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Lubrication System for a Low-Side Horizontal Scroll Compressor

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LUBRICATION SYSTEM FOR A LOW-SIDE HORIZONTAL SCROLL COMPRESSOR

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ABSTRACT

Low-side hermetic scroll compressors are designed to operate in the "vertical" orientation. Typically, the scroll elements are located above the drive motor. Lubrication for bearings and other rubbing surfaces is provided by pumping oil through the drive shaft from a sump located below the drive motor. In this orientation, the height of the compressor can be approximately twice its diameter. For many applications, minimizing the compressor height is an important consideration since the compressor can dictate the overall system package height. A large height reduction can be achieved by horizontal orientation of the compressor. The principal obstacle presented in the implementation of such a low-side horizontal scroll is the lubrication system. Accordingly, an oil pumping mechanism to facilitate horizontal compressor operation has been developed. This paper describes the lubrication concept and operating principles together with the results of both bench- and compressor-development tests.

NOMENCLATURE

Parameters:

f	Friction factor
m_c	Mass of fluid in pump chamber
t	Timing parameter
v_c	Pump chamber volume
A	Area
C	Loss coefficient
D	Oil gallery line diameter
L_1, L_2	Oil gallery line lengths
P	Pressure
S	Streamwise length
V	Velocity
Z	Height relative to datum
ϱ	Density
θ	Crank angle

Subscripts:

0–4	Station numbers (see Fig. 5)
a	Pump entrance location
b	Pump exit location
c	Pump chamber
p	Piston
T	Turn section

INTRODUCTION

A typical low-side scroll compressor is shown schematically in Fig. 1. Principal components include a drive motor, drive shaft with eccentric crank, a fixed scroll element, and a moving (orbiting) scroll element. Vapor compression is achieved as a fluid contained within pockets created by meshing of the fixed and orbiting scroll wraps is displaced from the scroll outer pockets to the inner pockets with continuously decreasing volume. Among the characteristics that are attractive vis-a-vis reciprocating compressors are high efficiency, fewer moving parts, low noise, and low vibration.

The compressor in Fig. 1 is shown in the "vertical" position. In this orientation, lubrication for the shaft bearings, orbiting scroll bearing, anti-rotation device, thrust surfaces, etc., is typically supplied using a passive centrifugal pump incorporated in the drive shaft. Oil from the sump enters

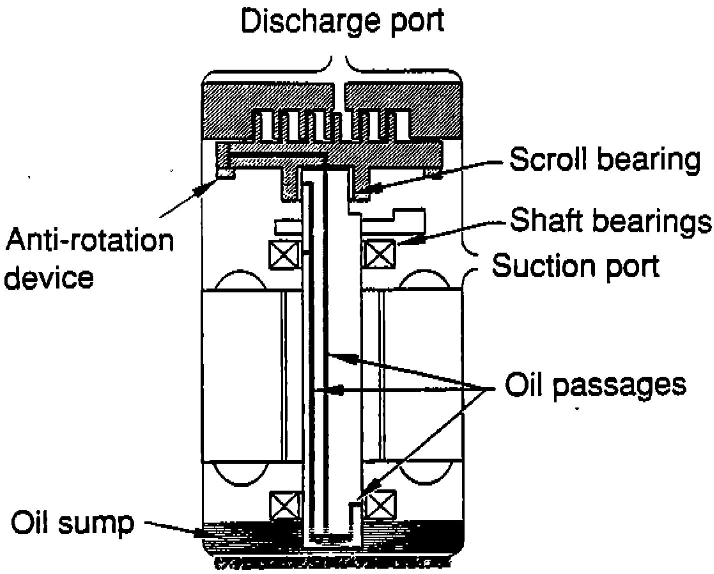

Fig. 1 Typical Scroll Compressor in Vertical Orientation

the pump through an orifice in the bottom of the shaft as shown in Fig. 1. A small increase in oil pressure is then generated as the oil is accelerated radially and enters the vertical oil delivery passages that supply the required oil to the components mentioned above. This relatively simple, passive lubrication system is an important reason why low-side scroll compressors are designed to operate in the vertical position. In this orientation, the compressor height-to-diameter ratio is typically two or greater. For many applications, the height of the compressor is a primary factor because of packaging considerations. Very often, the height of an air conditioning, refrigeration, or heat pump unit is much more important than width or depth. Thus, a distinct advantage is realized by installing the scroll compressor horizontally. The oil pump/lubrication system herein described facilitates scroll operation in a horizontal orientation.

