OIL FLOODED SCREW COMPRESSOR SYSTEMS FOR POWER GENERATION, PROCESS GAS COMPRESSION AND REFRIGERATION APPLICATIONS

PRESENTED BY:

MOHAMMAD A. GAUHAR

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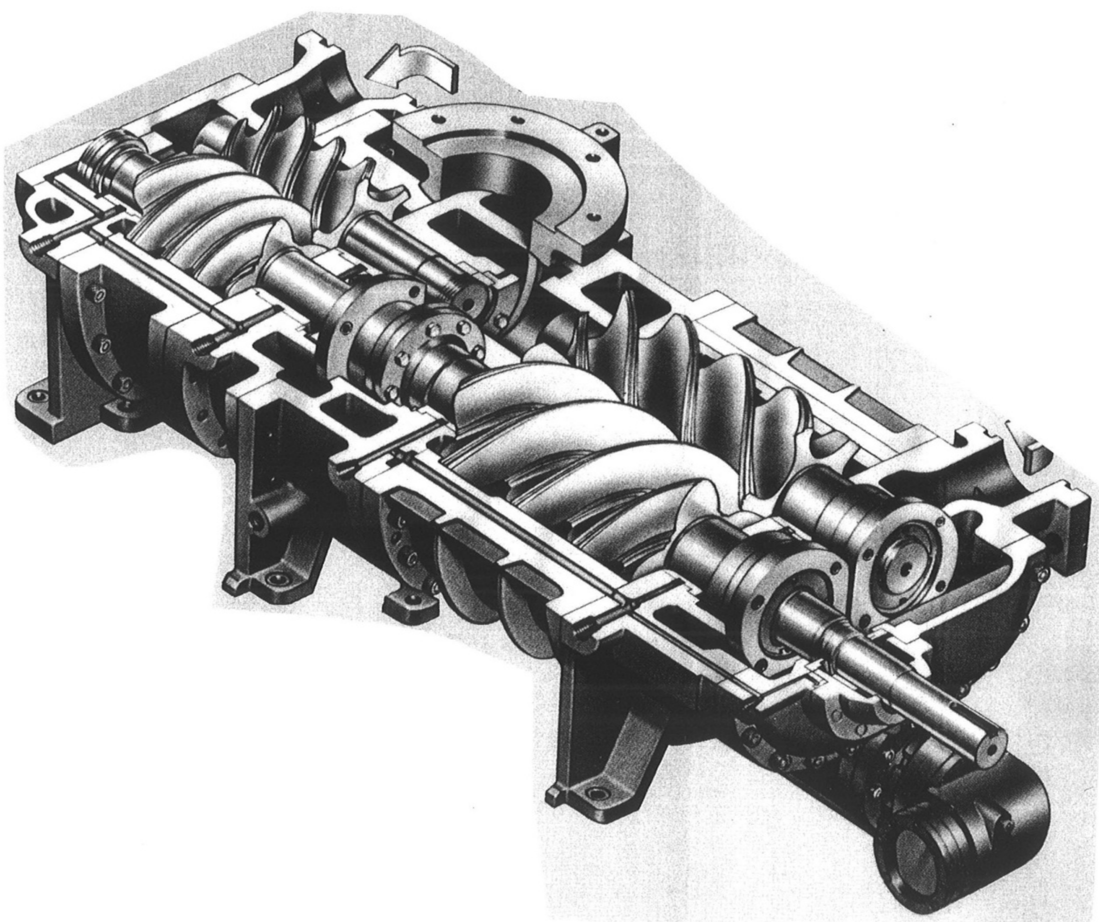
CHAPTER 1

AN INTRODUCTION TO OIL FLOODED SCREW COMPRESSORS



In this Chapter, the basic technology and the outstanding features of the oil flooded screw compressors are reviewed, and the principles governing the operation applications and performance are explained. Additionally, an attempt has been made to cover the fundamentals of the oil management system as well as the peripheral equipment which is essential to the operation of an oil flooded screw compressor.

Most details in this white paper apply to all compressors but certain features are applicable to the oil flooded screw compressors manufactured by Mycom, Howden, and Kobe Steel Ltd. The use of their figures, diagrams, and marketing literature is acknowledged with thanks.

CHAPTER 1

1.0 INTRODUCTION

1.1 EVOLUTION OF SCREW COMPRESSORS

The oil flooded rotary screw compressor in its present form was first introduced to the industry in the mid and late fifties. The initial introductions were mainly in Europe with applications restricted to refrigeration plants. It was not until the mid 60's that the industry felt the full impact of this machine. In the following ten years, the screw compressor gained a healthy share of the compression market. During this time, packaging concepts of screw compressors had also evolved steadily to a more streamlined shape and form. By the mid 70's, screw compressor engineers were gaining the confidence and the necessary know how to apply this machine to a variety of open cycle and non-conventional closed-loop applications. The importance of correct oil management in such applications was evident to the engineers from the onset. Much hard work was done to establish the screw compressor in refrigeration, HVAC, and gas compression fields. Today in the USA, Europe, Japan, Australia, and in the rest of the world, the screw compressor or the helical lobe machine (some of the earlier writers addressed the screw compressor by this name) occupies a most conspicuous place among the positive displacement compressors.

The screw compressor is characterized by high volumetric and good adiabatic efficiencies, stepless capacity modulation, wide compression range, and valveless, pulsation-free operation. The simplicity of rotary motion lends the screw compressor its high degree of reliability, performance, and predictable life.

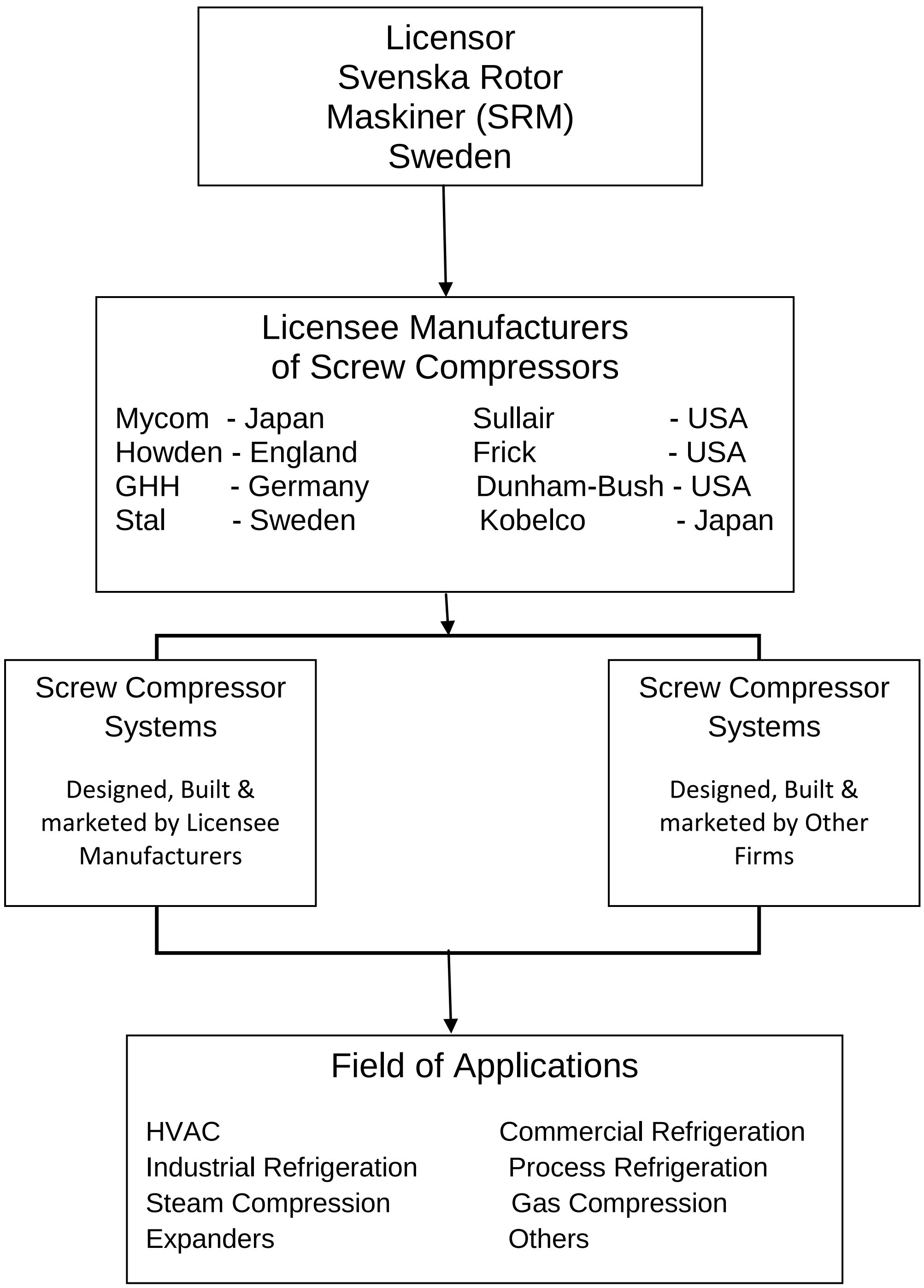
1.2 THE ROLE OF SVENSKA ROTOR MASKINER (SRM)

Oil injected screw compressors originated in Sweden about 60 years ago and were known to the gas and process industry well before its adaptation for refrigeration use. Earlier attempts to introduce screw compressors of non-lubricated types to gas compression and refrigeration duty proved futile since studies indicated high manufacturing costs, low operational efficiencies and noisy operation. These were a few of the reasons why this compressor was not pursued in America. In Europe, however, some engineers and firms continued their efforts towards an efficient, marketable screw compressor. In time a Swedish group called Svenska Rotor Maskiner (SRM) gathered, purchased, and developed a large pool of engineering and manufacturing know-how relating to design, fabrication, and performance of rotary screw compressors.

SRM developed the technique of introducing lubrication oil to the gas stream through the meshing lobes of the screw compressor. This technique drastically reduced the leakage which accounted for lower operational efficiencies of earlier machines. Through further work on the mathematics of rotor profile selection, SRM was able to reduce leakage losses still more. Rotors of non-symmetric profiles were designed which meshed more perfectly and uniformly. By the late 1950's, several efficiently-operating screw compressors were to be seen on refrigeration duty in Europe, and the screw compressor was well on its way.

SRM then protected their investments by patents and negotiated licensing agreements with manufacturers in Japan, Europe, and the USA to produce oil-injected screw compressors. These manufacturers then either packaged these compressors themselves or sold the bare compressors to other packagers for production of screw units. As can be readily sensed, excellent commercial decisions were incorporated in SRM's strategy to exploit the know-how they had acquired. Their strategy also accounts for the slow overall growth and the poor development and distribution of basic knowledge about the screw compressors.

SIMPLIFIED OVERALL STRUCTURE OF SCREW COMPRESSOR MARKET



2.0 PRINCIPLES OF OPERATION

2.1 GAS COMPRESSION MECHANISM

The screw compressor essentially consists of two mating, helically-grooved rotors (a male and a female) enclosed in a stationary housing with suitable inlet and outlet ports. The two rotors are not the same shape. Figure 1 shows a male and a female rotor. The male rotor is driven by the compressor driver. It has four lobes. The female meshes with and is driven by the male rotor. It has six interlobe spaces.

As shown in Figure 2, compression is obtained by direct volume reduction with pure rotary motion. Suction occurs when the lobes of the male rotor starts unmeshing from an interlobe space in the female rotor; a void is created which draws gas through the suction port. This trapped gas moves circumferentially at constant pressure until another lobe of the male rotor starts meshing with the interlobe space of the female rotor at the suction end. The interlobe space also decreases progressively. As a result, the gas pressure increases as the rotation continues. This compression continues until the interlobe space becomes exposed to the outlet port in the casing and the gas is discharged.

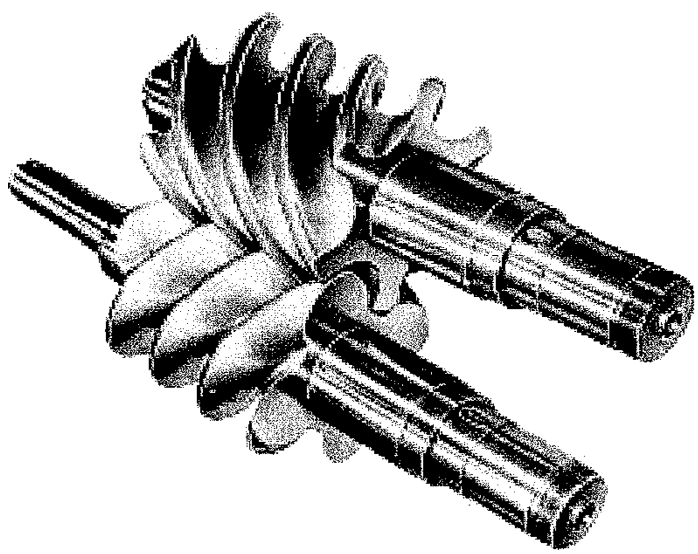
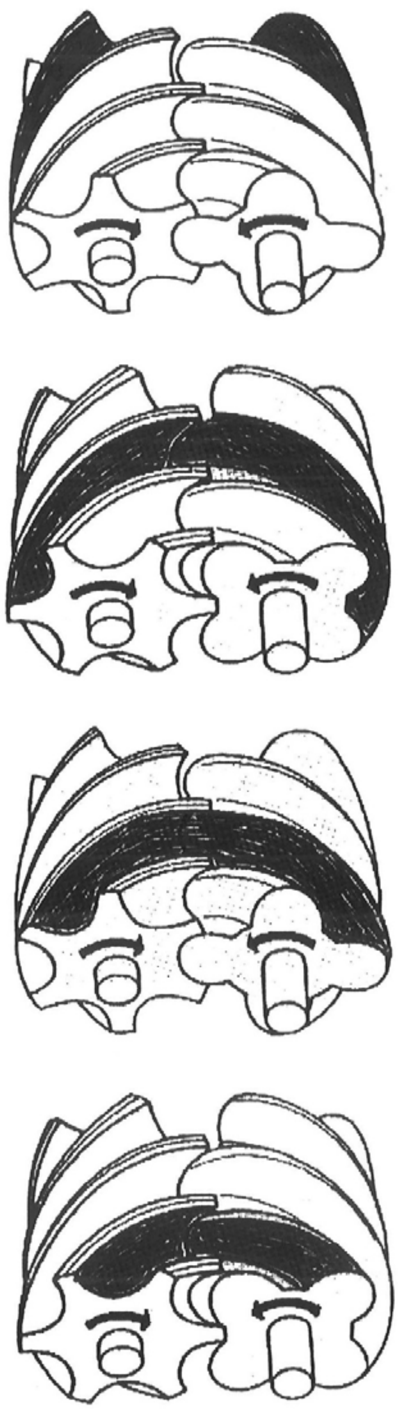


Figure 1

THE COMPRESSION CYCLE IN A SCREW COMPRESSOR



Gas is drawn in to fill the interlobe space between adjacent lobes.

As the rotors rotate, the interlobe space moves past the inlet port so sealing the interlobe space.

Continued rotation progressively reduces the space occupied by the gas causing compression.

When the interlobe space becomes exposed to the outlet port, the gas is discharged.

Figure 2

2.2 BUILT-IN VOLUME RATIO (Vi)

The compression process which is a result of progressively decreasing interlobe space can continue indefinitely if some relief is not provided at a particular point along the length of the casing. By providing a port for relief in the end or bearing housing of the compressor, the compressed gas will be permitted to pass into the discharge line. The amount of internal compression which occurs before release is, therefore, a characteristic which can be designed by the positioning of the discharge port to suit a particular duty. This feature is known as the "Built-In Volume Ratio" of the compressor and is usually denoted by the symbol "Vi."

$$Vi = \frac{\text{VOLUME OF TRAPPED GAS AFTER SUCTION PHASE IS COMPLETED}}{\text{VOLUME OF THE SAME AMOUNT OF GAS WHEN DISCHARGE PHASE BEGINS}}$$

$$Vi = (P2/P1)^{1/K} \text{ or } Vi = (C.R.)^{1/K} \text{ or } C.R. = (Vi)^{K}$$

Whereas:

Vi	=	Designed Volume Ratio
K	=	Ratio of Specific Heats for the Gas (Cp/Cv)
P2	=	Discharge Pressure, Psia
P1	=	Suction Pressure, Psia
C.R.	=	Designed Compression Ratio= P2/P1

Typical built-in volume ratios available from screw compressor manufacturers for the low, medium, and high compression ratio applications are as follows:

Compression Ratio (C.R.)	Volume Ratio (Vi)
Low	2.0-2.6
Medium	3.6-3.8
High	5.6-5.8

The correct built-in volume ratio should be chosen according to the operating conditions with careful consideration to all the variations of the actual operation. Considerable waste of energy occurs if the "Vi" is not properly matched to the operating conditions.

