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The Evolution of the High Efficiency Two-Pole Hermetic Compressor

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INTRODUCTION

During the last 30 years, there has been a 30% increase of efficiency of two-pole hermetic compressors. This improved performance has been in response to marketplace requirements and Government legislation. Throughout this evolution, there has been a noteworthy increase of the compressor reliability while sound and economic values have remained constant.

Presented in this paper is a tracking of this history. The increased efficiency has not been the result of improvements in any singular component. It has occurred due to small improvements in all components of the compressor. Discussed here-in are the changes of these components (ie. valving, lubrication, motors, flow path, materials, internal heat exchange and machining tolerances) and how these changes affected the overall performance of the compressor.

VALVING

The valving of a hermetic refrigeration compressor is probably the most important singular design item of the compressor. Although volumes of work have been applied to valve design, a direct approach for a universal solution has not been revealed. The early design goals were to achieve a reasonable life in bending fatigue, impact and fluid abuse while maintaining maximum flow rate requirements. The pressure drop, or efficiency of the valving system, was considered of less importance. Compressor designs with high bore/stroke ratios were favored because they resulted in sufficient cylinder cross-sectional areas for reasonable valve porting. Suction ports were normally two to three times larger than the discharge ports.

This ratio is approximately the same as the ratio of suction gas density to discharge gas density. Possibly ignored was the fact that the valve open period had a greater ratio (suction period to discharge period). This resulted in discharge valve systems having pressure drops nearly four to five times as great as the suction valve system pressure drops (1). Most early designs were either ring valves with the suction ports surrounding the discharge ports, or beam suction valves with cantilever discharge valves.

Valve design analytics were based upon the solution of second derivative displacement distribution strain equations (2). The equations ignored dampening effects (stiction) and valve inertia contributions. The flow analysis of the valving system was often based upon incompressionable fluid flow through a group of orifices. This type of analysis did not include gas dynamics or time dependent variable flow passages. These omissions, plus the affects of complex shapes, severely reduced the accuracy of the analysis.

Current valve design methodology utilizes finite element analysis techniques which result in improved predictions of dynamic valve stress levels and valve shape predictions. Shape predictions are extremely important in defining the actual, instantaneous flow areas and, thus, the pressure losses of the transient mass flow.

From these studies, it has become more and more evident that the perimeter of a valve may be of greater importance than the valve plate port area. The importance of the perimeter is readily evident when including valve lifted shape in determining the peripheral area for flow energy loss determination.

Valve action is controlled by aerodynamic drag forces, valve inertia, spring forces, fundamental resonances, gas dynamics, physical shape and stiction. Valve action analytical studies must include these items, as well as flow and dynamic performance predictions (3). Flow equations should predict the energy losses due to flow restriction. Dynamic equations should predict the time related motion of the valve. Dynamic predictions will also determine the maximum expected stress levels and instantaneous flow passages for flow energy loss predictions.

Modern valve analysis techniques for the above should include the following:

A. Modal analysis determination for natural mode shapes. These modal shapes determine the instantaneous flow passage areas.

B. Stiction of the valve to the valve plate and/or backing plate. Stiction is the dampening parameter of valve dynamic calculations. This dampening affect will the delay the valve opening and closing. These delays can result in back flow through the valve and loss of valve efficiency (4).

C. The inertia of the valve has major effects on opening and closing timing. Inertia has similar effects as stiction (ie. motion delays).

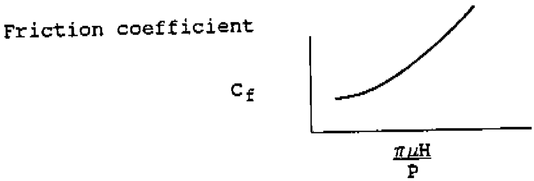
D. The valve contour is naturally needed when determining the modal shape. Alteration of the contour may be used to alter the modal shape and optimize the peripheral flow area between the valve and valve plate.

E. Though often very difficult to analyze, passage acoustical tuning can seriously affect the overall efficiency of a valving system. The timing of peak pressure waves with respect to valve opening can result in appreciable increases of cylinder entrance and exhaust flow.

LUBRICATION

The lubrication system of a compressor is, naturally, a very major item in the control of unwanted power consumption. The lubrication system will affect the peripheral losses by controlling the bearing friction. It also has a significant affect upon windage losses, cylinder friction losses and oil pump losses. Each of these items are more fully discussed in the following.

Bearing friction is a function of the oil viscosity, shaft speed and bearing load [(see figure below (5)].