DESCRIPTION OF PUMPING MECHANISM

The principal features of this simple, novel mechanism are

1. the utilization of the planar motion of the orbiting scroll to drive the pumping mechanism, and
2. the incorporation of oil distribution passages within the orbiting scroll and drive shaft such that no new external distribution lines are required.

The essential features of the device and the method of operation are described in the following paragraphs.

The upper portion of the scroll compressor is shown in Fig. 2 with the compressor re-oriented to the horizontal position. The pump mechanism, shown below the orbiting scroll, comprises a cylinder that is fixed to the shell (outer wall) of the hermetic enclosure and a piston that reciprocates within the cylinder. The assembly is positioned such that the piston is driven by the motion of the orbiting scroll. In Fig. 2, the orbiting scroll is shown in the full down position, and thus the piston is fully retracted within the cylinder. Also shown in Fig. 2 is the relocated oil sump

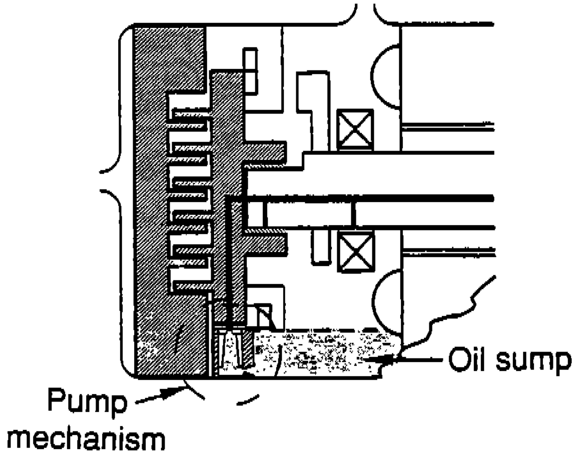

Fig. 2 Scroll Compressor in Horizontal Orientation

and the oil entrance passage to the lower side of the cylinder. A directional nozzle (or fluidic diode) is located within the entrance passage that allows oil to pass freely into the lower region of the cylinder but severely restricts flow in the reverse direction.

Enlarged views of the oil pump assembly corresponding to the fully retracted and fully extended piston positions are shown in Figs. 3a and 3b, respectively. During operation, oil is drawn

