

BASICS OF OPERATION AND APPLICATION OF OIL FLOODED ROTARY SCREW COMPRESSORS

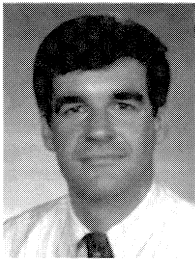
by

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ABSTRACT

Rotary screw compressors are widely used today in chemical and petrochemical refining, gas processing, and in the oil patch. Common applications include refrigeration using hydrocarbons, hydrofluorocarbons (HFCs), and ammonia refrigerants, vapor recover, gas gathering, and compression of fuel gas, natural gas, landfill gas, tail gas, CO₂, and helium. Simple in concept, the screw geometry is sufficiently difficult to visualize that many people using screws today have only a vague idea how they actually work. An understanding of the basics of their operation will help in applying them correctly, avoiding nuisance problems in operation, and achieving the best overall system designs.

CONSTRUCTION

A typical oil flooded twin screw compressor consists of male and female rotors mounted on bearings to fix their position in a rotor housing that holds the rotors in closely toleranced intersecting cylindrical bores (Figure 1). The rotor's basic shape is a screw thread, with varying numbers of lobes on the male and female rotors. The driving device is generally connected to the male rotor with the male driving the female through an oil film. In low and medium pressure gas compression, four or five lobed male rotors generally drive six or seven lobe female rotors to give a female rotor speed that is somewhat less than the male speed. Some designs connect the drive to the female rotor in order to produce higher rotor speeds, thus increasing displacement. However, this increases loading on the rotors in the area of torque transfer and can reduce rotor life. For this reason, female drive is not used on larger compressors.

Screw compressors are available in a wide variety of materials. For most refrigeration applications, the housings are made of grey cast iron, though nodular or ductile iron is available. A variety of cast steel housings are also available including LCB, LC-2, or LC-3 for improved ductility at lower application temperatures. Rotor

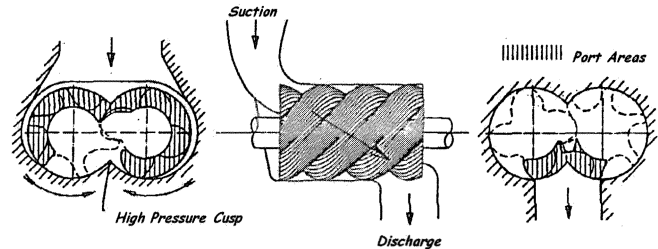


Figure 1. Screw Compressor Basic Geometry.

material is normally mild carbon steel. Heat treatment of the rotors is normally not required. Bearings can be ball and roller type in antifriction bearing equipped compressors, or conventional sleeve bearing materials in journal bearing equipped compressors. Neoprene or Buna-N O-rings and seals are common, with a wide variety of optional elastomers for special applications. Shaft seal designs are of the conventional mechanical types, with single, double, and wet and dry tandem arrangements usually available.

OIL INJECTION

Nonoil injected or dry screw compressors have been used on a variety of gas compression applications since the 1930s. The proliferation of engineered synthetic oils and coalescing oil separators in the last 20 years has caused a significant shift away from dry screws to the use of oil injection on many applications. Most common gas compression screws applied today use oil injection into the compression area for lubrication, sealing of leakage paths, and cooling. Injected oil quantities are approximately 10 to 20 gal/min per 100 hp. The use of such large oil quantities transfers most of the heat of compression to the oil and allows discharge temperatures to be very low even at high compression ratios. Running single stage at 20:1 compression ratio on methane would yield 650°F discharge temperatures in nonflooded types of compressors. With oil flooding, the screw discharge temperature does not exceed 230°F. Running screws at 20:1 or even higher, single stage, though not energy efficient compared with two stage systems, will not harm the compressor. Many such systems are running today. Oil injection quantity and oil temperature can be varied in order to maintain the discharge temperature above the dew point of water or hydrocarbons in the gas stream. This is essential to avoid dilution with hydrocarbons or accumulation of water in the high side oil sump on gas streams saturated with water.

FUNDAMENTALS OF OPERATION

A screw compressor is best described as a positive displacement volume reduction device. Its action is analogous to a reciprocating compressor (recip) more than any other common compressor type. It is helpful to refer to the equivalent recip process to visualize how compression proceeds in a screw. Gas is compressed by pure rotary motion of the two intermeshing helical rotors. Gas travels around the outside of the rotors, starting at the top and traveling to the bottom, while it is transferred axially from the suction end to the discharge end of the rotor area.

Suction Process

Suction gas is drawn into the compressor to fill the void where the male rotor rotates out of the female flute on the suction end of the compressor. Suction charge fills the entire volume of each screw thread as the unmeshing thread proceeds down the length of the rotor. This is analogous to the suction stroke in a reciprocating compressor as the piston is drawn down the cylinder (Figure 2).

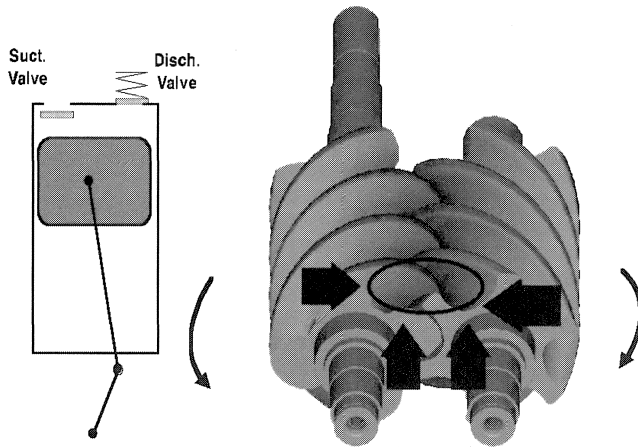


Figure 2. Suction Process.

The suction charge becomes trapped in two helically shaped cylinders formed by the screw threads and the housing as the threads rotate out of the open suction port. The volume trapped in both screw threads over their entire length is defined as the volume at suction (V_s). In the recip analogy, the piston reaches the bottom of the stroke and the suction valve closes, trapping the suction volume (V_s) (Figure 3).

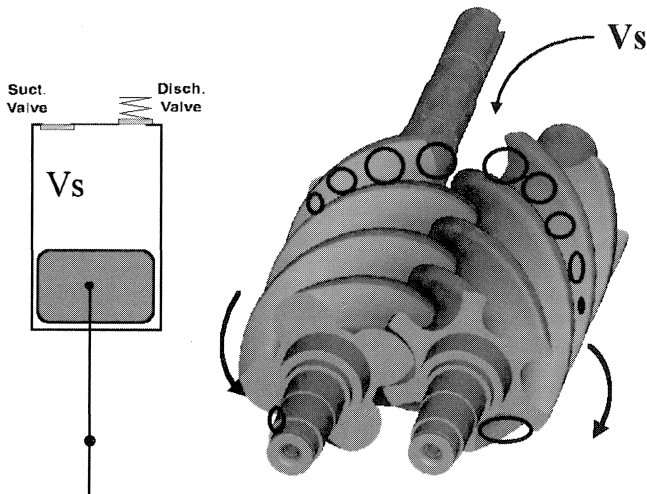


Figure 3. Maximum Suction Volume Trapped.

The displacement per revolution of the recip is defined in terms of suction volume by the bore times the stroke times the number of cylinders. The total displacement of the screw compressor is the volume at suction per thread times the number of lobes on the driving rotor.

Compression

The male rotor lobe will begin to enter the trapped female flute on the bottom of the compressor at the suction end, forming the back edge of the trapped gas pocket. The two separate gas cylinders in each rotor are joined to form a "V" shaped wedge of

gas, with the point of the "V" at the intersection of the threads on the suction end (Figure 4). Further rotation begins to reduce the trapped volume in the "V" and compress the trapped gas. The intersection point of the male lobe in the female flute is like the piston in the recip that is starting up the cylinder and compressing the gas ahead of it (Figure 5).

