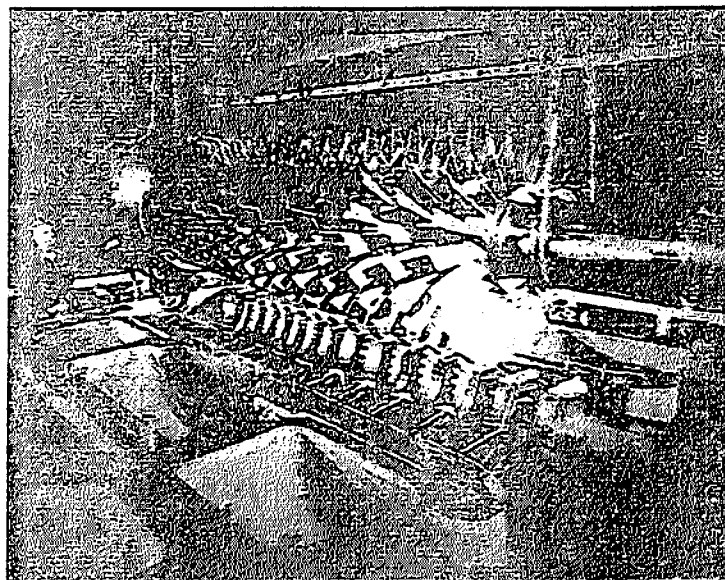
EDS-CD-147

The gas or air passes through stationary stator vanes between each row of rotor blades. The rotor vanes increase the velocity of the gas. This velocity energy is transformed into pressure energy as the gas passes through the stator blades.

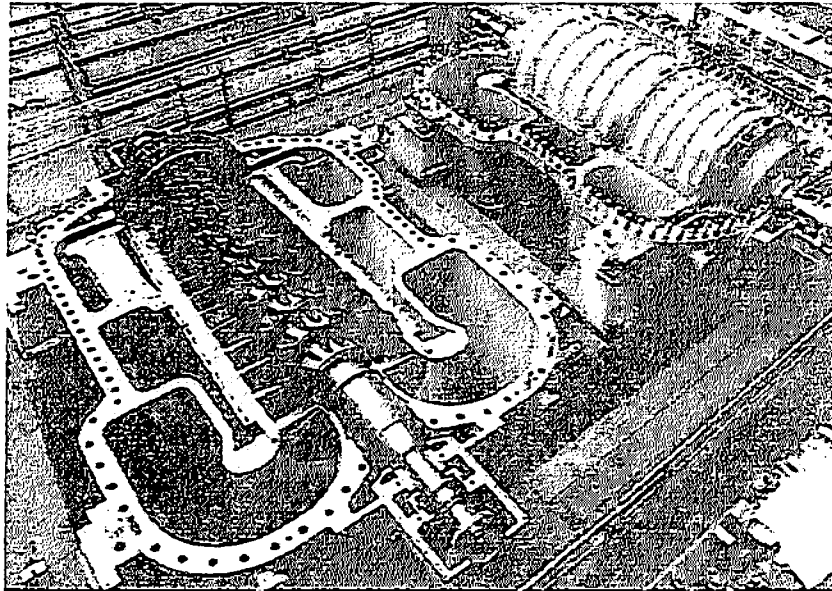


ECS/CO-148

Axial compressor with centrifugal wheel on back end. The centrifugal wheel flattens out the head-capacity curve and makes the rotor less prone to damage from operating in the choke region.



EDS-149



EDS-CD-150

Two axial compressors. The one on the right has the upper half of the stators installed.

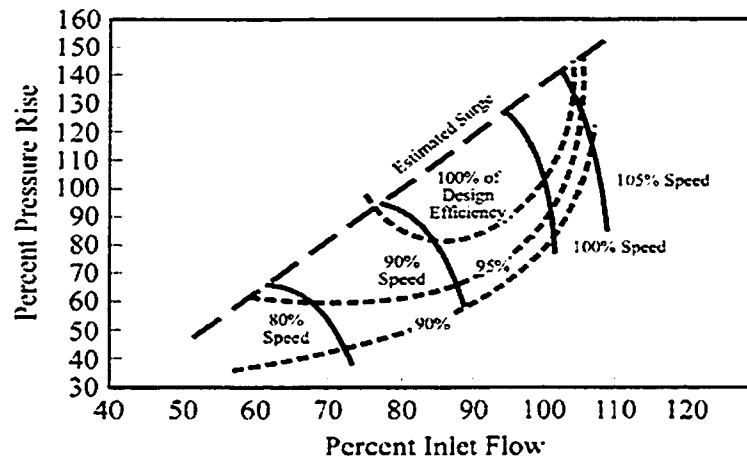
Axial Compressor Control

- Variable Speed
- Variable Stators
- Bypass (snort for air blower)

EDS-151

Axial compressors are controlled by variable speed, variable stators, and a bypass, or snort, for an air blower.

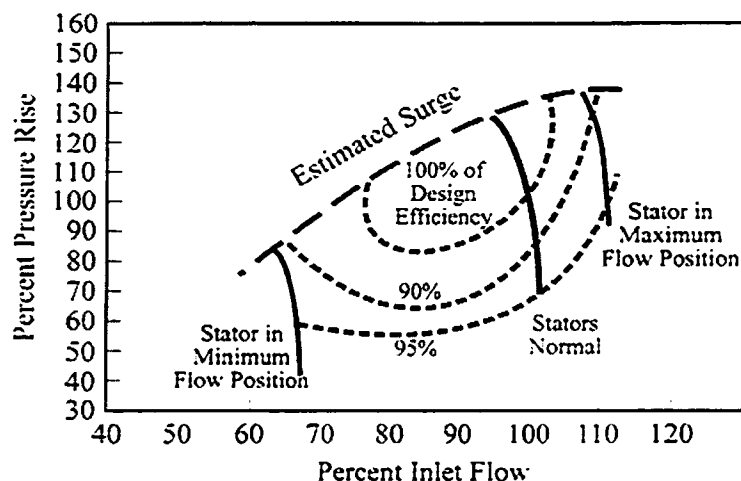
Axial Compressor Variable Speed Performance



EDS/CD-152
CD-R06-16

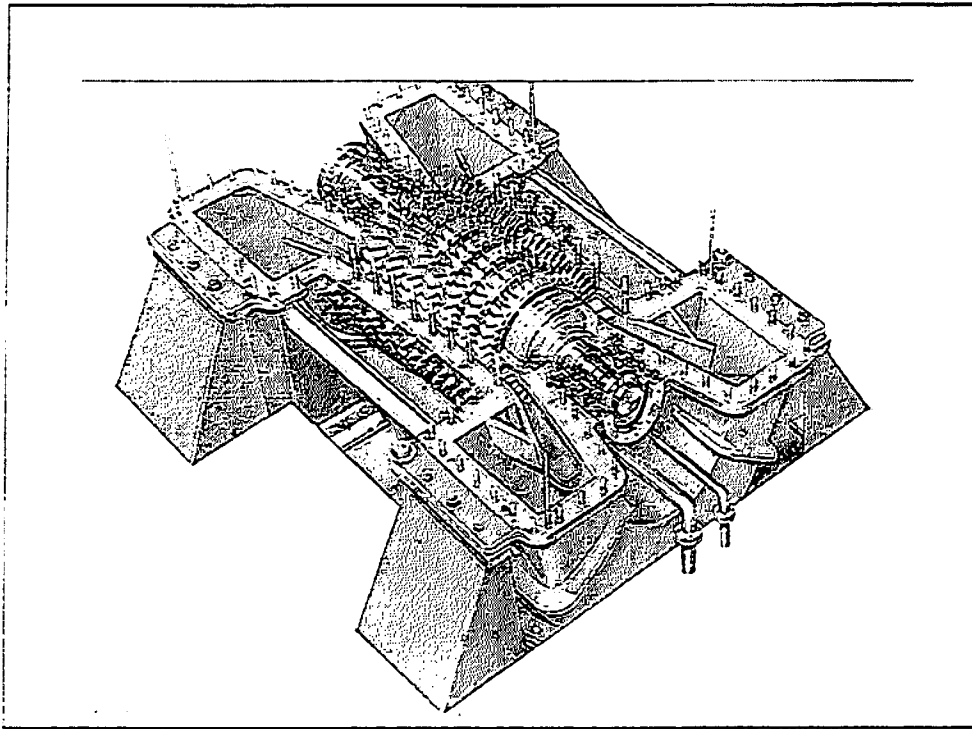
Compressor performance can be adjusted by changing speed with a variable speed driver, similar to a centrifugal compressor.

Axial Compressor Variable Stator Performance

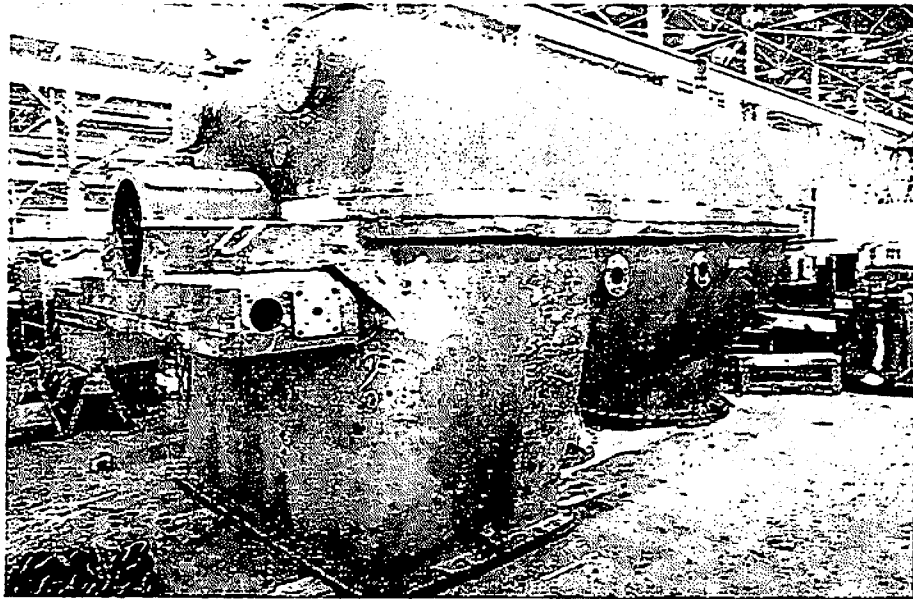


EDS-CD-153
CD-R00-17

Axial compressor performance can also be changed with variable angle stator blades or guide vanes. The angle of the first few rows of stator blades are adjustable. By changing the stator angle, the flow can be throttled. If a FCC Main Air Blower has a power recovery hot gas expander and a motor-generator, the whole compressor train (expander, steam turbine, axial air blower, motor-generator) is run at synchronous speed. Therefore, with no speed control, the only process control is with variable guide vanes.

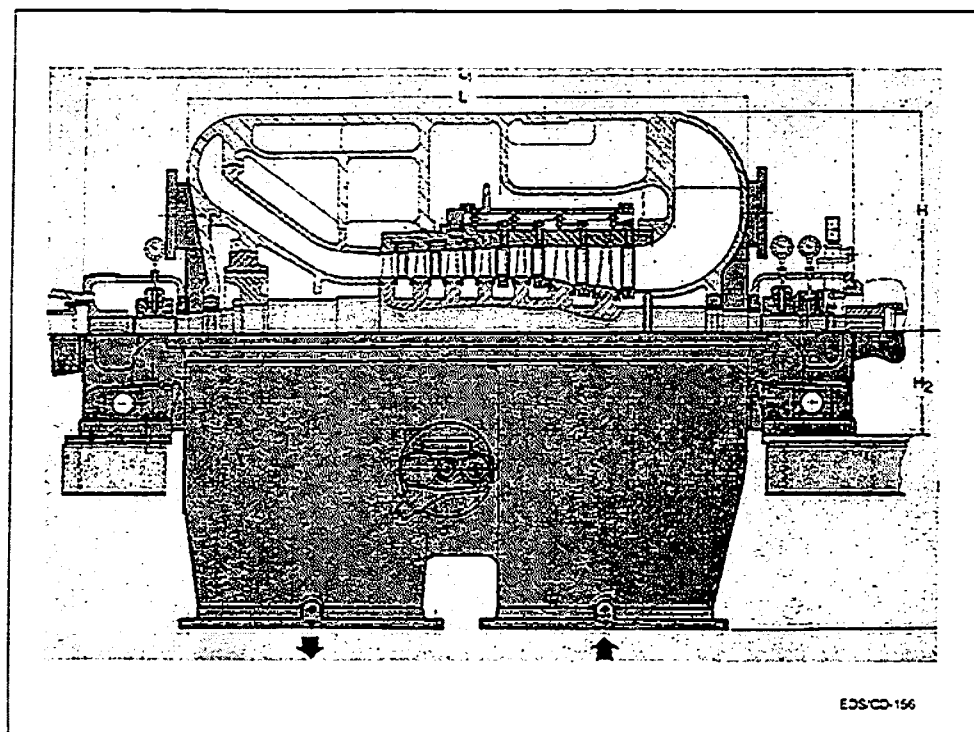


Axial compressor in bottom half of casing. Axial compressors are used primarily as FCC Main Air Blowers. The axial compressor is 5-10 efficiency points higher than a centrifugal compressor. It is also lighter and smaller than a centrifugal. They become cost effective at higher flow rates and horsepower. UOP has limited experience with axial compressors on process gas. The axial blades are not as rugged as centrifugal wheels and more susceptible to damage from condensate and particles in the gas and to surging.

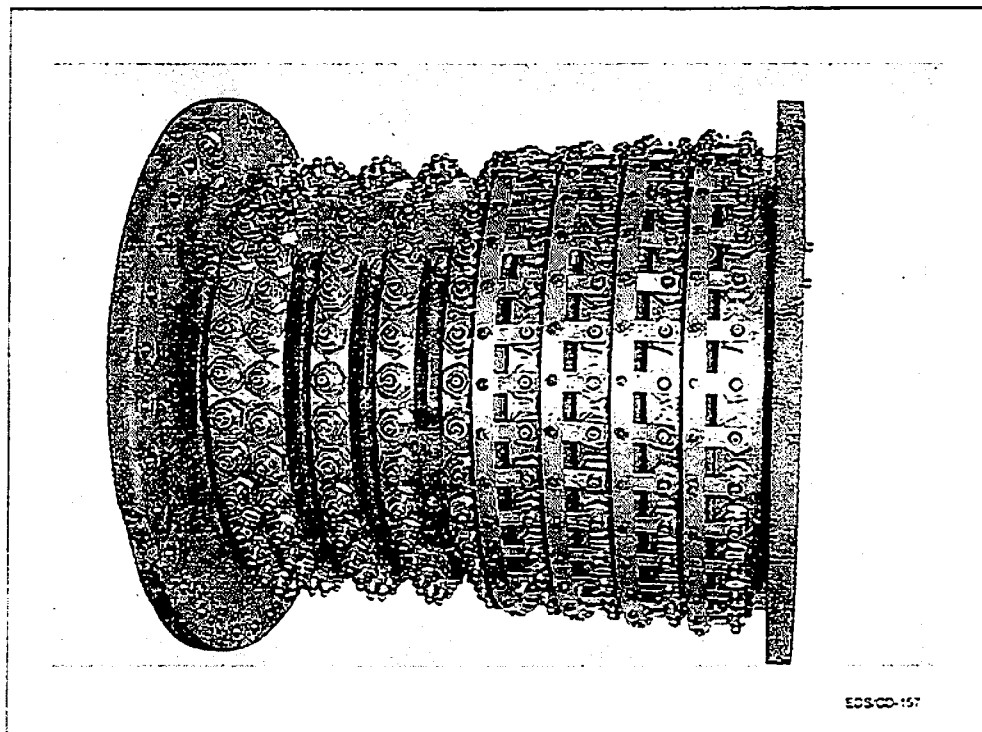


EDS/CD-155

Horizontally split, axial compressor casing. The axial compressor may have a radial (as shown) or axial inlet.



In this example, the first four stators are adjustable.



Axial compressor stator. Note how the first four are adjustable. Adjustable stators are used in clean service only.

Axial Compressor Control

- **Variable Speed**
- **Variable Stators**
- **Bypass (snort for air blower)**

EDS/CO-158

Axial Compressor Advantages

- **High Efficiency**
- **Smaller Relative Casing Size (Lower Cost)**
- **Low Maintenance - High Reliability**
- **Speed Match With Power Recovery Turbine**
- **Constant Volume for Wide Pressure Range**
- **Compatibility With Driver**
- **Smooth Flow**

EDS CD-155

Advantages of axial compressors over centrifugal and reciprocating compressors.

Many FCC air blowers are coupled to a power recovery hot gas expander. Since many larger axial compressors run at relatively low speeds, they are a good speed match for the expander.

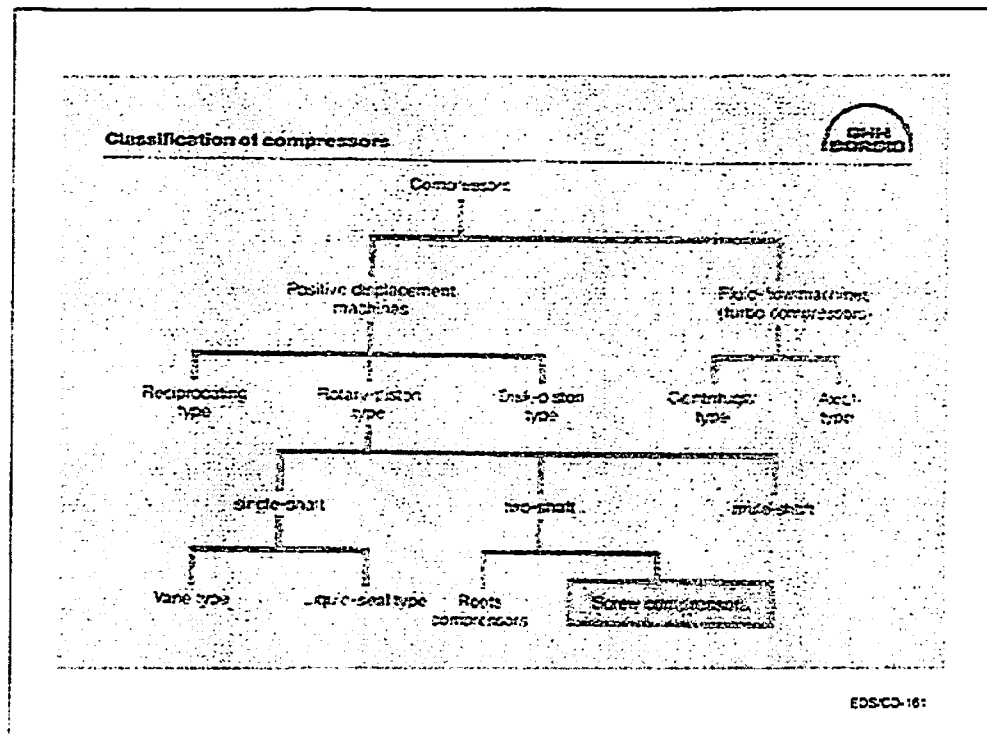
Like centrifugals, axial compressors have smooth flows.