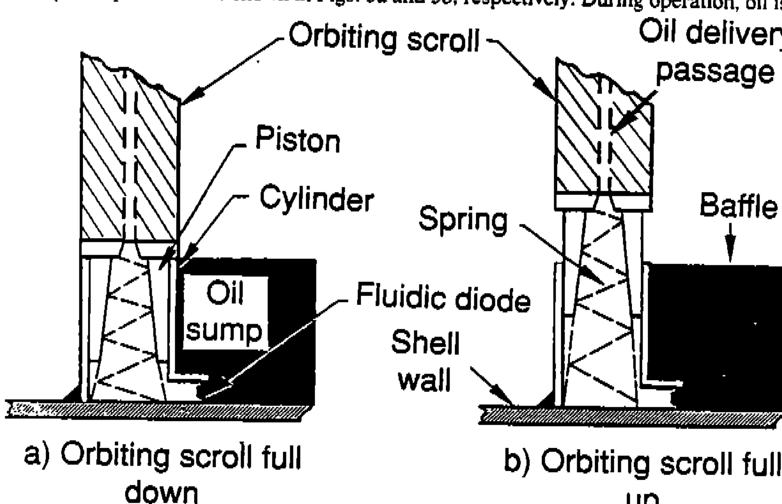

Fig. 3 Oil Pump Mechanism

into the lower region of the piston through the suction passage as the orbiting scroll moves from the full down to the full up position, Figs. 3a and 3b, respectively. Since there is no physical attachment of the piston to the orbiting scroll, a light spring is incorporated to assist the piston movement during this part of the cycle to ensure contact with the orbiting scroll. As shown, orifices are located in both the piston and the orbiting scroll such that, as the orbiting scroll moves from the full up to the full down position (Figs. 3b and 3a, respectively) the oil is forced through the orifices in the piston and into the mating oil delivery passage in the orbiting scroll. During this part of the cycle, oil is restricted from flowing back through the suction passage by means of a fluidic diode. The oil delivered to the orbiting scroll passage is now under a small positive pressure and can be routed to provide lubrication to the scroll thrust surface, anti-rotation device, and to the top of the drive shaft as shown in Fig. 2. From this location, the oil is delivered to the orbiting scroll bearing and the shaft support bearings in a manner similar to that used in a vertical compressor installation where the oil is pumped from the bottom of the shaft, as explained previously.

Since the motion of the orbiting scroll is two-dimensional, the passages in the piston and orbiting scroll must be properly positioned. Partial end-views of the pump assembly are shown in Fig. 4 to illustrate the passage location and additional operational details. This view shows that an enlarged "foot" has been included on the piston to accommodate the passage orifices and to increase the sliding contact surface area that interfaces with the flat surface of the orbiting scroll. In Fig. 4a, the orbiting scroll is shown at 0-deg. crank angle corresponding to the full down position of the orbiting scroll; i.e., Fig 3a. At this crank angle, the piston is fully retracted and the orifices in the piston and orbiting scroll are mis-mated; i.e., the passages are valved off. As the orbiting scroll moves toward 90-deg. crank angle (Fig. 4b), oil is drawn into the lower region of the piston, as

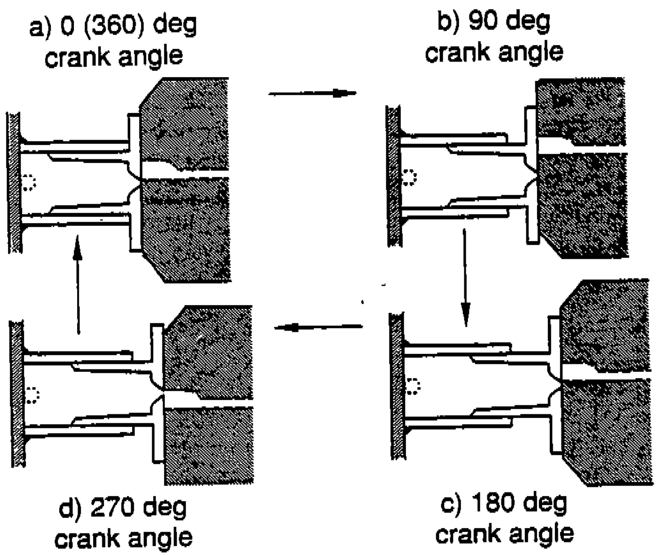

Fig. 4   Partial End View of Oil Pump Mechanism

previously explained. The passages remain mis-mated during this process to ensure that the lower region of the piston is charged with oil taken from the oil sump rather than pulled back from the oil delivery system. The oil charging process continues as the orbiting scroll moves to the 180-deg. crank angle position (Fig. 4c) at which point the scroll is in its full up position and the piston is fully extended, as in Fig. 3b. Further motion of the orbiting scroll towards the 270-deg. crank angle position (Fig. 4d) will cause the passages in the piston and orbiting scroll to open as the piston is driven into its cylinder, causing oil to be pumped into the delivery system. The oil pumping cycle continues as the scroll moves to the 360 (or 0) deg. crank angle position (Fig. 4a) and the cycle is repeated.

The system described above supplies lubrication under a positive pressure during approximately 180-deg. of each 360-deg. cycle. If desired, the pump can be positioned circumferentially within the compressor such that the pressurized oil delivery occurs during the portion of the cycle when the bearing loads are highest. Two pumps could be included to provide a continuous supply of pressurized oil. Likewise, the volume of oil delivered per cycle can easily be controlled by proper sizing and positioning of the piston/cylinder assembly and/or by varying the efficiency of the diodes and piston orifice.