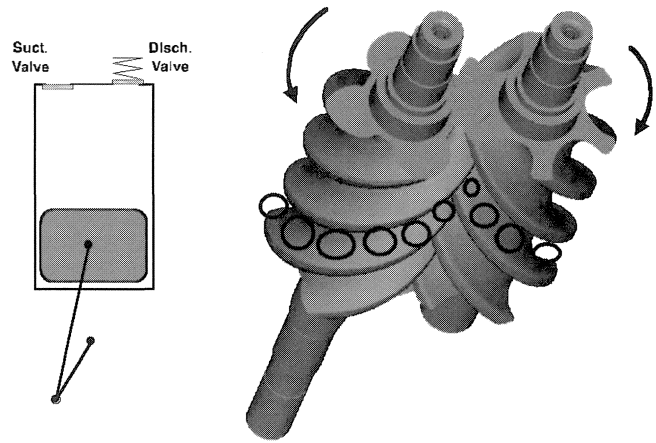


Figure 4. Compression.

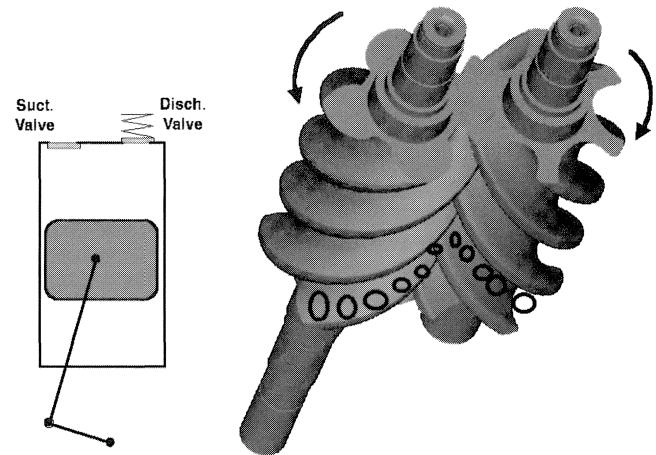


Figure 5. Continuation of Compression.

Discharge Process

In the recip compressor, the discharge process starts when the discharge valve first opens. As the pressure in the cylinder exceeds the pressure above the valve, the valve lifts, allowing the compressed gas to be pushed into the discharge manifold. The screw compressor has no valves. Only the location of the discharge port determines when compression is over (Figure 6). The volume of gas remaining in the "V" shaped trapped pocket at discharge port opening is defined as the volume at discharge (V_d).

A radial discharge port is used on the outlet end of the slide valve and an axial port is used on the discharge end wall. These two ports provide relief of the internal compressed gas and allow it to be pushed into the discharge housing. Positioning of the discharge ports is very important as this controls the amount of internal compression.

In the recip, the discharge process is complete when the piston reaches the top of the compression stroke and the discharge valve closes. The end of the discharge process in the screw occurs as the trapped pocket is filled by the male lobe at the outlet end wall of the compressor (Figure 7). The recip always has a small amount of gas (clearance volume) that is left at the top of the stroke to expand on the next suction stroke, taking up space that could have been

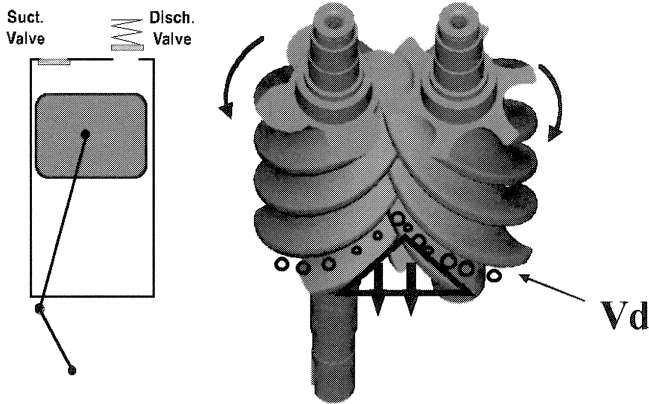


Figure 6. Beginning of Discharge.

used to draw in more suction charge. At the end of the discharge process in the screw, no clearance volume remains. All compressed gas is pushed out the discharge ports. This is a significant factor that helps the screw compressor to be able to run at much higher compression ratios than a recip.

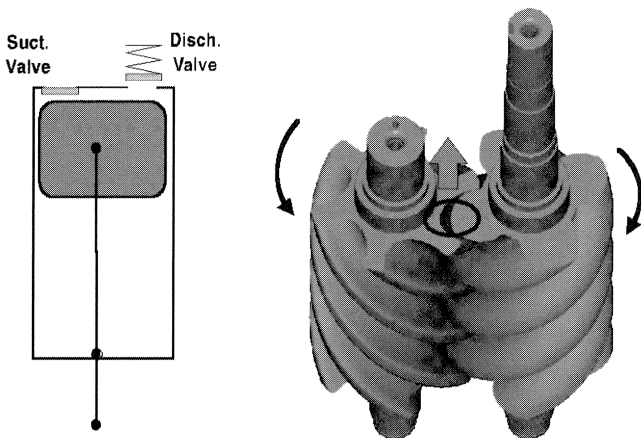


Figure 7. End of Discharge.

Volume Ratio

In a reciprocating compressor, the discharge valves open when the pressure in the cylinder exceeds the pressure in the discharge manifold. Because a screw compressor does not have valves, the location of the discharge port determines the maximum discharge pressure level that will be achieved in the screw threads before the compressed gas is pushed into the discharge pipe.

Volume ratio is a design characteristic of all screw compressors because the compressor is essentially a volume reduction device. The comparison of the volume of trapped gas at suction (V_s) to the volume of trapped gas remaining in the compression chamber when it opens to discharge (V_d), defines the internal volume reduction ratio of the compressor. This volume index, or “ V_i ,” determines the internal pressure ratio of the compressor, and the relationship between them can be approximated as follows:

$$V_i = V_s/V_d \tag{1}$$

where:

- V_i = Volume ratio or index
- V_s = Volume at suction
- V_d = Volume at discharge

$$P_i = V_i^k \tag{2}$$

where:

- P_i = Internal pressure ratio
- k = Specific heat ratio of the gas being compressed

Only the suction pressure and the internal volume ratio determine the internal pressure level in the trapped pocket before opening to the discharge port. However, in all systems, the suction and discharge pressures are determined by the needs of the process, not by the compressor.

If the internal volume ratio of the compressor is too high for a given set of operating conditions, the discharge gas will be kept trapped in the screw threads too long and be raised above the discharge pressure in the piping. This is called over compression and is represented in the pressure-volume curve in Figure 8. In this case, the gas is compressed above discharge curve pressure and when the port opening occurs, the higher pressure gas in the screw thread expands out of the compressor into the discharge line. This takes more energy than if the compression had been stopped sooner when the internal pressure was equal to the system discharge pressure.

Over Compression
P-V Diagram

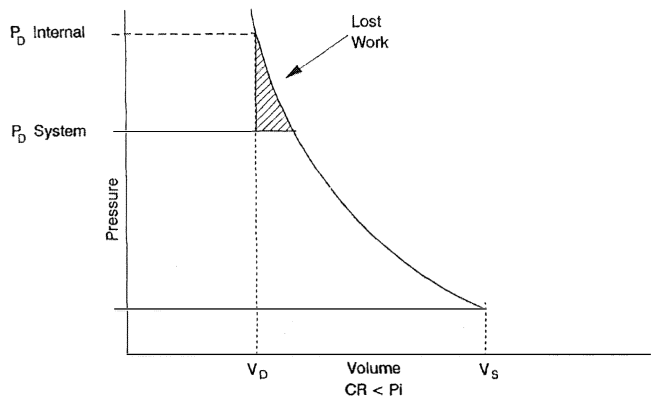


Figure 8. Over Compression P-V Diagram.