Axial Compressor Disadvantages

- **Decay of Performance With Blade Fouling**
- **Sensitivity to Surging**
- **Sophisticated Technology**
- **Costs**
- **Limited Experience With Hydrocarbons**

EDS-CD-160

Axial compressor disadvantages compared to a centrifugal. Currently, there are a few axial reactor effluent compressors in operation and more are in production. We are slowly gaining axial compressor experience using hydrocarbon gas. So far, experience is good with particular vendors.



Screw compressors are positive displacement, two-shaft compressors.

Screw Compressors

- No spare - 99% on stream reliability
- vs. Centrifugal - can be more expensive, noisy, less efficient
- vs. Recip - no spare, usually less expensive compared to two recips, less efficient, less maintenance

EOS/CO-162

Screw compressors have been used on a limited basis. They are a good hydraulic fit for Pacol recycle gas compressors, Styrene off gas compressors, and PSA product compressors. They have been considered for Platforming Net Gas services but to date none are in operation.

Screw Compressors

- Oil Flooded
 - 6000 acfm
 - 250 psi differential pressure
 - 350 psi discharge pressure

EDS/CD-163

Screw Compressors

- Oil Flooded
 - Slide valve control
 - Oil carryover
 - No timing gear
 - Lower cost than dry screw
 - PSA units

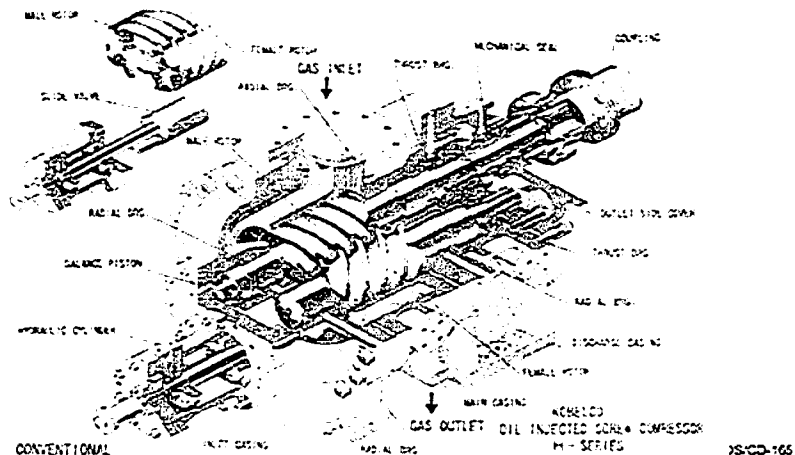
EDS/CO-164

Capacity on oil flooded compressors is with an internal spillback mechanism or slide valve.

Some oil will be carried downstream with the process.

High Reliability
Simple Structure, Principle, System Design, etc.

■ **Screw Compressor (Single Stage)**



Oil flooded screw compressor with hydraulic slide valve for capacity control.

Screw Compressors

■ Dry Screw

- 30,000 acfm
- 250 psi differential pressure
- 200 discharge pressure (over 20,000 acfm)
- 500 discharge pressure (under 10,000 acfm)

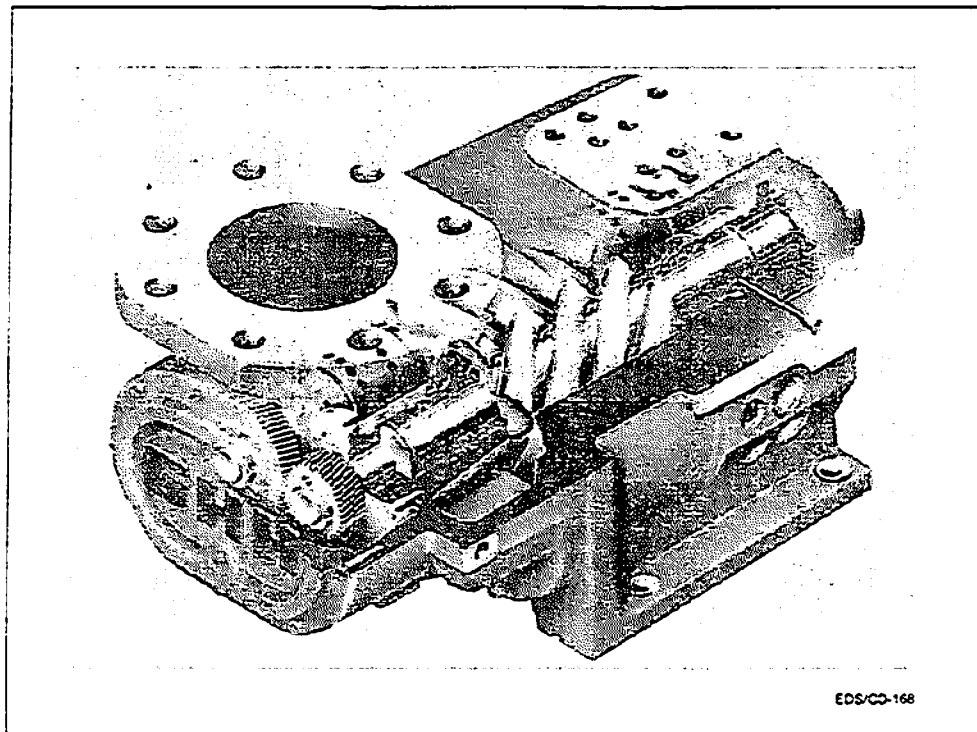
EDS/CO-166

Screw Compressors

■ Dry Screw

- No oil carryover
- Noisy - enclosure required
- Flow control - variable speed and spillback
- Styrene Off Gas, Pacol Recycle

EDS/CO-167

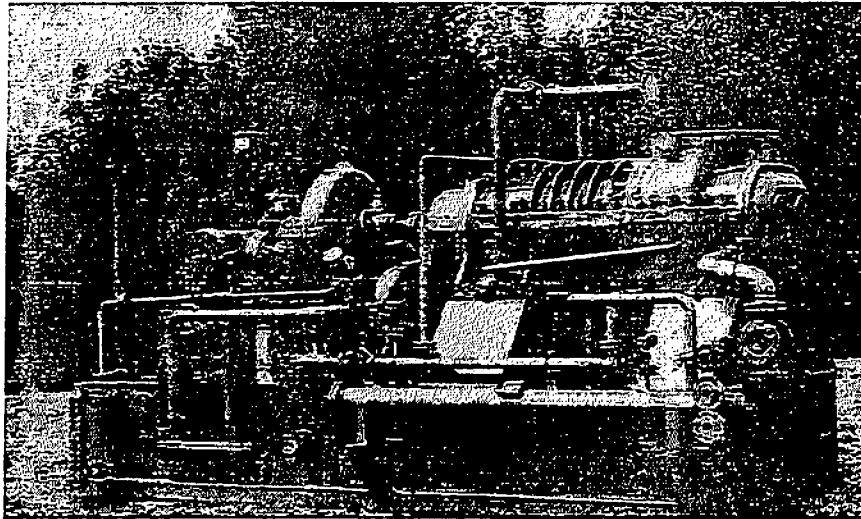


Dry screw compressor with timing gear. Dry screw compressors do not have slide valves so the only capacity control is with variable speed (steam turbine or variable speed motor) or with spillback.

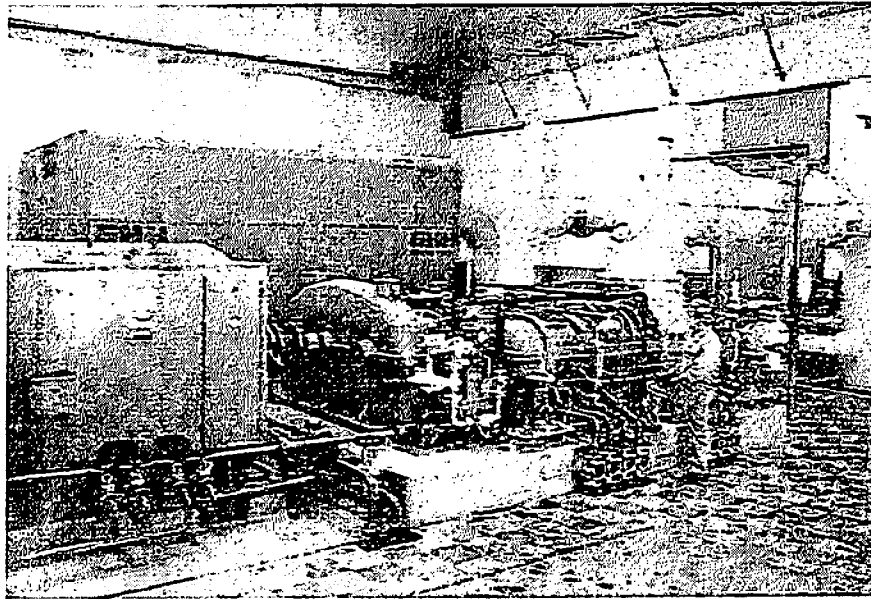


EDS CO-169

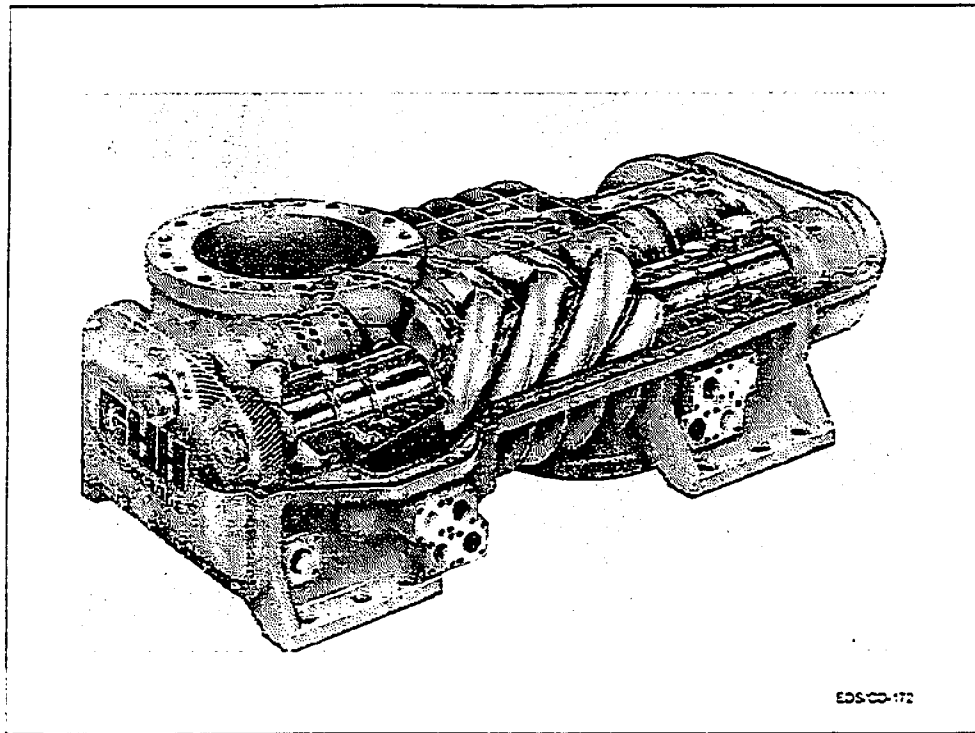
Two-stage screw compressor on common frame with a common driver and gear speed increaser.



EOS/CO-170

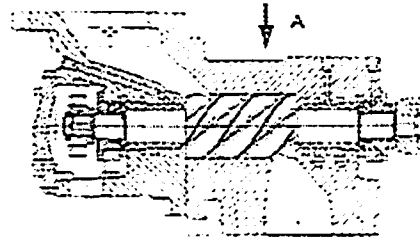


EOS-CD-171

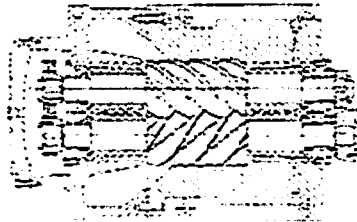


Dry screw compressor. Inlet is on top. Outlet on the bottom.

Process-gas screw compressors
Compressor Type CP



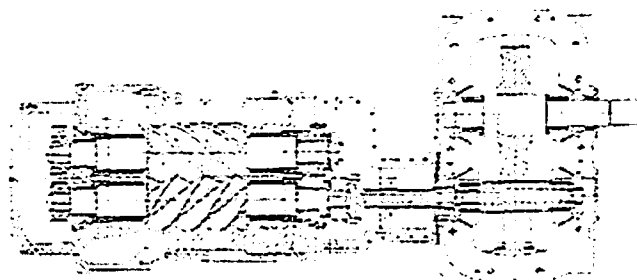
View A



Design Concept

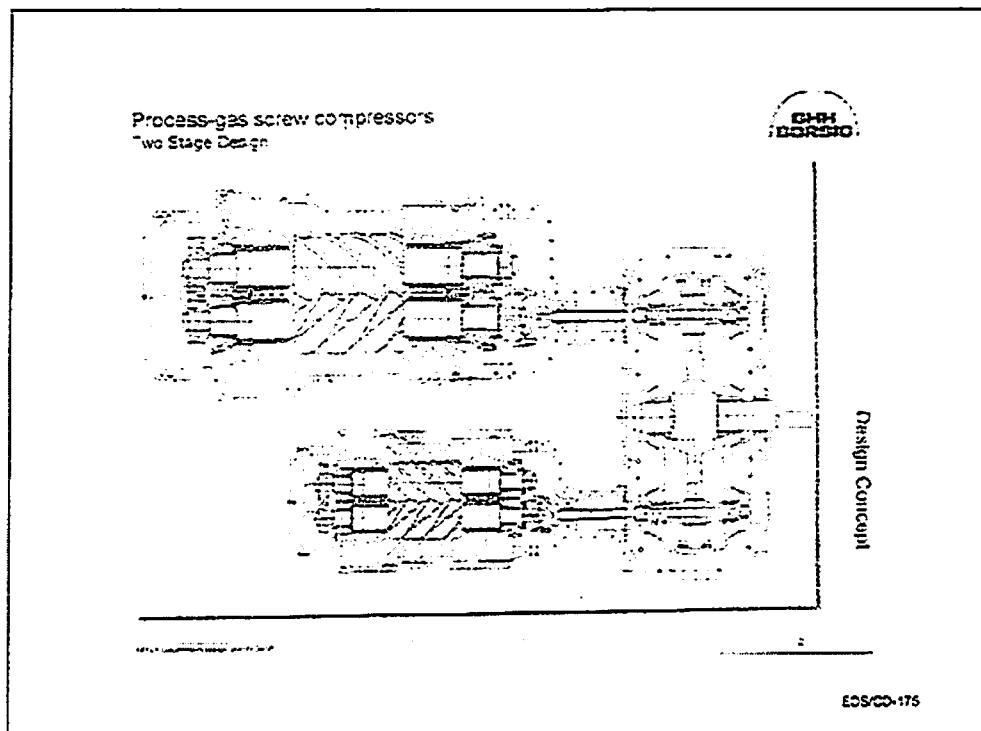
EDS/CD-173

Process-gas screw compressors
One Stage Design

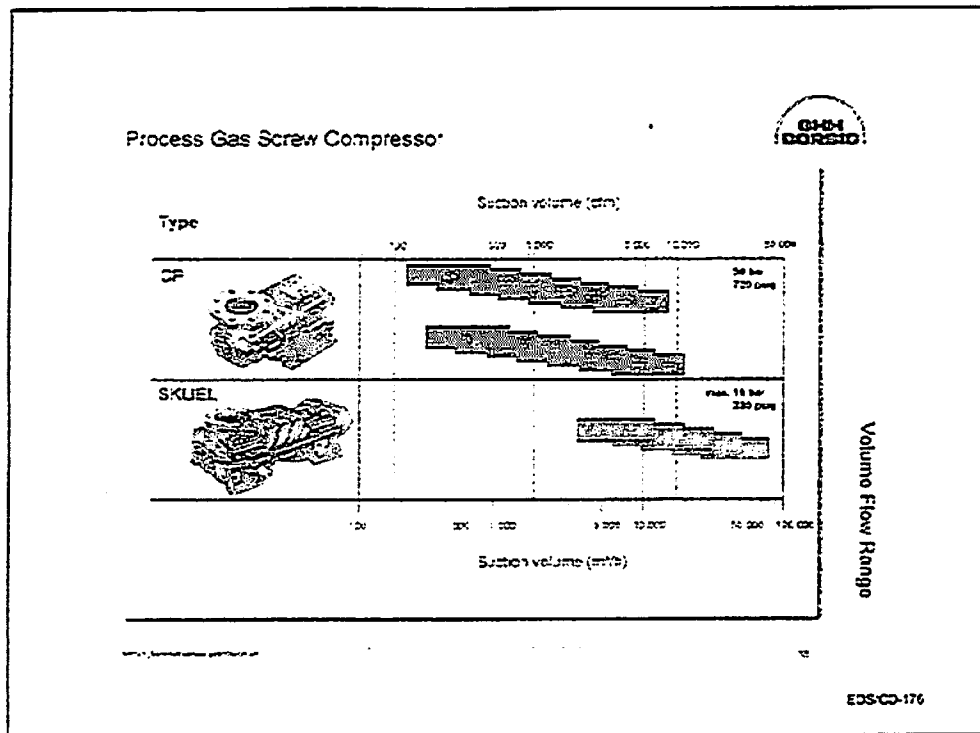


Design Concept

EDS/CO-174



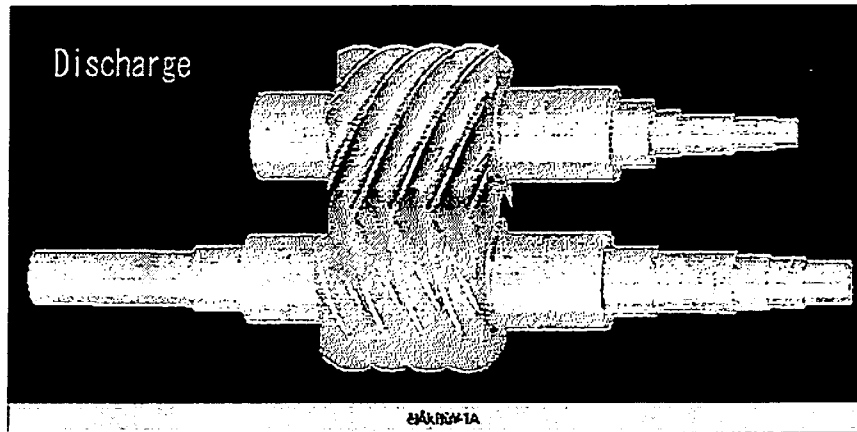
Two-stage dry screw compressor.