The lubrication device described above is intended to be representative of the basic concept. It is apparent that numerous variations could be formulated.

ANALYSIS OF PUMPING MECHANISM

An analytical model has been developed for the horizontal scroll lubrication system in order to identify key parameters and their influence on performance. A schematic diagram of the model is shown in Fig. 5. The conservation laws of mass and energy were used throughout to characterize

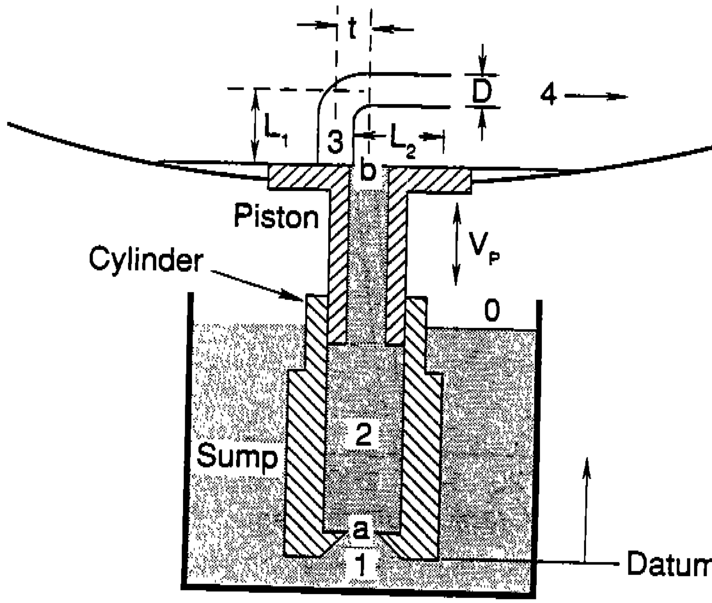

Fig. 5 Schematic Diagram for Pump Model

the internal behavior of the pumping system. The system features the use of hydraulic resistance components in order to reduce the number of moving parts and overall complexity. A directional nozzle type fluidic diode is positioned at the suction side of the pump and acts as a flow regulator. This component allows lubricating fluid to enter the pump chamber during the suction stroke while resisting the flow of fluid back to the oil sump during the discharge stroke. This action provides the requisite lubricating fluid to the oil gallery as described previously. However, some leakage back through the diode occurs during the pumping stroke decreasing the net efficiency. The purpose of this investigation was to develop a computer model that can be used to optimize the pump design and maximize overall efficiency.

The major assumptions used in the analysis are as follows:

1. The working fluid is incompressible.
2. Lateral fluid inertia is neglected in the discharge line.
3. The friction factors are based on steady state conditions.
4. The sump fluid level remains constant.

With reference to Fig. 5, the continuity equation for the pump can be expressed as follows: Note that in equation (1) the piston exit area A_b, is a function of crank angle, i.e., $A_b = A_b(\theta)$. For a given piston and oil gallery line size, this area variation can be controlled by offsetting the gallery

$$\varrho V_a A_a + \varrho V_b A_b = -\frac{dm_c}{dt} \quad (1)$$

center line from the piston center line according to a timing parameter, t, as shown in Fig. 5. The pump exit area characteristics for two values of this timing parameter are shown in Fig. 6. As previously mentioned, the motion of the orbiting scroll causes the gallery to pass across the piston, thereby increasing the exit flow area until a maximum is reached corresponding to their complete

