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Horizontal Scroll Compressor for Transport Applications

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ABSTRACT

With the need for increasing vehicle efficiency, reliability and refrigerant containment, vehicle manufactures are considering electric powered accessories as a replacement for current engine driven components. Electric hermetic and semi-hermetic air conditioning and refrigeration compressors are well suited for this application by providing the potential for both improved efficiency and capacity control along with the long life required by many transport applications. To meet the needs of this demanding application, a horizontal hermetic scroll compressor has been developed and tested over a broad range of operating conditions and speeds. The compressor design detailed in this paper utilizes a brushless permanent magnet (BPM) motor to provide high motor efficiency over a broad speed range while significantly reducing the size and weight of the motor. In addition the design utilizes many of the same components as a vertical scroll compressor but combines a suction gas cooled motor with a high-pressure oil sump. A mathematical model is presented for estimating the effect of high-to-low oil flow on compressor performance versus size, speed and operating condition. Volumetric efficiency is compared with a comparable high performance vertical scroll compressor.

1. INTRODUCTION

Compressors used for transportation applications are required to have a high resistance to shock and vibration, operate at both high ambient temperature and high pressure ratio test conditions, and be small and lightweight. In some applications such as automotive, a 2000 hour product life may be sufficient but for commercial vehicles and transport refrigeration, a product life of 10,000 to 40,000 hours may be required. To meet these higher end requirements, a compressor design generally requires a consistent oil supply and hydrodynamic versus splash or boundary lubricated bearings. Also, with increasing energy efficiency and environmental standards, both energy efficiency and refrigerant leakage control are becoming of increasing importance. To address both these needs, electric motor hermetic and semi-hermetic compressors should be considered as alternatives to typical open drive engine-mount compressors. In addition, when coupled with an electronic drive package, these designs will operate over a broad speed and load range with efficient refrigerant capacity modulation. To achieve both high efficiency and compact size, BPM motors have been shown by Perevozchikov and Pham (2004) to provide excellent efficiency in a scroll type product. The compressor used in their study was a vertical type scroll whereas the subject of this investigation is a horizontal scroll compressor.

2. HIGH SIDE SUMP COMPRESSOR

The primary technology discussed in this paper will be referred to as High Side Sump (HSS) to denote the use of a high side or high pressure oil sump in conjunction with the use of a low side or low pressure shell containing a suction gas cooled motor. Most scroll compressors in production today are low side in that both the motor and oil sump are in the low pressure side of the compressor. A few scroll designs as well as a number of rotary compressors use a high pressure shell where both the motor and oil sump are in the high pressure side of the compressor. The HSS compressor design lies between these two existing approaches since the motor remains suction gas cooled for lower operating temperature, and the oil sump resides in the high side of the machine. To make this work effectively, it is generally necessary to place the scroll compressor in a horizontal orientation versus its normal vertical position. Also, it is necessary that the inlet to the scroll elements is positioned downward such that oil entering the low side of the compressor can be returned to the high side oil sump through the natural aspiration of oil with the gas entering the scroll elements. In this way oil in the low side of the compressor reaches an equilibrium or residual level while the bulk of the oil resides in the high side oil sump.

HSS technology has been utilized in the past for compressors used with high heat-of-compression (high specific heat ratio) gases such as air, helium and natural gas. Both screw and rotating vane compressors, for example, inject a high quantity of oil into the compression process to control discharge gas temperature to approximately 100ºC maximum. The heated high pressure oil is then typically passed through an oil cooler before being returned to the compressor. For refrigerant applications of these compressor types, the oil flow is reduced to the amount needed to lubricate bearings, and an oil cooler is not generally required. A scroll compressor developed by Elson and Butler (2003) utilized a HSS design to compress natural gas coming from oil and gas wells. In this case the high pressure oil sump was separate from the compressor rather than integral to the compressor as will be described below.