Capacity and pressure ranges for dry screw compressor

Principle of Compression

■ Oil Flooded Screw

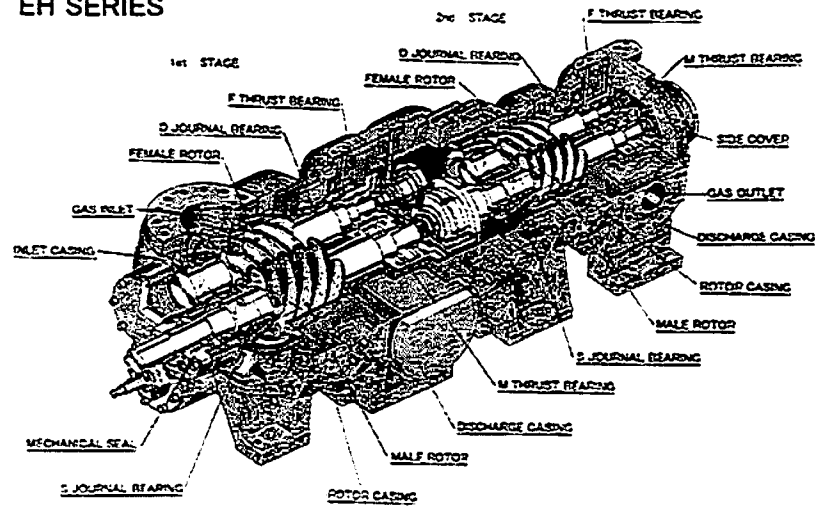


EDS-CD-177

Oil flooded screw compressor. There are no timing gears. Unlike a dry screw compressor the drive screw contracts and turns the idling screw. With the metal to metal contact the oil flood is required.

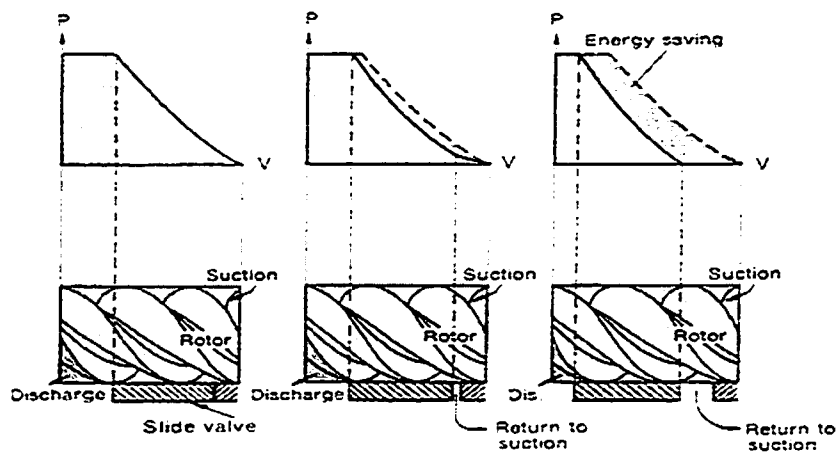
Screw Compressor (Tandem - Two Stages) for high pressure ratio case

EH SERIES



Two stage oil flooded screw compressor

Slide Valve Mechanism



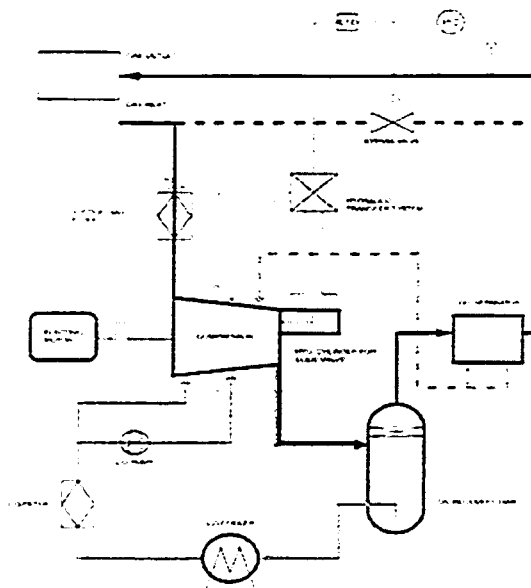
Capacity control by slide valve

Oil flooded screw compressor slide valve. Capacity can be controlled between 100% and 15% of rated with the slide valve which is an internal spillback. On the left the valve is closed so the capacity is 100%. In the middle the slide valve is slightly open. If this setting allows 5% capacity through the slide valve the compressor capacity will be 95%. On the right the slide valve is further open allowing more gas to recycle and further reducing the capacity.

Typical System (Flow Diagram)

KOBELCO OIL-INJECTED SCREW COMPRESSOR
COMPRESSOR CONTROL SYSTEM

- Red: Equipment
- Blue: Gas
- Yellow: Oil
- Oil Carryover:
- less than 1 ppm
- by weight



Oil flooded screw compressor schematic. Oil is pumped into the compressor and mixes with the process gas. The oil is recovered and goes through a cooler and exchanger before being pumped back into the compressor. The process gas leaves the recovery tank with some oil. The oil is further separated from the process gas in a separator or coalescer depending upon process requirements.

Process-gas Screw Compressors
Advantages in Comparison with Reciprocating Compressors



- ☐ No valves, requiring maintenance.
- ☐ No piston rings, which can wear.
- ☐ No mixed friction on such important parts as crossheads, piston pins, support plates, etc.
- ☒ No oscillating masses, better vibration behaviour.
- ☐ No low frequency pulsations of the gas at the discharge side.
- ☐ High availability, up to 99 %.

EDS/CO-161

Process-gas Screw Compressors
Advantages in Comparison with Centrifugal Compressors



- ☐ Nearly vertical performance curve, no surge line.
- ☐ Excellent part load behaviour with variable speed drive:
for instance: 50 % flow means 50 % speed at only 50 % power consumption.
- ☐ Compression of light molecular weight gases with fewer stages.
- ☐ Due to the rotor shape the 1st lateral critical speed is above the operating speed (rigid design).
- ☐ Clearly defined direction of relatively high bearing forces:
oil critical speeds are unknown.

EDS/CO-182

Screw Compressors

	DRY SCREW	RECIPROCATING
Tolerance for dust or particulate	Good	Poor
Tolerance for liquids	Excellent	Poor
Tolerance for MW variations	Excellent	Excellent
Reliability	Excellent (99.5%)	Fair (93%)
Plot space	Good	Poor
Operating efficiency	Poor	Excellent
Capacity regulation	Poor (for single speed)	Fair
Gas pulsation	Good	Poor
Gas polymerization	Excellent	Poor

EDS/CD-183

Screw Compressors

	DRY SCREW	CENTRIFUGAL
Tolerance for dust or particulate	Good	Good
Tolerance for liquids	Excellent	Good
Tolerance for MW variations	Excellent	Poor
Surge	No	Yes
Reliability	Excellent (99.5%)	Excellent (99.5%)
Plot space	Good	Good
Operating efficiency	Poor	Good
Capacity regulation	Poor (for single speed)	Good
Gas pulsation	Good	Excellent
Gas polymerization	Excellent	Poor

EDS/CD-184



#

UOP LLC • 25 East Algonquin Road • Des Plaines, Illinois 60017-5017 USA

**CENTRIFUGAL COMPRESSORS
AND DRIVERS****PROJECT SPECIFICATION**

#REF! - 508

SHEET #

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

APPLICABLE TO: ☐ PROPOSAL ☐ PURCHASE ☐ AS BUILT

FOR

UNIT

SITE

SERIAL NO.

SERVICE

#REF!

ITEM NO.

#REF!

NO. REQUIRED

☒ CONTINUOUS☐ INTERMITTENT☐ STAND BY

DRIVER TYPE (3.1.1)

MANUFACTURER

MODEL

DRIVER ITEM NO.

NOTE: INFORMATION TO BE COMPLETED:

☐

BY PURCHASER

☐

BY MANUFACTURER

OPERATING CONDITIONS

(ALL DATA ON PER UNIT BASIS)

☐ GAS HANDLED (ALSO SEE PAGE 3)
☐ 0 (760 mmHg & 15°C) (WET)☐ WEIGHT FLOW 0 (WET)**INLET CONDITIONS**☐ PRESSURE 0☐ TEMPERATURE 0☐ RELATIVE HUMIDITY %☐ MOLECULAR WEIGHT☐ Cp/Cv (K₁)☐ COMPRESSIBILITY (Z₁)☐ INLET VOLUME 0 (WET)**DISCHARGE CONDITIONS**☐ PRESSURE 0☐ TEMPERATURE 0☐ Cp/Cv (K₂)☐ COMPRESSIBILITY (Z₂)☐ 0 REQUIRED (ALL LOSSES INCLUDED)☐ SPEED (RPM)☐ ESTIMATED SURGE. m³/hr (AT SPEED ABOVE)☐ POLYTROPIC HEAD 0☐ POLYTROPIC EFFICIENCY (%)☐ CERTIFIED POINT☐ PERFORMANCE CURVE NUMBER**PROCESS CONTROL**METHOD ☐ SUCTION THROTTLING ☐ VARIABLE INLET
FROM _____ GUIDE VANES
TO _____ (3.4.2.4)☐ SPEED VARIATION
FROM _____ %
TO _____ %☐ DISCHARGE
BLOWOFF
TO _____☐ COOLED BYPASS
FROM _____
TO _____

SIGNAL SOURCE (3.4.2.1)

TYPE ☐ ELECTRONIC ☐ PNEUMATIC ☐ OTHER
RANGE _____ mA _____ psig

ANTI-SURGE BYPASS

☐ MANUAL☐ AUTOMATIC☐ NONE

REMARKS:

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CENTRIFUGAL COMPRESSORS AND DRIVERS

REV

PROJECT SPECIFICATION

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SHEET #

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

CONSTRUCTION FEATURES

☐ SPEEDS:

MAX. CONT. _____ RPM TRIP _____ RPM

MAX. TIP SPEEDS: _____ FPS @ 100% SPEED

_____ FPS @ MAX. CONT. SPEED

☐ LATERAL CRITICAL SPEEDS (DAMPED) PER API-617

FIRST CRITICAL _____ RPM _____ MODE

SECOND CRITICAL _____ RPM _____ MODE

THIRD CRITICAL _____ RPM _____ MODE

FOURTH CRITICAL _____ RPM _____ MODE

☐ TRAIN LATERAL ANALYSIS REQUIRED (2.9.2.3)

☐ TRAIN TORSIONAL ANALYSIS REQUIRED

(TURBINE DRIVEN TRAIN) (2.9.4.5)

☐ TORSIONAL CRITICAL SPEEDS: PER API-617

FIRST CRITICAL _____ RPM

SECOND CRITICAL _____ RPM

THIRD CRITICAL _____ RPM

FOURTH CRITICAL _____ RPM

☐ VIBRATION:

ALLOWABLE TEST LEVEL _____ mils

(PEAK TO PEAK)

☐ ROTATION, VIEWED FROM DRIVEN END CW CCW

☐ MATERIALS INSPECTION REQUIREMENTS (4.2.2)

☐ SPECIAL CHARPY TESTING (2.11.3)

☐ RADIOGRAPHY REQUIRED FOR _____

☐ ULTRASONIC REQUIRED FOR _____

☐ MAGNETIC PARTICLE REQUIRED FOR _____

☐ LIQUID PENETRANT REQUIRED FOR _____

CASING:

☐ MODEL _____

CASING SPLIT _____

MATERIAL _____

THICKNESS mm _____ CORR. ALLOW. mm _____

MAX. WORKING PRESS _____

MAX DESIGN PRESS _____ ###

TEST PRESS : HELIUM _____ HYDRO _____

MAX OPER. TEMP. _____ 0 MIN. OPER. TEMP. _____ 0

MAX NO. OF IMPELLERS FOR CASING _____

MAX CASING CAPACITY M³/HR _____

CASING SPLIT SEALING (2.2.10) _____

☐ SYSTEM RELIEF VALVE SET PT. (2.2.4) _____ 0

☐ DIAPHRAGMS:

MATERIAL _____

☐ IMPELLERS:

NO. _____ DIAMETERS _____

NO. VANES EA. IMPELLER _____

TYPE (OPEN, ENCLOSED, ETC.) _____

TYPE FABRICATION _____

MATERIAL _____

MAX. YIELD STRENGTH _____

BRINNEL HARDNESS: MAX _____ MIN _____

SMALLEST TIP INTERNAL WIDTH mm _____

MAX. MACH. NO. @ IMPELLER EYE _____

MAX. IMPELLER HEAD @ 100% SPD (FT-LBS/LB) _____

☐ SHAFT:

MATERIAL _____

DIA @ IMPELLERS mm _____ DIA @ COUPLING mm _____

SHAFT END: ☐ TAPERED ☐ CYLINDRICAL

MAX. YIELD STRENGTH 0 _____

SHAFT HARDNESS (BNH)(Rc) _____

MAX TORQUE CAPABILITY (N·m) _____

☐ BALANCE PISTON:

MATERIAL _____ AREA mm² _____

FIXATION METHOD _____

☐ SHAFT SLEEVES:

AT INTERSTG. CLOSE CLEARANCE POINTS MATL _____

AT SHAFT SEALS MATL 0 _____

☐ ACCESSIBLE(2.8.2)

☐ LABYRINTHS:

INTERSTAGE

TYPE _____ MATERIAL _____

BALANCE PISTON

TYPE _____ MATERIAL _____

SHAFT SEALS:

☐ SEAL TYPE (2.8.3) _____

☐ SETTLING OUT PRESSURE _____

☐ SPECIAL OPERATION (2.8.1) _____

☐ SUPPLEMENTAL DEVICE REQUIRED FOR CONTACT

SEALS (2.8.3.2) TYPE _____

☐ BUFFER GAS SYSTEM REQUIRED (2.8.7) MANIFOLD (3.5.1.6)

☐ TYPE BUFFER GAS _____

☐ BUFFER GAS CONTROL SYSTEM SCHEMATIC BY VENDOR

☐ PRESSURIZING GAS FOR SUBATMOSPHERIC SEALS (2.8.8)

☐ TYPE SEAL _____

☐ INNER OIL LEAKAGE GUAR _____

BUFFER GAS REQUIRED FOR:

☐ AIR RUN-IN ☐ OTHER _____

☐ BUFFER GAS FLOW (PER SEAL):

NORM: _____ KG /MIN @ _____ FALSE D P _____

MAX. _____ KG /MIN @ _____ FALSE D P _____

☐ BEARING HOUSING CONSTRUCTION:

TYPE (SEPARATE, INTEGRAL) _____ SPLIT _____

MATERIAL _____

###

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#REF!	#REF!	#REF!					

REV

CONSTRUCTION FEATURES (CONTINUED)**BEARINGS AND BEARING HOUSINGS**

RADIAL	INLET	EXHAUST	THRUST	ACTIVE	INACTIVE
☐ TYPE			☐ TYPE		
☐ MANUFACTURER			☐ MANUFACTURER		
☐ LENGTH mm			☐ UNIT LOADING (MAX 0)		
☐ SHAFT DIA. mm			☐ UNIT LOAD (ULT.) (0)		
☐ UNIT LOAD (ACT/ALLOW)			☐ AREA mm ²		
☐ BASE MATERIAL			☐ NO. PADS		
☐ BABBIT THICKNESS mm			☐ PIVOT: CENTER / OFFSET, %		
☐ NO. PADS			☐ PAD BASE MATL		
☐ LOAD: BTWN/ON PAD			LUBRICATION: ☐ FLOODED ☐ DIRECTED		
☐ PIVOT: CTR/OFFSET, %			THRUST COLLAR: ☐ INTEGRAL ☐ REPLACEABLE		
☐			MATERIAL		
☐ BEARING SPAN mm					

BEARING TEMPERATURE DEVICESSEE ATTACHED
API-670 DATA SHT

☐ THERMISTORS

☐ TYPE POS TEMP COEFF NEG TEMP COEFF

☐ TEMP SWITCH & INDICATOR BY: PURCH MFR

☐ THERMOCOUPLES

☐ SELECTOR SWITCH & INDIC. BY: PURCH MFR

☐ RESISTANCE TEMP DETECTORS

☐ RESISTANCE MATL OHMS

☐ SELECTOR SWITCH & INDIC. BY: PURCH MFR

☐ LOCATION-JOURNAL BRG

NO. EA PAD EVERY OTH PAD PER BRG

OTHER

☐ LOCATION-THRUST BRG

NO. EA PAD EVERY OTH PAD PER BRG

OTHER

NO. (INACT) EA PAD EVERY OTH PAD PER BRG

OTHER

MONITOR SUPPLIED BY (3.4.7.4)

LOCATION ENCLOSURE

MFR ☐ MODEL

☐ SCALE RGE ALARM ☐ SET @ 0

SHTDWN ☐ SET @ 0 TIME DELAY SEC

VIBRATION DETECTORS:

SEE ATTACHED API-670

☐ TYPE ☐ MODEL

☐ MFR

☐ NO. AT EA SHAFT BEARING TOTAL NO.