When the compressor volume ratio is too low for the system operating pressures, this is called under compression and is represented in Figure 9. In this case, the discharge port opening occurs before the internal pressure in the compressor trapped pocket has reached the system discharge pressure level. The higher pressure gas outside the compressor flows back into the lower pressure pocket, raising the thread pressure immediately to the discharge pressure level. The compressor then has to pump against this higher pressure level, rather than pump against a gradual buildup to discharge pressure level if the volume ratio had been higher, keeping the trapped pocket closed longer.

In both cases, the compressor will still function, and the same volume of gas will be moved, but more power will be required than if the discharge ports are correctly located to match the compressor volume ratio to what the system needs. Automatically variable volume ratio compressors are commonly used in order to optimize discharge port location and minimize compressor power (Pillis, 1983).

The fact that screws have an internal volume ratio and internal compression sets them apart from screw pumps or roots blowers, even though their structure may look similar. A roots blower traps gas and moves it to the discharge, but only raises pressure by pushing against an external resistance. A roots blower always takes more power to compress gas than a screw, because it always runs undercompressed with a built in volume ratio of 1:1.

CAPACITY CONTROL

Capacity control is used in screw compressors to vary the amount of gas drawn into the compressor. This is necessary in

Under Compression P-V Diagram

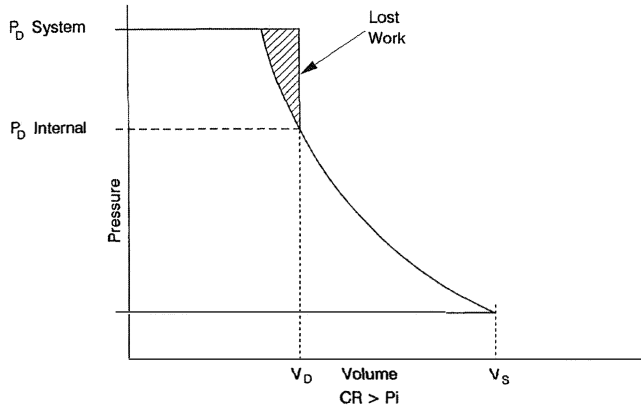


Figure 9. Under Compression P-V Diagram.

order to provide accurate balancing of the compressor capacity to the gas compression needed in the system. Common capacity control methods are:

- Slide valve controlling discharge port.
- Slide valve controlling discharge port and volume ratio.
- Slide valve not controlling discharge port.
- Plug valves.
- Variable speed.

Slide valves controlling the discharge port are a very common type of capacity control device used in screw compressors. They are popular because they can give infinitely adjustable control of capacity, often from 10 percent to 100 percent. This type of slide valve works by opening a recirculation passage in the high pressure cusp, which allows a portion of the trapped gas in the "V" shaped compression chamber to be recirculated back to the suction cavity before it begins compression (Figure 10). This method offers good efficiency at part load for two reasons. First, the recirculated gas only has to overcome a slight pressure drop in order to bypass back to suction, since the recirculation slot opens to the trapped pocket before compression has started, avoiding a precompression loss. Second, as the slide valve moves, the radial discharge port is also being moved. As the trapped volume at suction is decreased, the discharge port opening is also delayed, thus maintaining approximately the same volume ratio at part load as at full load for optimum part load efficiency.

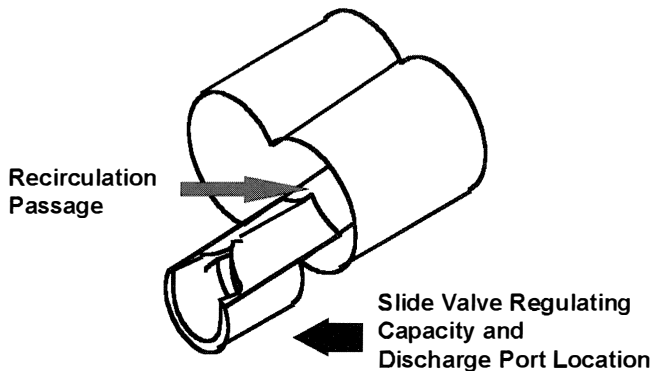


Figure 10. Slide Valve Capacity Control Mechanism.

A compressor designed to control capacity and volume ratio is shown in Figure 11. In this design, a movable slide stop is adjustable in the same bore as the slide valve, and the discharge port position and the recirculation slot position can both be adjusted. This allows an infinite number of adjustable positions for both valves, which provides volume ratio and capacity adjustment from full load to approximately 40 percent load, with continuing capacity adjustment down to 10 percent load. This arrangement offers the potential for improved energy efficiency at full and part load when compared with a conventional slide valve.

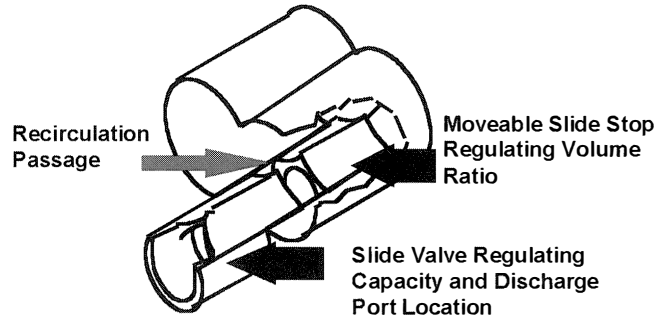


Figure 11. Slide Valve Capacity and Volume Ratio Control Mechanism.

Slide valves that do not control the discharge port come in several varieties. The most common is a round slide valve intersecting with slots in the rotor bore (Figure 12). This type of unloader still gives good reduction of capacity, but not as good a reduction in part load power because it does not maintain the volume ratio during unloading. There can also be some leakage across the slots in the rotor bore, which can hurt performance at all loads. These devices are lower in cost than conventional slide valves and used in some smaller compressors.

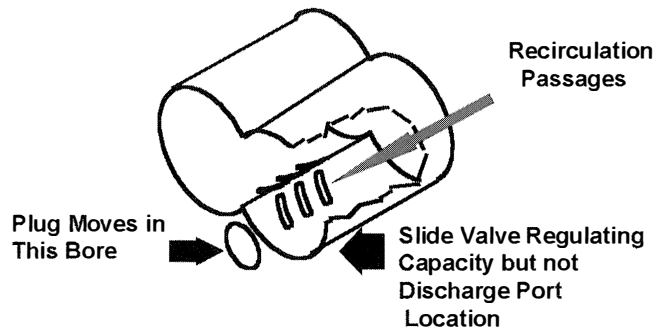


Figure 12. Herringbone Slot Slide Valve Capacity Control Mechanism.

Plug valves are radial or axial devices, which lift to open a recirculation passage from the trapped pocket back to suction. They will typically give unloading in steps of 75 percent, 50 percent, and 25 percent of full load, as each progressive plug is opened. These devices also do not give part load volume ratio correction like the first slide valves. Thus, part load efficiency is comparable to slide valves that do not regulate the discharge port. Plug valves also tend to be lower cost and simple in control method (Figure 13).