A main advantage of the HSS scroll compressor is its use as a horizontal compressor in applications such as transportation where compressor size and height limitations exist, and system packaging constraints allow minimal air cooling of the compressor shell. With a suction gas cooled motor, the HSS scroll compressor can perform without air cooling during both high ambient and high load operation while high side shell compressors have more difficulty operating due to the motor cooling coming from hot discharge gas.

With both HSS scroll compressors and with other HSS compressors, the oil flow for lubrication is dependant on the compressor operating pressure differential and not on the speed of the compressor. In fact, the oil supply pressure in a HSS compressor is typically several orders of magnitude more than the oil pressure of a typical centrifugal oil pump found in a low side sump compressor. This insures a high pressure lubrication supply to the inlet of the eccentric shaft lubrication passageway, and subsequent lubrication of the compressor bearings over the broad speed range required to meet the highly variable system capacity needs of a transportation system. The supply pressure does vary some with compressor operating condition but it is still many times greater than that produced by centrifugal designs in vertical compressors. The scroll compressor used in this study has demonstrated reliable operation over the speed range of 1200 rpm to 6000 rpm, for a turn down ratio of five.

A final consideration for the application of HSS technology is the control of the amount of oil allowed to flow from the high to low side of the compressor. This oil flow will result in both a loss in thermal efficiency due to suction gas heating and a power increase due to the pumping of oil from the low to high side. The oil pumping power loss is small relative to the potential loss of thermal efficiency but both losses can be controlled by understanding the impact of oil flow on compressor efficiency as a function of compressor size, speed, operating condition and refrigerant choice.

3. OIL FLOW AND LUBRICATION

A schematic illustrating the refrigerant and oil flow process is shown in Figure 1 below.

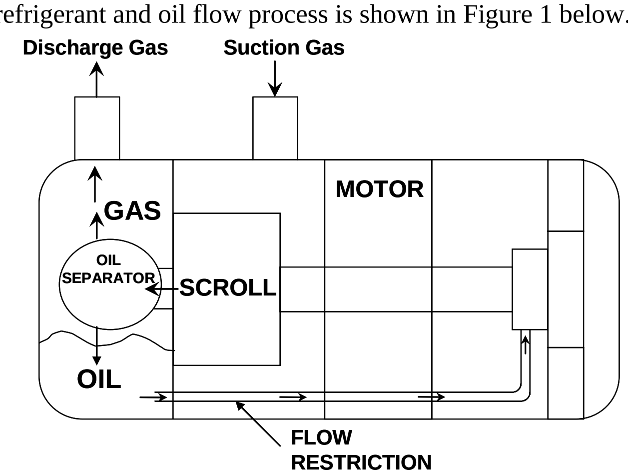

Figure 1: Schematic of Oil Flow

Starting with the oil contained in the high pressure oil sump, oil flows through a restriction in the oil passageway to the end bearing of the scroll compressor. The oil flow rate is dependant on both the operating compressor pressure differential and the designated restriction in the oil passageway. Since oil flow is limited, oil entering the oil passageway of the compressor shaft is not restricted, allowing the oil pressure inside the shaft to approach the compressor suction pressure. Once oil is in the shaft, centrifugal pressure generated by the rotating shaft (assumes an eccentric oil passageway) moves oil to the scroll bearings as is standard practice with low side sump vertical compressors. After this oil serves the function of lubricating all required bearing surfaces, it returns to the bottom of the compressor shell where the oil is scavenged or aspirated by the refrigerant gas and subsequently carried through the compression process. At discharge this oil/gas mixture is then separated such that the oil flow leaving the compressor is less than 0.5% by weight. A low oil circulation rate is important for two reasons: 1.) oil leaving the compressor will ultimately transfer heat to the system evaporator and reduce system efficiency, and 2.) high oil loss from the compressor will deplete the oil sump necessary to maintain a stable oil supply to bearings. In a compressor oil system designed to meet this criteria, the oil flow through the scroll elements may be 10 or more times the oil allowed to exit with the discharge gas.