☐ OSCILLATOR-DETECTORS SUPPLIED BY

☐ MFR ☐ MODEL

MONITOR SUPPLIED BY (3.4.7.2)

☐ LOCATION ENCLOSURE

☐ MFR ☐ MODEL

☐ SCALE RGE ALARM ☐ SET @ MILS

☐ SHTDWN: ☐ SET @ MILS ☐ TIME DELAY SEC

AXIAL POSITION DETECTOR:SEE ATTACH. API-670
DATA SHEET

☐ TYPE ☐ MODEL

☐ MFR ☐ NO. REQUIRED

☐ OSCILLATOR-DEMODULATOR SUPPLIED BY

☐ MFR ☐ MODEL

MONITOR SUPPLIED BY (3.4.7.2)

☐ LOCATION ENCLOSURE

☐ MFR ☐ MODEL

☐ SCALE RGE ☐ ALARM ☐ SET @ MILS

☐ SHTDWN: ☐ SET @ MILS ☐ TIME DELAY SEC

CASING CONNECTIONS

CONNECTION	DESIGN APPROVAL REQ'D (2.11.2.10)	☐ ASME ☐ API605 ☐ OTHER SIZE	☐ FACING	POSITION (2.4.2.1)	☐ FLANGED OR STUDDED (2.4.2.1)	MATING FLG & GASKET BY VENDOR (2.4.2.3)	☐ GAS VELOCITY m/s
INLET							
DISCHARGE							

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**CENTRIFUGAL COMPRESSORS
AND DRIVERS**

REV

☐ OTHER CONNECTIONS

SERVICE:

NO.

SIZE

TYPE

LUBE-OIL INLET

LUBE OIL OUTLET

SEAL-OIL INLET

SEAL-OIL OUTLET

SEAL GAS INLET

SEAL GAS OUTLET

CASING DRAINS

STAGE DRAINS

VENTS

COOLING WATER

PRESSURE

TEMPERATURE

SOLVENT INJECTION

PURGE FOR:

BRG. HOUSING

BTWN BRG & SEAL

BTWN SEAL & GAS

☐ INDIVIDUAL STAGE DRAINS REQUIRED (2.4.3.2)☐ VALVED & BLINDED☐ VALVED & BLINDED & MANIFOLD

KEY PHASOR REQUIRED

☐ COMPRESSOR ☐ GEAR ☐ DRIVER

Provide for Complete Drainage

PROJECT SPECIFICATION

#REF! - 508

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DATE

BY

APVD

REV

DATE

BY

APVD

#REF!

#REF!

#REF!

☐ ALLOWABLE PIPING FORCES AND MOMENTS:

INLET

DISCHARGE

FORCE

FORCE

MOMT

MOMT

N

N

N-m

N-m

N

N

N-m

N-m

AXIAL

VERTICAL

HORIZ. 90

AXIAL

VERTICAL

HORIZ. 90

☐ CASING VIBRATION TRANSDUCER (3.4.7.5)☐ SEE ATTACHED API-670 DATA SHEET☐ TYPE _____ MODEL _____☐ MFR _____ NO. REQUIRED _____☐ LOCATION _____☐ OSCILLATOR-DEMODULATORS SUPPLIED BY☐ MFR _____ ☐ MODEL _____☐ MONITOR SUPPLIED BY (3.4.7.6)☐ LOCATION _____ ENCLOSURE _____☐ MFR _____ ☐ MODEL _____☐ SCALE RANGE _____ ALARM ☐ SET @ _____ mm/s²☐ SHTDWN ☐ SET @ _____ mm/s² ☐ TIME DELAY _____ SEC**ACCESSORIES****COUPLING AND GUARDS (3.2)**

NOTE: SEE ROTATING ELEMENTS - SHAFT ENDS

☐ SEE ATTACHED API-671 DATA SHEET☐ KEYLESS HYDRAULIC☐ KEYED

COUPLING FURNISHED BY _____

MANUFACTURER _____

TYPE _____

MODEL _____

COUPLING GUARD FURNISHED BY: _____

TYPE: ☐ FULLY ENCLOSED☐ SEMI-OPEN

OTHER _____

COUPLING DETAILS☐ MAX O.D. _____ mm☐ HUB WEIGHT _____ kgs☐ SPACER LENGTH _____ mm☐ SPACER WEIGHT _____ kgs☐ VENDOR MOUNT HALF COUPLING

LUBRICATION REQUIREMENTS:

☐ NON-LUBE ☐ GREASE☐ CONT. OIL LUBE☐ OTHER

QUANTITY PER HUB _____ kgs

MOUNTING PLATES☐ BASEPLATES: FURNISHED BY (3.3.1.1) _____☐ COMPRESSOR ONLY (3.3.2.1) ☐ DRIVER ☐ GEAR☐ OTHER _____☒ DRIP RIM☒ LEVELING PADS (3.3.2.2)☐ COLUMN MOUNTING (3.3.2.3)☐ SUB-SOLE PLATES REQ'D (3.3.3.2)☐ STAINLESS STEEL SHIM THICKNESS _____ mm

PRIMER FOR EPOXY GROUT REQ'D (3.3.1.2.9)

TYPE _____

☐ BASE PLATE WILL BE ON CONCRETE FOUNDATION (3.3.2.5)

MACHINED MOUNTING PADS REQ'D. (3.3.2.6)

SOLEPLATES: FURNISHED BY: _____

☐ THICKNESS _____ mm☐ SUBSOLE PLATES REQ'D (3.3.3.2)☐ LEVELING BLOCKS REQ'D☐ STAINLESS STEEL SHIM THICKNESS - _____ mm☐ DRIVER ☐ GEAR ☐ COMPRESSOR☐ PRIMER FOR EPOXY GROUT REQ'D (3.3.1.2.9)

TYPE _____

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**CENTRIFUGAL COMPRESSORS
AND DRIVERS****PROJECT SPECIFICATION**

#REF! - 508

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REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

UTILITIES☐ **UTILITY CONDITIONS:**

STEAM:	DRIVERS	HEATING
INLET MIN	## °C	### °C
NORM	## °C	### °C
MAX	## °C	### °C
EXHAUST. MIN	## °C	### °C
NORM	## °C	### °C
MAX	## °C	### °C

ELECTRICITY: (3.4.6.1)

	DRIVERS	HEATING	CONTROL	SHUTDOWN
VOLTAGE				
HERTZ				
PHASE				
COOLING WATER:				
TEMP. INLET		MAX RETURN		
PRESS NORM		DESIGN		
MIN RETURN		MAX ALLOW D P		
WATER SOURCE				
INSTRUMENT AIR:				
MAX PRESS		MIN PRESS		

TOTAL UTILITY CONSUMPTION:

COOLING WATER	m ³ /hr
STEAM, NORMAL	kg/hr
STEAM, MAX	kg/hr
INSTRUMENT AIR	m ³ /hr
HP (DRIVER)	kW
HP (AUXILIARIES)	kW
HEATERS	kW
PURGE (AIR OR N ₂)	m ³ /hr

MISCELLANEOUS:

- ☒ RECOMMENDED STRAIGHT RUN OF PIPE DIAMETERS BEFORE SUCTION
- ☐ NOMOGRAPHS REQUIRED FOR EACH SELECTION (3.3.1.5)
- ☒ VENDOR'S REVIEW & COMMENTS ON PURCHASERS PIPING & FOUNDATION (3.5.3.2)
- ☐ COMPRESSOR TO BE SUITABLE FOR FIELD RUN IN ON AIR (2.1.17)
- ☐ PROVISION FOR LIQUID INJECTION (2.1.11)
- ☐ VENDOR'S REVIEW & COMMENTS ON PURCHASERS CONTROL SYSTEMS (3.4.1.1)
- ☐ EXTENT OF PROCESS PIPING BY VENDOR (3.5.3.1)
- ☐ SHOP FITUP OF VENDOR PROCESS PIPING (4.4.3.11)
- ☐ WELDING HARDNESS TESTING (4.2.1.6)

VENDOR'S REPRESENTATIVE SHALL (2.1.14)

- ☐ OBSERVE FLANGE PARTING
- ☐ CHECK ALIGNMENT AT TEMPERATURE
- ☐ BE PRESENT AT INITIAL ALIGNMENT

☐ **WEIGHTS (kg):**

COMPR.	GEAR	DRIVER	BASE
ROTORS:	COMPR.	DRIVER	GEAR
COMPRESSOR UPPER CASE			
SOUR SEAL OIL TRAPS			
L.O. CONSOLE		S.O. CONSOLE	
OVERHEAD SEAL OIL TANKS			
MAX. FOR MAINTENANCE (IDENTIFY)			
TOTAL SHIPPING WEIGHT			

☐ **SPACE REQUIREMENTS (M & MM):**

COMPLETE UNIT:	L	W	H
L.O. CONSOLE:	L	W	H
S.O. CONSOLE:	L	W	H
SOUR SEAL OIL TRAPS			
OVERHEAD SEAL OIL TANKS			

REMARKS: Fouling Factor:

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#REF!	#REF!	#REF!					

REV

LUBE OIL AND SEAL SYSTEMS**OVERALL SYSTEMS SCHEMATICS**DRY GAS SEAL (UOP STANDARD DRAWING S-109) ☐COMBINED LUBE, SEAL, AND CONTROL OIL SYSTEM (API 614 FIG. A1) ☐SEPARATE SEAL OIL SYSTEM (API 614 FIG. A2-A6) ☐WITH BOOSTER PUMPS (API 614 FIG. A2-A3) ☐LUBE/CONTROL OIL SYSTEMS (API 614 FIG. A8-A9) ☐

WASTE OIL TRAPS VENT TO

☐ ATMOSPHERE AT SAFE LOCATION☐ FLARE☐ COMPRESSOR SUCTION☐ COMPRESSOR SUCTION DURING PROCESSING AND TO
FLARE DURING CATALYST REGENERATION☐ OTHER _____

MAIN OIL PUMP

STANDBY OIL PUMP DRIVER

RESERVOIR HEATING

☐ TURBINE☐ MOTOR☐ TURBINE☐ MOTOR☐**INSTRUMENTATION**

COMPRESSOR SHUTDOWN SYSTEM

ENERGIZE TO TRIP COMPRESSOR

DE-ENERGIZE TO TRIP COMPRESSOR

LOCAL INSTRUMENT PANEL

☐ SEE UOP STANDARD SPECIFICATION 5-15, FIGURE 1☐ DETAIL A☐ DETAIL B☐**THERMOWELLS**

BALANCE LINE(1), DISCHARGE LINE (3) IF REQUIRED

MATERIAL

☐ 316 SS☐ MONEL☐

FLANGE RATING, FACING

Class 300, RF

FABRICATION (LENGTH = 10" (250 mm))

☐ UOP DRAWING 6-105, DETAIL A

VIBRATION MONITORING SYSTEM

☐ UOP STANDARD SPECIFICATION 5-19

INSTALLATION DRAWINGS AND STANDARDS

PRESSURE GAUGES

☐ UOP DRAWING 6-101

PRESSURE INSTRUMENTS

☐ UOP DRAWING 6-102

THERMOWELLS

☐ UOP DRAWING 6-108

LEVEL INSTRUMENTS

☐ UOP DRAWING 6-109

GAUGE GLASSES

☐ UOP DRAWING 8-101

THERMOMETER ASSEMBLIES

☐ UOP DRAWING 6-106

CONTROL VALVES

☐ UOP STANDARD SPECIFICATION 6-14

FLOW METERS

☐ UOP STANDARD SPECIFICATION 6-18

REMARKS:

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#REF!	#REF!	#REF!					

REV

ALARMS AND SHUTDOWNS

THE FOLLOWING COMPRESSOR ALARM AND SHUTDOWN DEVICES SHALL BE PROVIDED BY THE VENDOR.

	ALARMS	SHUTDOWNS (SEE NOTE E)
LUBE OIL LOW PRESSURE	☐	☐
CONTROL OIL LOW PRESSURE	☐	☐
LUBE OIL HIGH TEMPERATURE	☐	☐
LUBE OIL RESERVOIR LOW LEVEL	☐	☐
LUBE OIL FILTER HIGH PRESSURE DIFFERENTIAL	☐	☐
LUBE OIL STANDBY PUMP RUNNING	☐	☐
COMPRESSOR BEARING HIGH TEMPERATURE	☐	☐
COMPRESSOR SHAFT EXCESS VIBRATION	☐	☐
COMPRESSOR AXIAL POSITION	☐	☐
MOTOR BEARING HIGH TEMPERATURE	☐	☐
TURBINE BEARING HIGH TEMPERATURE	☐	☐
TURBINE SHAFT EXCESS VIBRATION	☐	☐
TURBINE AXIAL POSITION	☐	☐
TURBINE SHAFT SEAL EDUCTOR HIGH PRESSURE	☐	☐
HIGH SPEED	☐	☐
COMPRESSOR DISCHARGE HIGH TEMPERATURE	☐	☐
MOTOR STATOR HIGH TEMPERATURE	☐	☐
HIGH MOTOR AMPERAGE	☐	☐

THE FOLLOWING COMPRESSOR ALARM AND SHUTDOWN DEVICES SHALL BE PROVIDED BY THE CONTRACTOR:

HIGH LIQUID LEVEL IN SUCTION DRUM (S) (SEE PROJECT SPECIFICATION 612)

☐☐

NOTES:

- (A) EACH DEVICE SHALL BE FURNISHED IN A SEPARATE HOUSING. THE CONTRACTOR SHALL SPECIFY ELECTRICAL RATING.
- (B) ALARM AND SHUTDOWN SWITCH CONTACTS SHALL OPEN ON ABNORMAL CONDITION, DEENERGIZING THE CIRCUIT TO INITIATE ALARMS OR SHUTDOWNS.
- (C) SINGLE-POLE, DOUBLE-THROW, HERMETICALLY SEALED SWITCHES SHALL BE USED.
- (D) ANNUNCIATOR SYSTEM (BY CONTRACTOR) SHALL BE IN ACCORDANCE WITH UOP PROJECT SPECIFICATION 623, THIS PROJECT.
- (E) REFER TO API STANDARD 617 AND UOP STANDARD SPECIFICATION 5-15, FIGURE 1 FOR ADDITIONAL REQUIREMENTS
- (F) CONTRACTOR TO COORDINATE DETAILS OF SHUTDOWN CIRCUITRY.

MINIMUM DESIGN METAL TEMPERATURE

P001

P001

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UOP		#		PROJECT SPECIFICATION			
UOP LLC • 25 East Algonquin Road • Des Plaines, Illinois 60017-5017 USA				#REF! - 509		SHEET 10	
REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					
RECIPROCATING COMPRESSORS AND DRIVERS							
☒ CONSTRUCTION FEATURES (CONTINUED)							
☐ FABRICATED CYLINDER, HEADS, & CONNECTION SKETCHES FOR DESIGN REVIEW BY PURCHASER. (2.14.5.2.8)				☐ BUFFER GAS PACKING ARR. Ref: Appendix I Figures I-1, I-2 & I-3 ☐ OIL WIPER PACKING PURGE ☐ INTERMEDIATE PARTITION PURGE INERT BUFFER PURGE GAS: ☐ N ₂ ☐ OTHER ☐ VENT. DRAIN. PURGE PIPING BY MFGR ☐ NO ☐ YES			
☐ COUPLING(S) ☐ LOW-SPEED ☐ HI-SPEED Between Compressor & Driver or Gear Between Driver & Gear ☐ BY MANUFACTURER ☐ MODEL ☐ TYPE API-671 APPLIES ☐ YES ☐ NO				☐ V-BELT DRIVE ☐ DRIVEN SHEAVE (Compressor Shaft) ☐ DRIVE SHEAVE (Driver Shaft) RPM (EXPECTED) PITCH DIA. (millimeters) ☐ QTY & GROOVE X-SEC. POWER TRANSMITTD DRIVER NAMEPLATE HP RATING ☐ CENTER DISTANCE (millimeters) ☐ QTY, TYPE, X-SEC., & LENGTH BELTS ☐ BELT SERVICE FACTOR (RELATIVE TO DRIVER NAMEPLATE HP RATING) Incl. Belt Losses			
☐ INSPECTION AND SHOP TESTS (REF. 4.1.3)				☒ CYLINDER LUBRICATION			
REQD WITN. OBSER.				☐ NON-LUBE ☐ STAGE(S)/SERVICE ☐ LUBRICATED ☐ STAGE(S)/SERVICE TYPE OF LUBE OIL: ☐ SYNTHETIC ☐ HYDROCARBON ☐ LUBRICATOR ☐ COMP. CRANKSHAFT, DIRECT DRIVE BY: ☐ CHAIN, FROM CRANKSHAFT ☐ ELECTRIC MOTOR ☐ OTHER LUBRICATOR MFR ☐ MODEL TYPE LUBRICATOR: ☐ SINGLE PLUNGER PER POINT (2.13) ☐ DIVIDER BLOCKS ☐ COMPARTMT, TOTAL QTY. ☐ PLUNGERS (PUMPS), TOTAL QTY. ☐ SPARE PLUNGERS, QTY. ☐ SPARE COMPARTMT W/OUT PLUNGERS ☐ HEATERS: ☐ ELECTRIC W/THERM.(S) ☐ STEAM			
*SHOP INSPECTION ☐ REQD ☐ WITN. ☐ OBSER. ACTUAL RUNNING CLEARANCES ☐ ☐ ☐ AND RECORDS ☐ ☐ ☐ MFG STANDARD SHOP TESTS ☒ ☐ ☐ CYLINDER HYDROSTATIC TEST ☒ ☐ ☐ CYLINDER PNEUMATIC TEST ☐ ☐ ☐ CYLINDER HELIUM LEAK TEST ☐ ☐ ☐ CYL. JACKET WATER HYDRO TEST ☒ ☐ ☐ *MECHANICAL RUN TEST (4 HR) ☒ ☐ ☐ BAR-OVER TO CHECK ROD RUNOUT ☒ ☐ ☐ *LUBE OIL CONSOLE RUN/TEST (4 HR) ☐ ☐ ☐ *COOLING H ₂ O CONSOLE RUN/TEST ☐ ☐ ☐ RADIOGRAPHY BUTT WELDS ☐ ☐ ☐ ☐ GAS ☐ OIL ☐ FAB CYLS. MAG PARTICLE/LIQUID ☐ ☐ ☐ PENETRANT OF WELDS ☐ ☐ ☐ SPECIFY ADDITIONAL REQUIREMENTS (4.2.1.3) ☐ ☐ ☐ QC OF INACCESSIBLE WELDS (2.14.5.2.4) ☐ ☐ ☐ SHOP FIT-UP OF PULSATION SUPPL. DEVICES & ALL ASSOCIATED GAS PIPING ☐ ☐ ☐ *CLEANLINESS OF EQUIP., PIPING, & APPURTENANCES ☐ ☐ ☐ *HARDNESS OF PARTS, WELDS & HEAT AFFECTED ZONES ☐ ☐ ☐ *NOTIFICATION TO PURCHASER OF ANY REPAIRS TO MAJOR COMPONENTS ☐ ☐ ☐ *SPECIFIC REQUIREMENTS TO BE DEFINED, FOR EXAMPLE, DISMANTLING, AUX EQUIPMENT OPERATIONAL & RUN TESTS. ☐ ☐ ☐ APPENDIX K COMPLIANCE: ☐ VENDOR ☐ PURCHASER				☐ ESTIMATED WEIGHTS AND NOMINAL DIMENSIONS ☐ TOTAL COMPR. WT, LESS DRIVER & GEAR _____ KGS ☐ WT. OF COMPLETE UNIT, (LESS CONSOLES) _____ KGS ☐ MAXIMUM ERECTION WEIGHT _____ KGS ☐ MAXIMUM MAINTENANCE WEIGHT _____ KGS ☐ DRIVER WEIGHT/GEAR WEIGHT _____ / _____ KGS ☐ LUBE OIL/COOLING H ₂ O CONS. _____ / _____ KGS ☐ FREE STANDING PANEL _____ SPACE REQUIREMENTS- METERS LENGTH WIDTH HEIGHT ☐ COMPLETE UNIT _____ ☐ LUBE OIL CONSOLE _____ ☐ COOLING H ₂ O CONSOLE _____ ☐ FREE STANDING PANEL _____ ☐ PISTON ROD REMOVAL DIST. _____ OTHER EQUIPMENT SHIPPED LOOSE (DEFINE) ☐ PULSATION SUPP., WEIGHT _____ KGS ☐ PIPING _____ KGS ☐ INTERSTAGE EQUIPMENT _____ KGS			