2.2.1 Case "A" (High Compression Ratio Operation Using A Compressor With Low "Built-In" Volume Ratio.)

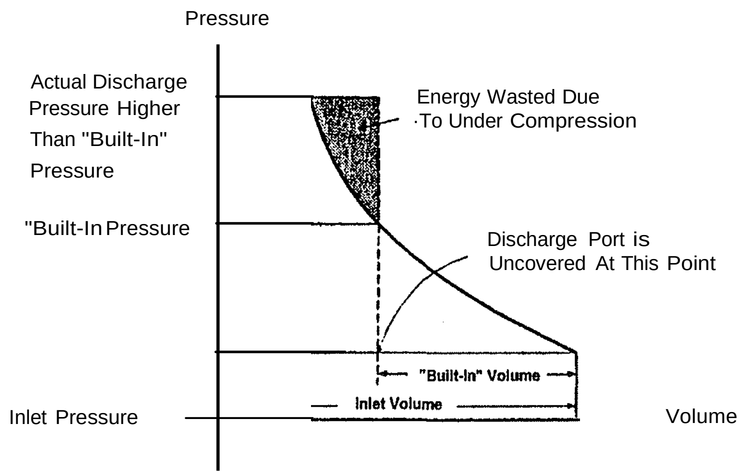


Figure 3

Figure 3 presents the case of high compression ratio using a compressor with low built-in volume ratio. The compressed gas arriving at the discharge port is not sufficiently compressed. The resulting difference in gas pressure on either side of the port produces a counterflow of gas towards the suction side.

Example:

P2	=	120 Psia
P1	=	20 Psia
C.R.	=	6.0
Vi	=	2.6
Gas	=	Nitrogen
K	=	1.4
C.R.	=	$(Vi)_{1tk}$

$$\frac{P2}{P1} = (2.6)^{1114} = (2.6)^{0.71} = 1.97$$

P2 = 20 x 1.97 = 39.4 Psia

Actual Operating Discharge Pressure = 120 Psia (given)
Under Compression= 120- 39.9 = 80.1 Psia

It can be clearly seen how much under compression occurred due to the mismatch of "Built-In" volume ratio to the actual operating pressures.

2.2.2 Case "8" (Low Compression Ratio Operation Using A Compressor With High "Built-In" Volume Ratio)

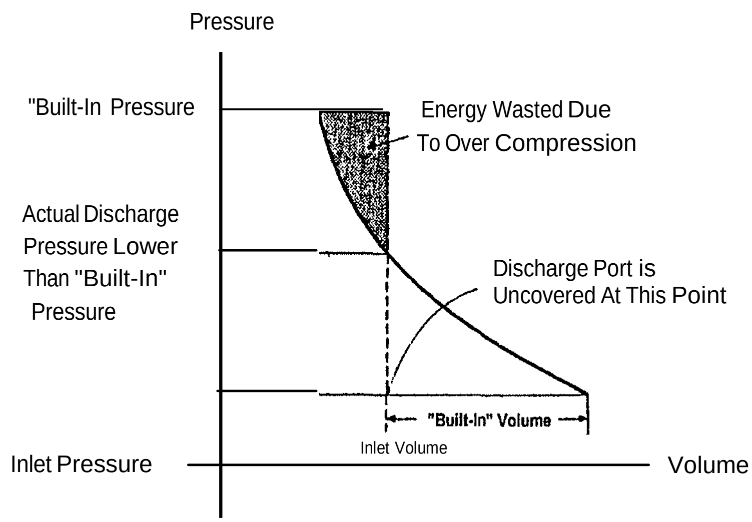


Figure 4

Figure 4 represents the case of low compression ratio operation using a compressor with a high built-in volume ratio. The gas releasing into the discharge port is excessively compressed, and it expands to occupy the space available at the discharge side. The compressor performs excess work unnecessarily.

Example:

P1	=	80 Psia
P2	=	175 Psia
Vi	=	**5.6**
K	=	**1.4**
C.R.	=	$(5.6)^{1/K} = (5.6)^{0.71} = 3.39$
$\frac{P2}{P1}$	=	**3.39**

P2 = 3.39 x 80 = 271.2 Psia

Actual Operating Pressure = 175 Psia (given)
Over Compression = 271.2 - 175 = 96.2 Psia

It can be clearly seen how much over compression occurred due to the mismatch of "Built-In" volume ratio to the actual operating pressures.

2.3 POSITIVE DISPLACEMENT OPERATION OF SCREW COMPRESSORS

If a discharge valve is closed and the compressor is allowed to operate, the discharge pressure will increase indefinitely along with a corresponding increase in absorbed power. This is a simple definition of a positive displacement compressor. An oil-flooded rotary screw compressor is a positive displacement machine and as such the power absorbed is directly proportional to the discharge pressure. In case of refrigeration machines, this feature provides the savings in energy due to varying condensing temperatures. Such is not the case in a centrifugal compressor.

Another advantage of the positive displacement compressor is that this type of compressor cannot surge and, therefore, complex control schemes and unloading by-passes are avoided. When an integral slide valve is used to unload this type of compressor, it provides considerable energy savings because full compression is avoided. External by-pass in other types of compressors has no energy savings because the by-passed gas is fully compressed.

2.4 MOVING PARTS

The only moving parts of an oil-flooded rotary screw compressor are:

A. Male and Female Rotors.

B. Slide Valve for Capacity Modulation.

2.5 VALVES

Oil-flooded rotary screw compressors have no suction or discharge valves. These valves are installed beyond the compressor as part of the system.

2.6 OPERATING SPEED

In most cases, the compressor is directly coupled to its driver-typically an electric motor. At 50 hertz operation, the synchronous speed is 3000 RPM; whereas at 60 hertz, it is 3600 RPM. Gearboxes are seldom used; although for special applications, it is a viable approach.

3.0 TYPES OF SCREW COMPRESSORS AND THEIR APPLICATIONS

3.1 TYPES OF SCREW COMPRESSORS

Screw compressors can be divided into two distinct categories:

3.1.1 Oil Free or "Dry" Compressors.
3.1.2 Oil Flooded or "Wet" Compressors.

Both types operate on the same principle except that in one case the gas is compressed completely oil-free. Whereas in the other case, the oil is injected into the gas during compression which brings about the combined effect of sealing, lubrication, and noise reduction. This oil is separated from the gas after it leaves the compressor.

Only the oil-flooded compressor types are discussed in this paper.

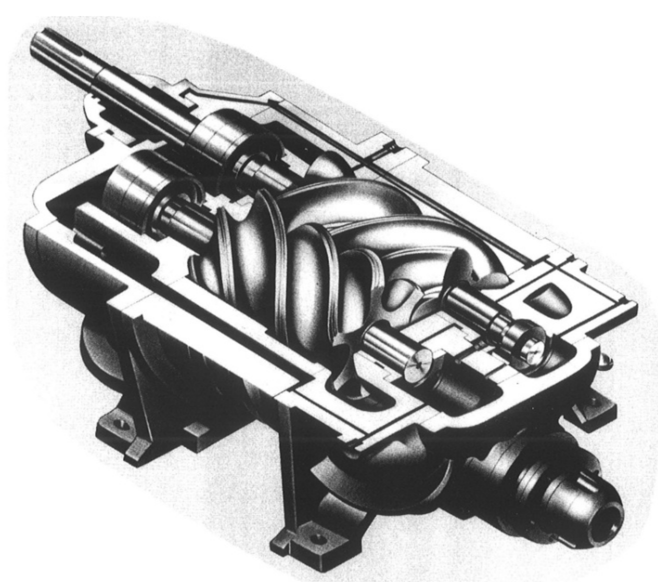
3.2 VARIATIONS OF OIL-FLOODED SCREW COMPRESSORS

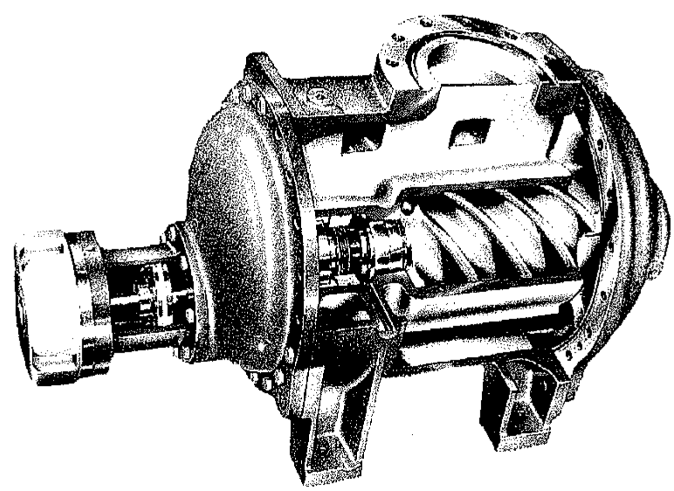
The oil-flooded compressor is now available in the following variations:

3.2.1 Single Stage Fixed Volume Ratio Compressor (Figure 5 and Figure 6).
3.2.2 Single Stage Variable Volume Ratio Compressor (Figure 7 and Figure 8).
3.2.3 Compound Screw Compressor with Fixed Volume Ratios (Figure 9).

The following pictures illustrate these three types of oil-flooded screw compressors.

3.2.1 SINGLE STAGE FIXED VOLUME RATIO COMPRESSORS

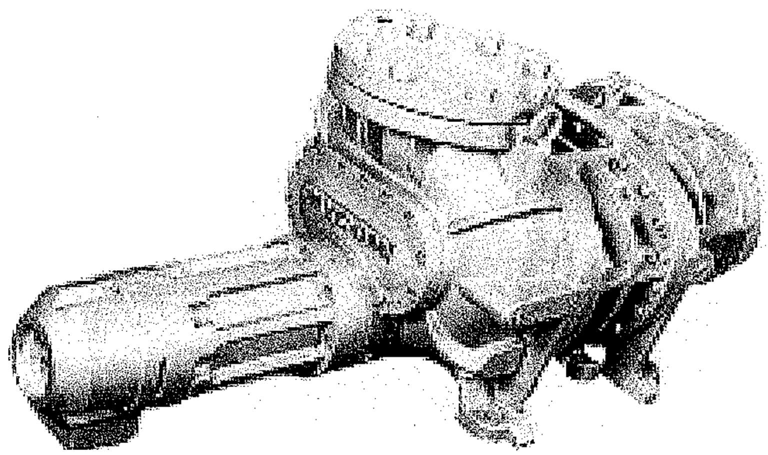




Howden Compressor
Figure 6

Mycom Compressor
Figure 5

3.2.2 SINGLE STAGE VARIABLE VOLUME RATIO COMPRESSOR



MYCOM COMPRESSOR
Figure 7

VARIABLE Vi MECHANISM

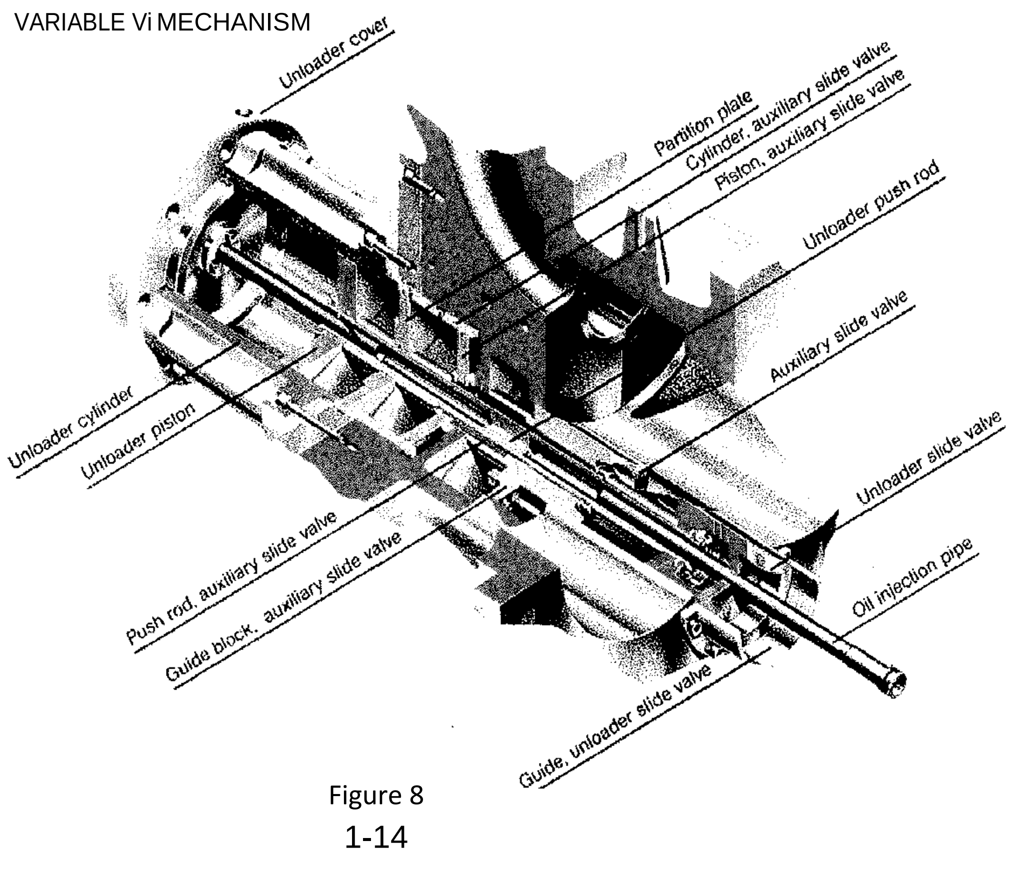


Figure 8

3.2.3 MYCOM COMPOUND SCREW COMPRESSOR WITH FIXED VOLUME RATIO

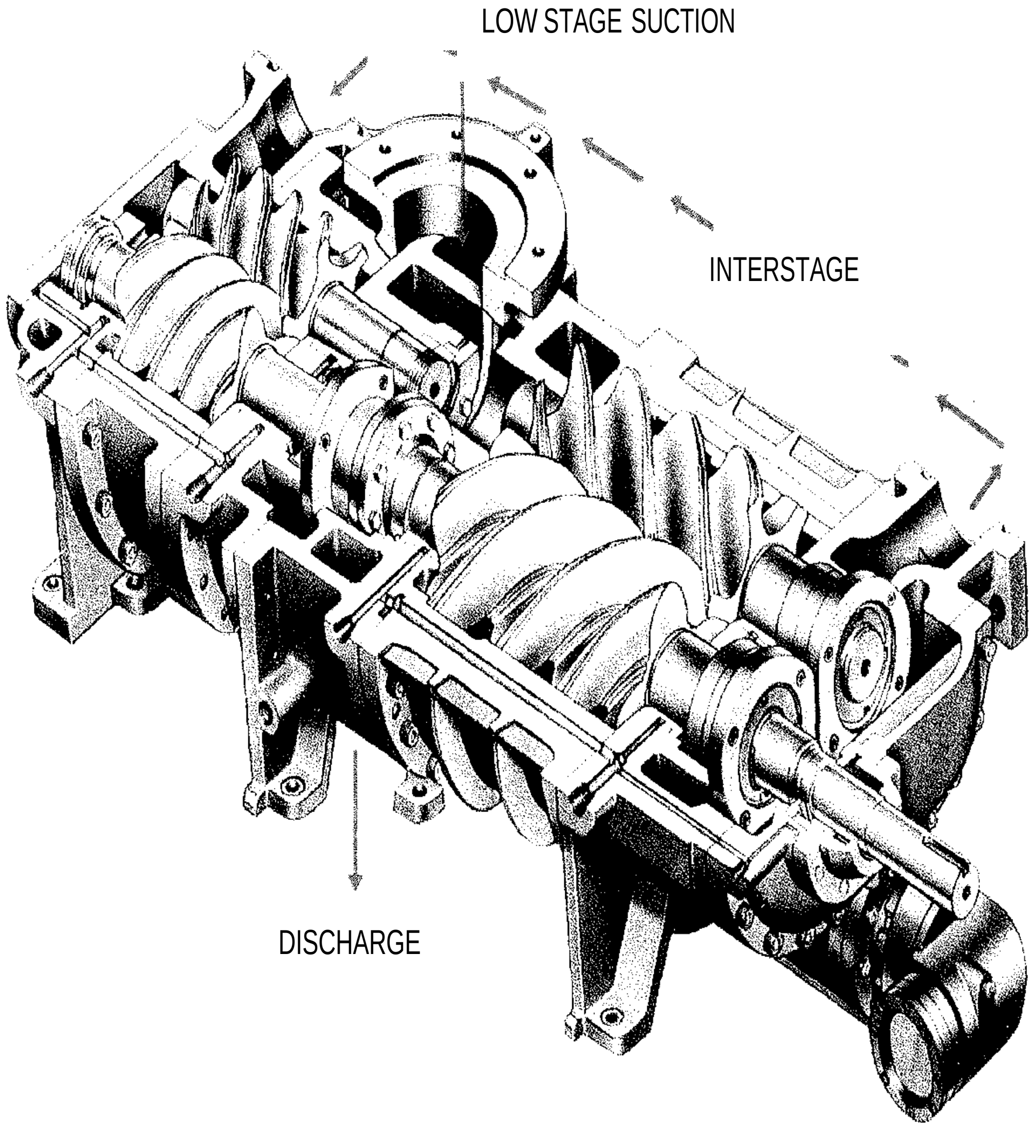


Figure 9

A compound screw compressor has two stages of compression built in one casing. Compression ratios of 22:1 are possible with only a single driver without intercooling.