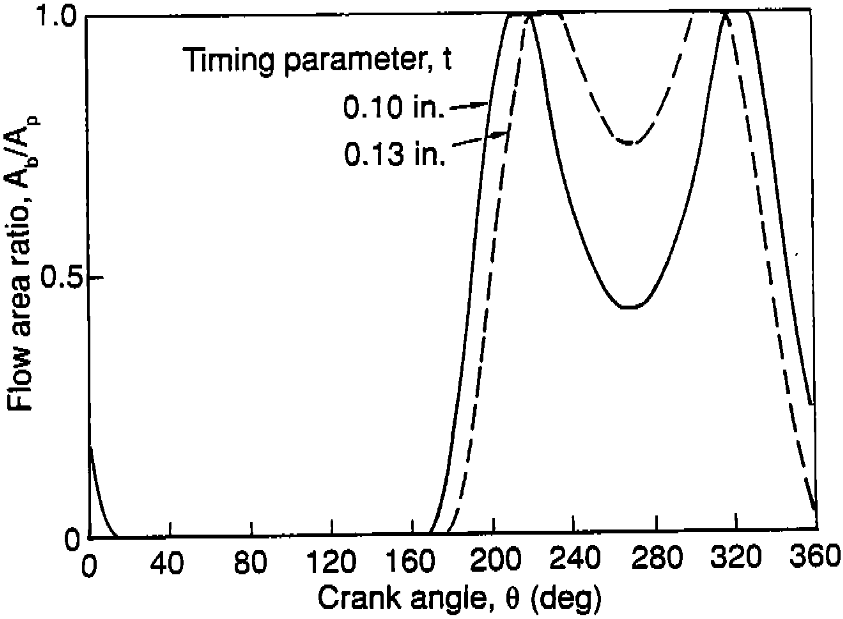

Fig. 6 Pump Exit Flow Area Characteristics

alignment. The effective exit flow area is then decreased as the motion continues until the cycle is reversed. It should be noted that the saddle in the area distribution curve can be eliminated by proper selection of the timing parameter if so desired. Defining the flow rate as positive out of the pump chamber, the rate of change of mass within the chamber is expressed as:

$$\frac{dm_c}{dt} = \frac{dm_c}{d\theta}\frac{d\theta}{dt} = \varrho\frac{dv_c}{d\theta}\frac{d\theta}{dt} \quad (2)$$

where v_c is the volume of the pump chamber.

The energy equation associated with the fluid diode is given as follows:

$$P_1 + \frac{1}{2}\varrho V_1^2 + \frac{1}{2}\varrho C_a V_a^2 = P_2 + \varrho g Z_2 \quad (3)$$

Similarly, the equation of motion for flow through the piston can be represented as:

$$P_3 + \frac{1}{2}\varrho V_3^2 + \varrho g Z_3 + \frac{1}{2}\varrho C_b V_b^2 = P_2 + \varrho g Z_2 \quad (4)$$

The parameters C_a and C_b in equations (3) and (4) are the loss coefficients associated with the fluid diode and the piston exit orifice respectively. C_a and C_b vary with flow direction and are estimated using experimental data and analytical approximations from Ref. 1.

For the discharge line, friction and unsteady effects must be accounted for, resulting in an energy balance of the form:

$$\int_3^4 \frac{\partial V_3}{\partial t} dS \pm \varrho \frac{f(L_1 + L_2)}{D} \frac{V_3^2}{2} + \frac{1}{2}\varrho C_T V_3^2 + P_4 = P_3 - \varrho g L_1 \qquad (5)$$

where $\pm$ denotes the flow direction and C_T is the turning loss coefficient.

Equations (1)-(5), along with the energy equation from 0 to 1, are solved to yield the flow rates into and out of the pumping chamber. The instantaneous flow rates are then integrated over one cycle to yield the net discharge flow from the pump.

Typical numerical results for the diode and orifice instantaneous flow rates are depicted in Fig. 7. During most of the suction stroke, the piston orifice is closed and the chamber is filled with