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**RECIPROCATING COMPRESSOR
AND DRIVERS****PROJECT SPECIFICATION**

#REF! - 509

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#REF!	#REF!	#REF!					

REV

☐ UTILITY CONSUMPTION**ELECTRIC MOTORS**

	NAMEPLATE kW	LOCKED ROTOR AMPS	FULL LOAD STEADY STATE AMPS	MAIN DRIVER NON-STEADY STATE AMPS AT COMPRES- SOR RATED HORSEPOWER (Induction Motors Only)
For Induction Motors See Note of 3.1.2.5 and Motor Data Sheet				
◇ MAIN DRIVER				AMPS
◇ MAIN LUBE OIL PUMP				@ COMPRESSOR RATED
◇ AUX LUBE OIL PUMP				kW OF
◇ MAIN COOLING WATER PUMP				@ CURRENT PULSATIONS
◇ AUX COOLING WATER PUMP				OF %
◇ ROD PACKING COOLING PUMP				
◇ CYLINDER LUBRICATOR				

	WATTS	VOLTS	HERTZ
◇ FRAME OIL HEATER(S)			
◇ COOLING WATER HEATER(S)			
◇ CYL. LUBRICATOR HEATER(S)			

	FLOW	PRESSURE	TEMPERATURE	BACK PRESSURE
◇ MAIN DRIVER	KG/HR @	FALSE	°CTT TO	###
◇ FRAME OIL HEATER(S)	KG/HR @	FALSE	°CTT TO	###
◇ CYL. LUB. HEATER(S)	KG/HR @	FALSE	°CTT TO	###
	KG/HR @	FALSE	°CTT TO	###
	KG/HR @	FALSE	°CTT TO	###

	FLOW M ³ /HR	INLET TEMP °C	OUTLET TEMP °C	INLET PRESS FALSE	OUTLET PRESS FALSE	MAX PRESS FALSE
☐ CYLINDER JACKETS						
◇ INTERCOOLER(S)						
◇ AFTERCOOLER						
◇ FRAME LUBE OIL COOLER						
◇ ROD PRESSURE PACKING*						
◇ TOTAL QUANTITY, M ³ /HR						

*ROD PACKING COOLANT MAY BE OTHER THAN WATER

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**RECIPROCATING COMPRESSOR
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#REF!	#REF!	#REF!					

☐ BASIC LUBE OIL SYSTEM FOR FRAME:		☐ FRAME LUBE OIL SYSTEM	
REF: TYPE MAIN BEARINGS:		☐ SPLASH	☒ PRESSURE (FORCED)
☐ PRESSURE SYSTEM:		☐ TAPRD ROLL'R	☐ PRECISION SL'VE
☐ MAIN OIL PUMP DRIVEN BY:		☒ COMP. CRANKSHAFT	☐ HEATERS REQUIRED:
☐ AUX OIL PUMP DRIVEN BY:		☐ PSV FOR MAIN PUMP EXTERNAL TO CRANKCASE	☐ ELEC. W/THERMOSTAT(S)
☐ HAND OPERATED PRE-LUBE PUMP FOR STARTING		☐ ELEC. MOTOR	☐ STEAM
☐ API-614 LUBE SYSTEM:		☐ OTHER	☐ ELEC. MOTOR
☐ CONTINUOUS FLOW THROUGH OIL (3.7.2.7)		☐ OPERATIONAL TEST & 4 HOUR MECH RUN TEST	☐ OTHER
☐ SEP. CONSOLE FOR PRESS. LUBE SYS:		☒ ONE CONSOLE FOR EA. COMP.	☐ ONE CONSOLE FOR _____ COMPRESSORS
Note: Instrumentation to be listed on Instrumentation Data Sheets.		☒ CONSOLE TO BE OF DECK PLATE TYPE CONSTRUCTION SUITABLE FOR MULTI-POINT SUPPORT AND GROUTING WITH GROUT & VENT HOLES.	
☐ ELECTRICAL CLASSIFICATION: CLASS _____, GROUP _____, DIV _____		☐ NON-HAZARDOUS	
☐ BASIC SYS. REQ'MTS (NORM. OIL FLOWS & VOLUMES)			
☐ LUBE OIL	FLOW M ³ /HR	PRESSURE FALSE	SSU @ 37.8°C
☐ COMPRESSOR FRAME			SSU @ 98.8°C
☐ DRIVER			SUMP VOLUME M ³
☐ GEAR			
☐ SYSTEM PRESSURES:	☐ DESIGN _____ ###	☐ HYDROTEST _____ ###	
☐ PUMPING MATERIALS:	☐ PRESSURE CONTROL VALVE SETTING _____ ###	☐ PUMP REL'F VALVE(S) SET _____ ###	
☐ UPSTREAM OF PUMPS & FILTERS	☐ CARBON STEEL	☐ STAINLESS STEEL WITH SS FLANGES	☐ STAINLESS STEEL WITH CARBON STEEL FLANGES
☐ DOWNSTREAM OF FILTERS	☐	☐	☐
☐	☐	☐	☐
☐	☐	☐	☐
☐ PUMPS (Gear or Screw Type Only)	RAT'D FL'W M ³ /HR	PRESSURE ###	COLD START REQ'D kW
☐ MAIN			DRIVER kW
☐ AUXILIARY			SPEED RPM
☐ PUMP CASING MATERIAL (Ref. 2.12.3.1):			COUPLING REQ'D
☐ GUARD(S) REQ. FOR COUPLING(S):	☐ MAIN PUMP	☐ AUX PUMP	☐ MECH. SEAL REQ'D
☐ AUXILIARY PUMP CONTROL:	☐ MANUAL	☐ AUTOMATIC	☐ ON-OFF-AUTO SEL. SWITCH:
			☐ WIRING TO TERMINAL BOX:
			☐ SWITCHES
			☐ RTD'S/THERMOCOUPLES
☐ COOLERS:	☐ SHELL & TUBE	☐ SINGLE	☐ DUAL W/TRANSFER VALVE
	☐ REMOVABLE BUNDLE	☐ WATER COOLED	☐ MFG'S STD.
	☐ W/BYPASS & TEMP CONTROL VALVE:	☐ MANUAL	☐ AUTO
			☐ SEE SEPARATE HEAT EXCHANGER DATA SHT
☐ FILTER(S)	☐ SINGLE	☐ DUAL W/TRANSFER VALVE	☐ ASME CODE DESIGN
	☐ DESIGN PRESSURE, _____ ###	☐ Δ P CLEAN, _____ ###	☐ ASME CODE STAMPED
	☐ MICRON RATING, _____	☐ CARTRIDGE MATERIAL, _____	☐ Δ P COLLAPSE, _____ ###
	☐ BONNET MATERIAL, _____	☐ CASING MATERIAL, _____	☐ CARTRIDGE P/N _____
☐ SYS. COMPONENT SUPP.	MANUFACTURER	MODEL	MANUFACTURER
☐ MAIN PUMP			☐ OIL COOLER(S)
☐ AUXILIARY PUMP			☐ TRANSFER VALVE(S)
☐ MECHANICAL SEALS			☐ PUMP COUPLING(S)
☐ ELECTRIC MOTORS			☐ SUCTION STRAINER(S)
☐ STEAM TURBINES			☐ CHECK VALVE(S)
☐ OIL FILTER(S)			



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PROJECT SPECIFICATION

#REF! - 509

SHEET 13

REV	DATE	BY	APVD	REV	DATE	BY	APVD
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RECIPROCATING COMPRESSOR AND DRIVERS

#REF!	#REF!	#REF!							

REV	☐ BASIC COOLING SYS. FOR: ☐ COMPRESSOR CYL(S) ☐ INTERCOOLER(S) ☐ AFTERCOOLER ☐ OIL COOLER(S)										1
	☐ HEATERS REQ'D FOR PRE-HEATING: ☐ ELEC., W/ THERMOSTAT(S) ☐ STEAM										2
	☐ PRESSURE FORCED CIRCULATING SYS.: ☐ OPEN, PIPING BY: ☐ PURCH. ☐ MFR ☐ CLOSED, PIPING BY MFR.										3
	MAIN WATER PUMP DRIVEN BY: ☐ ELEC. MOTOR ☐ STEAM TURBINE ☐ OTHER										4
	AUX WATER PUMP DRIVEN BY: ☐ ELEC. MOTOR ☐ STEAM TURBINE ☐ OTHER										5
	☐ SEP. CONSOLE FOR COOLING WATER SYS.: ☐ ONE CONSOLE FOR EA. COMP. ☐ ONE CONSOLE FOR _____ COMPRS										6
	NOTE: Instrumentation to be Listed on Instrumentation Data Sheets ☐ CONSOLE TO BE OF DECK PLATE TYPE CONSTRUCTION SUITABLE FOR MULTI-POINT SUPPORT AND GROUTING WITH GROUT & VENT HOLES.										7
	☐ ELECTRICAL CLASSIFICATION: CLASS _____, GROUP _____, DIV _____ ☐ NON-HAZARDOUS										8
	BASIC SYS. REQ'TS (NORM. COOLING WATER FLOW DATA)										10
		FORCED COOL'G	THERMO SYPHON	STAND PIPE	FLOW M ³ /HR	COOL'G WATER TO BE PRESSURE ###	% ETHYLENE GLYCOL INLET TEMP °C	OUTLET TEMP °C	SITE FLOW INDTR	11	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	12	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	13	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	14	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	15	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	16	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	17	
	CYLINDER(S), _____ STAGE	☐	☐	☐	_____	_____	_____	_____	☐	18	
	PISTON ROD PACK'G TOTAL	☐	☐	☐	_____	_____	_____	_____	☐	19	
	INTERCOOLER(S) TOTAL	☐	☐	☐	_____	_____	_____	_____	☐	20	
	AFTERCOOLER	☐	☐	☐	_____	_____	_____	_____	☐	21	
	OIL COOLER(S)	☐	☐	☐	_____	_____	_____	_____	☐	22	
	TOTAL FLOW	☐	☐	☐	_____	_____	_____	_____	☐	23	
	SYS. PRESSURES: ☐ DESIGN, _____, ### ☐ HYDROTEST, _____, ### ☐ RELIEF VALVE(S), SETTING _____										24
	WATER RESERVOIR: ☐ SIZE, _____ M IN DIA X _____ M IN HT. ☐ CAPACITY _____ M ³ @ Normal Operating Level										25
	RESERVOIR MATERIAL ☐ LEVEL GAUGE ☐ LEVEL SWITCH ☐ DRAIN VALVE ☐ INSPECTION & CLEAN-OUT OPENINGS										26
	☐ PUMPS: (Centrifugal Only) ☐ RATD FL'W M ³ /HR ☐ PRESS. ### ☐ REQD kW ☐ DRIVER kW ☐ SPEED RPM ☐ COUPLING REQD ☐ MECH. SEAL REQD										27
	MAIN _____										28
	AUXILIARY _____										29
	☐ PUMP CASING MATERIAL (Ref 2.12.3.1): MAIN PUMP _____ AUX PUMP _____										30
	☐ GUARD(S) REQ'D FOR COUPL'G(S) ☐ MAIN PUMP ☐ AUX PUMP ☐ GUARD TYPE OR CODE _____										31
	☐ AUX. PUMP CONTROL: ☐ MANUAL ☐ AUTO ON-OFF-AUTO SEL. SWITCH: ☐ BY PURCH. ☐ BY MANUFACTURER										32
	WIRING TO TERMINAL BOX: ☐ BY PURCH. ☐ BY MANUFACTURER										33
	COOLING WATER HEAT EXCH.: ☐ SHELL & TUBE ☐ SINGLE ☐ DUAL W/TRANSFER VALVE ☐ TEMA C ☐ TEMA R (API-660 Data Shets Attached)										34
	☐ AIR COOLED EXCHANGER W/AUTO TEMP CONTROL (API-661 Data Sheets Attached)										35
	☐ W/BYPASS & TEM. CONTROL VALVE ☐ MANUAL ☐ AUTO ☐ LOUVERS FOR AIR EXCH.										36
	☐ SEE SEPARATE COOLER DATA SHEET FOR DETAILS; SPECIFY % GLYCOL ON BOTH SIDES OF SHELL & TUBE										37
	SYS. COMPONENT SUPP.										38
	MANUFACTURER	MODEL		MANUFACTURER	MODEL					39	
	☐ MAIN PUMP	_____	☐ TEMP CONTROL VALVE(S)	_____						40	
	☐ AUXILIARY PUMP	_____	☐ TRANSFER VALVE(S)	_____						41	
	☐ MECHANICAL SEALS	_____	☐ PUMP COUPLING(S)	_____						42	
	☐ ELECTRIC MOTORS	_____		_____						43	
	☐ STEAM TURBINES	_____		_____						44	
	☐	_____		_____						45	
	☐	_____		_____						46	
	☐	_____		_____						47	
	☐	_____		_____						48	
	☐	_____		_____						49	
	☐	_____		_____						50	
	☐	_____		_____						51	
	☐	_____		_____						52	
	☐	_____		_____						53	



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RECIPROCATING COMPRESSOR AND DRIVERS

PROJECT SPECIFICATION

#REF! - 509

SHEET 14

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

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APPLICABLE TO: ☐ PROPOSALS ☐ PURCHASE ☐ AS BUILT		
FOR/USER _____		
SITE/LOCATION _____		AMBIENT TEMPERATURE MIN/MAX _____ 0 / 0 0
COMPRESSOR SERVICE ### _____	NUMBER OF COMPRESSORS _____	
COMPRESSOR MFG. _____	MODEL/TYPE _____	
SUPPRESSOR MFG. _____		
NOTE: ☐ Ind.Data Comp'd Purch. ☐ By Compr/Supp.Mfg.w/Proposal ☐ By Mfg(s) after order ☒ By Mfg(s)/Purchaser as Applicable		
TOTAL NUMBER OF CRANKTHROWS _____ STROKE _____ IN. RPM _____		
☐ ASME CODE STAMP ☐ GOVERNMENTAL CODES OF _____ CODE REGULATIONS APPLY		
☐ OTHER APPLICABLE PRESSURE VESSEL SPEC. OR CODE _____		
☐ LUBE SERVICE ☐ NON-LUBE SERV. ☐ NO OIL ALLOWED INTERNALLY ☐ DRY TYPE INTER.CORR.COATING ☐ YES ☐ NO		
RADIOGRAPHY (X-RAY OF WELDS): ☐ NONE ☐ SPOT ☐ 100% ☐ IMPACT TEST ☐ SPECIAL WELDING REQUIREMENTS		
☐ SHOP INSPECTION ☐ WITNESS HYDROTEST ☐ OUTDOOR STORAGE OVER 6 MONTHS ☐ SPECIAL PAINT SPEC _____		
☐ WITNESSED ☐ OBSERVED		
SERVICE #REF# _____ STAGE NO. _____		
☐ COMPRESSOR MANUFACTURER'S RATED CAPACITY	KG/HR _____	M ³ /HR _____ 0
☒ LINE SIDE OPERATING PRESSURE	INLET, _____ 0	DISCHARGE, _____ 0
☒ OPERATING TEMP. WITHIN SUPPRESSORS	INLET, _____ 0	DISCHARGE, _____ 0
☐ ALLOWABLE PRESSURE DROP THROUGH SUPPRESSORS	ΔP _____ 0 %	ΔP _____ 0 %
☐ SUPPRESSOR TAG NUMBER ☐ COMBINATION INLET SUPP SEPARATOR/INTERNAL ☒ NO. (QTY) OF INLET & DISCH. SUPP. PER STAGE ☐ ALLOWABLE PEAK-PEAK PULSE @ LINE SIDE NOZZLE ☐ ALLOWABLE PEAK-PEAK PULSE @ CYL FLANGE NOZZLE ☐ DESIGN FOR FULL VACUUM CAPABILITY ☐ MIN. REQ'D WORKING PRESSURE & TEMPERATURE NOTE: After design, the actual Mawp & temp are to be determined based on the weakest component and stamped on the vessel. The actual Mawp is to be shown on pg.14 line 12 and on the UIA Forms. ☐ INITIAL SIZING VOL. PER FORMULA OF 3.9.2.2.2 NOTE: This is a Reference ☐ AS BUILT VOLUME (FT ³)	INLET SUPPRESSOR	
	DISCHARGE SUPPRESSOR	
	☐ YES ☐ NO / ☐ YES ☐ NO / ☐ YES ☐ NO	
	0 / % 0 / %	
	0 / % 0 / %	
	☐ YES ☐ NO ☐ YES ☐ NO	
	0 @ 0 0 @ 0	
	M ³ M ³	
	M ³ M ³	