3.2.4 KOBE COMPOUND SCREW COMPRESSOR

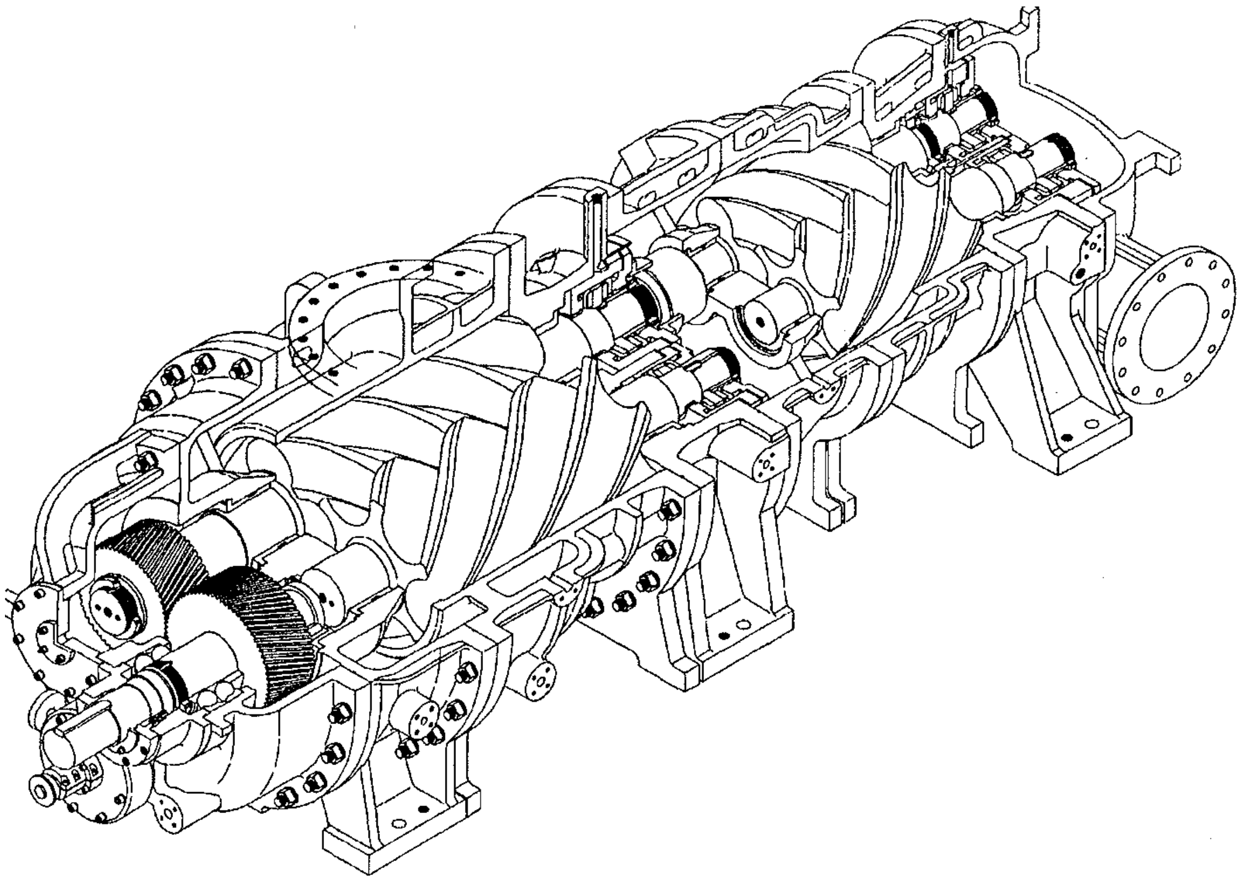
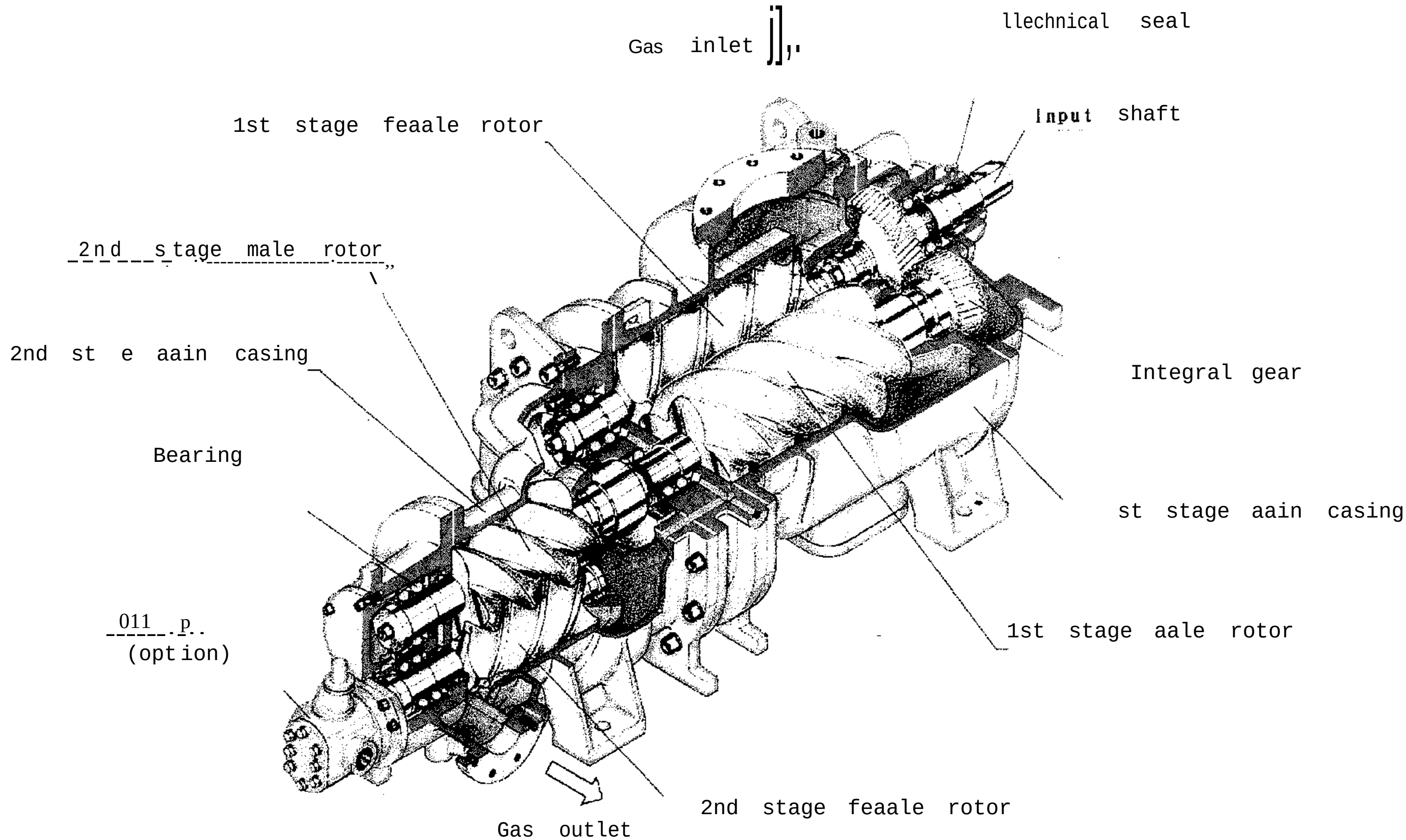
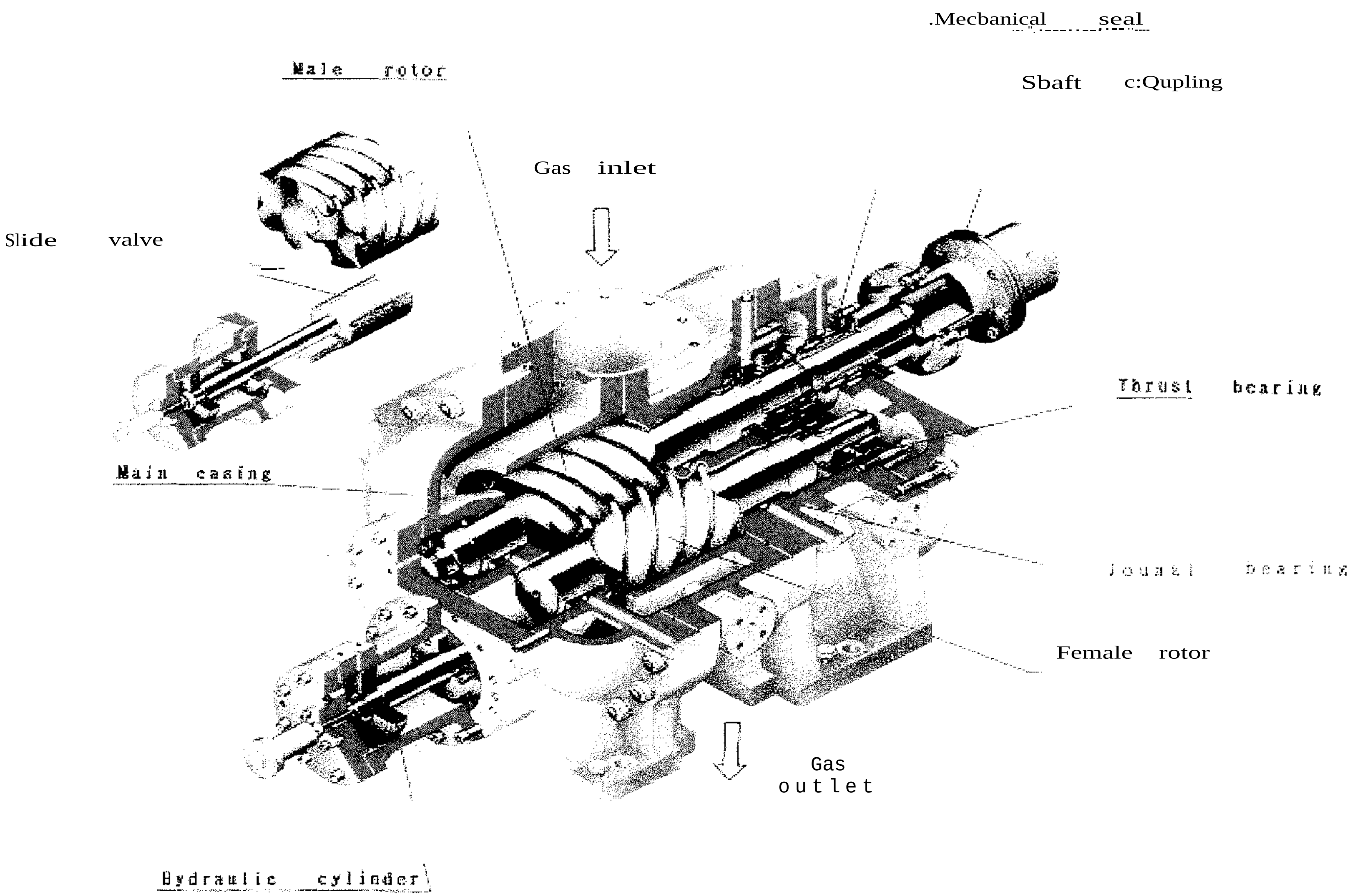


Figure 10

(GEARS SHOWN ARE NOT TIMING GEARS BUT ARE USED FOR VARYING OPERATIONAL SPEEDS)



COMPOUND SCREW COMPRESSORS WITH INTEGRAL GEARS (KOBE STEEL LIMITED)



SINGLE OIL-FLOODED SCREVV COMPRESSOR (KOBE STEEL Lltv11TED)

3.3 APPLICATIONS OF THE OIL-FLOODED SCREW COMPRESSOR

The applications of the oil-injected screw compressor can be categorized primarily into two divisions.

3.3.1 Gas Compression Systems

The oil-flooded screw compressor has been successfully applied in the compression of the following gases:

Argon	Refinery Off Gas
Hydrogen	Town Gas
Helium	Syngas
Nitrogen	Membrane Feed Gas
Landfill Gas	Cracked Gas
Natural Gas	Air
Coke Oven Gas	Instrument Air
Methyl Chloride	H2S Laden Hydrocarbon Streams (Sour Gas)
Hydrogen Chloride	LPG Vapors
Hexane	Mixed Gas Streams
PSA Tail & Feed Streams	LPG Vapors,

3.3.2 Refrigeration Systems

The following refrigerants have been successfully compressed by oil-flooded screw compressors.

Halocarbon,
R-12, R-11, R-22, R-13, R-134A, etc.

Azeotrope,
R-500, R-502, etc.

Hydrocarbons,
R-50, R-170, R-290, etc.

Inorganic Compounds,
R-717, R-744, R-764, etc.

Unsaturated Organic Compounds,
R-1150, R-1270, etc.

Diagrams on the following pages represent some of the oil-flooded screw compressors' applications in a simplified format.

TYPICAL SCHEMATIC DIAGRAM FOR A SINGLE STAGE GAS COMPRESSION SYSTEM WITH AN OIL-INJECTED SCREW COMPRESSOR AND AN API-614 CLASS LUBE OIL MANAGEMENT

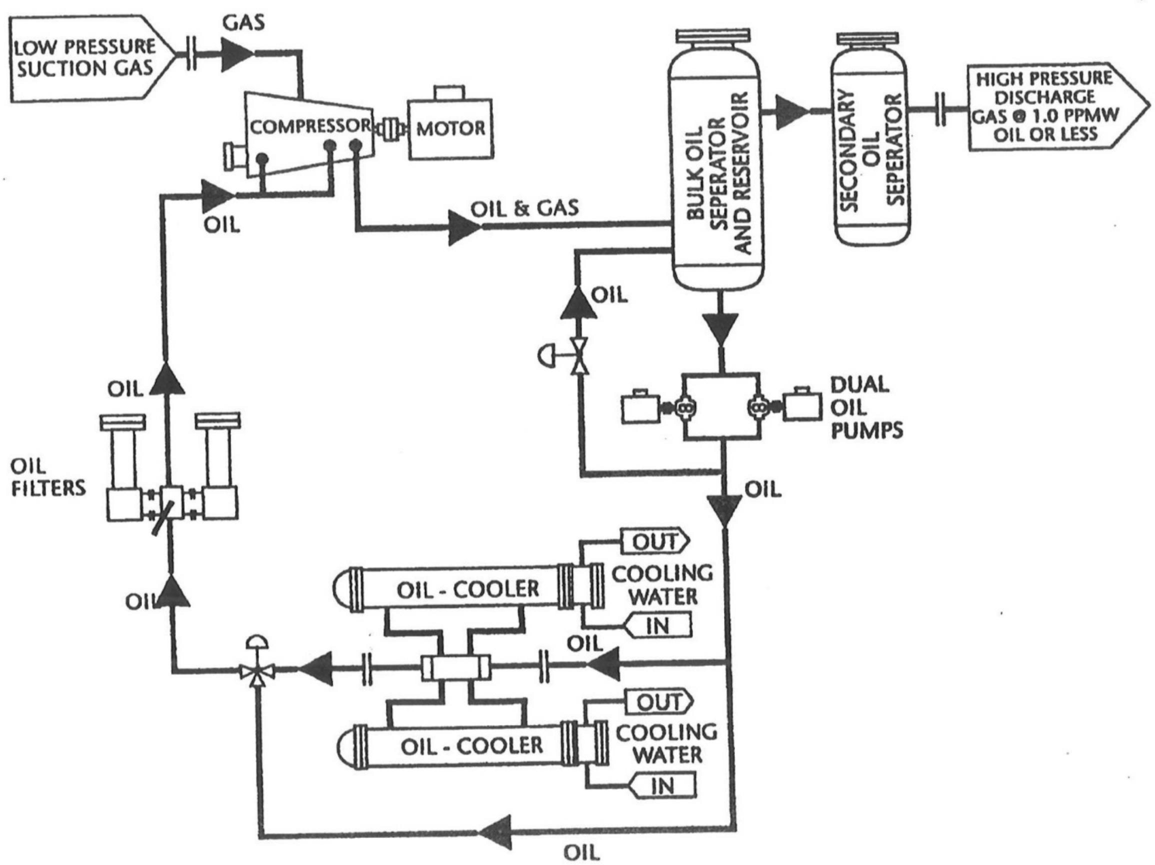


Figure 11

TYPICAL SCHEMATIC DIAGRAM FOR A THREE TWO-STAGE OIL-INJECTED SCREW COMPRESSOR ON A COMMON API-614 CLASS LUBE OIL SYSTEM WITH GAS PURIFICATION TO 1 PPBw OF OIL CARRYOVER