4. THERMAL MODEL

To understand the impact of return oil flow on the thermal efficiency of the compressor, a thermal model was developed to estimate the net heating effect of high side oil entering the compressor low side and mixing with the much cooler refrigerant gas returning through the compressor inlet. This model includes the prediction of temperature for the gas and oil mixture both entering the scroll elements and exiting the discharge port of the scroll. The model is intended to assess only the effect of the oil and is not intended to be a complete thermal model for the compressor. For this reason each solution of the proposed model is limited to a specific operating condition of the compressor where measured performance data can be obtained for the case of zero oil flow. In this analysis, we use the comparable vertical compressor test data to define the zero oil flow case.

To model the affect of oil flow on compressor performance, a heat balance was formulated to compute the heat transfer to the refrigerant gas from the oil and other internal heat sources illustrated in Figure 2 below.

$$Q_R = Q_O + Q_I - Q_S \tag{1}$$

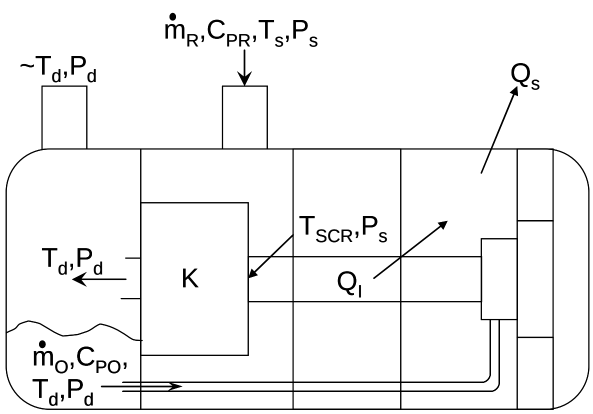

Figure 2: Heat Transfer Model

Where Q_R, Q_O, Q_I and Q_S represent respectively the heat added to the refrigerant, the oil heating, internal heat from motor and bearings and the heat loss through the shell. Q_I-Q_S can be approximated as a constant and identified for each test condition using vertical compressor data as the zero oil flow case. Assuming the refrigerant behaves as an ideal gas, the following may be written for a vertical, or zero oil flow compressor.

$$Q_I - Q_S = Q_R = \dot{m}_R C_{PR}(T_{SCR} - T_S) \tag{2}$$

Where $\dot{m}_R$, C_{PR}, T_{SCR} and T_S represent mass flow rate, specific heat, and temperature of the refrigerant at the locations designated in Figure 2. From the vertical compressor baseline, the volumetric efficiency due to thermal heating can be measured as:

$$\eta_{VOL} = \frac{\dot{m}_{R(ACTUAL)}}{\dot{m}_{R(THEORETICAL)}} \cong \frac{T_S}{T_{SCR}} \qquad \textbf{(3)}$$

Where $\dot{m}_{R(THEORETICAL)}$ designates the theoretical mass flow at the given condition that would equate to 100% volumetric efficiency.

Then, Equation (2) becomes:

$$Q_I - Q_S = \dot{m}_R C_{PR} T_S \left(\frac{1}{\eta_{VOL}} - 1 \right) = C_1 \qquad \textbf{(4)}$$

Where C_1 is a constant for each operation condition.
Equation (1) becomes:

$$\frac{\dot{m}_{R(THEO.)} T_S}{T_{SCR}} C_{PR}(T_{SCR} - T_S) = \dot{m}_O C_{PO}(T_d - T_{SCR}) + C_1 \qquad \textbf{(5)}$$

Where $\dot{m}_O$, C_{PO} and T_d represent mass flow rate, specific heat, and temperature of the oil at the locations designated in Figure 2. In this equation, it is assumed that the oil temperature is equal to the discharge temperature, T_d. Since Equation (5) contains 2 unknowns, T_{SCR} and T_d, an additional equation is needed to define the temperature of the refrigerant/oil mixture during compression. During this compression it is assumed the oil and gas are the same temperature due to instantaneous heat transfer from the gas to the oil. Then, for the compression process:

$$\dot{m}_O C_{PO}(T_{SCR} - T_d) = \dot{m}_R C_{PR}(\Delta T_R) \qquad \textbf{(6)}$$

Where ΔT_R is the temperature increase or decrease relative to the compression temperature obtained from the zero oil flow or vertical compressor base case.