Variable Speed

Variable speed is occasionally used as a method of capacity control with screws. This can be provided with speed controlled engines, steam turbines, or variable frequency electric drives. Compressor power does not decrease linearly with speed

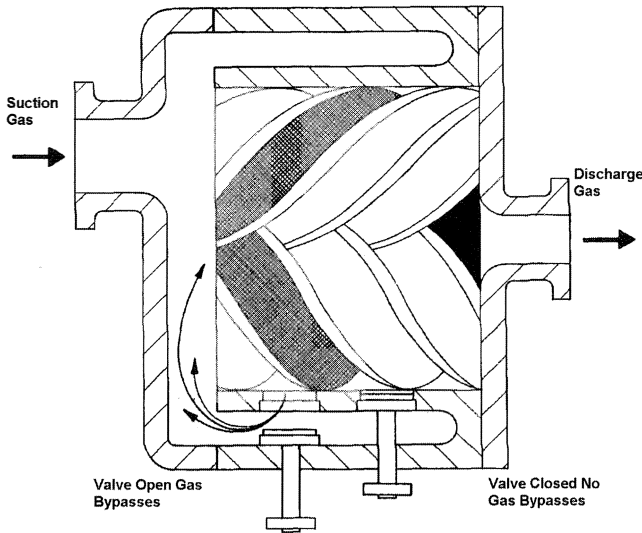


Figure 13. Two Step Plug Valve Unloading Mechanism.

reduction, but rather decreases as a function of rotor tip speed and operating compression ratio. In general, the compressor part load efficiency will be slightly better at low compression ratio and significantly better at high compression ratio with reduced speed compared to slide valve control, but this is before taking into account the losses in the driver at reduced speed.

Typical pulse width modulated variable frequency drives will have a loss in efficiency of approximately 2 percent at full load. They will also cause an additional loss of motor efficiency of 2.5 percent due to the effects of nonsinusoidal power on the motor. This gives a full load efficiency loss of about 4.5 percent. With some drives, this loss may stay at the same relative horsepower level as the drive speed is reduced, effectively becoming a larger percentage loss at lower speeds. If a compressor will operate at part load and at high compression ratio for many hours per year, the cost of the drive may be justified. However, if a compressor operates near full load for a high percentage of time, or operates primarily at low compression ratios, it is unlikely that there is any advantage with a variable speed drive, or that the cost of the drive can be justified against slide valve control. Typical curves for differences in power consumption are shown in Figures 14 and 15. These curves are calculated assuming drive and motor losses are a fixed percentage of power, not fixed kW at reduced speed.

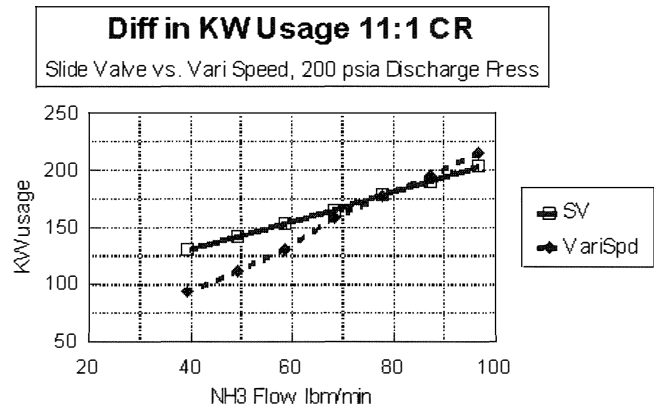


Figure 15. Part Load Power Consumption at High C.R., Variable Speed Versus Slide Valve Control.

Variable speed control with screws should not be implemented without consulting the compressor manufacturer. There are lower speed limits for compressors below which bearings may fail due to inadequate bearing lubrication. Large compressors will have lower minimum speeds than small compressors. Minimum speed considerations will typically limit unloading by speed control to 35 percent to 50 percent of full load, depending on compressor size. Many small compressors may be able to accommodate drive speeds above the input line frequency, but separator limits, oil cooler size, and other package limitations must be investigated. It is also possible to fill a compressor with oil and cause failure, if the speed is reduced below an acceptable range with the compressor unloaded. Many of these limits are not published but should be investigated early in a variable speed proposal or study.

OIL SYSTEMS—SEPARATION AND COOLING

As pointed out, oil injection performs many useful functions in the screw compressor; however, since oil is not desirable in other parts of the process system, oil flooded screws bring with them the need for oil separators. One type of oil separator design is seen in Figure 16. The mixture of discharge gas and oil leaving the compressor is directed against one head of the oil separator, where it experiences a change in direction and a large reduction in velocity. The larger oil particles are drawn to the oil sump by gravity, with the smallest particles or oil smoke being carried into the coalescing filters. Here, these small particles impact on the internal fibers in the filters and coalesce into larger oil droplets that can then be collected in the dry end sump and returned to a low-pressure area in the compressor. Besides removing oil from the gas stream, the separator also gives the oil in the main sump time for any condensed gas to absorb heat and vaporize or entrained bubbles to rise to the surface of the oil, giving higher quality oil for reinjection in the compressor.

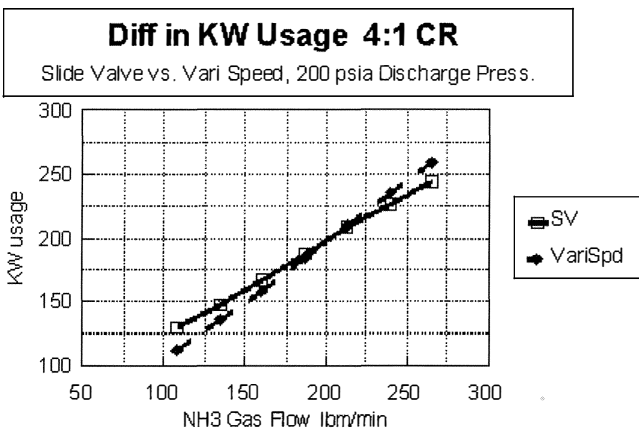


Figure 14. Part Load Power Consumption at Low C.R., Variable Speed Versus Slide Valve Control.

Whether variable speed is justified as a part load method, one must consider the load profile and operating conditions that are expected in a particular application.

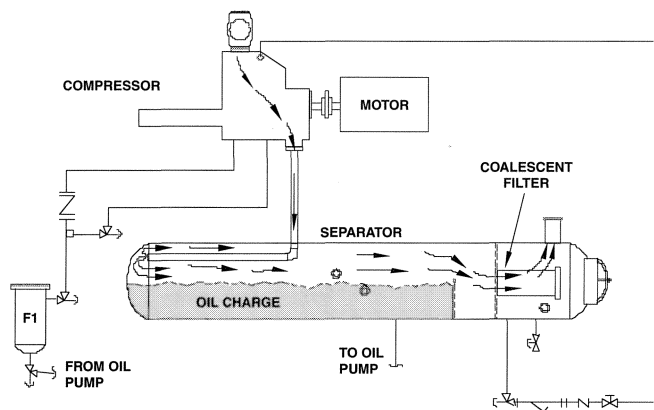


Figure 16. Screw Compressor Oil Separator Function.

Since most of the heat of compression is transferred to the oil during compression, an oil cooling system must remove this heat. The three most common systems are water-cooled, thermosyphon-cooled, or liquid injection. Air-cooled oil coolers are also used, but not as popular as the above three systems and will not be addressed here.

Water-Cooled Oil Cooling

Referring to Figure 17, the hot oil leaves the oil separator through a strainer into the oil pump. The oil is pumped through a shell and tube or plate type heat exchanger where the heat is rejected to circulating water, or glycol. The cool oil is then filtered and returned to the compressor for reinjection. Primary disadvantages of this system involve the initial cost and maintenance of the water or glycol system, and risks of tube rupture or plugging if proper water condition is not maintained. It is generally preferred to leave water flow to the cooler at a fixed rate and use a temperature regulated mixing valve on the oil side of the cooler to blend hot and cold oil to the desired reinjection temperature. This will help to minimize waterside fouling of the tubes by maintaining a higher minimum water velocity. The use of closed loop glycol eliminates fouling risks.