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**RECIPROCATING COMPRESSOR
AND DRIVERS****PROJECT SPECIFICATION**

#REF! - 509

SHEET 15

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV	PULSATION SUPPRESSION DEVICES FOR RECIPROCATING COMPRESSORS (CONT'D)		SERVICE	1
	THESE SHEETS TO BE FILLED OUT FOR EACH SERVICE AND/OR STAGE OF COMPRESSION		STAGE NO.	2
	CONSTRUCTION REQUIREMENTS & DATA		INLET SUPPRESSOR	3
	☐ SUPPRESSOR TAG NUMBER			4
	☐ BASIC MATERIAL REQUIRED, CS, SS, ETC.			5
	☐ ACTUAL MATERIAL DESIGNATION SHELL/HEAD		/	6
	☐ SPECIAL HARDNESS LIMITATIONS, Rc YES NO		SHELL & HEADS WELDS	7
	☒ CORROSION ALLOWANCE, MM REQUIRED		MM	8
	☐ WALL THICKNESS, MM SHELL/HEAD		MM / MM	9
	☐ NOM. SHELL DIA X OVERALL LGTH. (MM/M')		x MM / M'	10
	☐ PIPE OR ROLLED PLATE CONSTRUCTION		PIPE ROLLED PLATE	11
	☐ ACT. MAX ALLOW. WORKING PRESS. AND TEMPERATURE		0 @ 0	12
	☐ MINIMUM DESIGN METAL TEMP (2.14.8)		0	13
	☐ IN. SUPPRESS. TO BE SAME MAWP AS DISCHRG SUPPRESS.		☐ YES ☐ NO	14
	☐ MAX EXPECTED PRESSURE DROP(Δ P, PSI%) LINE PRESS		Δ P 0 / %	15
	☐ WEIGHT (KGS EACH)		KGS	16
	☐ INSUL NUTS & ALLOW. FOR INSULATION REQUIRED (X)			17
	☐ EXPECTED P-P PULSE @ LINE SIDE/CYL FLG, % LINE PRESS BASED ON FINAL SUPPRESSOR DESIGN		%/ %	18
	☒ SUPPORTS, TYPE/QUANTITY			19
				20
				21
	☐ LINE SIDE FLANGE. SIZE/RATING/FACING/TYPE			22
	☐ COMP CYL FLANGE(S), QTY/SIZE/RATING/FACING/TYPE			23
	☐ FLANGE FINISH, ☐ PER 3.9.3.15 ☐ SPECIAL (SPECIFY) >125 <250 ☐ PER ANSI 16.5			24
	☐ INSPECTION OPENINGS REQUIRED		☐ YES ☐ NO ☐ BLINDED	25
	☐ SPEC. QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			26
	☐ * QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			27
	☐ VENT CONNECTIONS REQUIRED		☐ YES ☐ NO	28
	☐ SPEC. QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			29
	☐ * QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			30
	☐ DRAIN CONNECTIONS REQUIRED		☐ YES ☐ NO	31
	☐ SPEC. QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			32
	☐ * QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			33
	☐ PRESSURE CONNECTIONS REQUIRED		☐ YES ☐ NO	34
	☐ SPEC. QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			35
	☐ * QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			36
	☐ TEMPERATURE CONNECTIONS REQUIRED		☐ YES ☐ NO	37
	☐ SPEC. QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			38
	☐ CYL NOZZLE ☐ MAIN BODY			39
	☐ * QTY. SIZE. 2722 KG NPT CPLG/FLG TYPE & RATING			40
				41
				42
				43
				44
				45
	OTHER DATA AND NOTES			46
	☐ COMPRESSOR MFG'S SUPP. OUTLINE OR DRAWING NO.			47
	☐ SUPP. MFG'S OUTLINE OR DRAWING NO.			48
	NOTES *- AS BUILT			49
	NOTE: MAIN PULSATION SUPPRESSION DEVICE CONNECTIONS FOR THE COMPRESSOR AND PROCESS LINE MUST BE FITTED WITH WELD NECK FLANGES. ANY DEVIATIONS MUST BE APPROVED BY UOP IN WRITING.			50
				51

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SHEET ##

RECIPROCATING COMPRESSOR
AND DRIVERS

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV

INSTRUMENTATION

PURCHASER TO FILL IN (☐ ☐ ☐) AFTER COMMODITY TO INDICATE: ☐ BY COMP. MFR. ☐ BY PURCH. ☐ BY OTHERS

INSTRUMENT & CONTROL

☒ ONE FOR EA. UNIT ☐ ONE COMMON TO ALL UNITS

PANEL ():

☐ MACHINE MTD☐ FREE STANDING (OFF UNIT)☐ LOCAL☐ REMOTE☐ OUTDOORS☐ PNEUMATIC☐ ELEC.☐ ELECTRONIC☐ HYDRAULIC☐ PROGRAMMABLE CONTR'L☐ NEMA 7, CLASS _____, GROUP _____, DIVISION _____☐ INTRINSICALLY SAFE☐ I/S BARRIERS (☐ ☐ ☐)☐ NEMA 4, WATERTIGHT & DUSTTIGHT ☐ PURGED TO NFPA 496 TYPE☐ X☐ Y☐ Z☐ OTHER NEMA _____☐ LOW PURGE PRESS.☐ ALARM☐ SHUTDOWN☐ VIB. ISOLATORS☐ STRIP HEATERS☐ PURGE CONN.☐ EXTRA CUTOUTS☐ ANNUNCIATOR W/FIRST-OUT INDICATION LOCATED ON CONTROL PANEL☐ PURCHASER'S CONN. BROUGHT OUT TO TERMINAL BOX BY VENDOR☐ INSTRUMENTATION SUITABLE FOR: ☐ INDOORS ☐ OUTDOORS ☐ OTHER☐ PREFERRED INSTRUMENT SUPPLIERS, (TO BE COMPLETED BY PURCHASER), OTHERWISE MFR'S STANDARD APPLIES

PRESSURE GAUGES	MFR	SIZE & TYPE	MTL
TEMPERATURE GAUGES	MFR	SIZE & TYPE	MTL
LIQUID LEVEL GAUGES	MFR	TYPE	MTL
DIFF. PRESSURE GAUGES	MFR	SIZE & TYPE	MTL
PRESS. TRANSMITTERS	MFR	TYPE	MTL
LIQUID LEV. TRANSMITTER	MFR	TYPE	MTL
PRESSURE SWITCHES	MFR	TYPE	MTL
TEMPERATURE SWITCHES	MFR	TYPE	MTL
LIQUID LEVEL SWITCHES	MFR	TYPE	MTL
DIFF. PRESSURE SWITCHES	MFR	TYPE	MTL
CONTROL VALVES	MFR	TYPE	MTL
PRESSURE SAFETY VALVES	MFR	TYPE	MTL
SIGHT FLOW INDICATORS	MFR	TYPE	MTL
VIBRATION MONITORS & EQUIP.	MFR	TYPE	MTL
THERMOCOUPLES	MFR	TYPE	MTL
RTD'S	MFR	TYPE	MTL
SOLENOID VALVES	MFR	TYPE	MTL
ANNUNCIATOR	MFR	MODEL & (QTY SPARE POINTS)	()
PROGRAMMABLE CONTROLLER	MFR	TYPE	MTL
	MFR	TYPE	MTL
	MFR	TYPE	MTL

☐ PRESSURE GAUGE REQUIREMENTS☐ LIQUID FILLED PRESSURE GAUGES:☐ YES☐ NO

FUNCTION	LOCALLY MOUNTED	PANEL MOUNTED	LOCALLY MOUNTED	PANEL MOUNTED
LUBE OIL MAIN PUMP DISCH.	(☐ ☐ ☐)	(☐ ☐ ☐)	PROCESS GAS: INLET PRESS.	(☐ ☐ ☐)
LUBE OIL AUX. PUMP DISCH.	(☐ ☐ ☐)	(☐ ☐ ☐)	@ EA. STAGE	(☐ ☐ ☐)
LUBE OIL PRESS. AT FRAME HEADER (	(☐ ☐ ☐)	(☐ ☐ ☐)	DISCH. PRESS.	(☐ ☐ ☐)
LUBE OIL FILTER Δ P	(☐ ☐ ☐)	(☐ ☐ ☐)	@ EA. STAGE	(☐ ☐ ☐)
COOLING H ₂ O INLET HEADER	(☐ ☐ ☐)	(☐ ☐ ☐)		(☐ ☐ ☐)
	(☐ ☐ ☐)	(☐ ☐ ☐)		(☐ ☐ ☐)
	(☐ ☐ ☐)	(☐ ☐ ☐)		(☐ ☐ ☐)

REMARKS:

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RECIPROCATING COMPRESSOR AND DRIVERS

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SHEET ##

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

INSTRUMENTATION (CONT'D)

				LOCALLY MOUNTED	PANEL MOUNTED	GAUGE W/ CAPIL'RY	THERMO CPL SYS	RTD SYS	I/S SYS
LUBE OIL	INLET TO	OUT OF	FRAME	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
LUBE OIL	INLET TO	OUT OF	COOLER	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
MAIN JRNL BEARINGS	(THERMOCOUPLES OR RTD'S ONLY)			(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
MOTOR BEARING(S)	(THERMOCOUPLES OR RTD'S ONLY)			(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
COOLING WATER HEADER:	INLET	OUTLET		(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
CYL. COOLING WATER:	INLET	OUTLET	EA. CYL	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
PROCESS GAS:	INLET	DISCH.	EACH CYL	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
INTERCOOLER(S)	INLET	GAS	WATER	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
	INLET	GAS	WATER	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
AFTERCOOLER:	INLET	GAS	WATER	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
	INLET	GAS	WATER	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
COOLING WATER	INLET	OUTLET/COOLED PKG CASE(S)		(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
PRESS. PKG CASE, CYL PIST ROD	(THERMOCPLS OR RTD'S ONLY)			(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
COMPRESSOR VALVES	SUCT.	DISCH.	TC'S OR RTD'S ONLY	(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
				(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐
				(☐ ☐ ☐)	(☐ ☐ ☐)	☐	☐	☐	☐

ALARM & SHUTDOWN SWITCH REQ'TS

NOTE: ALARM & SHUTDOWN SWITCHES SHALL BE INDIVIDUALLY SEPARATE

ANNUNCIATION POINTS

FUNCTION	ALARM						SHUTDOWN				TOTAL NO. OF POINTS		
	IN			IN CTL			IN		IN CTL				
	PNL			ROOM			PNL		ROOM				
	BY	PANEL	MFR	OTHERS	BY	PANEL	MFR	OTHERS	BY	PANEL		MFR	OTHERS
LOW LUBE OIL PRESS. @ BEARING HEADER	(	☒	☐	☐	(	☒	☐	☐	(	☒	☐	☐	
HIGH LUBE OIL Δ P ACROSS FILTER	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
LOW LUBE OIL LEVEL, FRAME	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
AUX LUBE OIL PUMP, FAIL TO START	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
CYL LUBE SYSTEM PROTECTION	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
COMPR. VIBRATION, SHUTDOWN ONLY	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
VIBRATION, W/ CONTINUOUS MONITORING	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
ROD DROP DETECTOR, CONTACT TYPE(1/CYL)	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
ROD DROP PROXIMITY PROBE (1/CYL)	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
OIL TEMP OUT OF FRAME	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HIGH GAS DISCH. TEMP EACH CYLINDER	(	☐	☒	☐	(	☐	☐	☐	(	☐	☐	☐	
HIGH JACKET WATER TEMP., EA. CYL	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
LOW SUCTION PRESS., FIRST STG INLET	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HI DISCH. PRESS. FINAL EA STG	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HI CYL. GAS Δ P, EACH STAGE	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HI LIQ. LEV., EA. MOISTURE SEPARATOR	(	☐	☒	☐	(	☐	☒	☐	(	☐	☐	☐	
LOW PURGE GAS PRESS, DISTANCE PIECE(S)	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HI X-HD PIN TEMP	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
PRESS PKG CASE (PISTON ROD TEMP)	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HIGH BEARING TEMPERATURE	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
HIGH PACKING TEMPERATURE	(	☐	☐	☐	(	☐	☐	☐	(	☐	☐	☐	
TOTAL NUMBER OF ANNUNCIATION POINTS													

SWITCH CONTACT OPERATION

NOTE: EACH SWITCH SHALL BE MINIMUM SPDT ARRANGEMENT

ALARM CONTACTS SHALL:

☒ OPEN (DE-ENER.) TO SOUND ALARM & BE ENERGIZED WHEN COMPR. IS IN OPERATION

☐ CLOSE (ENERGIZE) TO SOUND ALARM & BE DE-ENERGIZED WHEN COMPR. IS IN OPERATION

SHUTDOWN CONTACTS SHALL:

☒ OPEN (DE-ENERGIZED) TO SHUTDOWN & BE ENERGIZE WHEN COMPR. IS IN OPERATION

☐ CLOSE (ENERGIZE) TO SHUTDOWN & BE DE-ENERGIZE WHEN COMPR. IS IN OPERATION

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Note: The information in this document is confidential and the property of UOP LLC and must not be disclosed to others or reproduced in any manner or used for any purpose whatsoever without its written permission.



#

UOP LLC • 25 East Algonquin Road • Des Plaines, Illinois 60017-5017 USA

RECIPROCATING COMPRESSOR AND DRIVERS

PROJECT SPECIFICATION

#REF! - 509

SHEET ##

REV	DATE	BY	APVD	REV	DATE	BY	APVD
#REF!	#REF!	#REF!					

REV

☒ INSTRUMENTATION (CONT'D)

MISCELLANEOUS INSTRUMENTATION

SIGHT FLOW IND. (COOLING H₂O ONLY)

(☐ ☐ ☐)

FOR:

INTERCLR(S)

AFTERCLR

OIL CLR

H₂O CLR

CYL JACKET WATER

ROD PRESS. PKG CASES

PNEUMATIC PRESSURE TRANSMITTERS

(☐ ☐ ☐)

FOR:

PRESSURE TRANSMITTERS (ELEC. OUTP.)

(☐ ☐ ☐)

FOR:

PNEUMATIC LEVEL TRANSMITTERS

(☐ ☐ ☐)

ALARM HORN & ACKNLMT TEST BUTTON

(☐ ☐ ☐)

CONDUIT & WIRING W/JUNCT. BOXES

(☐ ☐ ☐)

TEST VALVES

(☐ ☐ ☐)

FOR:

DRAIN VALVES

(☐ ☐ ☐)

FOR:

GAUGE GLASS(ES)

(☐ ☐ ☐)

FOR:

TACHOMETER

(☐ ☐ ☐)

SPEED RANGE

TO

RPM

CRANKSHAFT KEY PHASER

(☐ ☐ ☐)

FOR:

AND TRANSDUCER

(☐ ☐ ☐)

SEPARATE LUBE OIL CONSOLE INSTRUMENTATION:

PURCH. TO LIST REQ'MTS IN ADDITION TO ANY ABOVE REQ'MTS

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

SEPARATE COOLING WATER CONSOLE INSTRUMENT:

PURCH. TO LIST REQ'MTS IN ADDITION TO ANY ABOVE REQ'MTS

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

☒ RELIEF VALVES

LOCATION

BY

MANUFACTURER

TYPE

◇ SIZE

◇ SETTING

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

(☐ ☐ ☐)

NOTES: SEE MOTOR DATA SHEET FOR ADDITIONAL MOTOR INSTRUMENTATION REQUIREMENTS

FOR TURBINE DRIVERS USE APPLICABLE API DATA SHEETS

ELECTRICAL & INSTRUMENTATION CONNECTIONS SHALL BE MADE DIRECTLY BY THE PURCHASER TO INDIVIDUAL INSTRUMENTS ON THE COMPRESSOR

ADDITIONAL INSTRUMENTATION REMARKS/SPECIAL REQUIREMENTS:

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PLATFORMING NET GAS – PROBLEM C

	1st Stage	2nd Stage
MMSCFD	74.76	77.27
P ₁ PSIA	49.35	107.81
P ₂ PSIA	115.06	243.50
T ₁	106	95
MOL WT	10.3	9.9
C _p /C _v	1.257	1.267
Z ₁	1.000	1.000
Z ₂	1.002	1.004

Centrifugal compressor (condensing steam turbine driven) vs. 3-50% reciprocating compressors (motor driven). Which is better?