To model the compression process, an ideal gas equation for a polytropic process is used to "fit" the actual compression process with a polytropic coefficient determined from the zero oil flow case. The following equation then approximates the discharge temperature of the two-phase compression process.

$$T_d = T_{SCR} \left(\frac{P_d}{P_S} \right)^{\frac{k-1}{k}} + \Delta T_R \qquad \textbf{(7)}$$

Combining Equations (3) and (6) with Equation (7) yields the following expression for discharge gas temperature.

$$T_d = \left(\frac{T_S}{\eta_{VOL}} \right) \left(\frac{P_d}{P_S} \right)^{\frac{k-1}{k}} + \left(\frac{\dot{m}_O C_{PO}(T_{SCR} - T_d)}{\dot{m}_R C_{PR}} \right) \qquad \textbf{(8)}$$

Where P_d, P_S and k represent respectively the absolute discharge pressure, absolute suction pressure and a polytropic compression coefficient. For each operating condition, k is computed by setting $\Delta T_R = 0$ and using vertical compressor test data to define η_{VOL}. .

To assess the affect of increasing oil flow for each operating condition Equations (4) and (8) are used to compute the vertical compressor constants C_1 and k. Next, $\dot{m}_O$ is varied and Equations (5) and (8) are solved by iteration, with the help of Refprop 7.0 (2002), for T_{SCR} and T_d assuming T_{SCR} equal to vertical data for the first iteration. Finally, using Equation (3), a new value for the horizontal compressor volumetric efficiency can be obtained for each value of $\dot{m}_O$. This new value for $\dot{m}_R$ is compared to the preceding value for $\dot{m}_R$ until convergence is attained for each value of $\dot{m}_O$.

5. RESULTS – THEORY AND EXPERIMENT

The volumetric efficiency reduction in a horizontal compressor as compared to a vertical compressor is largely dependent on two variables, the amount of oil flow and the temperature of this oil. This affect is shown in Figure 3 below as a plot of volumetric efficiency versus the dimensionless parameter, R.

$$R = \dot{m}_O C_{PO} \big/ \dot{m}_R C_{PR} \qquad \textbf{(9)}$$

Shown in Figure 3 is the theoretical affect of R at several common air conditioning operating conditions using refrigerant R134a. Also shown in this figure is a best fit line to experimental data showing accuracy to within about 1%. In this chart, vertical compressor volumetric efficiency is represented by the vertical axis where R equals zero. Obviously, a large value for R is not desirable but Figure 3 does show the affect of increasing R is less with a lower system temperature difference as is typical for both high efficiency systems and average ambient operation versus maximum ambient temperatures.

The selection of a value for R requires a compromise between best performance and sufficient lubrication flow to the compressor bearings. An analysis can be performed to estimate the oil flow required for each bearing of a specific compressor design, and then use this result to set the allowed level for oil mass flow. However, as a practical matter, variations in bearing clearance and refrigerant dilution and disruption of oil will require more oil than the minimum to insure reliable compressor operation for all modes of operation. In this regard, theoretical estimates of oil requirements and compressor life testing have demonstrated values of R as low as 0.03 can allow reliable compressor operation at the highest operating speed.