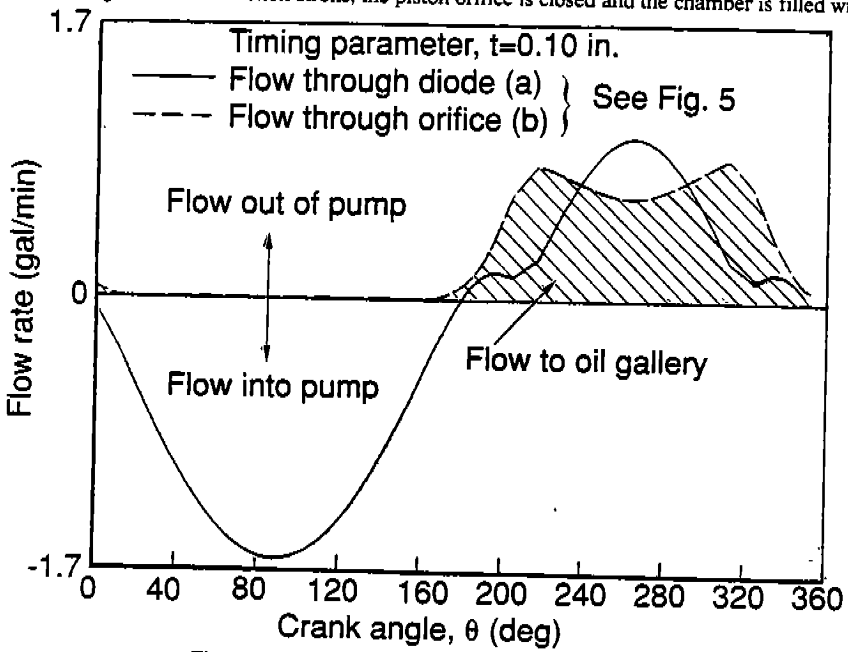

Fig. 7 Pump Mass Flow Rate Characteristics

lubricating fluid from the sump. The pressure losses through the resistance components dictate the flow rates for the pumping stroke. The volumetric efficiency is defined as the net discharge flow divided by the ideal flow based on the pump displacement volume. For the timing used, a volumetric efficiency of 30% is calculated. This parameter can be used in the optimization of the pump design. Instantaneous chamber pressures were calculated numerically and are shown in Fig. 8.

EXPERIMENTAL PROGRAM

For the experimental development program, the baseline oil pumping system was designed to deliver approximately the same oil flow rate as that of the conventional (vertical) centrifugal pump with an overall pumping efficiency of at least twenty percent. Primary emphasis was placed on achieving a simple design with a minimum of moving parts that would be easy to manufacture and convenient to package. Since the design was thought to be sensitive to the pump entrance nozzle (diode) design, three diodes were machined with area ratios of 0.15, 0.20, and 0.25. Subsequently, 'simple' diodes were fabricated by simply punching them out of flat sheet stock and limited hand

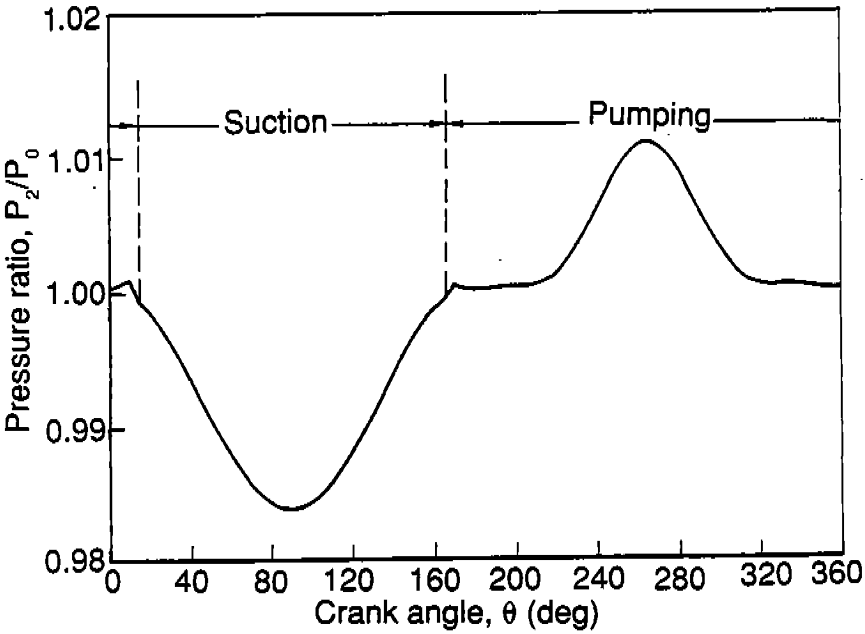

Fig. 8 Pump Pressure Characteristics From Model

forming of the divergent section. The pump piston was made from a low friction polyamide material. The pump discharge nozzle is formed by alignment of an axial hole drilled in the center of the piston and a radial hole drilled in the orbiting scroll base; no special nozzle shape was used. A groove was cut in the orbiting scroll base to provide a guide for the piston and a flat piston/scroll interface. A spring was designed to maintain piston contact with the scroll at all times with minimum lateral loading and minimum working stress.