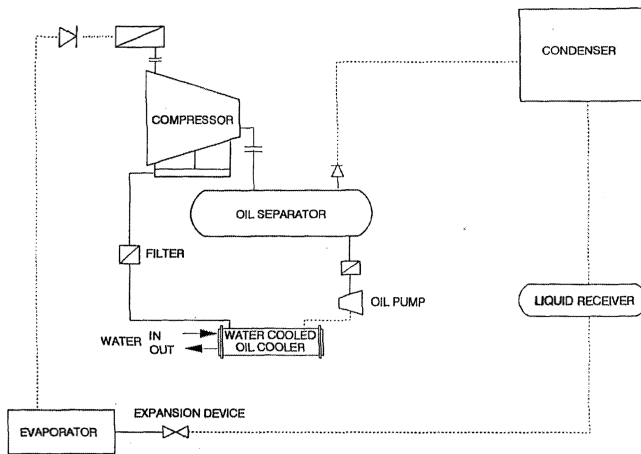


Figure 17. Screw Compressor with Water Cooled Oil Cooler.

Thermosyphon Oil Cooling

Thermosyphon oil cooling is used primarily in refrigeration applications in industrial plants where the compressed gas is condensed to a liquid as part of the cycle. A typical thermosyphon system is shown in Figures 18, 19, and 20. The thermosyphon system is similar to the water-cooled system, except the water is replaced by refrigerant boiling on the tube side of the oil heat exchanger. The thermosyphon system is basically a flooded evaporator fed by the gravity head of liquid refrigerant in a thermosyphon receiver, elevated above the level of the heat exchanger. As hot oil enters the shell side of the heat exchanger, refrigerant boils in the tube side with the bubbles rising in a return line back to the thermosyphon receiver. Vapor generated by this process is vented back to the condenser inlet, where it gives up heat and returns with the system liquid. In effect, the thermosyphon oil cooler is a gravity-flooded evaporator, with its evaporating temperature set by the pressure at the condenser.

Thermosyphon systems are popular because they require virtually no maintenance and they do not degrade compressor performance.

Liquid Injection Oil Cooling

Liquid injection is another oil cooling system that is used primarily in refrigeration applications where the compressed gas is liquefied as part of the cycle. Liquid injection cools the oil by

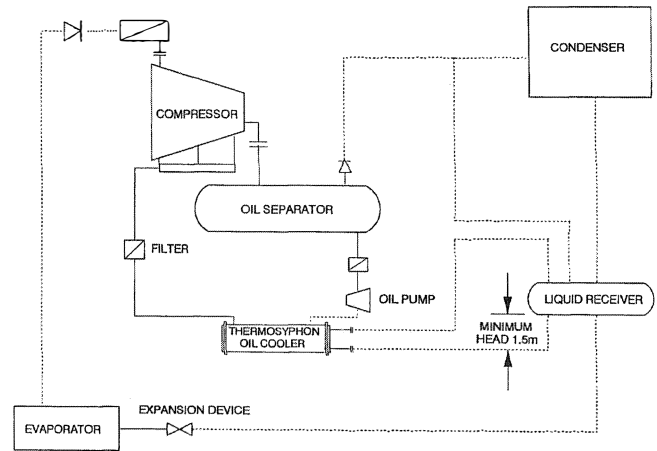


Figure 18. Screw Compressor with Thermosyphon Oil Cooler.

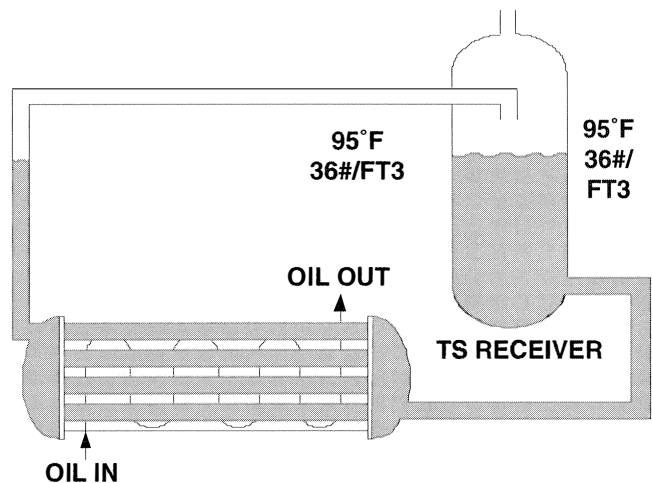


Figure 19. Thermosyphon Oil Cooling with Compressor Not Running.

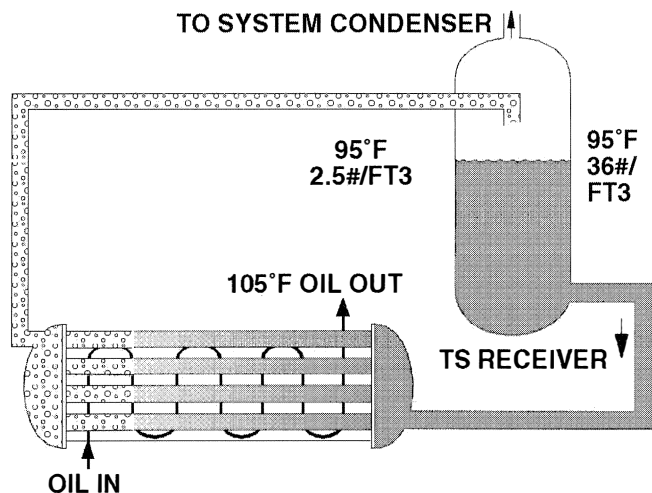


Figure 20. Thermosyphon Oil Cooling with Compressor Running.

direct injection of condensed liquid into a low-pressure screw thread, part way down the compression process, as seen in Figure 21. The oil and the discharge gas are controlled to the desired temperature by a thermal expansion valve. The valve controls the injected liquid flow to maintain the temperature of a thermal bulb installed in the compressor discharge line.

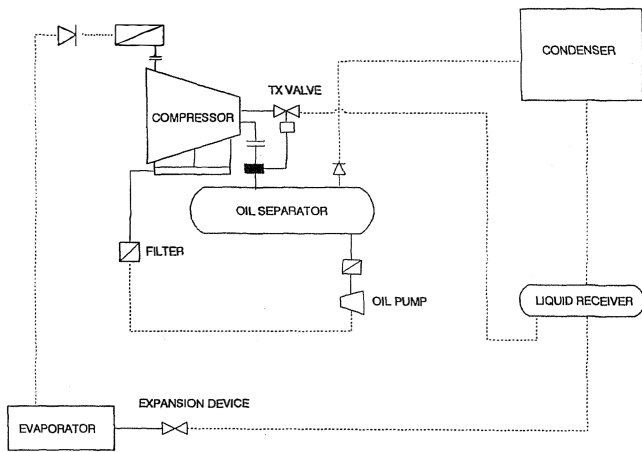


Figure 21. Screw Compressor with Liquid Refrigerant Injection Oil Cooling.

Some of the injected liquid expands through the expansion valve into the screw threads, requiring additional power to compress it. All the remaining liquid mixes with the oil and is carried down the discharge line with the discharge gas. The injected liquid remains in the compressor less than 0.01 sec. As heat transfer takes time, much of the cooling takes place in the discharge line and the oil separator, where the liquid has time to absorb heat and be evaporated. Some of the liquid mixed with the oil will leak through internal compressor clearances to the suction. Oil leaked to suction has little effect. However, the liquid will expand into the suction as vapor with a large increase in specific volume, reducing the amount of suction charge that can be drawn into the compressor. This causes a reduction in capacity with liquid injection, with larger effect at high compression ratios.

Liquid injection cooling is attractive because of low initial cost and low maintenance; however, the power and capacity penalties make it unattractive for most applications with high heat of compression. Liquid injection is also not recommended for systems with even temporary conditions of high suction and low head, where inadequate differential across the expansion valve causes erratic liquid feed.