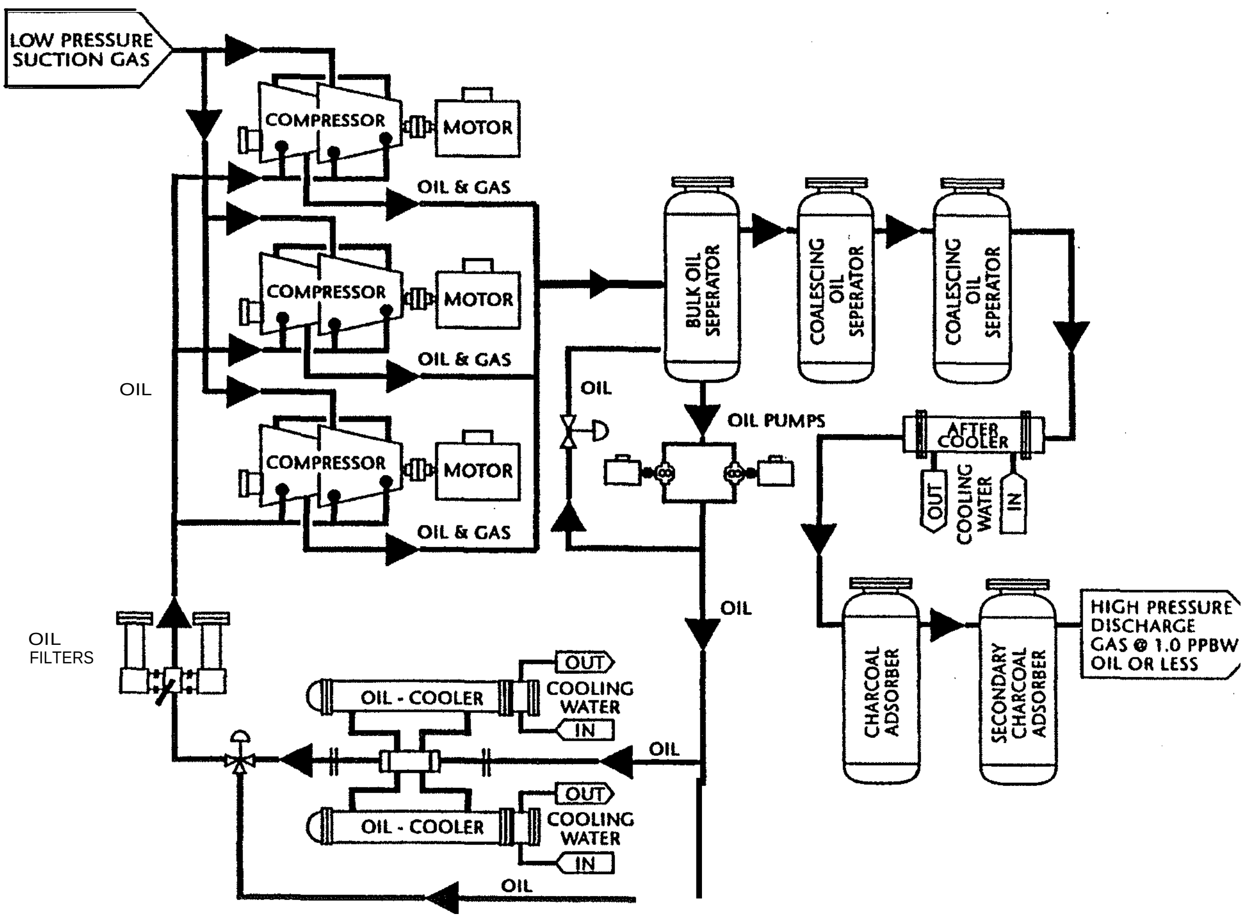
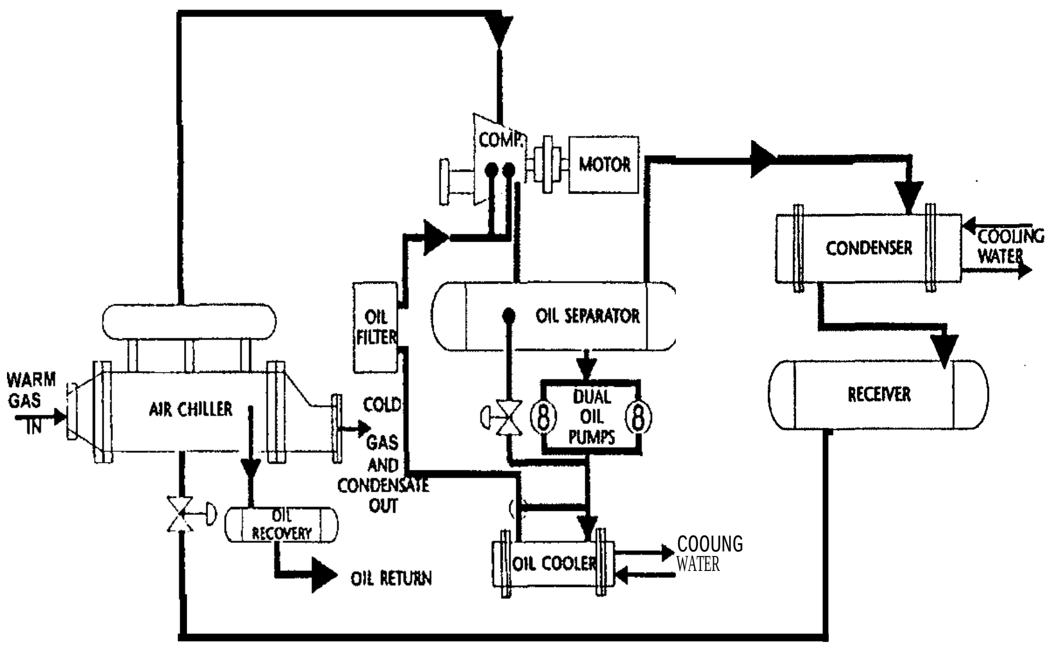


Figure 12

TYPICAL APPLICATION OF AN OIL-INJECTED SCREW COMPRESSOR
IN A REFRIGERATION SYSTEM



4.0 THE DESIGN OF THE OIL-INJECTED SCREW COMPRESSOR

4.1 GENERAL DESCRIPTION

Figure 14 and Figure 15 show the basic design features of a typical oil-injected screw compressor. Each major component is described as follows:

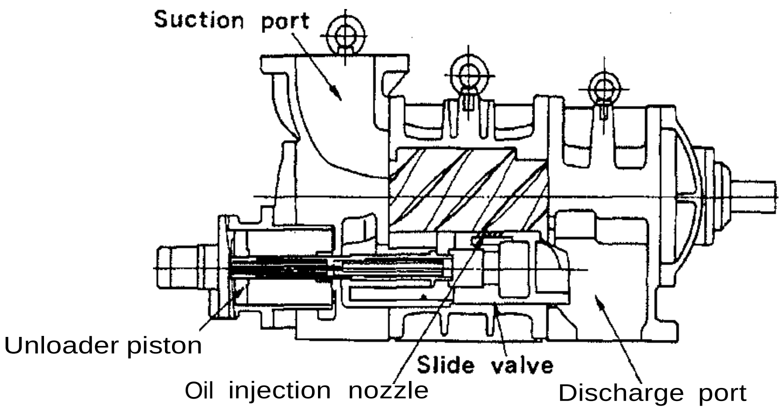


Figure 14

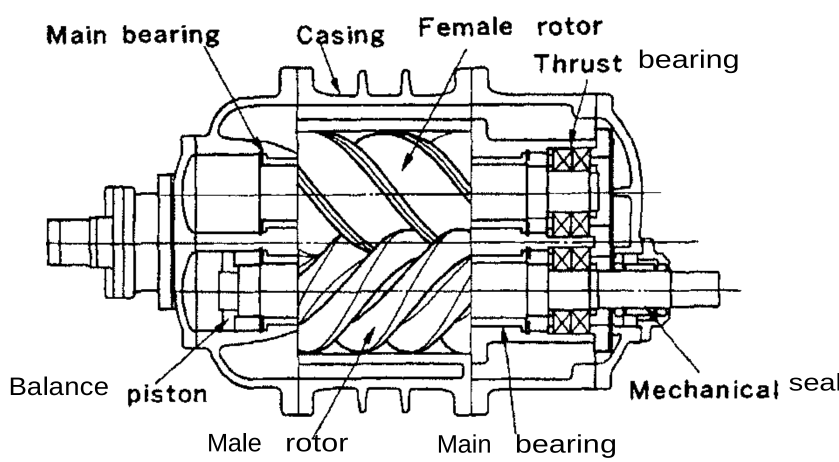


Figure 15

4.1.1 Casing

The casing is usually an iron casting. Optional materials of ductile iron, steel, stainless steel, etc. are available for petro-chemical applications. Double casing or ribbed design provides the necessary structural strength. These casings are bored precisely to close radial, axial, and angular tolerances.

4.1.2 Rotors

Figure 16 below shows the rotors as they appear in the end view from the suction end.

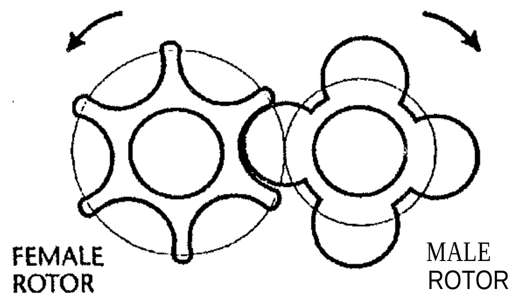


Figure 16

For the rotors to operate without binding and to fit in a SLJitable housing, proper clearances are maintained between the rotor tips and the casing. However, unlike reciprocating compressors, these clearances cannot be sealed by a simple piston ring. These clearances are responsible for the poor efficiency of the dry compressor. With oil injection, these clearances are effectively sealed. Thus, the wet screw compressor becomes a very efficient machine.

The rotor having the four convex lobes is called the male rotor, and the rotor with the six concave lobes is called the female rotor. At 60 hertz operation, the male rotor is driven by a 2-pole motor at 3600 RPM, and the female rotor follows at 2400 RPM. At 50 hertz operation, the male rotor turns at 3000 RPM while the female rotor follows at 2000 RPM.

The rotor profiles are unsymmetrical which gives high compression efficiency. Oil flooding prevents wear on the rotors, and there is no mechanical contact between the rotor and the casing.

Rotors are typically cast from ductile iron. Forged steel rotors are optional.

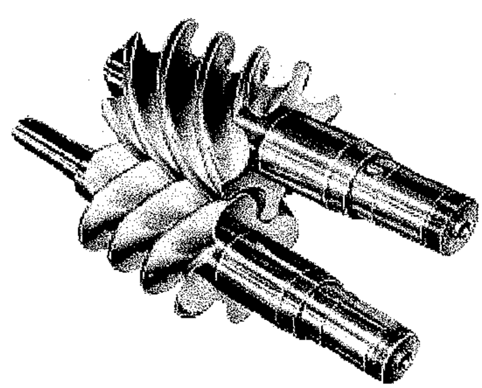


Figure 17

4.1.3 Journal Bearings (Shown in Figure 15 as Main Bearings)

The male and the female rotors are supported by generously-sized journal bearings with large oil pockets to ensure oil film under all working conditions. The bearings are not isolated from the compression chamber by shaft seals. In fact, the bearings serve the dual function of supporting the rotors and oil sealing the compression chamber. In order to accomplish this, oil is fed to the bearings at a pressure of 30-50 psi above the compressor discharge. By locating the bearings on either end of the compression chamber, the most rigid arrangement is obtained for minimizing the rotor deflection under gas compression loads.

Materials of construction for journal bearings is Babbit Lining on steel backing. Silver, graphitar, or other linings have been used on special applications.

4.1.4 Thrust Bearings

Considerable axial thrust is developed across the rotors which is countered by a pair of angular contact, anti-friction bearings on each rotor. The use of angular contact ball bearings also minimizes the gas leakage since minimum axial end clearance between the rotor and the casing can be maintained at the discharge end.

Tilting pad-type thrust bearings are available in most sizes as optional thrust bearings. Special considerations due to the hydro-dynamic nature and the end-play characteristics of such bearings have to be accounted for in such applications.

4.1.5 Balance Pistons

The thrust bearing's life is enhanced by incorporating a thrust balance piston which has the oil pump pressure (30-50 psi above discharge) on one side and compressor suction on the other side.

This thrust balance piston counters the axial thrusts of the rotors and relieves the bearings of full axial load. Thus, the thrust bearings carry a lighter load giving them a long life.

4.1.6 Shaft Seals

Single and double seals of balanced or unbalanced types are used. A double mechanical seal of the balanced type is generally recommended for applications involving low-molecular- weight gases such as helium and hydrogen.

4.2 <u>THE OIL INJECTION SYSTEM</u>

The injection of oil into the compression chamber of a screw compressor performs the following functions:

4.2.1 Lubrication

Oil film prevents the metallic contact of the rotating elements. The lube oil returns to the compression chamber and is exhausted with injection oil into the discharge manifold.

4.2.2 Sealing

The clearances between the male and the female rotors and between the rotors and the casing are effectively sealed by oil; thus, minimizing gas leakage from high pressure to low pressure zones. This means that the screw compressor can maintain a high volumetric efficiency over a wide range of compression ratios.

4.2.3 Cooling

The power required to compress the gas from the suction to the discharge pressure appears as heat in the gas. As an example, if dry compression of hydrogen is accomplished over a pressure ratio of 10:1, the final discharge temperature would be well over 356°F.

A small volume of oil injected into the working chamber of the compressor can bring the final discharge temperature in the range of 170-175°F. This enables the application engineer to apply the screw compressor without temperature limitations. The low discharge temperatures also minimizes potential deterioration of the oil and deformation of the rotors due to thermal expansion.

4.2.4 Noise Reduction

Oil injection considerably reduces the sound pressure level of screw compressors.

4.3 THE OIL FLOW PATH

Figure 18 shows the typical oil flow path through an oil-injected screw compressor. Figure 19 depicts the same in the form of a simplified line diagram.