Two series of tests were conducted to investigate the concept feasibility, document operating characteristics, and demonstrate the system on an operating scroll compressor.

Concept Feasibility Bench Tests

The bench tests were conducted using existing scroll compressor hardware installed in a bolted shell configuration which was open to atmosphere on the suction side (i.e., end closure removed). The test assembly is shown in Fig. 9. In order to maintain test flexibility and accommodate interchangeable components, the pump assembly was installed below the shell envelope as shown in Fig. 9. (A production design could be packaged within the shell envelope.) A temporary oil dam was attached to retain the oil in the shell; the pump support and cover assembly can be seen located on the underside of the shell. This assembly consists of a pump platform, which is bolted and sealed to the shell beneath the scroll, the pump cylinder support plate which mounts on the platform in such a way as to allow lateral adjustment of the cylinder centerline relative to the compressor shaft axis, and a pump cover attached to the support plate. This cover also functioned as a secondary sump for the oil at the suction orifice for the pump. By isolating this sump from the main sump and connecting it to an auxiliary sump outside the shell (see Fig. 9) it was possible to measure the oil flow through the pump by measuring the oil level in the auxiliary sump as a function of time. Instrumentation for this test series included a thermocouple and an ultraminiature pressure transducer in the main shaft bearing to monitor the condition of the bearing and its oil pressure characteristics.

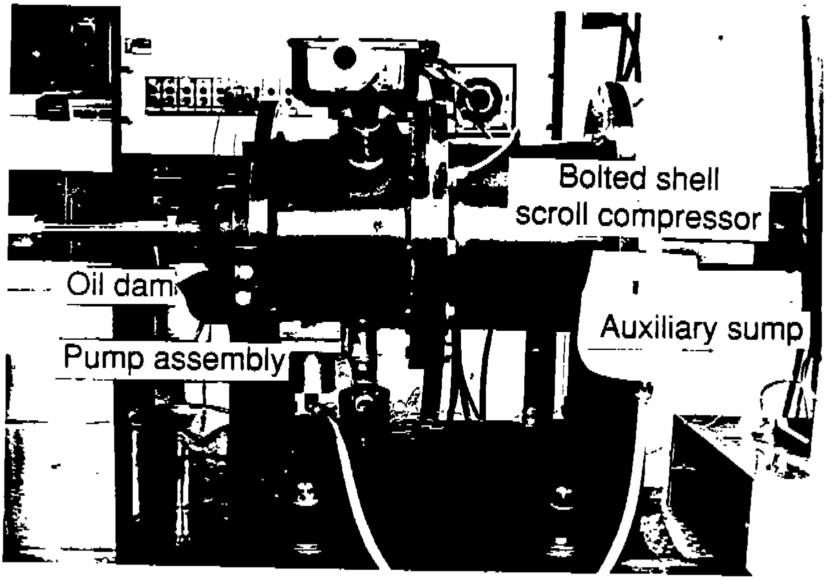

Fig. 9 Bench Test Configuration

Typical oil flow rate measurements are shown in Fig. 10 for various pump timing positions (see Fig. 5) and for various pump entrance diode area ratios and diode types (machined or hand-formed). Also shown in Fig. 10 are the predicted flow rates for a diode area ratio of 0.12.