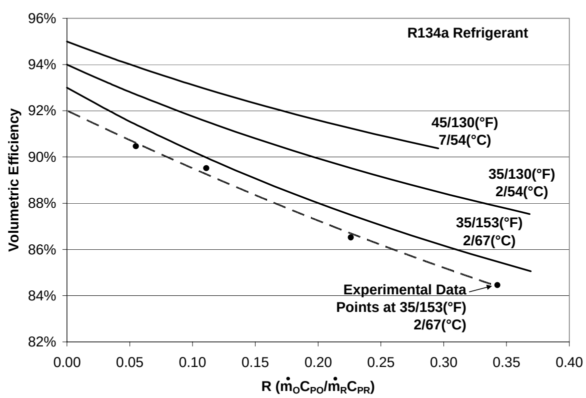

Figure 3:
Affect of Oil Flow and Heating on Volumetric
Efficiency at Maximum Speed

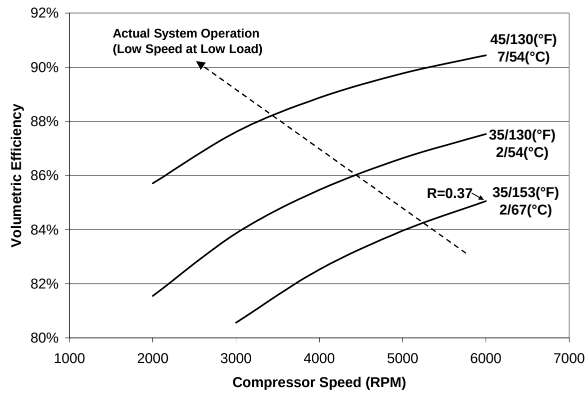

Figure 4:
Affect of Speed on Volumetric Efficiency -
Constant Oil Flow at an Excess Level

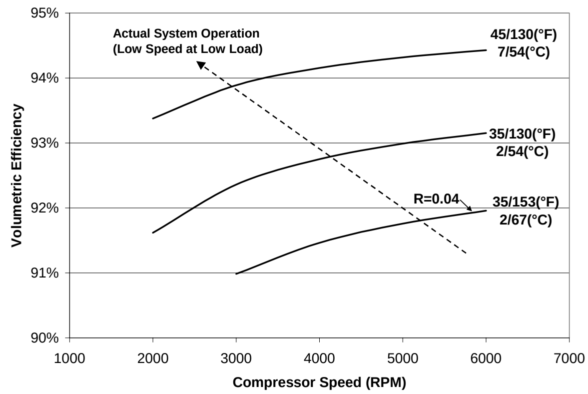

Figure 5:
Affect of Speed on Volumetric Efficiency -
Constant Oil Flow at an Optimum Level

Two figures are presented to illustrate the impact of oil flow on volumetric efficiency. Figure 4 shows a worst case, excess oil, scenario with a value of 0.37 selected for R at high speed and at maximum operating temperature difference. A similar plot with R restricted to an optimum level of 0.04 shows a dramatic improvement in compressor efficiency in Figure 5. The oil flow rates in both cases were held constant throughout speed and operating condition variations since this parameter is relatively independent of both operating speed and operating condition. Figure 4 shows efficiency reduction is significant with both higher operating temperature difference and lower speed but the low speed loss is less at lower temperature difference where low compressor speed is needed to match a reduced cooling requirement. Also, any loss of volumetric efficiency helps to further improve the effective turndown ratio for the compressor. The same types of efficiency reductions are seen at the optimum oil flow levels shown in Figure 5, but to a much lesser extent.

In addition to its affect on volumetric efficiency, the oil flow rate also changes the gas discharge temperature. This temperature change is not significant at low values of R and is also relatively minor at high values of R. This is due to excess oil flow first increasing the gas temperature entering the scrolls and then subsequently cooling the compressed gas during the compression process. Thus, the temperature of the discharge gas in the horizontal compressor is not significantly different from that of the vertical compressor as shown in Figures 6 and 7.