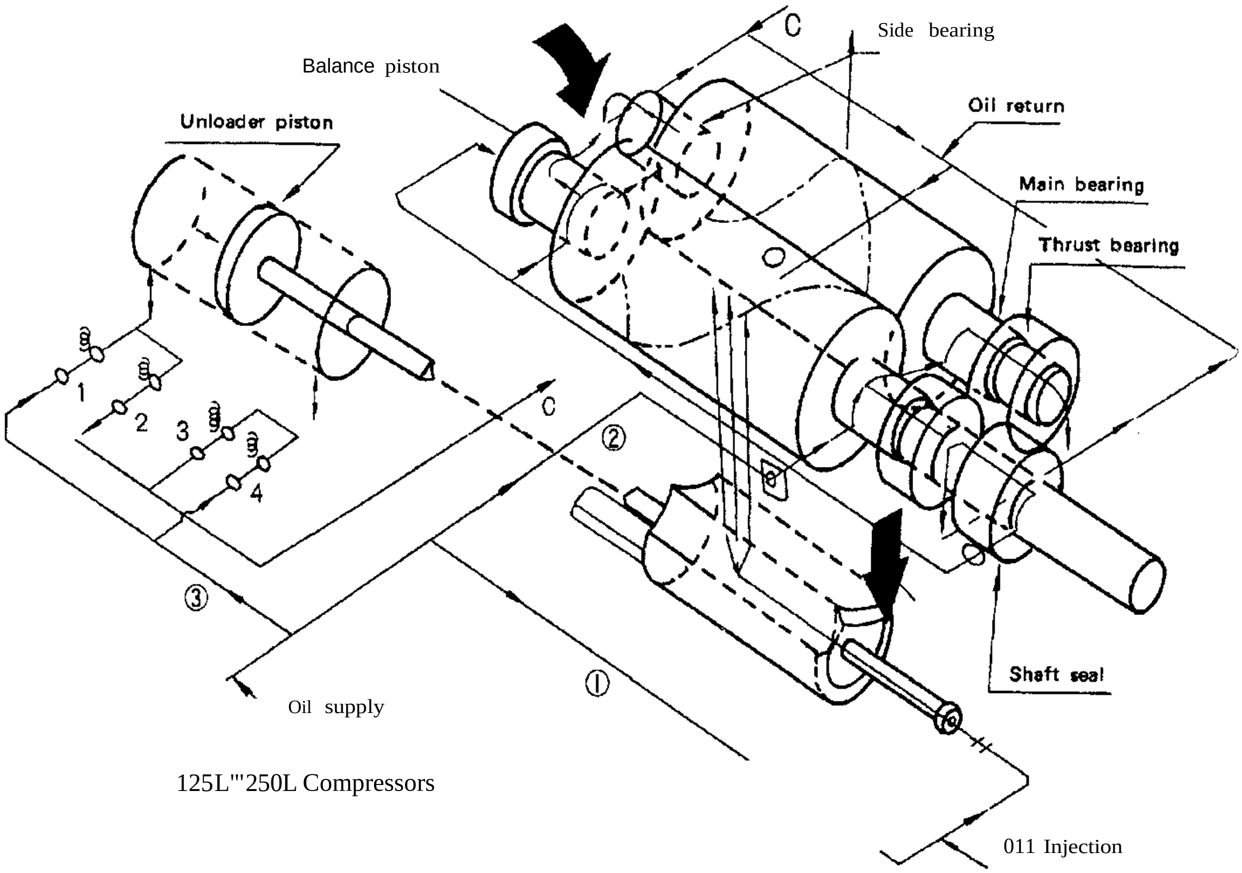
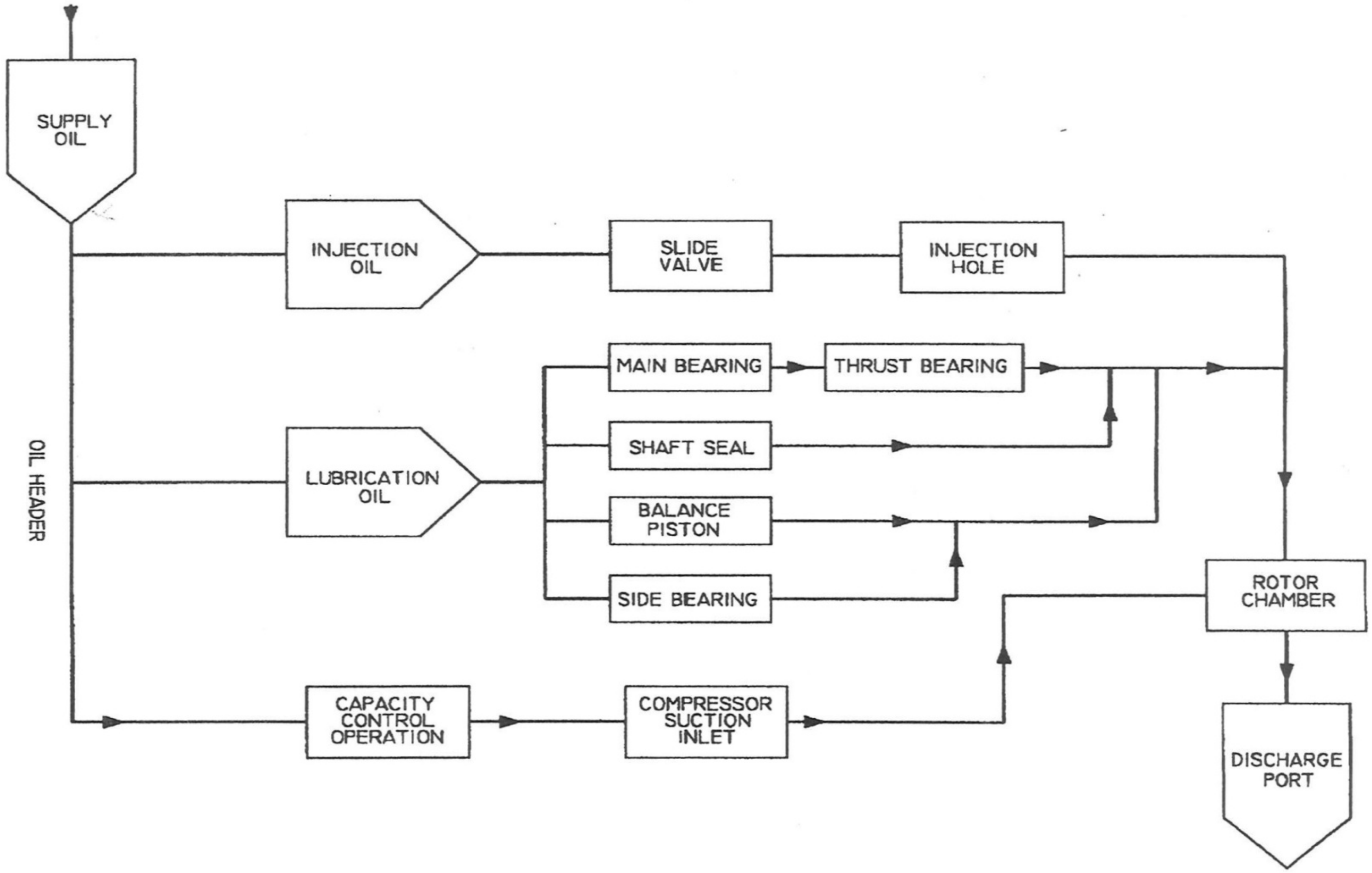


Figure 18



OIL FLOW SCHEMATIC

Figure 19

4.4 CAPACITY CONTROL SYSTEM

The capacity control in a screw compressor is accomplished by a slide valve. This slide valve moves parallel to the rotors axially and controls the capacity by altering the working length of the rotors being used for compression.

The action of the slide valve is to provide an opening for the low pressure gas at suction pressure trapped between the rotors and the casing to flow back into the suction manifold before the compression begins. In this, minimal work is done on the released gas, and, since the amount of the gas recycled is in direct proportion.to the position of the slide. valve, hence .the stepless capacity modulation.

The graphical representation of the slide valve's operation is best illustrated by Figure 20 and Figure 21. The movement of the slide valve is accomplished by feeding oil to one side or to the other side of a double-acting hydraulic cylinder which is integral with the compressor. The oil flow is controlled by solenoid valves. The position of the slide valve is indicated on an external stationary scale at the end of the hydraulic cylinder. The scale markings are a percent of the slide valve stroke and not a percent of the compressor output. Slide valve indicators are equipped with micro switches for performing functions such as confirming full unloaded and full loaded positions of the slide valve.

The unloaded position confirmation is another noteworthy feature of the slide valve arrangement providing for unloaded starting of the screw compressor. When the compressor is stopped, the slide valve remains at the fully open position; and therefore, the compressor can be started with a normal torque motor. To reduce the starting torque still further, a starting by-pass can be used.

OIL INJECTED SCREW COMPRESSOR CAPACITY CONTROL

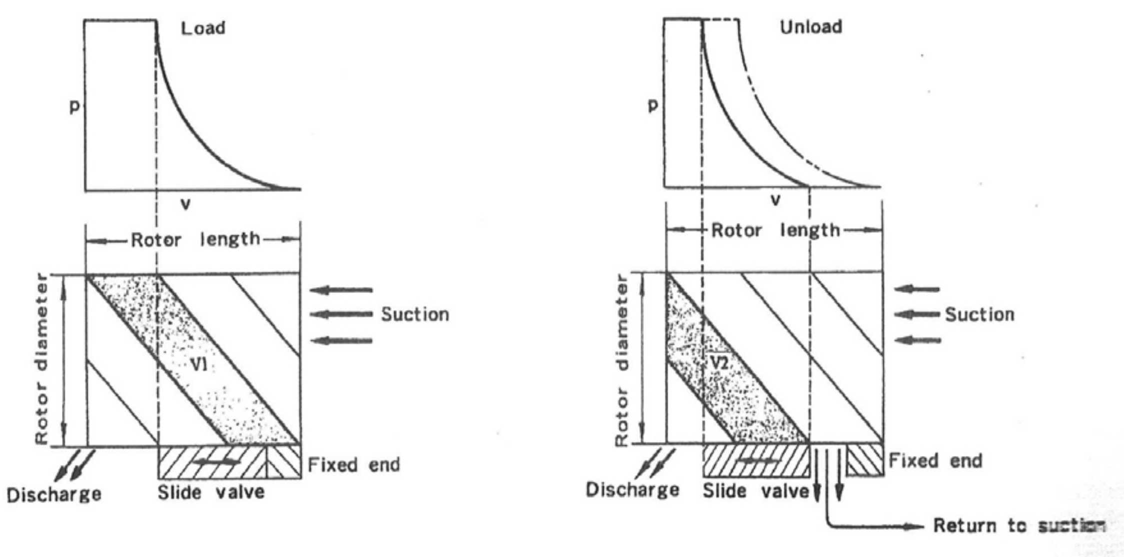
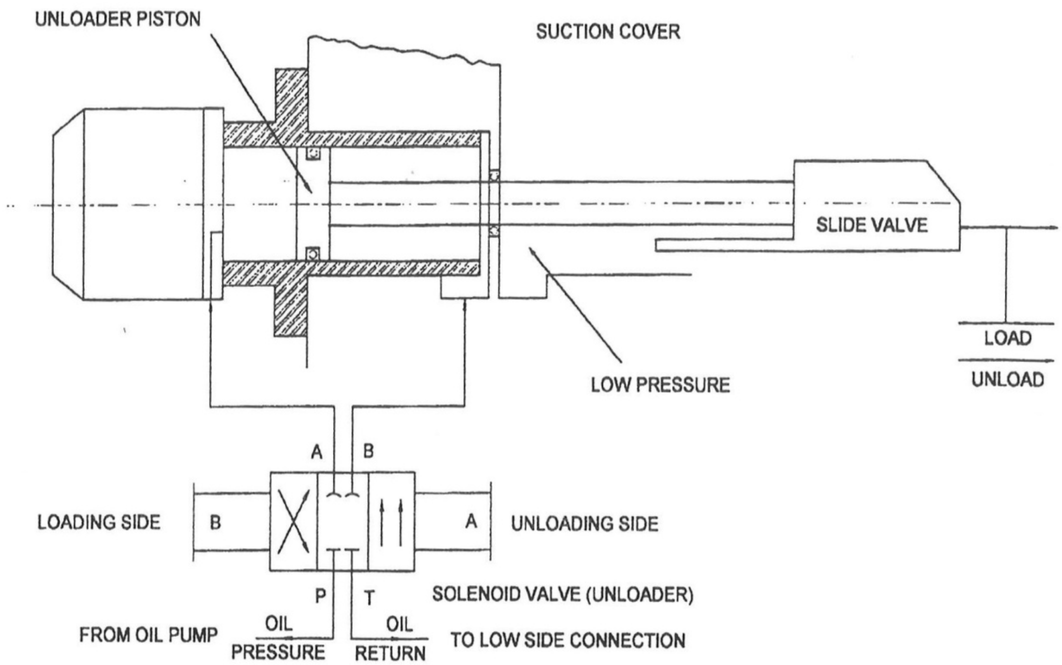


Figure 20

THE UNLOADING MECHANISM



CAPACITY CONTROL
Figure 21

The function of the control system is to vary the compressor capacity by means of the slide valve in accordance with the load. Any time the compressor capacity deviates considerably from the system load, unsatisfactory operating conditions will result. For re-establishing system equilibrium at design operating conditions, a variety of control systems are in use. The simplest control system is a direct hand control in which the compressor output is varied by push buttons located on the control panel. Figure 21 depicts the use of a 4-way spool valve for varying the capacity of an oil-injected screw compressor.

Another type employs pressure switches which are actuated by suction or discharge pressure. These pressure switches will control capacity by energizing or de-energizing solenoids. Loading is done by energizing the appropriate solenoid valve which permits oil pressure to move the hydraulic piston in the required direction. Unloading is accomplished by energizing solenoid valves which then move the hydraulic piston in the opposite direction. Similar capacity modulation can be obtained by sensing compressor suction temperature rather than pressure or the signal for capacity modulation may come from process temperature or pressure.

Yet another type of control system is the "time proportioning" system. In this, the plant temperature or pressure is monitored at regular intervals, and when this varies by a present amount from the set condition, the compressor's capacity is changed by a fixed percentage. This is a very simple system but it takes a little time to adjust and establish equilibrium. Pressure or temperature controls are utilized in this system.

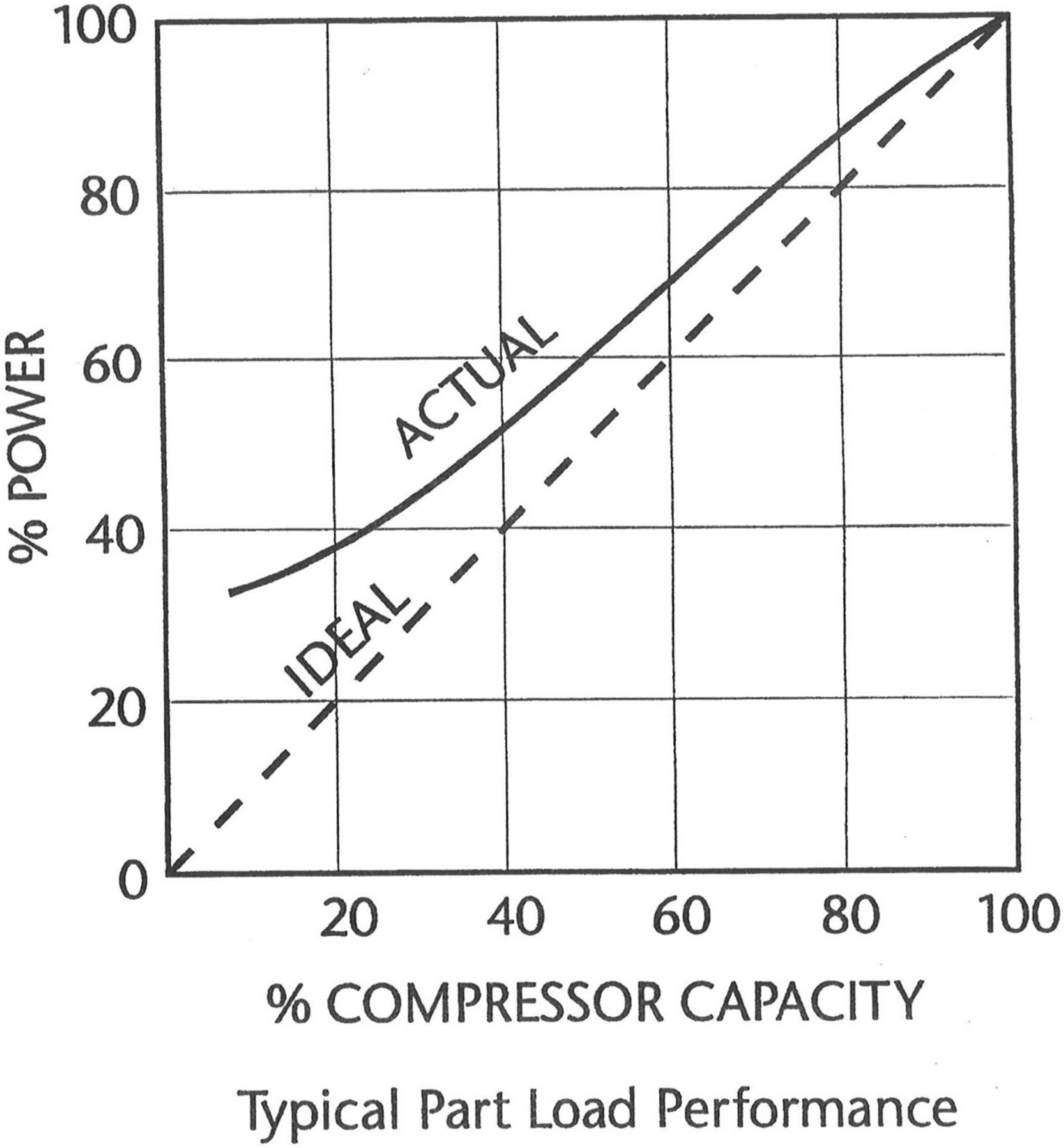
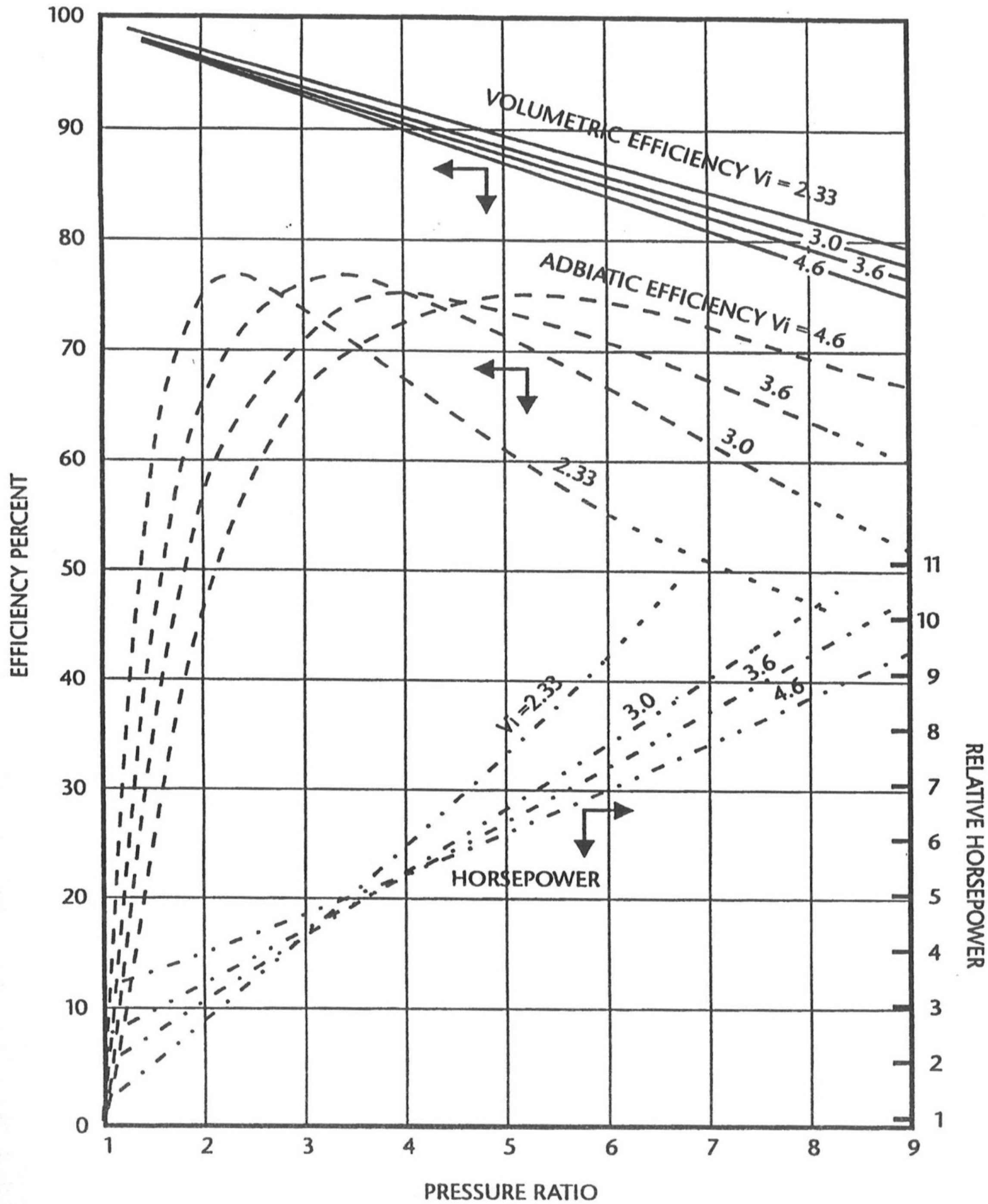