ECONOMIZERS AND SIDeloadS

Since the pressure in a screw compressor is gradually increasing along the length of the rotor, it is possible to locate holes down the rotor bore at any pressure between suction and discharge. An economizer port is a hole located at a fixed volume ratio from suction that can be used as a secondary suction port on the compressor. Additional gas can be drawn into the screw thread while it is open to the economizer port, raising the pressure in the thread, with the combined massflow being compressed together to discharge.

In the case of a sideload, the additional gas generally comes from a second evaporator. The refrigerant pressure at the evaporator must be higher than the economizer port pressure by a sufficient amount to overcome pressure drop and be able to force gas into the screw thread. As an example, if the economizer port is located at a pressure of 1.9 times suction pressure, Figure 22 gives a comparison of how much sideload capacity would be available at various suction pressures.

An economizer is a special case of a sideload used in refrigeration applications. A portion of the liquid from the condenser is evaporated, with the flash gas going to the economizer port in order to subcool the remaining condensed liquid to the saturated temperature at the port pressure (Figure 23). This allows an increase in capacity because the liquid going to the main evaporator is colder. This also allows an increase in overall

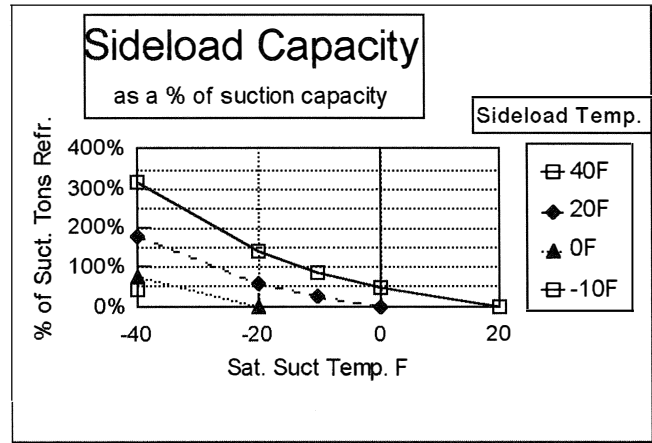


Figure 22. Screw Compressor Sideload Capacity at Varying Suction and Sideload Temperatures.

efficiency because of the thermodynamic advantage of subcooling and the fact that the compression of vapor at the economizer port is generally very efficient.

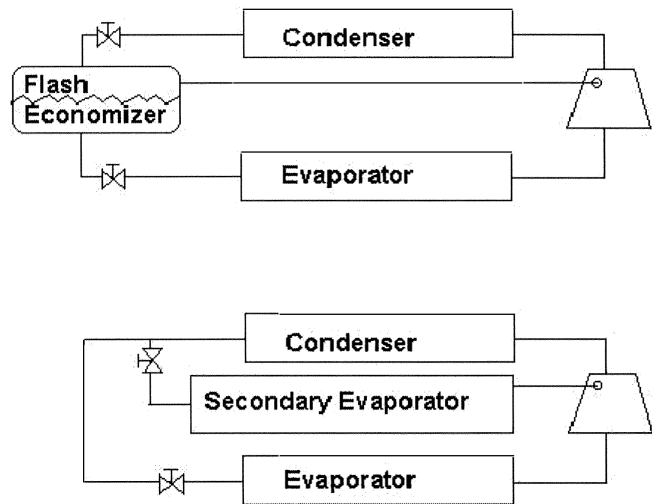


Figure 23. Piping Diagram for Economizer and Sideload Application.

When the compressor is spinning, there are certain sources of inefficiency that must always be overcome: friction in the bearings and at the walls, shearing of the oil films, leakage of a certain amount of oil from thread to thread, and friction at the shaft seal. The addition of the economizer gas does not significantly increase any of these losses, and thus the economizer compression is almost like it is performed in a frictionless, "loss-less" compressor.

MAINTENANCE

Follow manufacturer's recommended maintenance schedules, as requirements vary between products. However, some general comments can be made that should fit most applications.

Oil analysis is an important part of any compressor maintenance program. It is most important to watch for water content (Karl Fischer test) and viscosity change over time, which would indicate oil breakdown or dilution. Excessive water can damage compressors if allowed to remain in the system. Metals analysis may detect some problems, but usually indicates problems rather late in the failure cycle.

Vibration analysis is the best method for monitoring bearing condition. It is very effective with antifriction bearings, giving an indication of bearing deterioration in the very early stages of

failure. A good vibration monitoring program or onboard vibration monitoring, employed with antifriction bearing compressor designs, can eliminate the need for routine disassembly inspection if properly implemented, and greatly minimize the risk of catastrophic compressor failure.

MARKET TRENDS

Screw compressors have earned a reputation for reliability, flexibility, low maintenance, and good efficiency in many industries. Screws have a large market share in the portable and stationary air markets, industrial refrigeration, air-conditioning, and in the oil, gas, and petrochemical refrigeration markets. As gathering pressures in the gas patch decrease, the same arguments that have allowed screws to take large market shares in other industries are becoming obvious to packagers and operators. We have seen a large increase in screw compressor usage in natural gas gathering in the last five years.

Screw compressors have effectively displaced reciprocating compressors in the medium displacements and are today the most widely used compressor type in their size range. With the introduction of larger displacement screws, they are encroaching further on the lower end of the centrifugal sizes.

The Montreal Protocol (Montreal Protocol, 1989) has dictated the elimination of all chlorine containing refrigerants, with chlorofluorocarbons (CFCs) already banned in most countries and hydrochlorofluorocarbons (HCFCs) slated for phaseout in new equipment sold after 2010. Oil flooded screw compressors are uniquely suited to compression of a wide range of new and conventional refrigerant gases because of positive displacement and low discharge temperature. Screws have already been applied on many of the new alternative blend refrigerants, with numerous systems in operation on R-404a, R-507, R-134a, R-410a, and R-407c. As new refrigeration plants are installed, the life cycle of the plants makes consideration of long-term refrigerant availability an issue to be considered. The flexibility of screws gives the advantage that they do not have to be redesigned to handle the new gases.

CONCLUSION

It is certain that change will continue in the petrochemical, chemical, and gas businesses. The screw compressor has proven to be a strong central component in many of our systems because of its inherent efficiency, safety, and flexibility. It is certain that continuing innovation in screw compressor design and proper application of screws in new processes will be a key in improving system designs for the future.

APPENDIX A

CASE STUDY #1—SCREW COMPRESSORS FOR LOW TEMPERATURE REFRIGERATION

A producer of specialty chemicals had a need to condense vent gas from a reactor in one of their plants. This vent gas was made up of numerous compounds including phosgene. Aside from minimizing the release of hazardous materials to the atmosphere, liquefying these vapors recovered salable product. In order to condense the vent gas to a liquid, it was necessary to cool it to a rather low temperature. A mechanical refrigeration system, with a refrigerant evaporating temperature of -100°F , was chosen as the best means to accomplish this.

A "cascade" refrigeration system employing oil-flooded, rotary screw compressors was selected for this application (Figure A-1). Two separate working fluids (refrigerants) are employed, with each fluid contained in a separate cooling loop. The lower temperature working fluid absorbs heat from the process and rejects it to the higher temperature working fluid in a heat exchanger. Refrigerant 23 was chosen as the working fluid in the low temperature cooling loop because it operates at positive pressure at the very low refrigerant temperature (-100°F) required to cool the vent gas.

The R-23 vaporizes as it absorbs heat from the vent gas and enters the suction of the R-23 screw compressor at approximately 23 psia. This compressor boosts the R-23 vapor to approximately 183 psia and discharges to a heat exchanger. In this heat exchanger, the R-23 vapor condenses as it rejects heat to a second refrigerant (ammonia). The ammonia, in turn, vaporizes as it absorbs heat from the R-23 and flows to the suction of a second screw compressor. The pressure of the ammonia vapor is increased from approximately 20 psia to 250 psia in the compressor. The heat originally absorbed from the vent gas (along with heat of compression) is then rejected in a shell and tube condenser to cooling tower water as the ammonia vapor condenses.