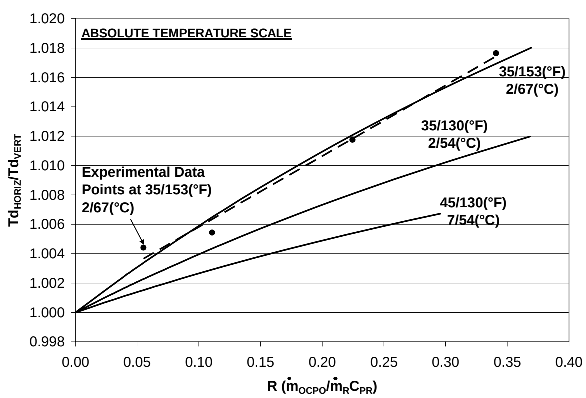

Figure 6:
Affect of Oil Heat on Compressor
Discharge Temperature

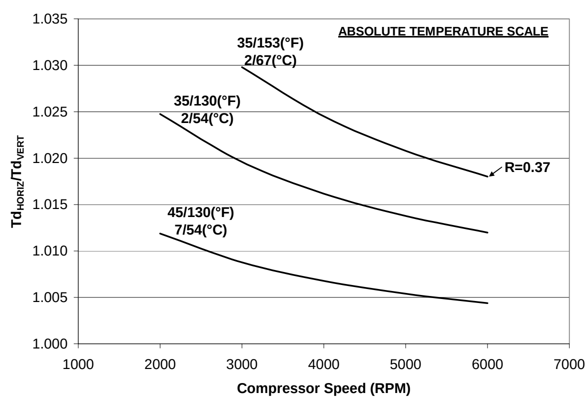

Figure 7:
Affect of Oil on Discharge Temperature
Constant Oil Flow at an Excess Level

6. APPLICATION CONSIDERATIONS

A low profile horizontal hermetic compressor is beneficial for transportation applications where electric power is available but space and height are limited. For example, a horizontal compressor can be built as a rooftop packaged unit where the compressor, condenser and evaporator are together in one unit thus eliminating refrigerant lines for a remote compressor. This approach results in both minimal refrigerant charge and a factory charged unit requiring only a power supply. Low vibration and noise is also important for this application due the proximity of the system to vehicle passengers. Scroll compressors in general have inherent low vibration with minimal torque reaction at low speed. Also, with the HSS design presented above, the oil and oil separator reduce discharge gas related noise below that of the comparable vertical design.

For transportation applications, shock and vibration loads require a compressor with rigid internal construction, and an external mounting system insensitive to vehicle motion. In addition, tilt and roll up to 30º, or abrupt start/stop operation can occur with some mobile applications. This requires the compressor lubrication to be robust against loss of lubrication including loss of oil pumping. With the HSS design presented above, the oil supply to bearings is dependant on the operating pressure differential of the system and has been demonstrated to continue to supply oil for tilt and roll angles up to 30º.

Liquid refrigerant tolerance is another requirement of all compressors including transportation applications. With the HSS horizontal compressor presented here, liquid refrigerant returning to the low side of the compressor does not dilute the oil feeding the bearings as is the case with a conventional low side sump. Particularly with positive displacement oil pumps, refrigerant flood back can result in “vapor lock” in the oil pump and a loss of continuous oil pumping. For the case of a compressor starting with liquid refrigerant migration to the oil sump, the low side oil sump will see more oil dilution than the high side sump. Initial life testing with liquid refrigerant has shown the HSS design compressor to be capable when dealing with typical system refrigerant induced problems.

7. SUMMARY

A horizontal scroll compressor concept has been presented for primary use with transportation versus stationary air conditioning applications. The concept utilizes both a high side oil sump and a low side shell for optimal suction gas cooling of the hermetic motor. The high side oil sump allows a modified vertical scroll compressor to be built as a horizontal model with the low profile advantage required by some mobile applications. A mathematical model is presented and employed to estimate the impact on compressor volumetric efficiency of the high side oil returning to the compressor low side. The control of this return oil is discussed and experimental and theoretical results are presented to demonstrate a good correlation of the model with theory. Model results are given for compressor volumetric efficiency versus oil flow, compressor speed and operating condition. With consideration for broad operating range of a variable speed compressor, good performance is shown for the horizontal compressor relative to the vertical compressor. A parameter for relative oil flow versus refrigerant flow is also identified to help define the best compromise to achieve both bearing lubrication and high performance in the horizontal design.

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