Figure 22

Figure 22 shows the part load performance of the screw compressor. As can be seen from the curve, the power reduction in the most important upper region is essentially proportional to the decrease in capacity for a wide range of operating conditions.



Efficiencies of Screw Compressors

Figure 23

5.0 RELIABILITY

5.1 COMMENTS ON RELIABILITY AND MAINTENANCE

Screw compressors are one of the simplest types of compressors to operate. The two rotors are the only moving parts. Bearings are very moderately loaded. The shaft seal is mechanical with forced lubrication and pressure balanced. These important features lend a high degree of respectability to screw compressors in an age when maintenance costs are skyrocketing.

Screw compressors have now proved successful in a number of applications, and new possibilities are opening up. The ability of the screw compressor to handle high pressure rations (10:1) makes it an obvious choice in low temperature or high pressure ratio applications previously considered reserved for two stage systems. At still higher ratios, a two stage screw compressor system is certainly a better choice than a conventional two stage system. In a compound screw system, there is no need for an intercooler.

The continuous capacity regulation in screw compressors is a very desirable feature as far as applications in the food processing industry are concerned where widely varying loads are very frequently encountered. In the chemical industry, shutdowns can become very expensive; here high reliability, ruggedness, and low maintenance makes the screw compressor a better choice. In addition to the already mentioned features, the ability of the screw compressor to tolerate some liquid slugging and its vibration-free operation can be added to the list of advantages in using a screw compressor.

As a general rule, a single oil flooded screw compressor provides the same degree of reliability and availability as two reciprocating compressors with one operational and the other standby.

CHAPTER 2

OIL-FLOODED ROTARY SCREW COMPRESSOR SYSTEMS FOR GAS TURBINE FUEL GAS BOOSTING APPLICATIONS

1.0 INTRODUCTION

Since the 1990s the author has designed, built and supplied over 300 systems with oil-flooded screw type compressors. To his credit, has the unique privilege of supplying some of the largest compressor systems of oil-flooded screw type in the world without any operating or oil carry-over problems. Most of such specialized compression systems are in hydrogen service with natural gas compressors closely following hydrogen applications.

The purpose of this chapter is to prove to the technical decision makers at large that if an oil-flooded screw compressor system is designed correctly for a gas turbine power plant application, it is the most suitable compressor for the fuel gas boosting among all available options.

The above conclusion is based on intimate knowledge the author has acquired over the years through extensive experience and advancements made in the oil management and operation of screw compressors. This know-how is a valuable intellectual property and this body of knowledge far exceeds the know-how possessed and acquired by manufacturers of screw compressor proper itself.

This advance know-how regarding these oil-flooded screw compressor systems is based upon some of the following Key areas:

2.0 OIL SEPARATION IN FUEL GAS BOOSTER (FGB) COMPRESSORS

Recommended design limits the oil carryover to 5 ppbw (parts per billion by weight). This oil carry-over limit far exceeds any stipulation of oil carry-over by turbine manufacturers. Typical stipulations by turbine manufacturers is 0.1 - 0.5 ppmw (aerosol). Author's design guarantees a much lower oil carry-over of 5 ppbw in both aerosol and vapor phase. Turbine manufacturers guidelines do not address vapor phase oil.

Author designs, manufactures and tests their own coalescing elements and oil separation equipment and has been supplying these for over 25 years with hundreds of finite oil removal systems in operation all over the world in critical open cycle compression systems. Finite oil removal systems with oil carry-over as low as 1.0 ppbw in both aerosol and vapor phase in low molecular weight gases (He and H_2) is the unique strength and expertise behind this technical paper.

3.0 AUTO-IGNITION OF OIL

Air and fuel gas is mixed and in dry, low NOx (DLN) Combustor with temperature reaching 750 °F. Auto-ignition point of oil is around 480°F. Oil in aerosol form must be avoided in the combustor. Many gas turbine manufacturers specify oil mist size of 0.3 micron or less to protect the hot spots damage on the gas turbine blades. We have gone above and beyond turbine manufacturers' stipulations by eliminating all mist form oil and reducing the vapor phase oil to 5 ppbw or less.

Finite oil separation technology proposed in this paper completely eliminates hot spot problem on turbine blades and further safe-guards against back-fire.

4.0 OIL FILTRATION

The author designs, manufactures and tests their own oil filtration elements, transfer valves and filter assemblies. There has been supplied critical products for over 25 years, and similar to coalescing elements, hundreds of filtration systems are in operation all over the globe.

We recommend oil filtration designed to filter all oil down to 5 microns absolute. This enhances the bearing life by 100%.

5.0 GAS QUALITY

Gas quality after the "FGB" compressor is far superior than the pipeline quality or boosted by any other types of compressors.

6.0 SUMMARY

Proposed Fuel Gas Boosting compressors can contribute to increase in reliability and efficiency of the power plant. Unique characteristics of our “FGB” Compressors are highlighted below:

- Quick Start (2 – 3 minutes)
- Precise Pressure Control
- Step-Less Capacity Control
- High Efficiency
- Power Reduction Proportional to Pipeline Pressure Increase and Gas Turbine Load Changes
- No Surging
- No External Gear
- No Seal Gas Unit
- No Nitrogen Required
- No Passing of Critical Speed
- Limited Interlocks
- No influence due to Mol. Wt. Change
- Can Operate at Pipeline Pressure
- Oil Free Compressed Gas 1-5 ppbw Carry-over in Vapor Phase
- Zero Oil Mist
- Guaranteed No Hot-spot Problems on Turbine Blades
- Guaranteed No Back-fire
- Gas Quality Better than Pipeline or Downstream of Any Other Type of Compressor
- Precise Compressor Discharge Pressure & Temperature Control
- Upon Turbine Trip Compressor in 1-3 Seconds Goes into the Re-cycle Mode and Within 30-45 Seconds Compressor Unloads to Least Power Consumption While Being in the Stand-alone Mode of Operation
- Withstands all Suction Pressure Fluctuations
- No Impact due to Dirty Gas

7.0 CONCLUSION

We strongly recommend that the state of the art oil flooded screw compressors designed and built by with criteria outlined above are the best suited compressor systems for fuel gas boosting. Reciprocating or centrifugal compressors must be avoided in these applications because these compressors will compromise the power plant operation and will lead to extended periods of shutdowns.

8.0 COMPARISON OF COMPRESSORS AS FUEL GAS BOOSTERS

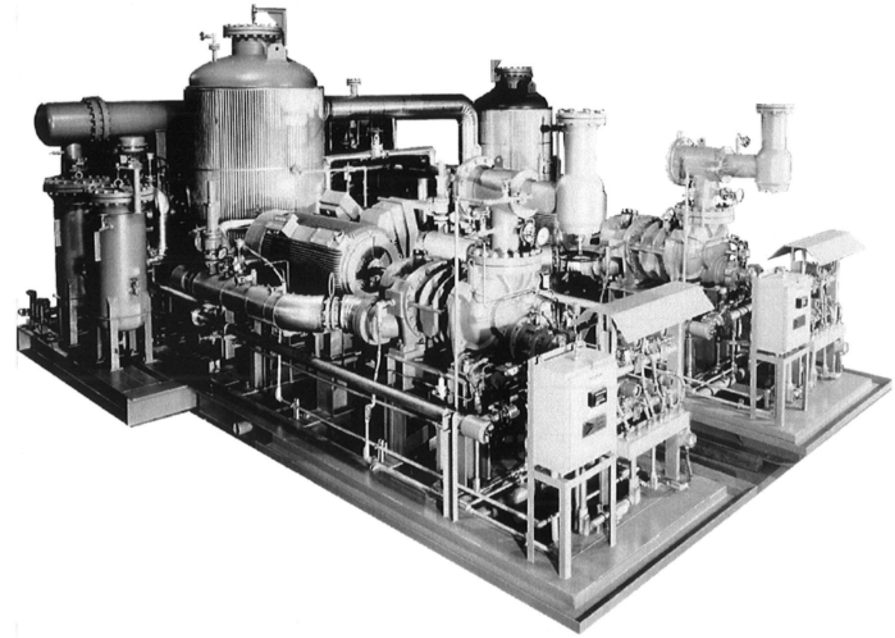
ITEM	OIL-FLOODED SCREW	RECIPROCATING	CENTRIFUGAL
MARKET SHARE	Over 90% Share for Applications up to 500 psig. Over 75% share for Application up to 870 psig.	2 – 5 % Share (For High Pressure Applications)	2 – 5 % Share (For Low Pressure Ratio and High Capacity Applications)
COMPRESSOR DISCHARGE PRESSURE CONTROL (INLET PRESSURE TO GAS TURBINE) (CONSTANT INLET GAS PRESSURE IS MANDATORY FOR PROPER TURBINE OPERATION.	Discharge Pressure of the Compressor is controllable to ±1 psi by Combination of Slide valve and Spillback. No Additional Volume Tanks Required Due to the Presence of Bulk and Coalescing Oil Separators	Discharge Volume Tank Required to Control Discharge Pressure of the Compressor	Discharge Volume Tank Required to Control Discharge Pressure of the Compressor
LOAD SHEDDING OF GAS TURBINE (UNLOADING OF GAS TURBINE)	No Surge, Discharge Pressure is Constant While Compressor Slide Valve Unloads the Compressor	No Surge, But Pulsations Must be Absorbed	Surging Protection Required
FLUCTUATIONS IN SUCTION PRESSURE	Slide Valve can Control Capacity with Energy Saving in Case of Higher Suction Pressure	Suction Pressure Must be Regulated to Avoid Excessive Energy Consumption and Prevent Motor Overload. Step Capacity Control Can Save Some Energy.	Surge Control Required. No Energy Saving.
PRINCIPLE OF COMPRESSION	Rotary Positive Displacement Type	Reciprocating Positive Displacement Type	Rotary Non-Positive Displacement (Aerodynamic Type)
PERFORMANCE	Good	Better	Large Type: Good Small Type: Poor
CAPACITY CONTROL	100 – 15% Continuous	100-75-50-25% in Steps	100-70% Continuous
COMPRESSION RATIOS	8 - 10	2.5 – 3.0	1.1 – 1.5 (Low MW) 1.5 – 3.0 (High MW)

PULSATION	None	Considerable	None
OIL CARRY-OVER IN DELIVERY GAS	0.1 PPMW with Two Separators in Series 1 – 5 PPBW with Charcoal Adsorber	Oil Injection Type ….More than 100 PPMW None Lube Type …None	Low Mole Wt. (2-20) …1.1 – 1.5 High Mole Wt. (20-50) …1.5 – 3.0
SLIDING PART	None. Thus No Change in Performance	Piston, Piston Ring, Rod, Cylinder, Rod Packing, Etc. Which Work to Cause Performance Drop	None. Thus No Change in Performance
WORKING VALVE	None. Thus no trouble and No Change in Performance	Suction and Discharge Valve. Thus, Trouble in Valve Plate and Its Spring	None. Thus, No Trouble and No Change in Performance
VIBRATION	Almost None Because of Rotary Type	Vibration Remains Due to Unbalanced Load	Almost None Because of Rotary Type
SURGING	None	None	Found
FOOT PRINT	Compact & Light Weight	Larger in Size and Heavier	Compact & Light Weight
EFFECTS OF DIRTY GAS	No Effect on Performance	Increase in Trouble of Valves and Associated Parts, Increase in Wear of Piston Ring, Cylinder Inner Surface, Rod and Rod Packing Cleaning Required for Valve Passage Continuous Running Hours Shortened Excessively	Foreign Substances Stay on Impeller, Causing Adverse Effects Upon Performance and Vibration Foreign Substances stay on Diffuser, Causing a Drop in Performance Cleaning Required
CRITICAL SPEED	Higher than Normal Speed and No Problem Solved	Higher Than Normal Speed and No Problem Involved	Critical Speed Lower Than Normal Speed and Care Must be Taken in Starting and Stopping
OPERATION	Very Easy. Easy for Remote Automatic Operation	Skill Required to Find Trouble in Valve and Associated Fittings	Care Must be Taken About Surging Point

CONTINUOUS OPERATION	One-Year Continuous Operation Possible as Minimum, and Two of Three-Year Continuous Operation in Some Cases	One-Year Continuous Operation Not Permitted and Stand-By Set Necessary	One Year Continuous Operation Possible as Minimum
MAINTENANCE	Minimum Wear Parts Overhaul Required Once in One or Two Years Low Part Weight	Many Wear Parts High Frequency of Removal of Valve Plate, etc High Part Weight Control Necessary for Oil Drain from Drain Separator High Cost of Inside Lubrication Oil Which is High Quality Cylinder Oil	Few Wear Parts High Part Weight
ELECTRIC MOTOR	High Speed Induction Type Motor Giving Good Power Factor and Efficiency Small GD^2 and Small Size Permitted Small Starting Torque	Low Speed Induction Type Giving Poor Power Factor and Efficiency Large GD^2 and Large Size Required Large Starting Torque	High Speed Induction Type Giving Good Power Factor and Efficiency Large GD^2 and Large Size Required Large Starting Torque
FOUNDATION	Simple, Compact Foundation (Suitable for Rotary Type Machine, Dynamically Balanced)	Large, sturdy Foundation Required Because of Unbalanced Forces	Simple Foundation (Suitable for Rotary Type Machine)
RELIABILITY & DURABILITY	Excellent	Less Reliable Because of Sliding Part and Work Valve	Excellent

CHAPTER 3

EXAMPLES OF PROCESS GAS COMPRESSION APPLICATIONS WITH OIL-FLOODED SCREW COMPRESSOR



APPLICATION
PSA Tail Gas

EDTI PROJECT NUMBER
ES - 159

DESIGN CRITERIA

Skid Inlet Pressure, PSIA	19.5
Skid Outlet Pressure,PSIA	101
Skid Inlet Temperature, °F	100
Skid Outlet Temperature, °F	185
Gas	Tail Gas
Flow Rate, MMSCFD	8.4
Molecular Weight (Average)	14.03
K Value (Cp/Cv)	1.3
Acceptable Oil Carryover, PPMw	1.0
Prime Mover,	Electric Motor
Main Power	6600 VAC 3PH 60 Hz
Small Motor Power	440 VAC 3PH 60 Hz
Control Voltage,	120 VAC - 1PH-60 Hz
Electric Area Class	CL.1, GR. B,C,D DIV. II
Panel Class	NEMA - 4
Oil Cooling By,	Water @ 90 °F

COMPRESSOR PERFORMANCE

Model	320 MUD-M
Load Control, %	100-0 (Stepless)
Displacement, CFM	2795.8
Suction Pressure, PSIA	19.5
Discharge Pressure, PSIA	106
Suction Temperature, °F	100
Discharge Temperature, °F	185
Gas Flow, MMSCFD	4.2
Shaft Power, BHP	655
Oil Temperature, °F	130
Oil Heat Rejection, BTUH x 1000	2518

COMPRESSOR TYPE
Rotary Screw

DRIVERS
Two ,800 HP, 3560 RPM, 6600 VAC, TEFC Motors

MECHANICAL CONFIGURATION
Two 2795.8 CFM oil-flooded rotary screw compressors are driven by 3560 RPM, 800 HP electric motors thereby compressing 8.4 MMSCFD of PSA tail gas from 19.5 to 106.0 PSIA. Primary oil separator is a 60" diameter vessel whereas the secondary oil separator is a 48" diameter vessel. Both separators are coalescing type with oil carryover limited to 1.0 PPMw. Gas at the skid limit is delivered at 101.0 PSIA and 185.0 °F. Oil management system consists of dual oil pumps, dual oil coolers and dual oil filters. Control oil for the actuation of slide valve is common with the main lube oil system. Capacity variation is 100 % with slide valve and spill back arrangement. Vibration monitoring is integrated with PSA control system with velocity and axial displacements displayed on remote "CRT" and interfaced with DCS.