Cost of electricity	\$0.05/kw-hr
Cost of steam	\$5.00/1000 lb
Steam rate	7.6 lb/hp-hr

Cost of 2 body compressor, condensing steam turbine including oil console and gas seal console:
\$4,250,000

Spare compressor and turbine rotors: \$1,065,000

Cost of 2 stage (6 cylinder) reciprocating compressor: \$3,327,000

Assumptions: Installation: Centrifugal – 20% of compressor cost
Reciprocating – 50% of compressor cost

Maintenance: Centrifugal – 0/hp/yr
Reciprocating – \$35/hp/yr

Figure 25

Specific Gravity Correction

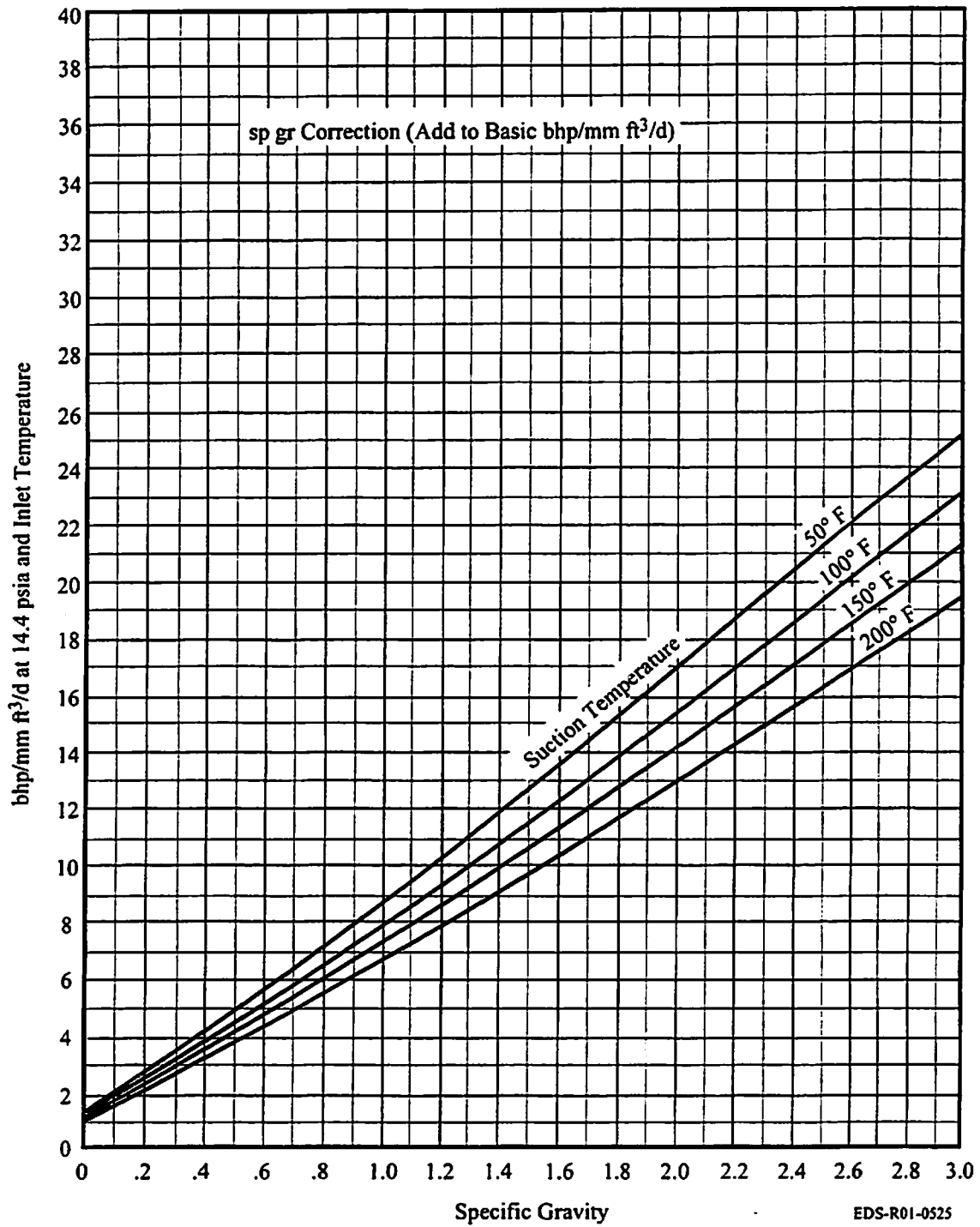
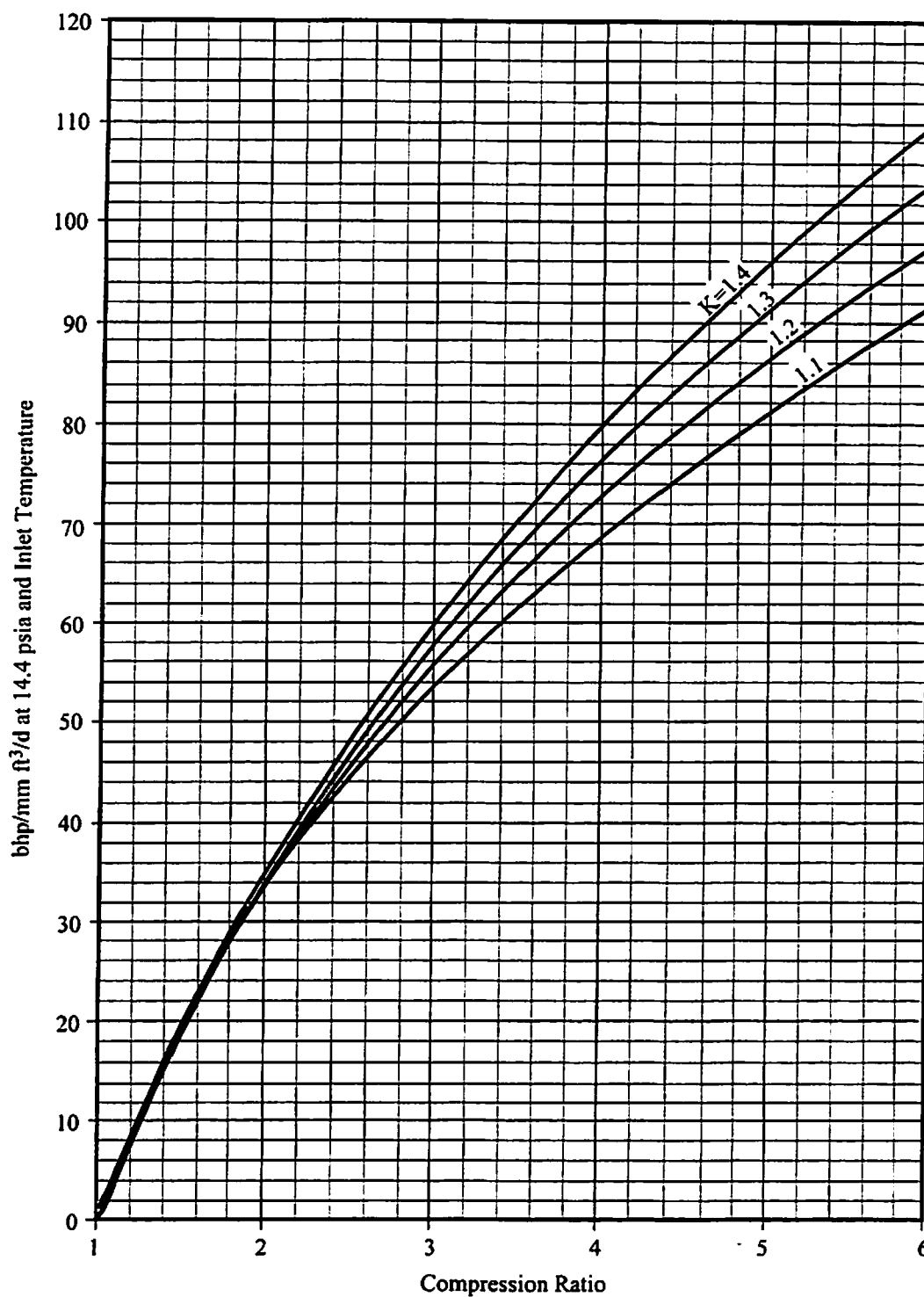


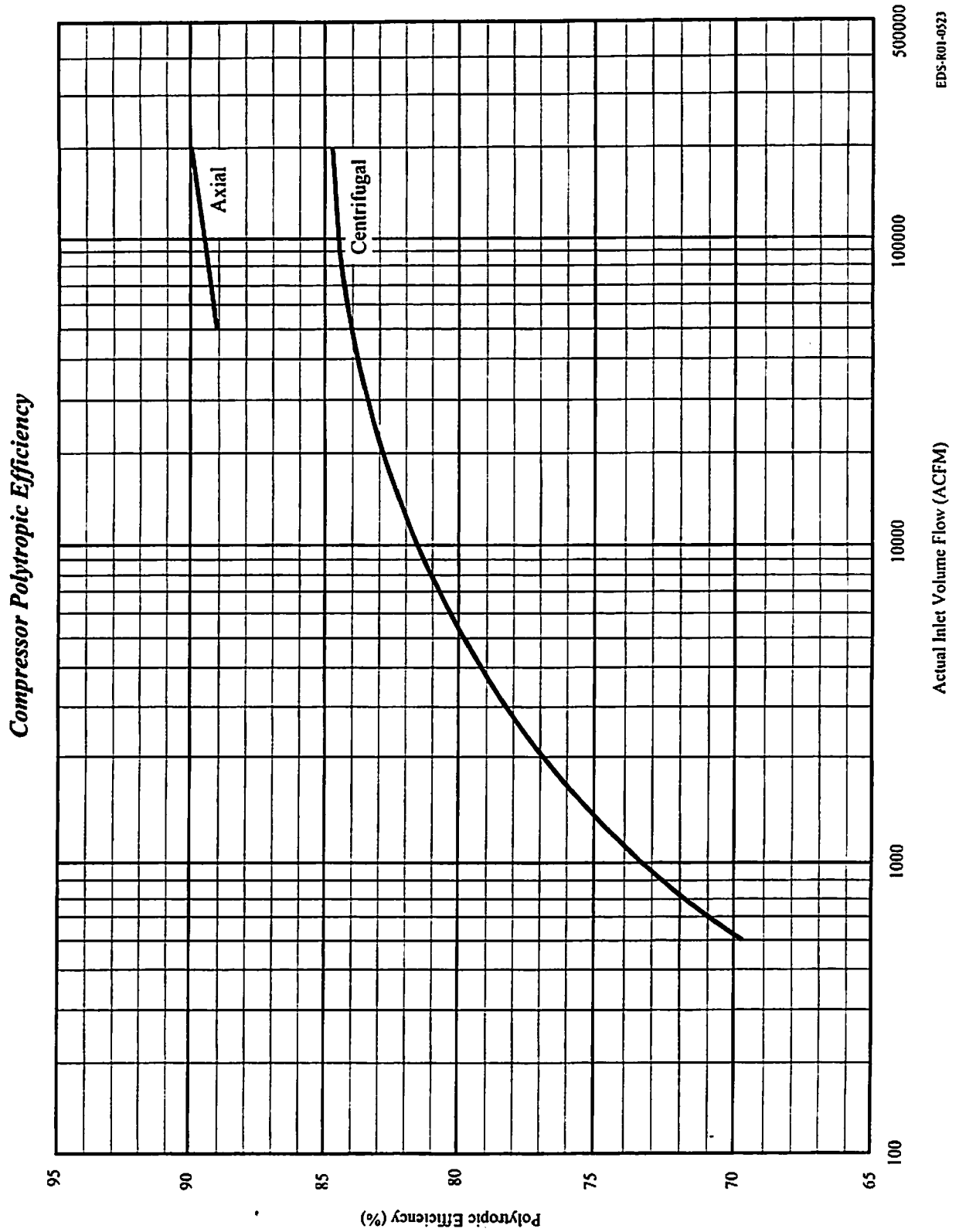
Figure 24

Basic bhp/mm ft³/d



EDS-R01-0524

Figure 23



Subject: CENTRIFUGAL COMPRESSORS-BHP	Date
For: Sample Problem B	By

SYMBOL

Q_b	Million SCFD	34.54		
T_s	$^{\circ}R$	555		
P_s	psia	14.4		
P_d	psia	51.2		
r	P_d/P_s			
mol wt	Molecular Weight	28.49		
R	$\frac{1545}{\text{mol wt}}$			
k	C_p/C_v	1.4		
Q_s	$\text{ft}^3/\text{min @ Suction}$			
lb/min wt flow	$\frac{(Q_b)(\text{mol wt})}{546000}$			
Z_s		1.0		
Z_d		1.0		
Z_{avg}		1.0		
E_h	Polytropic Efficiency			
	$\frac{k-1}{k}$			
M	$\frac{k-1}{k E_h}$			
	$(r)^M - 1$			
Beta	$\frac{(r)^M - 1}{M}$			
Head	$(Z_{avg})(R)(T_s)(\text{Beta})$			
Gas hp	$\frac{(\text{Wt Flow})(\text{Head})}{(33000)(E_h)}$			
Losses Frictional Seal	1 pct of Gas hp			
Gear				
Total bhp	(If Used) 3 pct of hp			
T_d	$(T_s)(r)^M$			

Subject: RECIPROCATING COMPRESSORS-BHP	Date
For: Sample Problem A	By

SYMBOL

Q_b	Million SCFD	30.55			
T_s	$^{\circ}R$	580			
P_s	<i>psia</i>	243			
P_d	<i>psia</i>	387			
r	P_d/P_s				
k	C_p/C_v	1.311			
<i>mol wt</i>	Molecular Weight	7.6			
<i>sp gr</i>	Specific Gravity				
Z_s		1.001			
Z_d		1.003			
Z_{avg}		1.002			
Q_s	$\frac{\text{ft}^3/\text{min @ Suction}}{(1440)(P_s)(520)} \times (14.7)(T_s)(Z_s)$				
	$\frac{k-1}{k}$				
T_d	$r^{k-1/k} T_s$				
(A)	BHP/imm @ 14.4 psia and Suction Temperature				
(B)	<i>sp gr</i> Correction				
(C)	$\frac{(Q_b)(1.02)(T_s)}{520}$				
(D)	Add 5 pct if N.L.				
(E)	Z_{avg}				
(F)	Gear Loss 3 pct (if Gear is Used)				
BHP Req' d	$(A+B)(C)(D)(E)(F)$				

SOLUTION – PROBLEM A

Subject: RECIPROCATING COMPRESSORS-BHP			Date		
For: Recycle Gas			By		
SYMBOL					
Q_b	Million SCFD	30.55			
T_s	$^{\circ}R$	580			
P_s	psia	243			
P_d	psia	387			
r	P_d/P_s	1.59			
k	C_p/C_v	1.311			
mol wt	Molecular Weight	7.6			
sp gr	Specific Gravity	0.26			
Z_s		1.001			
Z_d		1.003			
Z_{avg}		1.002			
Q_s	$\frac{\text{ft}^3/\text{min @ Suction}}{(1440)(P_s)(520)} \times (14.7)(T_s)(Z_s)$	1433			
	$\frac{k-1}{k}$	0.237			
T_d	$(T_s) \left(r^{\frac{k-1}{k}} \right)$	187°F			
(A)	BHP/mm @ 14.4 psia and Suction Temperature	23			
(B)	sp gr Correction	3			
(C)	$\frac{(Q_b)(1.02)(T_s)}{520}$	34.76			
(D)	Add 5 pct if N.L.	—			
(E)	Z_{avg}	1.002			
(F)	Gear Loss 3 pct (if Gear is Used)	—			
BHP Req'd	$(A+B)(C)(D)(E)(F)$	906			

COMPRESSOR PROBLEM SOLUTIONS – PROBLEM B

Centrifugal – Main Air Blower

$$Aft^3/min = \frac{std\ ft^3/d}{minutes/day} \times \frac{14.7\ psia}{P_s\ psia} \times \frac{T_s^\circ R}{520^\circ R} \times Z_s$$

$$Aft^3/min = \frac{34.54 \times 10^6}{1440} \times \frac{14.7}{14.4} \times \frac{555}{520} \times 1.0 = 26134\ ft^3/min\ @\ Inlet$$

$$lb/min\ wt\ flow = \frac{34.54 \times 10^6}{1440} \times \frac{28.49\ lb/lb\ mol}{379.48\ std\ ft^3/lb\ mol} = 1800.8\ lb/min$$

$$\frac{k-1}{k} = 0.2857$$

$$M = \frac{0.2858}{0.832} = 0.343$$

$$(comp.\ ratio)^M - 1 = 3.556^{0.343} - 1 = 0.546$$

$$\frac{0.546}{M} = Beta = 1.590$$

$$Polytropic\ Head = Z_{avg}RT_s B = (1.0)(1545/28.49)(555)(1.590) = 47846\ ft\ lb\ f/lb\ m$$

$$GHP = \frac{Head \times lb/min}{33000 \times E_h} = \frac{(48376)(1800.8)}{(33000)(0.832)} = 3138$$

$$BHP = 3138(1.01) = 3170\ HP$$

$$T_{disch} = T_s \times r^m = 555(1.546) = 858^\circ R = 398^\circ F$$

SOLUTION – PROBLEM B (cont.)

