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DESIGN OF THE OIL INJECTED REFRIGERATION SYSTEM AND ITS EFFECT ON RELIABILITY AND MAINTENANCE

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INTRODUCTION
This paper gives details of the refrigeration screw in general but certain experience and design features apply specifically to the screw compressors manufactured by Howden Godfrey Ltd., Glasgow, Scotland and marketed in America by Lewis Refrigeration Co., Woodinville, Washington.

While this paper basically describes this compressor applied to refrigeration, it is also widely used in other industries handling air, propane, natural gas and other hydrocarbons.

PRINCIPLES OF OPERATION
The screw compressor consists basically of two rotors enclosed in a casing. The rotors differ in shape and are identified as "female" and "male". Figure 1 shows a female and male rotor.

If these rotors are rotated, a space is formed (Figure 2(a)) due to the helix angle of the rotor lobes. This space is enclosed by the rotors and the casings.

This volume increases as the rotors rotate and gas is drawn in to fill the depression formed (Figure 2(b)).
A useful analogy in understanding this process is to consider the male rotor lobes as pistons and the female flutes as cylinders. Rotation of the rotors causes the piston to slide along the cylinder and draw in gas. Continued rotation results in the lobes remeshing and expelling the gas out of the far end of the casing. However, if a cover is placed over the discharge end of the machine then the remeshing of the lobes results in the volume of gas which has been drawn into the set being compressed between the meshing rotors and the casing.

It is necessary of course to provide some form of escape for the gas, otherwise, in theory at least, the gas would be reduced to zero volume by the meshing of the rotors and the pressure would rise to infinity.

At a particular point along the length of the casing a discharge port can be positioned which will permit the gas to pass out of the compressor.

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 Compression Ratio" of the set and is usually denoted by the symbol

It can be seen that the principle of the compressor is positive displacement, rotary and the only moving components are the rotors themselves. There are no valves at either suction or discharge.

**Screw Compressor Types**

Screw compressors can be divided into two distinct categories, oil free or 'dry' compressors and oil injected or 'wet' compressors.

The principle of operation of both types is exactly the same but in the one case the gas is compressed completely oil free whereas in the other oil is injected into the gas during compression and is removed from it after leaving the compressor.

In this paper the oil injected compressor is discussed as this is the type whose features are most suited to refrigeration applications.

**General Description of Oil Injected Refrigeration Compressor Design**

The basic compressor has been described as consisting of the male and female rotor rotating in mesh in appropriate casings. In end view these rotors appear as shown in Figure 3.

These rotors must be supported on bearings and as oil is injected into the gas during compression there is no requirement to isolate the bearings from the compression chamber by shaft seals. In fact the bearings can serve the dual function of supporting the rotors and also sealing the compression chamber. In order to do this oil is fed to the bearings at a pressure higher than compressor discharge pressure and in this way a full oil film within the bearings is maintained at all times.

As the bearings are next to the compression chamber as shown diagrammatically in Figure 4, they ensure the most rigid arrangement possible thus minimizing rotor deflection under gas compression loads and permitting high operating pressure differences across the compressor.

The geometry of the screw compressor rotors is arranged so that the female rotor absorbs only 15% of the input torque and the male rotor the remaining 85%. Thus the rotors can be considered to act as fully lubricated gears transmitting only 15% of the compressor input power. Taking into account the long length of the rotor relative to this low torque transmission, it can be seen that wear is quite negligible and the full compressor performance is maintained over long operating periods.

As one end of the rotors is at compressor discharge pressure and the other at compressor suction pressure an axial thrust is developed along the rotors. As the pressure difference across the refrigeration compressor is high the thrust developed is correspondingly high. However, the high discharge pressure necessitates the rotor to casing axial clearance at the outlet end to be maintained at as small a value as possible to minimize gas leakage and thus ensure high compressor efficiency. The best method of locating the rotors for this purpose is to use antifriction (i.e. ball and roller) bearings. However under the full axial
thrust the bearing life would be short. This problem is overcome in this type of compressor by incorporating a thrust balance piston to unload the angular contact ball thrust bearings. The balance piston has oil pressure (i.e. discharge pressure plus 30 p.s.i.(2 kg/cm²)) on one side and compressor suction pressure on the other. This arrangement has the added advantage that as rotor thrust is a function of discharge pressure and oil pressure is also related to the same discharge pressure, the balancing thrust is self compensating for varying discharge pressures that occur with changing condensing temperatures. The thrust bearings therefore carry a low load and have a long life thus reducing compressor maintenance requirements.

A horizontal section of a compressor with the above design features is shown in Figure 5.

![Image](image-url)

**OIL INJECTION**

Oil injection performs a number of important functions in a screw compressor:-

1. **Lubrication**
   Firstly and most obviously it lubricates the rotors in their operation.