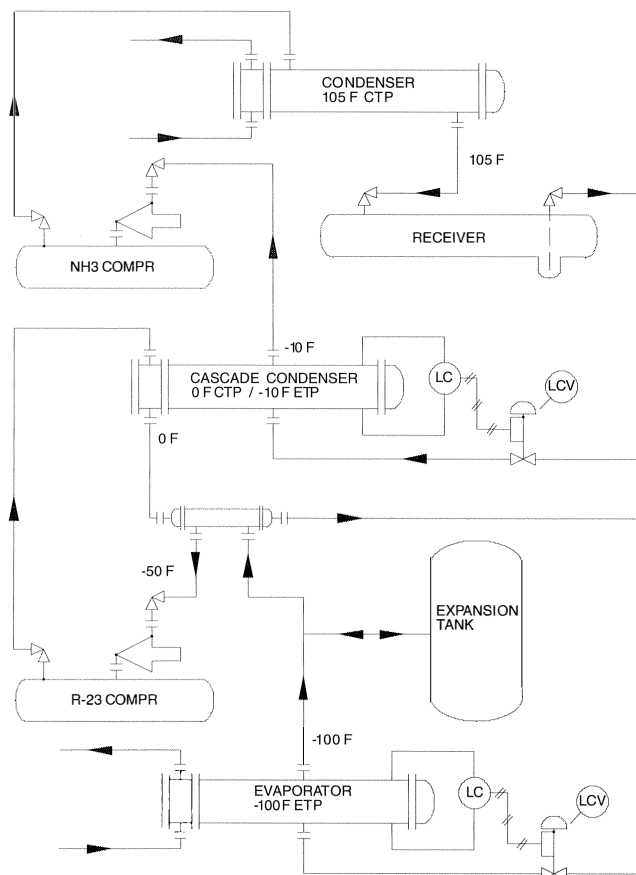


Figure A-1. Case #1 Piping Diagram for Cascade R23/ NH_3 Refrigeration System.

R-23 and ammonia were chosen as the lower temperature and higher temperature working fluids in a "cascade" arrangement to ensure that operating pressures in all parts of the system were above atmospheric, yet low enough to allow the use of standard refrigeration screw compressors and components. Note that the R-23 vapor is heated from -100°F to -50°F before it enters the screw compressor.

The rotary screw compressors for this system are directly driven by electric motors (150 hp for the R-23 compressor, 350 hp for the ammonia compressor).

In addition to the cooling load associated with condensing the vent gas, this system handles another refrigeration load at a higher temperature. An aqueous calcium chloride solution is cooled by the ammonia refrigerant in another heat exchanger and circulated to other process users in the plant.

CASE STUDY #2—SCREW COMPRESSORS FOR HYDROCARBON REFRIGERATION

A producer of commodity petrochemicals wished to replace two 20-year-old refrigeration systems that cooled ethylene dichloride

in one of their plants. Ethylene dichloride (EDC) is an intermediate product used in the manufacture of chlorinated and fluorinated compounds, as a raw material for vinyl chloride monomer, and as a solvent.

The existing refrigeration systems employed reciprocating compressors in a compound arrangement (low-stage and high-stage compressors). They used Refrigerant-12, which is no longer permitted in new equipment. The plant owners wanted the new systems to use a working fluid that would be available for the foreseeable future, and ultimately selected propylene (C_3H_6).

This was the first time that a hydrocarbon refrigerant had been used at this facility (Figures A-2, A-3, and A-4). Although the plant was accustomed to handling flammable fluids, they wanted to limit the size of the refrigerant charge to 1000 lb of propylene per system. In order to achieve this, the shell and tube heat exchangers in which the EDC was cooled were of a somewhat unconventional design. Whereas the more typical approach would be to evaporate the propylene refrigerant outside the heat exchanger tubes, these units were designed with the propylene evaporating inside the tubes.

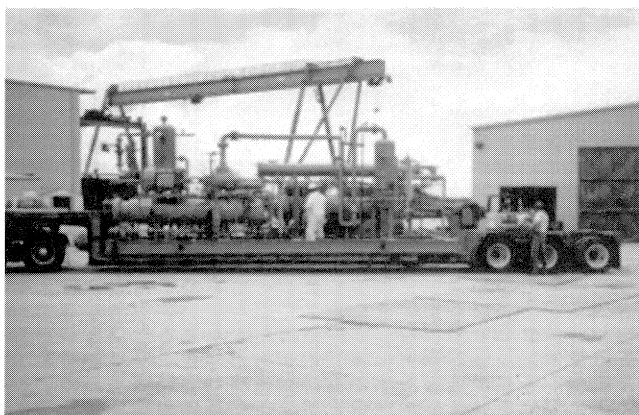


Figure A-2. Case #2 Ethylene Dichloride Chiller Utilizing Propylene as the Refrigerant (During Shipping).

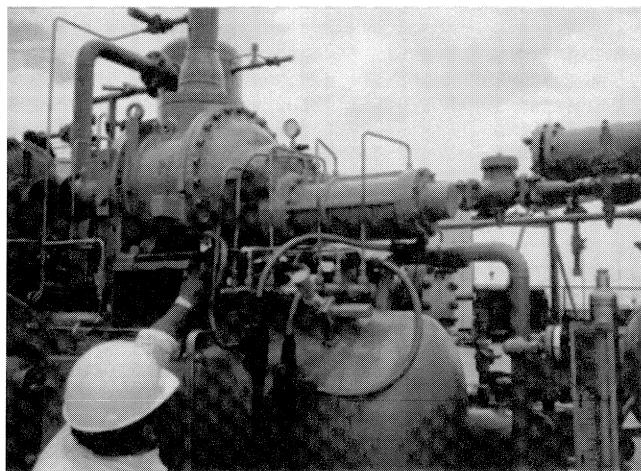


Figure A-3. Case #2 Case Steel Screw Compressor for Propylene Compression.

Oil-flooded rotary screw compressors were chosen to replace the aging reciprocating compressors. Unlike the reciprocating compressors, the screw compressors are able to compress the refrigerant vapors from the evaporator conditions of 6.9 psig, $-38^{\circ}F$ to condensing conditions 260 psig, $115^{\circ}F$ in a single stage. An “economizer” was employed to improve refrigeration cycle efficiency.

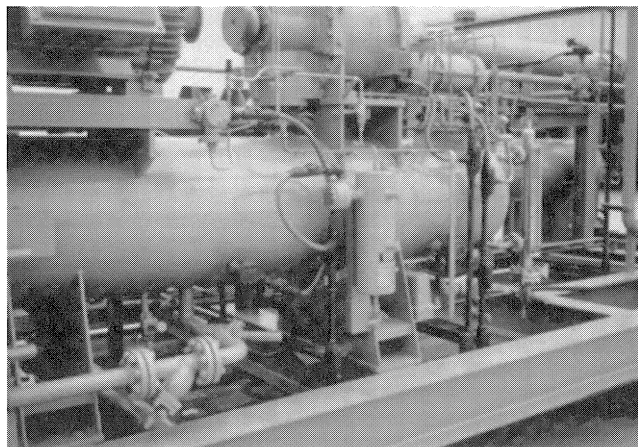


Figure A-4. Case #2 Tandem Seal Oil Recovery System.

The screw compressors for this propylene refrigeration service were supplied with cast steel casings and tandem mechanical shaft seals. Each compressor is directly driven by a 200 hp electric motor.