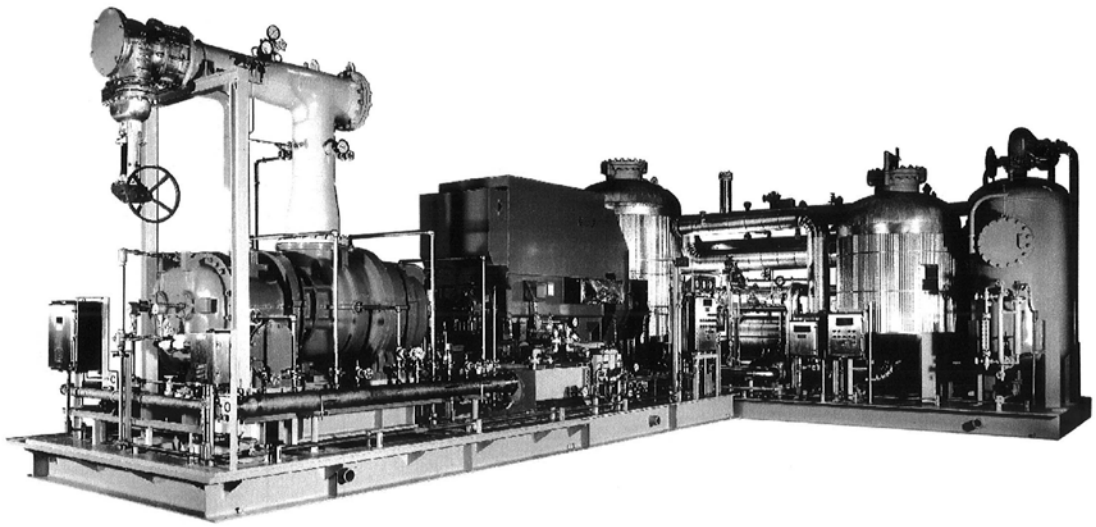
CONTROL SYSTEM CONFIGURATION
PLC based control system with Rosemount smart transmitters and NEMA-4X local control panel. Control system is fully integrated into the PSA control system.

NUMBER OF SKIDS
03

SHIPPING DATE
August, 1996

START UP DATE
February, 1997

PROJECT LOCATION
Ulsan, Korea (TKI Co. Ltd.)



APPLICATION
PSA Tail Gas Compressor

EDTI PROJECT NUMBER
ES-218

DESIGN CRITERIA

Skid Inlet Pressure, PSIA	15.7
Skid Outlet Pressure,PSIA	86
Skid Inlet Temperature, °F	90
Skid Outlet Temperature, °F	113
Gas	PSA Tail Gas
Flow Rate, MMSCFD	10.7
Molecular Weight (average)	19.9
K Value (Cp/Cv)	1.30
Acceptable Oil Carryover, PPMw	1.0
Prime Mover	Electric Motor
Main Voltage	13200 VAC 3PH 60Hz
Small Motors/Heaters Voltage	460 VAC 3PH 60 Hz
Control Voltage	120VAC-1PH-60 Hz
Electric Area Class	CL1, GR B, DIV II
Panel Class	NEMA - 4 X
Oil Cooling By,	Water @ 90 °F

COMPRESSOR PERFORMANCE

Model	KS 50 LV
Displacement, CFM	8700@1770 RPM
Load Control, %	100 - 0 (Stepless)
Suction Pressure, PSIA	15.7
Discharge Pressure, PSIA	92
Suction Temperature, °F	90
Discharge Temperature, °F	192.9
Gas Flow, MMSCFD	10.7
Shaft Power, BHP	1639
Oil Temperature, °F	113
Oil Heat Rejection, BTUH x 1000	2835.2

COMPRESSOR TYPE
Screw, Oil Injected Type

DRIVER
2000 HP, 1770 RPM, 13200 VAC, TEFC Motor

MECHANICAL CONFIGURATION
8700 CFM oil-flooded rotary screw compressor is driven by a 1770 RPM, 2000 HP electric motor to compress 10.7 MMSCFD of PSA tail gas from 15.7 PSIA to 92.0 PSIA. Primary oil separator is a 72" diameter vessel, whereas the secondary oil separator is a 60 " diameter vessel. Both separators are coalescing type with oil carryover limited to 1.0 PPMw. Gas at the skid limit is delivered at 86.0 PSIA and 113 °F. Gas cooling is accomplished in a shell and tube exchanger with water at 90 °F. Oil management system consists of dual oil pumps, dual oil coolers and dual oil filters. Control oil system for slide valve is independent of the main lube oil system. Capacity variation is 100% with slide valve and spill back arrangement. Vibration monitoring is in local Bently Nevada monitor with axial displacement, velocity and temperature measurements.

CONTROL SYSTEM CONFIGURATION
PLC based control system with Rosemount smart transmitters and NEMA-4X local control panel. Control system is fully integrated into the PSA control system.

NUMBER OF SKIDS
02

SHIPPING DATE
August, 1999

START UP DATE
November, 2000

PROJECT LOCATION
Cedereyta, Mexico (PEMEX, Refinery)

CHAPTER 4

EXAMPLES OF PROCESS REFRIGERATION APPLICATIONS

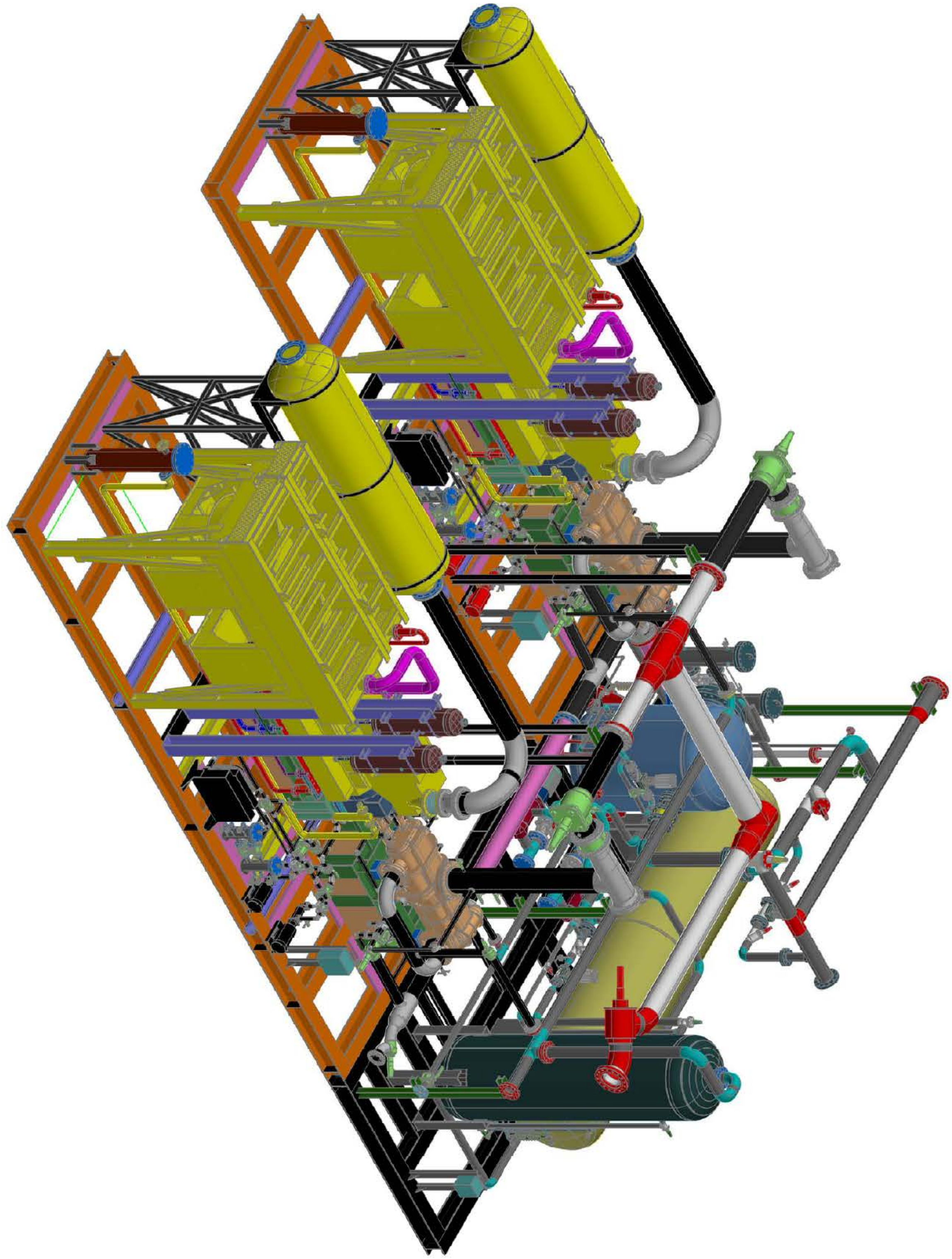
Please see attached drawings and

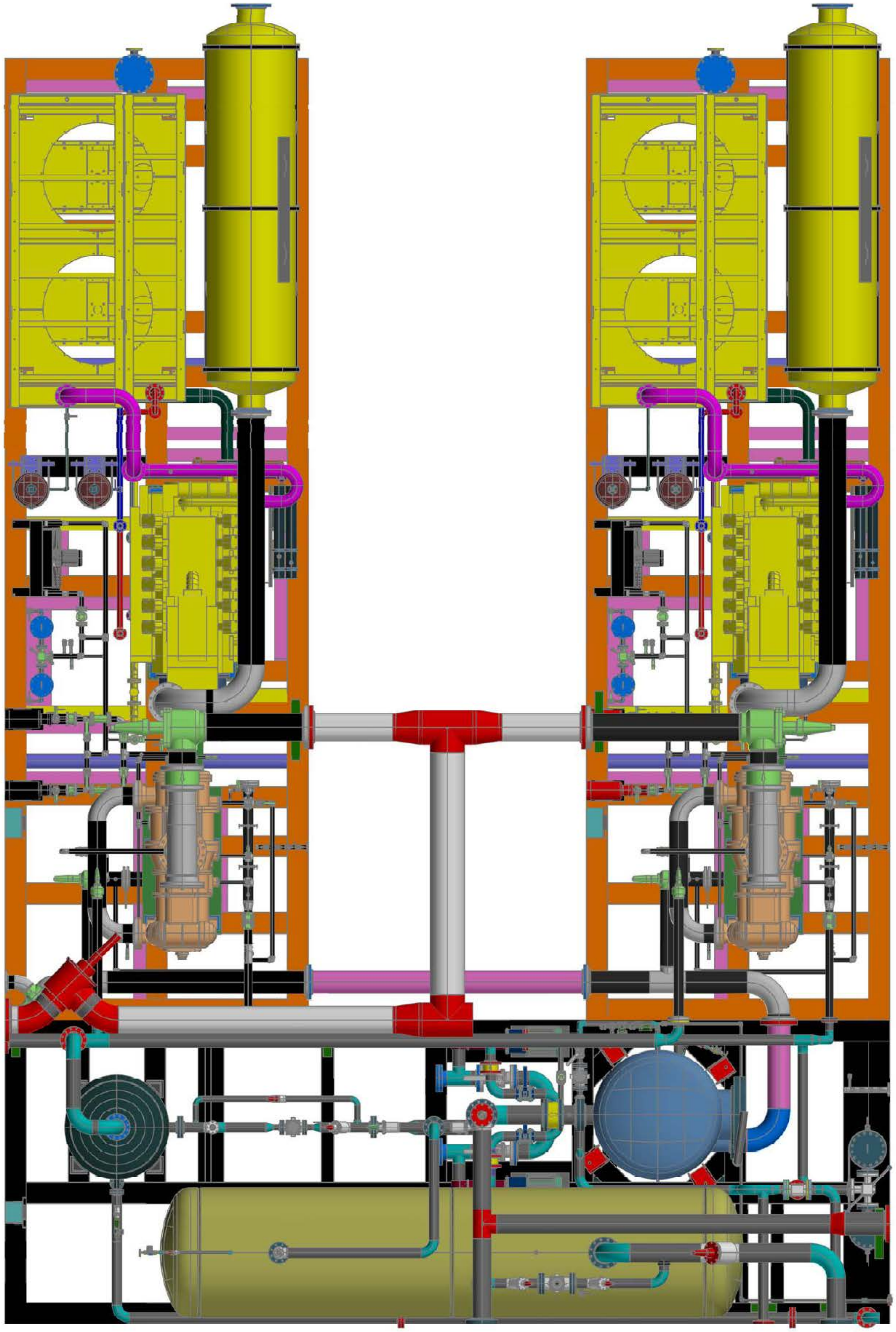
Pictures of Gas Engine Driven Screw Compressors

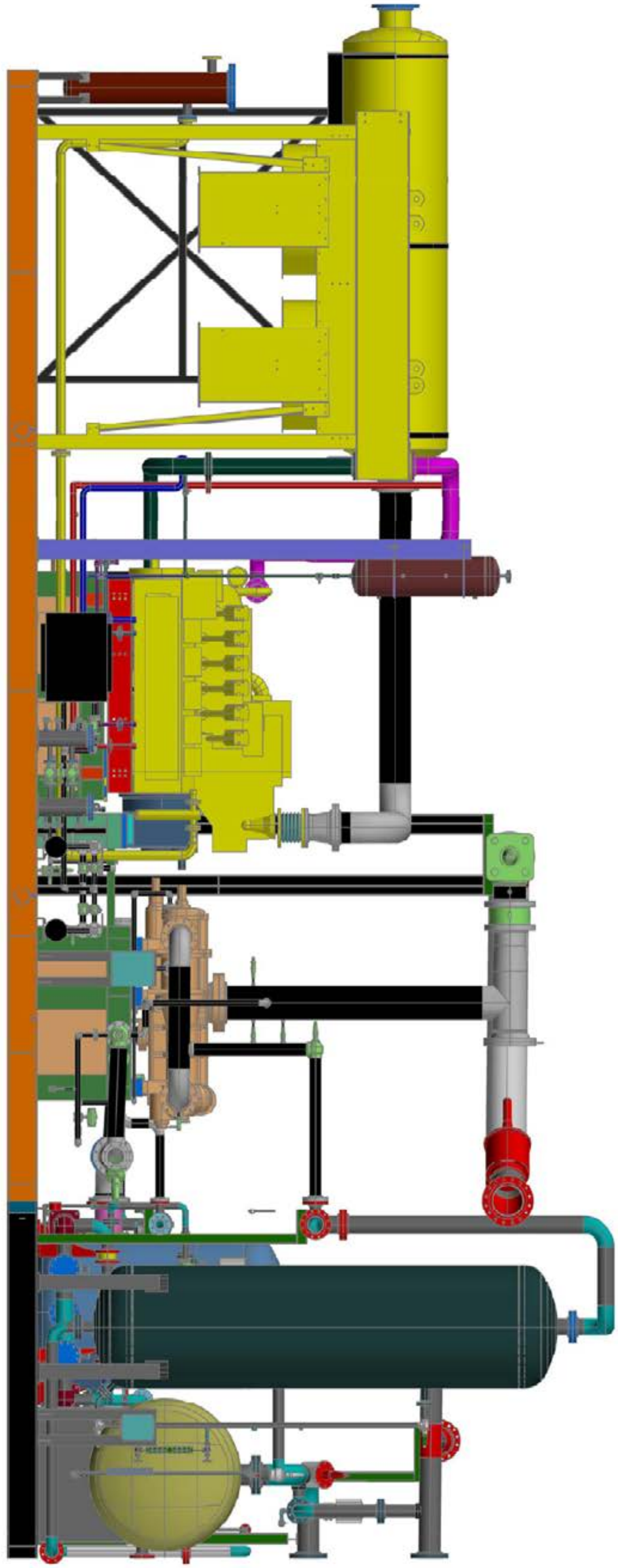
produced by IES. This particular system has been in operation