2. **Sealing**
   The rotors, in order to mesh smoothly and be capable of some temperature variations, must have clearances between them. In operating 'dry', gas would leak back through these clearances reducing the compressor performance. When oil is injected into the compressor these clearances are effectively sealed. This means that not only is the gas leakage reduced at the same conditions but much higher pressure differences are possible without affecting performance.

3. **Cooling**
   In compressing any gas the power absorbed in increasing the pressure level appears as heat in the gas. In 'dry' gas compression high temperatures result from reasonable compression ratios. As an example consider ammonia compression over a pressure ratio of 10/1, e.g. -109°F/105°F (-23/40°C). The adiabatic temperature rise is 315°F (175°C). If an adiabatic temperature efficiency of 85% is applicable the final temperature becomes 380°F (193°C).

In the case of an oil injected compressor the volume of injected oil is very small, normally less than 1% but its heat capacity is high as it is in the liquid phase and its mass flow is proportionately greater. Thus with this relatively small amount of approximately 1% by volume of oil the final discharge temperature is reduced to values of the order of 170°F (75°C). The exact value depends on the duty and refrigerant concerned but the system is designed to ensure that the discharge temperature never exceeds 212°F (100°C) regardless of the compressor duty within its design limits. This enables wide variation of suction and discharge pressures to be handled with complete freedom from temperature limitations. An example of the system flexibility which results from this is that it is common practice to pipe up a 2 stage system such that either the first or the second stage can operate over the total pressure ratio of the complete plant in the event of one or other stage being out of service for any reason. In this way a partial stand-by is always available.

4. **Simplifying Compressor Design and Increasing Reliability**
   As the rotors are fully lubricated and act as cycloidal gears as noted earlier, there is no requirement for timing gears as are used in oil free screw compressors.

Since the maximum temperature occurring anywhere in the compressor never exceeds 212°F (100°C) there is no requirement for casing cooling by water jacket or any other means. Similarly, rotor cooling is unnecessary and solid rotors are used.

Reliability of the compressor is therefore increased both due to the low operating temperatures and the reduction in compressor components. The reliability of the compressor is such that the recommended overhaul period is after 50,000 hours of operation.

**CAPACITY CONTROL**

In any refrigeration installation there are fluctuations in demand, the magnitude depending on the process.

One of the many advantages of the refrigeration screw compressor is that stepless capacity control with almost proportional power saving is provided.

1. **Compressor Slide Valve**
   Capacity control is provided by an internal slide valve in the compressor. This slide valve moves parallel to the rotor axis and achieves capacity control by altering the effective length of the rotors being used for compression.

   Figure 6 shows the typical relationship between power and capacity for part load. This figure gives the results with maintained compressor discharge pressure. However, in a refrigeration plant the discharge pressure falls with reduced mass flow during part load and making allowance for this the typical part load curve appears as shown in Figure 7.
The action of the slide valve is to release the gas drawn into the compressor and trapped between the rotors and casings before the compression process starts. In this way no work is done on the released gas as it is recycled back to suction again. The amount of gas so released is in direct proportion to the position of the valve hence the stepless control.

The slide valve is modulated by feeding oil to one or other side of the double acting hydraulic cylinder which is integral with the compressor. The oil flow is controlled by solenoid valves. Figure 8 shows the principals of the system in a vertical section of the compressor.

The sliding contacts between the piston and cylinder and the piston rod and its seals are all fitted with P.T.F.E. ("Teflon") slipper seals backed by 'O' rings. These minimize friction forces and also reduce wear to negligible proportions. This arrangement also has the advantage that swelling of 'O' ring materials for any reason does not jam the operation as the 'O' rings are captive and not themselves in sliding contact with other components.

2. Control Systems

Control systems are designed to adjust the compressor capacity by means of the slide valve to match plant duty at all times.

These systems vary in sophistication depending on individual plant requirements. The simplest is direct hand control in which the compressor output is varied by direct operator control of pushbuttons.

Another type is the "time-base" system. In this the plant temperature or pressure is monitored at regular time intervals and when this varies by a preset amount from the set condition, the compressor is on or off loaded by a fixed percentage. The compressor capacity therefore is adjusted to match plant condition changes on a time base. This system is very simple but takes a little time to adjust to very rapid duty changes. Where this is important the proportional control system would be used. The proportional control system is shown diagrammatically in Figure 9. Its operation is as follows:--

A sensing device is fitted to the refrigeration plant which measures the temperature or pressure of refrigerant or the process material temperature as required. This device senses any variation in the preset condition and signals the change to the proportional control unit. This unit then operates the appropriate solenoid valve to apply oil pressure to the hydraulic piston to move it in the required direction. As the piston moves due to this oil pressure the movement is recorded by the potentiometer.
coupled to the capacity position indicator shaft and a signal is passed to the control unit. When the capacity control valve movement matches the original system variation signal the solenoid valve is closed and the system is once more in equilibrium. The system can use all electric controls or all pneumatic controls using standard components in both cases.