Subject: CENTRIFUGAL COMPRESSORS-BHP			Date	
For: Main Air Blower			By	
SYMBOL				
Q_b	Million SCFD	34.54		
T_s	$^{\circ}R$	555		
P_s	psia	14.4		
P_d	psia	51.2		
r	P_d/P_s	3.556		
mol wt	Molecular Weight	28.49		
R	$\frac{1545}{\text{mol wt}}$	54.23		
k	C_p/C_v	1.4		
Q_s	$\text{ft}^3/\text{min} @ \text{ Suction}$	26134		
lb/min wt flow	$\frac{(Q_b)(\text{mol wt})}{546000}$	1800.8		
Z_s		1.0		
Z_d		1.0		
Z_{avg}		1.0		
E_h	Polytropic Efficiency	0.832		
	$\frac{k-1}{k}$	0.2857		
M	$\frac{k-1}{k E_h}$	0.343		
	$(r)^M - 1$	0.546		
Beta	$\frac{(r)^M - 1}{M}$	1.590		
Head	$(Z_{avg})(R)(T_s)(\text{Beta})$	47846		
Gas hp	$\frac{(\text{Wt Flow})(\text{Head})}{(33000)(E_h)}$	3138		
Losses Frictional Seal	1 pct of Gas hp	32		
Gear	(If Used) 3 pct of hp	—		
Total bhp		3170		
T_d	$(T_s)(r)^M$	398		

SOLUTION – PROBLEM C

	Centrifugal	Reciprocating
Capital Cost	\$5.32	\$9.98
Installation	\$0.85	\$4.99
Utilities/Year	\$2.72	\$2.44
Maintenance	0	\$0.26
Total/1st Year	\$8.89	\$17.67

*17000
by compressor
found*

Savings with centrifugal compressor – \$8.78 million for first year

*not for cons
big*

CENTRIFUGAL COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD	74.76	77.27			
T_s	$^{\circ}R$	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
MW	Mol. Wt.	10.3	9.9			
R	1545/MW	150.0	156.1			
k	C_p/C_v	1.257	1.267			
Q_s	CFM @ Suct.	16832.556	7824.635			
#/Min Wt. Flow	$\frac{Q_b \times MW}{546000}$	1409.14	1399.89			
Z_s		1.000	1.002			
Z_d		1.002	1.004			
Z_{av}		1.001	1.003			
E_h	Poly Eff.	0.83	0.81			
	$k - 1/k$	0.204	0.211			
M	$k - 1/k E_h$	0.246	0.260			
	$(r)^M - 1$	0.232	0.236			
Beta	$\frac{(r)^M - 1}{M}$	0.941	0.908			
Head	$Z_{av} R T_s \text{ Beta}$	79991.486	78840.783			
GHP	$\frac{\text{wt Flow} \times \text{Head}}{33000 \times E_h}$	4115.351	4129.017			
Losses	1% of GHP	1.01	1.01			
Frict.	3% of GHP	1	1			
Seal						
Gear						
Total BHP		4156.504	4170.307			
T_d	$T_s(r)^M$	237.232	226.041			

RECIPROCATING COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD	37.38	38.64			
T_s	$^{\circ}R$	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
k	C_p/C_v	1.257	1.267			
MW	Mol. Wt.	10.3	9.9			
S.G.	Spec. Grav.	0.4	0.3			
Z_s		1.000	1.000			
Z_d		1.002	1.002			
Z_{ave}		1.001	1.001			
Q_s	CFM Suct	8416.278	3905.014			
	$k - 1/k$	0.204	0.211			
T_d	$\frac{k-1/k}{r} T_s$	212.948	198.960			
(A)	BHP/mm @ 14.4 psia & Suct T	41	41			
(B)	Sp. Gr. Corr	3.5	3.5			
(C)	$Q_b (1.02) T_s/520$	41.50	42.07			
(D)	Add 5% if N.L.	1	1			
(E)	Z_{av}	1.001	1.001			
(F)	Gear loss 3%	1	1			
BHP Req'd	(A+B) (C) (D) (E) (F) (G)	1848.616	1873.790			

11/1/00

RECIPROCATING COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD	37.38	38.64			
T_s	$^{\circ}R$	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
k	C_p/C_v	1.257	1.267			
MW	Mol. Wt.	10.3	9.9			
S.G.	Spec. Grav.	0.4	0.3			
Z_s		1.000	1.000			
Z_d		1.002	1.002			
Z_{ave}		1.001	1.001			
Q_s	CFM Suct	8416.278	3905.014			
	$k - 1/k$	0.204	0.211			
T_d	$\frac{k-1/k}{r} T_s$	212.948	198.960			
(A)	BHP/mm @ 14.4 psia & Suct T	41	41			
(B)	Sp. Gr. Corr	3.5	3.5			
(C)	$Q_b (1.02) T_s/520$	41.50	42.07			
(D)	Add 5% if N.L.	1	1			
(E)	Z_{av}	1.001	1.001			
(F)	Gear loss 3%	1	1			
BHP Req'd	(A+B) (C) (D) (E) (F) (G)	1848.616	1873.790			

CENTRIFUGAL COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD	74.76	77.27			
T_s	°R	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
MW	Mol. Wt.	10.3	9.9			
R	1545/MW	150.0	156.1			
k	C_p/C_v	1.257	1.267			
Q_s	CFM @ Suct.	16832.556	7824.635			
#/Min Wt. Flow	$\frac{Q_b \times MW}{546000}$	1409.14	1399.89			
Z_s		1.000	1.002			
Z_d		1.002	1.004			
Z_{av}		1.001	1.003			
E_h	Poly Eff.	0.83	0.81			
	$k - 1/k$	0.204	0.211			
M	$k - 1/k E_h$	0.246	0.260			
	$(r)^M - 1$	0.232	0.236			
Beta	$\frac{(r)^M - 1}{M}$	0.941	0.908			
Head	$Z_{av} R T_s \text{ Beta}$	79991.486	78840.783			
GHP	$\frac{\text{wt Flow} \times \text{Head}}{33000 \times E_h}$	4115.351	4129.017			
Losses	1% of GHP	1.01	1.01			
Frict.						
Seal	3% of GHP	1	1			
Gear						
Total BHP		4156.504	4170.307			
T_d	$T_s(r)^M$	237.232	226.041			

SOLUTION – PROBLEM C

	Centrifugal	Reciprocating
Capital Cost	\$5.32	\$9.98
Installation	\$0.85	\$4.99
Utilities/Year	\$2.72	\$2.44
Maintenance	0	\$0.26
Total/1st Year	\$8.89	\$17.67

Savings with centrifugal compressor – \$8.78 million for first year

SOLUTION – PROBLEM B (cont.)

Subject: CENTRIFUGAL COMPRESSORS-BHP			Date	
For: Main Air Blower			By	
SYMBOL				
Q_b	Million SCFD	34.54		
T_s	$^{\circ}R$	555		
P_s	psia	14.4		
P_d	psia	51.2		
r	P_d/P_s	3.556		
mol wt	Molecular Weight	28.49		
R	$\frac{1545}{\text{mol wt}}$	54.23		
k	C_p/C_v	1.4		
Q_s	$\text{ft}^3/\text{min @ Suction}$	26134		
lb/min wt flow	$\frac{(Q_b)(\text{mol wt})}{546000}$	1800.8		
Z_s		1.0		
Z_d		1.0		
Z_{avg}		1.0		
E_h	Polytropic Efficiency	0.832		
	$\frac{k-1}{k}$	0.2857		
M	$\frac{k-1}{k E_h}$	0.343		
	$(r)^M - 1$	0.546		
Beta	$\frac{(r)^M - 1}{M}$	1.590		
Head	$(Z_{avg})(R)(T_s)(\text{Beta})$	47846		
Gas hp	$\frac{(\text{Wt Flow})(\text{Head})}{(33000)(E_h)}$	3138		
Losses Frictional Seal	1 pct of Gas hp	32		
Gear	(If Used) 3 pct of hp	—		
Total bhp		3170		
T_d	$(T_s)(r)^M$	398		

COMPRESSOR PROBLEM SOLUTIONS – PROBLEM B

Centrifugal – Main Air Blower

$$A \text{ ft}^3/\text{min} = \frac{\text{std ft}^3/\text{d}}{\text{minutes/day}} \times \frac{14.7 \text{ psia}}{P_s \text{ psia}} \times \frac{T_s^\circ R}{520^\circ R} \times Z_s$$

$$A \text{ ft}^3/\text{min} = \frac{34.54 \times 10^6}{1440} \times \frac{14.7}{14.4} \times \frac{555}{520} \times 1.0 = 26134 \text{ ft}^3/\text{min} @ \text{ Inlet}$$

$$\text{lb/min wt flow} = \frac{34.54 \times 10^6}{1440} \times \frac{28.49 \text{ lb/lb mol}}{379.48 \text{ std ft}^3/\text{lb mol}} = 1800.8 \text{ lb/min}$$

$$\frac{k-1}{k} = 0.2857$$

$$M = \frac{0.2858}{0.832} = 0.343$$

$$(\text{comp. ratio})^M - 1 = 3.556^{0.343} - 1 = 0.546$$

$$\frac{0.546}{M} = \text{Beta} = 1.590$$

$$\text{Polytropic Head} = Z_{\text{avg}} R T_s B = (1.0)(1545/28.49)(555)(1.590) = 47846 \text{ ft lb f/lb m}$$

$$\text{GHP} = \frac{\text{Head} \times \text{lb/min}}{33000 \times E_h} = \frac{(48376)(1800.8)}{(33000)(0.832)} = 3138$$

$$\text{BHP} = 3138(1.01) = 3170 \text{ HP}$$

$$T_{\text{disch}} = T_s \times r^m = 555(1.546) = 858^\circ R = 398^\circ F$$

SOLUTION – PROBLEM A

Subject: RECIPROCATING COMPRESSORS-BHP			Date		
For: Recycle Gas			By		
SYMBOL					
Q_b	Million SCFD	30.55			
T_s	$^{\circ}R$	580			
P_s	psia	243			
P_d	psia	387			
r	P_d/P_s	1.59			
k	C_p/C_v	1.311			
mol wt	Molecular Weight	7.6			
sp gr	Specific Gravity	0.26			
Z_s		1.001			
Z_d		1.003			
Z_{avg}		1.002			
Q_s	$\frac{ft^3/min @ Suction}{(std ft^3/d)(14.7)(T_s)(Z_s)}$ $\frac{(1440)(P_s)(520)}{(1440)(P_s)(520)}$	1433			
	$\frac{k-1}{k}$	0.237			
T_d	$(T_s)\left(r^{\frac{k-1}{k}}\right)$	187°F			
(A)	BHP/mm @ 14.4 psia and Suction Temperature	23			
(B)	sp gr Correction	3			
(C)	$\frac{(Q_b)(1.02)(T_s)}{520}$	34.76			
(D)	Add 5 pct if N.L.	---			
(E)	Z_{avg}	1.002			
(F)	Gear Loss 3 pct (if Gear is Used)	---			
BHP Req' d	$(A+B)(C)(D)(E)(F)$	906			

SOLUTION – PROBLEM A

Subject: RECIPROCATING COMPRESSORS-BHP			Date		
For: Recycle Gas			By		
SYMBOL					
Q_b	Million SCFD	30.55			
T_s	$^{\circ}R$	580			
P_s	psia	243			
P_d	psia	387			
r	P_d/P_s	1.59			
k	C_p/C_v	1.311			
mol wt	Molecular Weight	7.6			
sp gr	Specific Gravity	0.26			
Z_s		1.001			
Z_d		1.003			
Z_{avg}		1.002			
Q_s	$\frac{\text{ft}^3/\text{min @ Suction}}{(1440)(P_s)(520)}$ $\frac{(\text{std ft}^3/\text{d})(14.7)(T_s)(Z_s)}{(1440)(P_s)(520)}$	1433			
	$\frac{k-1}{k}$	0.237			
T_d	$(T_s) \left(r^{\frac{k-1}{k}} \right)$	187°F			
(A)	BHP/mm @ 14.4 psia and Suction Temperature	23			
(B)	sp gr Correction	3			
(C)	$\frac{(Q_b)(1.02)(T_s)}{520}$	34.76			
(D)	Add 5 pct if N.L.	—			
(E)	Z_{avg}	1.002			
(F)	Gear Loss 3 pct (if Gear is Used)	—			
BHP Req' d	$(A+B)(C)(D)(E)(F)$	906			

COMPRESSOR PROBLEM SOLUTIONS – PROBLEM B

Centrifugal – Main Air Blower

$$A \text{ ft}^3/\text{min} = \frac{\text{std ft}^3/\text{d}}{\text{minutes/day}} \times \frac{14.7 \text{ psia}}{P_s \text{ psia}} \times \frac{T_s^\circ R}{520^\circ R} \times Z_s$$

$$A \text{ ft}^3/\text{min} = \frac{34.54 \times 10^6}{1440} \times \frac{14.7}{14.4} \times \frac{555}{520} \times 1.0 = 26134 \text{ ft}^3/\text{min} @ \text{ Inlet}$$

$$\text{lb/min wt flow} = \frac{34.54 \times 10^6}{1440} \times \frac{28.49 \text{ lb/lb mol}}{379.48 \text{ std ft}^3/\text{lb mol}} = 1800.8 \text{ lb/min}$$

$$\frac{k-1}{k} = 0.2857$$

$$M = \frac{0.2858}{0.832} = 0.343$$

$$(\text{comp. ratio})^M - 1 = 3.556^{0.343} - 1 = 0.546$$

$$\frac{0.546}{M} = \text{Beta} = 1.590$$

$$\text{Polytropic Head} = Z_{\text{avg}} R T_s B = (1.0)(1545/28.49)(555)(1.590) = 47846 \text{ ft lb f/lb m}$$

$$\text{GHP} = \frac{\text{Head} \times \text{lb/min}}{33000 \times E_h} = \frac{(47846)(1800.8)}{(33000)(0.832)} = 3138$$

$$\text{BHP} = 3138(1.01) = 3170 \text{ HP}$$

$$T_{\text{disch}} = T_s \times r^m = 555(1.546) = 858^\circ R = 398^\circ F$$

SOLUTION – PROBLEM B (cont.)

Subject: CENTRIFUGAL COMPRESSORS-BHP			Date	
For: Main Air Blower			By	
SYMBOL				
Q_b	Million SCFD	34.54		
T_s	$^{\circ}R$	555		
P_s	psia	14.4		
P_d	psia	51.2		
r	P_d/P_s	3.556		
mol wt	Molecular Weight	28.49		
R	$\frac{1545}{\text{mol wt}}$	54.23		
k	C_p/C_v	1.4		
Q_s	$\text{ft}^3/\text{min @ Suction}$	26134		
lb/min wt flow	$\frac{(Q_b)(\text{mol wt})}{546000}$	1800.8		
Z_s		1.0		
Z_d		1.0		
Z_{avg}		1.0		
E_h	Polytropic Efficiency	0.832		
	$\frac{k-1}{k}$	0.2857		
M	$\frac{k-1}{k E_h}$	0.343		
	$(r)^M - 1$	0.546		
Beta	$\frac{(r)^M - 1}{M}$	1.590		
Head	$(Z_{avg})(R)(T_s)(\text{Beta})$	47846		
Gas hp	$\frac{(\text{Wt Flow})(\text{Head})}{(33000)(E_h)}$	3138		
Losses Frictional Seal	1 pct of Gas hp	32		
Gear	(If Used) 3 pct of hp	-		
Total bhp		3170		
T_d	$(T_s)(r)^M$	398		

SOLUTION – PROBLEM C

	Centrifugal	Reciprocating
Capital Cost	\$5.32	\$9.98
Installation	\$0.85	\$4.99
Utilities/Year	\$2.72	\$2.44
Maintenance	0	\$0.26
Total/1st Year	\$8.89	\$17.67

Savings with centrifugal compressor – \$8.78 million for first year

CENTRIFUGAL COMPRESSORS

PROJECT NUMBER

11103

		1st	2nd			
Q_b	Million SCFD	74.76	77.27			
T_s	$^{\circ}R$	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
MW	Mol. Wt.	10.3	9.9			
R	1545/MW	150.0	156.1			
k	C_p/C_v	1.257	1.267			
Q_s	CFM @ Suct.	16832.556	7824.635			
#/Min Wt. Flow	$\frac{Q_b \times MW}{546000}$	1409.14	1399.89			
Z_s		1.000	1.002			
Z_d		1.002	1.004			
Z_{av}		1.001	1.003			
E_h	Poly Eff.	0.83	0.81			
	$k - 1/k$	0.204	0.211			
M	$k - 1/kE_h$	0.246	0.260			
	$(r)^M - 1$	0.232	0.236			
Beta	$\frac{(r)^M - 1}{M}$	0.941	0.908			
Head	$Z_{av} R T_s \text{ Beta}$	79991.486	78840.783			
GHP	$\frac{\text{wt Flow} \times \text{Head}}{33000 \times E_h}$	4115.351	4129.017			
Losses	1% of GHP	1.01	1.01			
Frict.						
Seal	3% of GHP	1	1			
Gear						
Total BHP		4156.504	4170.307			
T_d	$T_s(r)^M$	237.232	226.041			

RECIPROCATING COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD	37.38	38.64			
T_s	$^{\circ}R$	566	555			
P_s	psia	49.35	107.81			
P_d	psia	115.06	243.50			
r	P_d/P_s	2.33	2.26			
k	C_p/C_v	1.257	1.267			
MW	Mol. Wt.	10.3	9.9			
S.G.	Spec. Grav.	0.4	0.3			
Z_s		1.000	1.000			
Z_d		1.002	1.002			
Z_{ave}		1.001	1.001			
Q_s	CFM Suct	8416.278	3905.014			
	$k - 1/k$	0.204	0.211			
T_d	$\frac{k-1/k}{r} T_s$	212.948	198.960			
(A)	BHP/mm @ 14.4 psia & Suct T	41	41			
(B)	Sp. Gr. Corr	3.5	3.5			
(C)	$Q_b (1.02) T_s/520$	41.50	42.07			
(D)	Add 5% if N.L.	1	1			
(E)	Z_{av}	1.001	1.001			
(F)	Gear loss 3%	1	1			
BHP Req'd	(A+B) (C) (D) (E) (F) (G)	1848.616	1873.790			

CENTRIFUGAL COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD					
T_s	$^{\circ}R$					
P_s	PSIA					
P_d	PSIA					
r	P_d/P_s					
MW	Mol. Wt.					
R	$1545/MW$					
k	C_p/C_v					
Q_s	CFM @ Suct.					
#/Min Wt. Flow	$\frac{Q_b \times MW}{546000}$					
Z_s Z_d Z_{av}						
E_h	Poly Eff.					
	$k - 1/k$					
M	$k - 1/k E_h$					
	$(r)^M - 1$					
Beta	$\frac{(r)^M - 1}{M}$					
Head	$Z_{av} R T_s \text{ Beta}$					
GHP	$\frac{\text{wt Flow} \times \text{Head}}{33000 \times E_h}$					
Losses						
Frict.	1% of GHP					
Seal						
Gear	3% of GHP					
Total BHP						
T_d	$T_s(r)^M$					

RECIPROCATING COMPRESSORS

PROJECT NUMBER

		1st	2nd			
Q_b	Million SCFD					
T_s	$^{\circ}R$					
P_s	PSIA					
P_d	PSIA					
r	P_d/P_s					
k	C_p/C_v					
MW	Mol. Wt.					
$S.G.$	Spec. Grav.					
Z_s Z_d Z_{ave}						
Q_s	CFM Suct					
	$k-1/k$					
T_d	$\frac{k-1}{r} \frac{k}{T_s}$					
(A)	BHP/mm @ 14.4 psia & Suct T					
(B)	Sp. Gr. Corr					
(C)	$Q_b (1.02) T_s/520$					
(D)	Add 5% if N.L.					
(E)	Z_{av}					
(F)	Gear loss 3%					
BHP Req't	(A+B) (C) (D) (E) (F)					

[illegible]