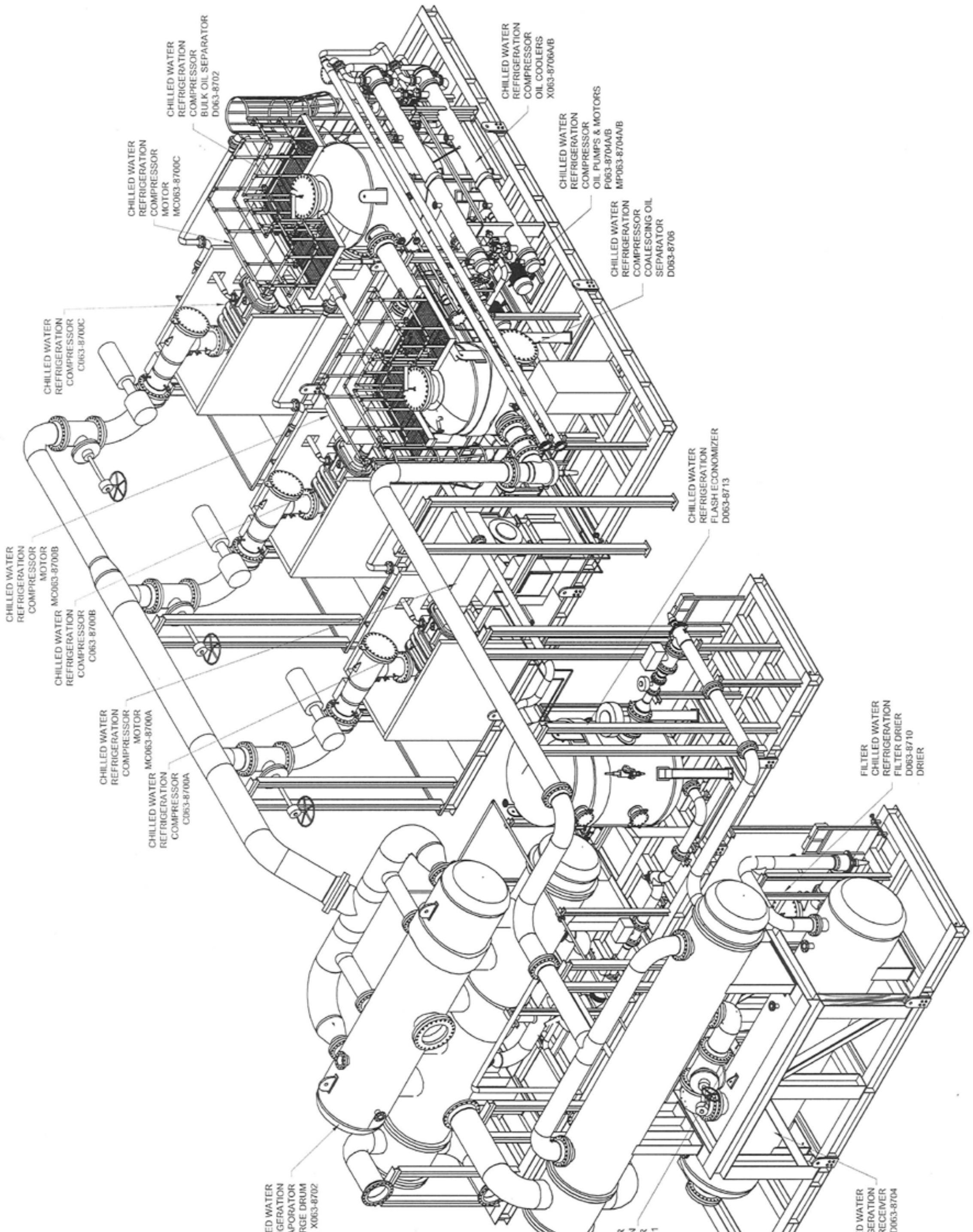
for the last 32 months in North Dakota (USA) in a

45 MMSCFD NGL Recovery System.



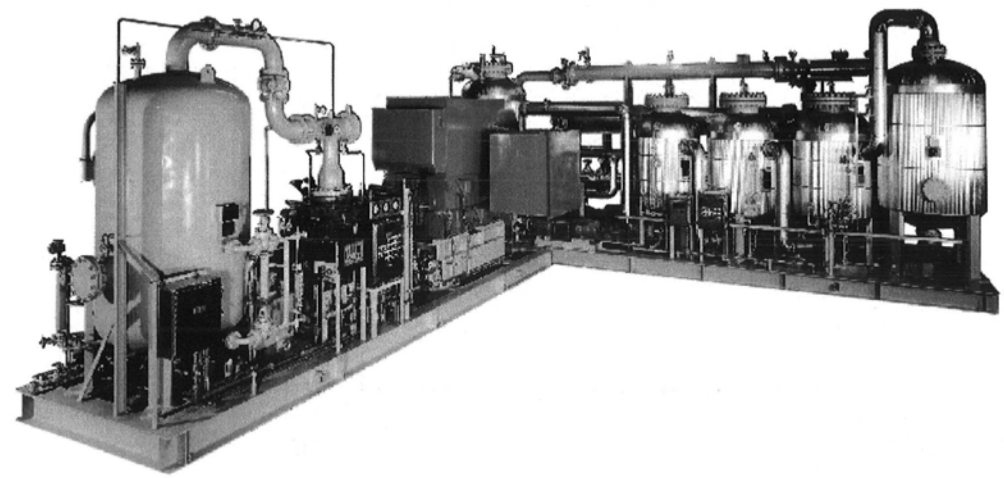





CHILLED WATER REFRIGERATION COMPRESSOR BULK OIL SEPARATOR D063-8702
CHILLED WATER REFRIGERATION COMPRESSOR MOTOR MC063-8700C
CHILLED WATER REFRIGERATION COMPRESSOR C063-8700C
CHILLED WATER REFRIGERATION COMPRESSOR OIL COOLERS X063-8706A/B
CHILLED WATER REFRIGERATION COMPRESSOR OIL PUMPS & MOTORS P063-8704A/B MP063-8704A/B
CHILLED WATER REFRIGERATION COMPRESSOR COALESCING OIL SEPARATOR D063-8706
CHILLED WATER REFRIGERATION COMPRESSOR MOTOR MC063-8700B
CHILLED WATER REFRIGERATION COMPRESSOR C063-8700B
CHILLED WATER REFRIGERATION FLASH ECONOMIZER D063-8713
CHILLED WATER REFRIGERATION COMPRESSOR MOTOR MC063-8700A
CHILLED WATER REFRIGERATION COMPRESSOR C063-8700A
FILTER CHILLED WATER REFRIGERATION FILTER DRIER D063-8710 DRIER
CHILLED WATER REFRIGERATION EVAPORATOR WITH SURGE DRUM X063-8702
CHILLED WATER REFRIGERATION R-134A CONDENSER X063-8700
CHILLED WATER REFRIGERATION INTELI-PURGER D063-8711
R-134A RECEIVCHILLED WATER REFRIGERATION R-134A RECEIVER D063-8704

CHAPTER 5

EXAMPLES OF PROCESS GAS COMPRESSOR WITH OIL-FLOODED SCREW COMPRESSOR AND FINITE OIL REMOVAL (1.0 PPBW)



APPLICATION:
PSA Feed Gas Compressor

EDTI PROJECT NUMBER
ES-216A

DESIGN CRITERIA

Skid Inlet Pressure, PSIA	100
Skid Outlet Pressure,PSIA	309
Skid Inlet Temperature, °F	105
Skid Outlet Temperature, °F	100
Gas	PSA Feed Gas
Flow Rate, MMSCFD	19.74
Molecular Weight (Average)	9.36
K Value (Cp/Cv)	1.36
Acceptable Oil Carryover, PPBw	50
Prime Mover	Electric Motor
Main Voltage	13,200 VAC 3PH 60Hz
Small Motors/Heaters Voltage	13,200 VAC-3PH-60 Hz
Control Voltage	120 VAC-1PH-60 Hz
Electric Area Class	Cl.1, GR. B, C, D Div. 2
Panel Class	NEMA - 4X
Oil Cooling By,	Water @ 90°F/106.5 Psia

COMPRESSOR PERFORMANCE

Model	320 MUD-L
Displacement, CFM	2795.8
Load Control, %	100-10 (Stepless)
Suction Pressure, PSIA	98
Discharge Pressure, PSIA	320
Suction Temperature, °F	105
Discharge Temperature, °F	213.2
Gas Flow, MMSCFD	19.74
Shaft Power, BHP	2055
Oil Temperature, °F	120
Oil Heat Rejection, BTUH x 1000	2853

COMPRESSOR TYPE
Screw, Oil-Injected Type

DRIVER
2250 HP, 3560 RPM, 13,200 VAC, TEFC Motor

MECHANICAL CONFIGURATION
Single 2750 CFM oil-flooded rotary screw compressor is driven by a 3550 RPM, 2250 HP electric motor to compress 19.74 MMSCFD of hydrogen rich mixed hydrocarbon feed gas stream from 98 PSIA to 320 PSIA. Primary oil separator is followed by three coalescing oil separators and a charcoal adsorber to limit the oil carry over in the compressed gas stream to 10-150 PPBw. Gas at the skid limit is delivered at 309 PSIA and 100 ° F. Gas cooling is accomplished in a shell and tube exchanger with water at 90 ° F. Oil management system consists of dual oil pumps, dual oil coolers and dual oil filters. Motor lubrication is forced type with an independent oil system. Capacity control is 100-10 % with internal slide valve. Vibration monitoring is in local Bently Nevada Monitor with axial displacement, velocity and temperature measurements.

CONTROL SYSTEM CONFIGURATION
PLC based control system with Rosemount smart transmitters and NEMA-4X local control. Control system is fully integrated into the PSA control system.

NUMBER OF SKIDS
02

SHIPPING DATE
August, 1999

START UP DATE:
November, 2000

PROJECT LOCATION
Cadereyta, Mexico (PEMEX, Refinery)

CHAPTER 6

MTBF AND MTTR ANALYSIS FOR AN OIL-INJECTED SCREW COMPESSOR AND AN OIL-FLOODED SCREW COMPRESSOR SYSTEM

The following statistical analysis proves 99.5% availability of “FGB” Compressors.

MTBF AND MTTR ANALYSIS FOR AN OIL-INJECTED SCRW COMPRESSOR

1.0 COMMENTS ON RELIABILITY AND MAINTENANCE

Screw compressors are one of the simplest types of compressors to operate. The two rotors are the only moving parts. Bearings are very moderately loaded. The shaft seal is mechanical with forced lubrication and pressure balanced. These important features lend a high degree of reliability to screw compressors in an age when maintenance costs are skyrocketing.

Screw compressors have now proven successful in a number of compression processes. The ability of the screw compressor (single stage) to handle high pressure ratios (10:1) makes it an obvious choice in low temperature or high pressure ratio applications previously considered reserved for two stage systems. At still higher ratios, a two stage screw compressor system is certainly a better choice than a conventional two stage system. In a compound screw system, there is no need for an intercooler.

The continuous capacity regulation in screw compressors is a very desirable feature in applications where widely varying loads are very frequently encountered. In the chemical industry, shutdowns can become very expensive; here high reliability, ruggedness, and low maintenance make the screw compressor a better choice. In addition to the already mentioned features, the ability of the screw compressor to tolerate some liquid slugging and its vibration free operation can be added to the list of advantages in using a screw compressor.

As a general rule, a single oil flooded screw compressor provides the same degree of reliability and availability as two reciprocating compressors with one in operation and the other standby.

2.0 SUMMARY OF MTBF AND MTTR DATA AS SHOWN ON THE FOLLOWING PAGES

Mean Time Between Failures For a Screw Compressor	5,725 Hours
Mean Time To Repair a Screw Compressor	4.74 Hours
Mean Time Between Failures For a Screw Compressor System	1,073 Hours
Mean Time To Repair System Failures	2.96 Hours
Number of system Failures Requiring 8 hours To Repair per 10^6 Hours	62.44 Hours
Mean time between failures requiring 8 hours to repair	16,015 Hours
Down Time Per 10^6 Hours for Repairs Requiring 8 Hours	500 Hours

3.0 "MTBF" AND "MTTR" DATA FOR AN OIL INJECTED SCREW COMPRESSOR

SCREW COMPRESSOR MAIN COMPONENTS	DESIGN LIFE (HRS)	FAILURE RATE PER 10^6 HRS λ	ESTIMATED REPAIR TIME T (HRS)	QTY N	TOTAL FAILURE RATE PER 10^6 HRS $N\lambda$	DOWN TIME PER10^6 HRS FOR REPAIR $DT = TN\lambda$
THRUST BEARINGS	50,000	20	6	2	40	240
RADIAL BEARINGS	100,000	10	16	2	20	320
SHAFT SEALS	15,000	67	2	1	67	134
ROTORS	200,000	4	2	2	8	16
LOAD/UNLOAD MECH.	100,000	10	4	2	20	80
GEAR COUPLING	50,000	20	2	1	<u>20</u> $\Sigma\lambda = 175$	<u>40</u> TOTAL = 830 HRS

Compressor Failure Rate per 10^6 Hours = $\Sigma N\lambda = 175$

Compressor Failure Rate Per Year = 1.53 (One Year = 8,760 Hours)= $\dfrac{175 \times 8760}{10^6} = 1.53$

Mean Time Between Failure (MTBF) = $\dfrac{8{,}760}{1.53}$ = 5,725 Hours

Mean Time to Repair (MTTR) = $\dfrac{DT}{\Sigma N\lambda} = \dfrac{830}{175} =$ 4.74 Hours

4.0 “MTBF” AND “MTTR” DATA FOR AN OIL INJECTED SCREW COMPRESSOR SYSTEM

SCREW COMPRESSOR MAIN COMPONENTS	FAILURE RATE PER 10^6 HRS λ	ESTIMATED REPAIR TIME T (HRS)	QTY N	TOTAL FAILURE RATE PER 10^6 HRS $N\lambda$	DOWN TIME PER 10^6 HRS FOR REPAIR $DT = TN\lambda$
COMPRESSOR	175	4.74	1	175	830
BULK OIL SEPARATOR	57	4.0	2	114	456
OIL PUMP(S) & MOTOR(S)	114	3.0	1	114	342
OIL COOLER(S)	7.6	4.0	1	7.6	30.4
AFTER COOLERS(S)	7.6	4.0	1	7.6	30.4
STOP.CHECK VALVE(S)	6	1.0	3	18	18
HAND OPERATED VALVES(S)	6	1.0	3	18	18
TEMP. CONTROL VALVE(S)	11	3.0	1	11	33
COALESCERS(S)	57	4.0	1	57	228
PRESSURE SAFETY VALVE(S)	6	4.0	3	18	72
FILTER(S)	--	--	--	--	--
TEMP. TRANSMITTERS	11.4	4	3	34.2	34.2
PRESSURE TRANSMITTERS	11.4	1.0	4	45.6	456
PRESSURE REGULATORS(S)	23	3.0	2	46	138
SOLENOID VALVE(S)	57	2.0	4	228	456
PRESSURE TRANSMITTER(S)	--	--	--	--	--
LOCAL CONTROL PANEL	38	1.0	1	<u>38</u>	38
				$\Sigma\lambda = 932$	TOTAL = 2762 (HRS)

Complete System Failure Rate per 10^6 Hours = $\Sigma N\lambda = 932$

Complete System Failure Rate per Year = 8.16 (One Year = 8,760 Hrs)

Mean Time Between Failure (MTBF) = $\frac{8{,}760}{8.16}$ = 1,073 Hours

Mean Time to Repair (MTTR) = $\frac{DT}{\Sigma\lambda} = \frac{2{,}762}{932} =$ 2.96 Hours

Probability of Repair Requiring 8 Hours = Error $^{-T/MTTR}$ = Error $^{-8/296}$ = Error $^{-2.7}$ = 0.67

MTBF (Requiring 8 Hours to Repair) = 1073/.067 = 16,015 Hours

Number of system Failures Requiring 8 Hours to Repair per 10^6 Hours = $\frac{10^6}{16{,}015}$ = 62.44

Down Time per 10^6 Hours for Repairs Requiring 8 Hours = 62.44 x 8 = 500 Hours

5.0 NOTATIONS

Filter	Dual, therefore no failure.
Pressure Transmitter	Even though there is a problem, can make the compressor run on manual.
Oil Cooler/Gas Cooler	Although every one or two years the tubes need to be cleaned, it is classified under routine maintenance. The tubes should have a lifetime minimum of 15 years.
Transmitters	The need to be simulated during maintenance period. In general, it can last a very long time, perhaps 10-15 years.
Primary & Secondary Oil Separator Element	Minimum life is two to three years.
Control Valves	Use 10 years life.
Pressure Safety Valve	If all shut-downs are operating properly, it should never pop. Use 20 years life.