3. Speed Variation

As the compressor is of the positive displacement type it has a wide band of operating speed over which the full range of pressure duties can be utilized.

Speed variation is a most efficient means of capacity control within the speed band of the compressor and is often used when the compressor is turbine or engine driven.

Variable speed electric motors are extremely expensive however and for motor drive the slide valve control is the feature used.

COMMENTS ON COMPRESSOR DESIGN

To summarize, the compressor operation utilizes only two moving parts, the rotors with entirely rotary balanced action. There are no out of balance forces, no sliding surfaces such as piston and cross head and no valves. Part load operation utilizes a slide valve, a lightly loaded smoothly operating component with a large supporting bearing surface fully lubricated by oil galleries.

The combination of the principles of operation of the oil injected screw compressor with the particular design features built in to them, ensures that operating life is at a maximum with an excellent proven record of maintenance free operation.

AUXILIARIES

The main auxiliaries (excluding the control system which has already been described) are the gas suction strainer, oil tank/separator, the oil pump, oil cooler, oil filter and oil tank heaters when fitted. The circuit diagram in Figure 10 shows the system incorporating these items.

1. Gas Suction Strainer

This is basically a fine mesh strainer whose purpose is to remove all large particles of pipe scale or other debris which might be carried through into the compressor. If they were allowed to pass through the compressor they would score the rotors and over a period of time, damage the seals on the rotor tips. The function of the strainer is therefore to ensure that the compressor performance is maintained in service.

2. Oil Tank/Separator

The oil tank/separator removes the injected oil from the refrigerant before it passes into the plant system.

Initial separation of the major quantity of the oil occurs by gravity in the bottom of the tank. The gas velocity is low and all the oil in droplet form simply falls down into the main body of oil.

The refrigerant then passes into the separator demister where almost all the oil remaining is collected by impingement on the very fine knitted wire from where it returns to the oil reservoir in the bottom of the vessel.

A most important point is that the discharge temperatures being low (170°F, 75°C) make the oil separation relatively simple. The oil carried over by reciprocating compressors is at much higher temperature levels and separation at these conditions is a much more difficult task.

This is true on two separate counts. Firstly, high temperatures reduce the oil viscosity and surface tension and this makes oil separators less effective due to the reduced tendency for the oil to be retained by the separator element.

Secondly, the amount of oil in the vapor phase is higher at higher temperatures. In any fluid/gas mixture a quantity of fluid is present in
the gas in the vapor phase and as much it acts with the properties of a gas. A typical example of this phenomenon is relative humidity, i.e., water/air mixtures. The actual proportion of fluid in the gas can be determined by using Dalton's law of partial pressures, and from this the following relationship can be derived.

\[
\frac{W_o}{W_g} = \frac{P_o}{P_v}
\]

where \( W_o \) = weight of oil, \( W_g \) = weight of gas, \( P_o \) = partial pressure of oil = vapor pressure of oil at system temperature.

i.e. the weight of oil in the gas in the vapor phase is proportional to the vapor pressure of the oil at the existing temperature conditions.

Consider the discharge conditions of an oil injected screw and a reciprocating compressor of 170°F (75°C) and 285°F (140°C) respectively.

Using a typical refrigeration oil the ratio of oil in the vapor phase between these two temperatures is 6 \( \times 10^{-5} \) i.e. a factor of 100 times.

In the case of the oil injected screw compressor effective oil separation is therefore a considerably easier function than with other types due to the low discharge temperatures.

Removal of oil in the vapor phase from a gas stream can only be effected by chemical rather than mechanical means although reduction of temperature reduces the quantity in the vapor phase.

A proportion of the oil in the liquid phase which is passed into the oil separator is in a very finely divided particle size, e.g., with dimensions of molecular scale. These small particles also tend to act as a gas and to pass through standard wire mesh separators, although fortunately their quantity is small and for normal applications the oil separation is quite acceptable.

Where very low oil carryover values are necessary (better than 10 p.p.m.) then a secondary separator is fitted of the agglomerator or coalescer type. These "coalesce" the very small particles into droplets and this enables them to be removed by conventional means. These separators normally require replacement of the cartridge or element at regular intervals to maintain full separation efficiency. Demistors or knitted wire separators require no routine maintenance.

3. Oil Pump

While the oil pump is required to deliver oil at discharge pressure + 30 p.s.i. (2 Kg/cm²) at the oil manifold, it draws the oil from the oil tank separator at discharge pressure and its duty is therefore relatively light. Either gear type or centrifugal pumps may be used.