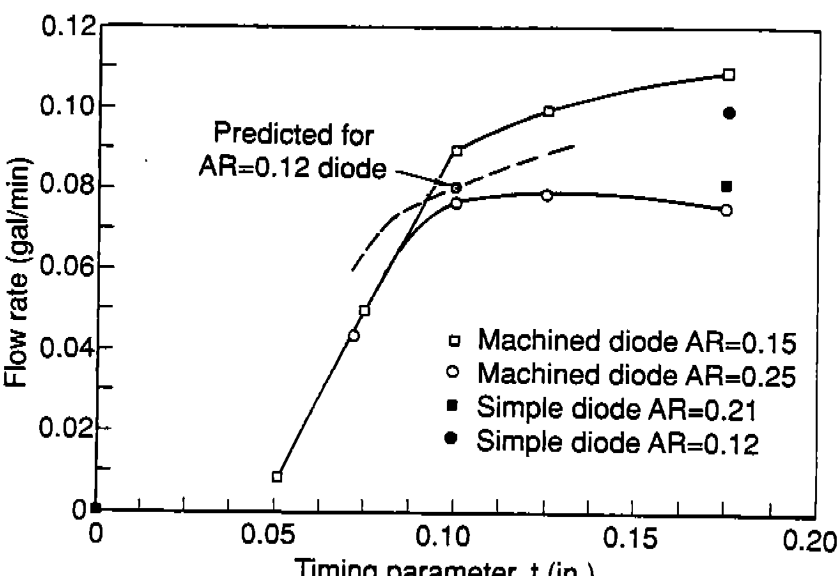

Fig. 10 Measured Flow Rate Characteristics

Both the magnitude and shape of the predicted curve are in substantial agreement with the experimental data.

Concept Demonstration Compressor Tests

Having shown the mechanical feasibility of the pumping device during bench testing, the same pump assembly was used for concept demonstration in an operating compressor configuration. The objectives were to exercise the pump for a significant time to check durability and wear and to monitor the pump performance and compressor efficiency at various compressor operating conditions. The same bolted shell arrangement was used except that the end closure piece was installed in place of the oil dam. Pressure transducers were installed in the main bearing and in the pump cylinder. The compressor was connected to a de-superheater test stand with its associated instrumentation.

During the initial tests, static pressure measurements within the shell showed an axial pressure differential of the order of 0.2 psi and excessive oil ingestion into the scroll elements. Therefore, a series of baffles were installed to control the oil level at the scroll station and reduce the oil ingestion to acceptable amounts. A typical trace of the pressure-time history measured in the pump chamber is shown in Fig. 11 for the compressor operating near the ARI condition. The suction and

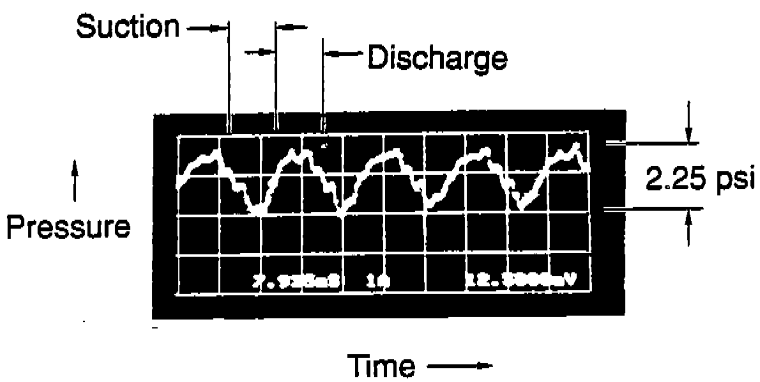

Fig. 11 Typical Pump Pressure Trace

discharge regions are indicated; the peak-to-peak pressure was approximately 2.25 psi. Comparison of the pump chamber pressure time histories from analysis and test, see Figs. 8 and 11, show an expected similarity in form and magnitude.

The compressor was subsequently operated over a wide range of suction and discharge conditions with no degradation in performance or evidence of wear.

CONCLUSIONS

1. A lubrication system has been developed to facilitate horizontal operation of a low-side scroll compressor. This results in a compact compressor that retains all the other advantages of the scroll compressor.

2. Bench tests and on-compressor tests have shown that the horizontal scroll lubrication system has lubrication characteristics equal to or better than the conventional 'vertical' oil system.

REFERENCES

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