Friction losses are the result of viscous shear in the bearing oil interface and may be further defined as: $H_p = \frac{2\pi^3 \mu N^2 D^3 L}{C}$

From this, we see that the shear losses are directly proportional to the oil viscosity, vary with the second power of the shaft speed and the third power of the shaft diameter. They are also inversely proportional to the diametrical clearance when the clearance is established with respect to the required minimum oil film thickness. The lower viscosities will have a negative affect upon the load carrying capacity of the bearing

Oil viscosity is the load-carrying element in the selection of a bearing. In each application, the viscosity of the oil at the operating condition must be determined. This operating oil viscosity will then reveal whether sufficient minimum film thickness is present to support the load without asperity contact. The lower the oil viscosity, the lower the bearing viscous losses. Higher bearing temperatures result in lower viscosities. Early compressors applied oils having viscosities as high as 450 SUS, where a more current compressor utilizes oils of 150 SUS.

The shaft speed in most current compressors is nearly constant (3,500 rpm) and is, therefore, not a consideration in performance improvement to date.

The bearing diameters may have major effects on the total bearing friction contribution. This is readily seen with the third order relationship to the bearing friction losses. Adjustment of bearing diameter and bearing length (first order losses) for a given load may well result in optimizing for minimum bearing losses.

Permissible minimum oil film thickness is generally considered as a function of the opposing surface finishes. The asperities of these surfaces must be separated by the oil to have minimum friction losses. The combined finish, times a factor of 4 to 10, is normally used for a minimum thickness guideline. The total area of the asperities is the primary supporting element of the bearing. This area, related with the oil film pressures (ie. viscosity), result in the support without contact of the bearing load.

The lubrication system also becomes a major factor in controlling the friction losses of the cylinder/piston operation. Sufficient oil must be present to control these cylinder losses. Operating loss studies of reciprocating engines have shown that these cylinder loses are "number one" in the overall engine losses (6). The major contributors to cylinder losses are piston ring and piston skirt friction.

Piston ring friction losses have been defined by Stribeck as a direct function of oil viscosity and piston velocity. It also has an inverse relationship with the ring load. Piston skirt (body) friction follows a very similar relationship as ring friction where the ring load is supplemented with piston side loading. Interfacial oil film shear losses will, again, be a direct function of surface finish and clearance.

Total lubrication system oil pumping power requirements are separate and additive to the internal compressor losses. Oil circulation controls the operating bearing temperatures and also affects the windage losses. Oil pump power is $H = PQ/K\mu$, where P is the pumping pressure rise, Q the oil flow rate, K the pump efficiency and μ the oil viscosity. Naturally, excessive oil pumping for either bearing viscosity control or for the control of compressor sound levels will result in unwanted power inputs.

Although internal cranking windage losses are not a major item in determining overall compressor losses, they should be included in design considerations. These losses have been defined not only as a function of the frontal areas of the moving components, but they are affected by the oil entrainment within the cranking cavity (6).

$$H_w = \frac{Pv^2CdA}{2}$$

The density of the cranking cavity gases is directly affected by the oil gas entrainment. An increase of 10% of the oil entrainment can result in major increases of windage losses. Windage losses may outweigh friction losses if the cranking cavity oil concentration is excessive.

Overall oil entrainment in the pumped mass flow also affects the total refrigerant pumping ability of the compressor. Also, high oil pumping rates (greater than 2%) will affect system performance by reducing the heat transfer rates of the heat rejection surfaces.

MOTORS

As mentioned in the introduction, improvments in motor efficiency have been a significant contributor in the improvement of overall compressor efficiency. This is especialy true for a suction gas cooled motor where the motor losses directly cause an increase of suction gas temperature. A 20% increase of motor losses (2% reduction of motor efficiency) will result in a 14% increase of suction gas heating and a significant drop in compressor efficiency.

Single-phase motor efficiencies of 78-82% were considered quite acceptable in the early 1970's. Current high efficiency compressors have motor efficiencies in the range of 86-90%. Motor improvements that have yielded this increase are discussed below:

1. Low Loss Lamination Steel - Improved quality and reduced thickness of motor lamination steel has resulted in reduction of core losses of the laminations. Standard lamination steel has a core loss of approximately 2.7 watt/lb, while a high quality steel has a core loss of 1.7 watt/lb.

2. Lamination Insulation - Improvements in the electrical insulation between core laminations (either oxides or bondings) have resulted in reduced core eddy current losses.

3. Phase Separator and Slot Liner Insulation - Improvements of these insulations have allowed for improvement of slot fill for higher density more efficient coils.

4. Coil Compaction - Improved manufacturing methods of placing and compacting magnet wire in the slots have yielded higher density fills. High density filling has resulted in higher flux densities for the same quantity of wire (constant electrical losses), resulting in improved efficiencies.

5. Improved Magnet Wire Coating - Improved wire varnishing systems to the current two coat polyesterimide coating has also contributed to the increased slot fill of Item 4.