CASE STUDY #3—METHANE COMPRESSORS

A compressor system was installed utilizing four 2500 hp oil flooded screw compressors for recovering primarily methane from a storage tank at 27.7 psia and $-87^{\circ}F$ suction, to 435 psia and $191.2^{\circ}F$ discharge. The compressors operated single stage at 16:1 compression ratio. They were equipped with cast steel compressor housings, cycling lube oil pumps, and WPII motors. The oil was cooled in separate air-cooled heat exchangers prior to injection in the compressors. The destination was in the Caribbean (Figures A-5, A-6, and A-7).

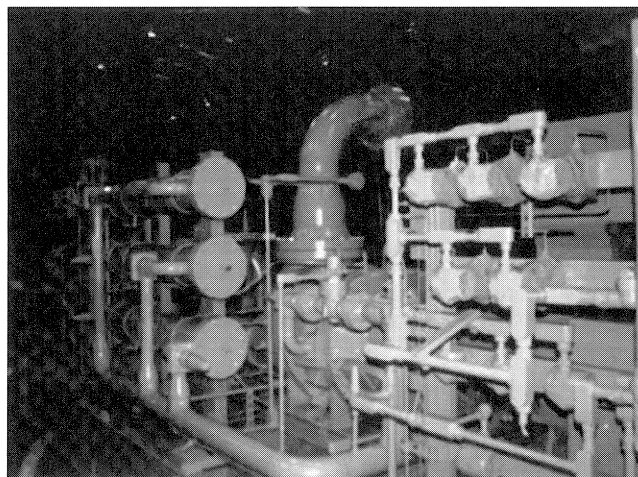


Figure A-5. Case #3 Oil System for Methane Compressors.

CASE STUDY #4—CHLOR ALKALI TERTIARY CHLORINE RECOVERY SYSTEMS

A cascade refrigeration system was supplied to condense chlorine from a process gas stream at approximately $-50^{\circ}F$ (Figure A-8). Two identical screw compressor systems were designed utilizing a low temperature CO_2 refrigeration loop condensed by an R-22 refrigeration loop. The R-22 loop was designed for 350 psig design pressure, and could accommodate R-507 (a hydrofluorocarbon (HFC)) in the future, if R-22 (an HCFC slated for elimination by the Montreal Protocol) becomes too expensive over the life of the equipment.

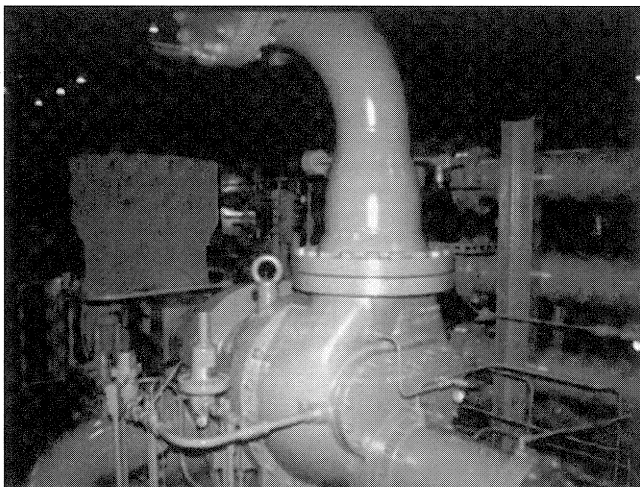


Figure A-6. Case #3 Cast Steel Screw Compressor for Methane Compression.

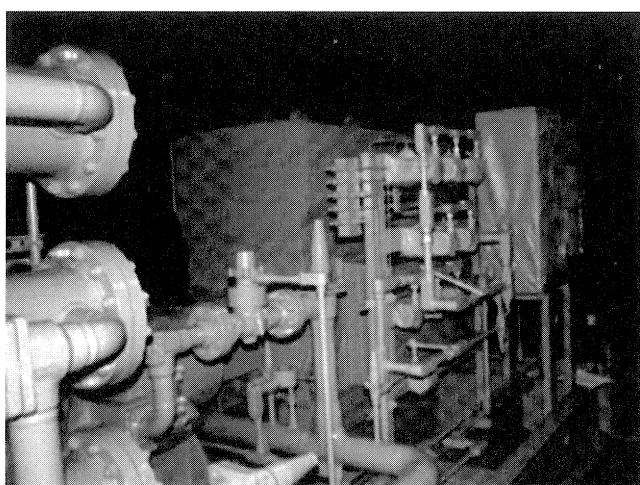


Figure A-7. Case #3 2500 HP WPII Electric Drive Motor.

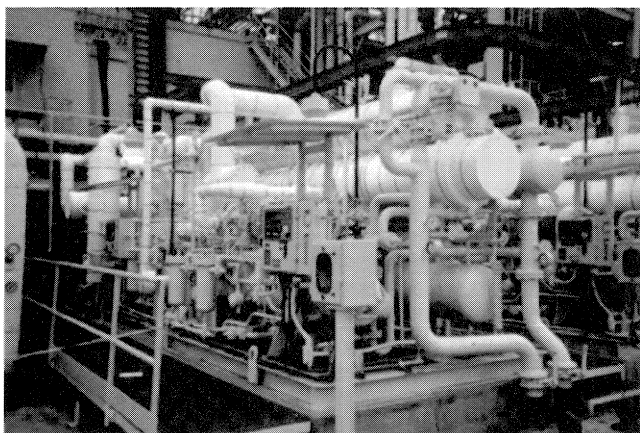


Figure A-8. Case #4 Cascade Refrigeration System.

The system was designed to utilize secondary oil coalescing technology on the CO₂ refrigerant side, in order to maintain the oil carryover to the CO₂ evaporator at or below approximately 0.5 ppm. The tertiary condensing load is up approximately 50 Ft in elevation, and therefore the liquid CO₂ has to be subcooled before going to the evaporator. The size of the system is around 70 TR (tons refrigeration) at -55°F, and is driven by a 100 hp electric motor on the CO₂ screw compressor, and 200 hp on the R-22 screw compressor.

Each of these systems replaced existing systems utilizing R-12 with multistage centrifugal compressors in a deep vacuum. The benefits include the following:

- With the efficient incremental turndown of the screw compressor units, the operations people have the flexibility of turning down to approximately 25 percent of design load and achieving reduced power consumption at the low rates.
- The use of CO₂ refrigerant in the tertiary chlorine condenser does not present a reaction problem. Chlorine reacts with most refrigerants. However, CO₂ is inert in a chlorine stream.
- The working fluids are environmentally friendly refrigerants that are readily available and inexpensive.
- This system does not operate anywhere in a vacuum, thus reducing long-term maintenance costs.
- The system is easier to operate than the previous system, and is not dependent upon constant operating conditions, completely eliminating any possibility of surging, and offering the flexibility to operate at any leaving temperature desired.

CASE STUDY #5— PROPYLENE STORAGE APPLICATION

This application is an open loop refrigeration system attached to a large atmospheric propylene storage facility. This facility stores polymer grade propylene, and the refrigeration system was designed to recover the flash that results from loading the propylene at 90°F liquid from a marine terminal, as well as accommodating the storage load.

Each of these systems utilizes two oil flooded screws compressing the propylene in two stages from around 15 psia up to 257 psia. The system is designed with two identical booster and high-stage compressor units used to recover the propylene under loading conditions. Each of these compressors utilizes about 1000 hp drive motor. In addition to the compound compressor system operating during loading times, an additional screw compressor with a 200 hp motor maintains the vessel pressure single stage, with an economizer cycle during the storage conditions in order to save energy.

REFERENCES

- Montreal Protocol, 1989, "Montreal Protocol on Substances that Deplete the Ozone Layer," Complete text on the United Nation Environmental Protection Website, <http://www.unep.org/unep/secretar/ozone/montreal.htm>
- Pillis, J. W., 1983, "Development of a Variable Volume Ratio Screw Compressor," IAR Annual Meeting.