4. Oil Cooler

The oil cooler is required to dissipate the heat of compression removed from the gas by the oil injected into the compressor. This involves a heat exchanger, but all the heat removed in this way does not have to be removed in the refrigerant condenser which is correspondingly smaller. Water/oil, air/oil or refrigerant/oil heat exchangers may be used.

5. Oil Filter

The oil system used with these compressors is arranged to filter all the oil being supplied to the compressor without a filter bypass. Experience has shown that this is necessary to insure that the compressor life is not reduced by bearing or shaft seal wear resulting from unfiltered oil. If the oil filter becomes dirty the pressure drop across it increases and the oil pressure relief valve positioned before the filter bypasses the oil directly back to the oil tank. If the oil filter is not changed at the recommended pressure drop level, the manifold oil pressure will fall and the compressor will be shut down and is therefore fully protected.

6. Oil Tank Heaters

Oil heaters are sometimes fitted to the oil tank. The purpose of these is to maintain the oil temperature at a level at which the oil viscosity is suitable for pumping through the oil system and filters. In outdoor installations, for example, it would be possible for the oil tank temperature to fall to very low temperatures in winter when the set was not in operation and without an oil heater there would be some difficulty in obtaining correct oil flows at start up under these conditions.

SCREW COMPRESSOR OPERATION

1. General

The screw compressor is one of the simplest types of compressor to operate. The injected oil is so effective in controlling compressor discharge temperature that wide variations in load can be handled by the compressor.

2. Wet Refrigerant

These compressors are capable of handling wet refrigerant or even random slugs of liquid without any effect on the compressor whatsoever. The compressor has no valves to be broken, there are no sliding surfaces to be washed clear of lubricant and there are no excessive loads applied to any component.

3. Oil Foaming

Foaming of the compressor oil system at start up does not occur. Unlike reciprocating compressors the oil tank is maintained at compressor discharge pressure and at start up, therefore, the pressure on the oil increases counteracting any tendency for the oil/refrigerant to foam.

4. Oil Temperature

The temperature of the oil supplied to the compressor should be maintained in the region of the manufacturers recommended level. This is not critical to the compressor within a wide band. However, if the oil is too cold the oil filter pressure drop, due to the higher viscosity, will
be higher than normal reducing the filter life. If the oil is too hot, the compressor discharge temperature may rise to an unacceptable level but also the compressor capacity will be slightly reduced by about 1 to 1.5% for every 10°C rise in oil temperature. The oil temperature is of course very easily controlled by either manual or thermostatic valves in the water supply to the oil cooler.

5. Routine Operation

In normal operation the only routine attention required for compressors of this type is to control oil filter pressure drop and change when necessary and to check oil level.

In view of this most screw compressor operators have found that full time plant operators are not required, therefore considerably reducing the plant operating costs.

6. Safety Protection

The oil injected screw compressor is a very robust, simple machine with large safety reserves on all operating parameters.

The compressor systems are normally fitted with protection to ensure the safety of the compressor in the event of any fault occurring. These are oil pressure, compressor discharge pressure and compressor discharge temperature. Other protective devices, such as low suction pressure and oil temperature, may be fitted in particular instances to suit the characteristics of the plants concerned.

COMPRESSOR OVERHAUL

1. General

The compressor running time before overhaul, the amount of overhaul required and the skill and equipment needed vary with each manufacturer and the particular design of compressor.

The comments given below must therefore be considered as applying to the specific compressors manufactured by the company by whom the author is employed.

2. Period of Service before Overhaul

The recommended period of service before overhaul of these oil injected refrigeration screw compressors is 6 years which is equivalent to approximately 50,000 hours in continuous operation.

This figure can be compared to the equivalent value for reciprocating compressors of at least once a year to demonstrate the effectiveness of the rotary screw principles with their few moving parts.

3. Skill Required for Overhaul

Oil injected refrigeration screw compressors are very simple machines with very few components. As a result any skilled tradesman is capable of fully dismantling and rebuilding them with the assistance of the manufacturers instruction manual.

All components are interchangeable in the field including rotors and casings.

4. Components Renewed at Overhaul

During overhaul it is standard practice for the thrust bearings to be renewed along with all 'O' ring seals and locking washers.

Antifriction bearings (i.e. ball and roller bearings) have a life which is controlled by operating speed and load. Those fitted to these compressors are lightly loaded due to the off-loading provided by the thrust balance piston but it is still advisable to renew them at major overhaul periods.

Any other components which have worn such as journal bearings of shaft seals would also be renewed.

It is important to note that rotors do not require renewal at overhaul periods.

As the compressors are standard for all refrigerants, the seal materials used are common for any application simplifying the spares requirements.