6. Increased Stack Height - The tendancy in higher efficiency compressors has been to apply longer stack motors for the same maximum running torque, which results in improved motor efficiency.

7. Reduced Rotor Fans - Improvements compressor efficiencies have reduced the amount of forced cooling required to maintain reasonable motor temperatures. With this reduced need for cooling, the motor windage losses have been reduced by the application of smaller rotor fans.

FLOW PATHS

Gas flow pressure losses through a compressor must be minimized to realize high efficiencies. Reduced flow restriction is often counter productive in achieving low sound levels and reduced pressure peaks. To have the best of both worlds, some of the following design techniques are being applied:

1. Through-flow side band and Helmholtz resonators for sound control. These are replacing the high flow resistance impedance muffling systems.
2. Through-flow resonance chambers for discharge pressure pulse control.
3. Increased valve travel for reduced value losses.
4. Enlarged valve ports for reduce passage losses.
5. Increased shockloop sizes for reduced pressure drops.
6. Tuned discharge systems for reducing discharge pressure pulse levels without high impedance losses.
7. Balanced flow losses of the section passages and discharge passages.
8. Non-flow restrictive motor cooling.
9. Non-restrictive liquid refrigerant scheduling.
10. Dynamic gas pressure recover.

MATERIALS

Although generally overlooked, the proper selection of materials is having important effects upon the improvement of compressor efficiency. Proper material selection can affect the motor efficiency, reduce internal heat transfer, reduce friction losses and affect dynamic responses. Some areas in which proper material selection has been effective in increasing the overall efficiency of the compressor are:

1. Low loss motor lamination steel.
2. Improved bearing materials (ie. bronze vs. aluminum; pheneolic thrust plate material; coated shafts).
3. Low thermal conductivity barriers between suction and discharge gas passages.
4. Low thermal conductivity valves.
5. Oil additives for reduced friction.
6. Improved motor insulation systems (wire coating, phase separators and slot liners).
7. Improved cutting tool material for improved finishes.

INTERNAL HEAT EXCHANGE

The transfer of heat within a compressor has been found to have major effects not only upon the compressor's capacity, but also upon its efficiency (7). Energy requirement per unit of mass flow increase with the temperature of the mass. With this in mind, control of the suction gas temperature is very essential in maintaining or increasing efficiency levels. Items which affect the suction gas temperature are:

1. Gas Passage Heat Interchange - The use of common manifolds or castings for routing the refrigerant to and from the cylinder may result in internal heat transfer from the discharge gas to the suction gas. This internal heat transfer increases the suction gas temperature, which results in an associated loss of compressor efficiency.

2. Discharge Shockloop Oil Submersion - Routing the internal shock loop through the oil sump (possibly for driving dissolved refrigerant from the oil) increases the oil sump temperature. The increased sump temperature results in an increase of heat transfer to the suction gas blanking the oil. This results in an increase of the suction gas temperature and a loss in compressor efficiency.

3. Minor Internal Leakage - Assumed "minor" internal gas leakage from the discharge plenum to the suction gas can have severe affects upon the overall compressor efficiency. Some possible sources of these minor leaks are:

   A. Valve plate porosity
   B. Gasket permeability
   C. Piston dome porosity
   D. Porous braze joints
   E. Valve seat imperfection or roughness
   G. Piston blow-by

4. Motor Efficiency - See the section on "Motor Improvements.

MACHINING TOLERANCES

Improved machining tolerances have resulted in significant improvments in the general efficiency of compressors. These tolerance improvments have involved surface finish improvmeents (bearings, journals, cylinder bores), crankshaft alignment improvements, and out-of-round control (cylinder bores, journals and bearings).

The most significant of these is the improvement of the surface finishes. The roughness of bearing journals has been reduced from an average value of 30 micro inches to a plus three sigma limit value of 16. This 16 limit results in a maintaining an average value between 10 and 12 micro inches. Surface finish reductions directly result in reducing the friction power losses as noted in the section on "Lubrication".

Production crankshaft alignment has also been improved through the application of modern automatic assembly techniques. Improved alignment results in reduced main bearing power consumption and, thus, improves compressor efficiency.

SYMBOL DIRECTORY

μ	=	Viscosity
N	=	Shaft speed
D	=	Bearing inside diamter
L	=	Bearing length
C	=	Diametrical clearance
H_p	=	Bearing heat loss
P	=	Bearing load
H_w	=	Windage losses
P	=	Gas density
v	=	Piston velocity
A	=	Piston frontal area
C_f	=	Friction coefficient
H	=	Oil pump power
Q	=	Oil pump flow
K	=	Oil pump efficiency
C_d	=	Drag coefficient
V	=	Velocity

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