Pressure Signatures of Damaged Valves

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PRESSURE SIGNATURES OF DAMAGED VALVES

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INTRODUCTION

A method of identifying failures in compressors was investigated. In particular, leakage failures due to poorly seated or cracked valves and worn cylinders were considered. It was determined that those failures could be detected and identified through the use of pressure transducer signatures. Specifically, the cylinder pressure signature profile for a satisfactory low side reciprocating compressor was compared to the profile of one with an induced failure. Not only could a failure be detected, but it could also be determined whether the failure was due to a faulty suction or discharge valve.

Since almost all refrigeration compressors are hermetically sealed, and are usually replaced when a failure is discovered, the use of pressure transducers to measure on-line cylinder pressure may or may not prove to be economical. However, larger and more expensive compressors stand to benefit from this investigation. For instance, the detection of a suction valve crack before occurrence of a catastrophic failure would be desirable by not only preventing further damage, but by adding to the safety of the machine.

EXPERIMENTAL APPROACH

Valve failures, due to incorrect seating or a crack were duplicated experimentally by drilling small holes in the valve reeds, as shown in Fig. 1 for a suction reed. Cylinder pressures were measured and compared. This was done for the discharge valve and the suction valve. Experiments were repeated for a number of different hole sizes. Holes were also drilled in the piston to simulate excessive piston leakage due to wear.

A typical hot gas loadstand as developed at the Herrick Laboratories in the late fifties was used, but a secondary or auxiliary compressor was installed in parallel to the primary one to keep the pressures at normal operating conditions for the hot gas cycle in the event of a failure in the primary compressor, as shown in Fig. 2. The reasoning behind this concept is that a compressor failure does not have an immediate effect on the refrigeration cycle due to the thermal load of the system. The hot gas cycle, on the other hand, will respond very quickly, thus giving different results. Furthermore, the primary compressor would not be able to reach the standard operating conditions because of substantial leakage.

FAILED DISCHARGE VALVE

The first test case was the compressor under normal operating conditions with no failures. The first failure was achieved by drilling a .035" hole in the discharge valve. The compressor was reassembled and run with the booster compressor by its side. The pressure-time relationship is shown in Figure 3 and compared to the trace of the perfect compressor. Furthermore, the pressure-volume curves are compared in Figure 4. The result can be explained with the help of the subsequent mathematical model or with intuition. As the cylinder gas compresses, the hole in the discharge valve leaks high pressure gas into the cylinder, thus adding more mass to the cylinder. The gain of mass in the compression cycle will contribute to a quicker rise in cylinder pressure. During the re-expansion phase, the continuing mass leakage into the cylinder sustains pressure longer.

The same effect, in a more pronounced form, is shown in Figure 3(b). In this case, the hole size was enlarged to about twice the area of the previous case (.053 inches diameter) in order to simulate a more advanced stage of failure.
Failed Suction Valve

The first induced failure for the suction valve was not to install a valve at all. This, of course, meant that the compressor could not pump to achieve the required operating conditions. Here, the secondary compressor kept the system pressures as prescribed. Needless to say, the cylinder pressure was unable to reach the discharge pressure and instead followed the suction pressure closely as shown in Fig. 5.

For the .035" hole in the suction valve, the cylinder signal had an identifiably narrower profile than the no failure case. This agreed with what one would expect since as the cylinder gas compresses, some of the gas escapes through the hole in the suction valve, thus reducing the cylinder pressure. The pressure-time and pressure-volume comparisons are shown in Figure 6(a) and Figure 7(a).

The more advanced stage of failure, simulated by a .053" hole shows the same effect as the .035" hole but more pronounced. The drop in amplitude compared to the no failure case is quite apparent. What was also noticed was that the cylinder pressure dropped below the discharge and then started to rise again, only to drop before reaching it. The plots appear in Figures 6(b) and 7(b).

Piston Leakage

A hole of .053" was drilled into the piston of the compressor to simulate the gas leakage due to excessive wear of the cylinder or piston. One would expect the same or similar results to the suction valve failure case since both eventually leak back into the low side shell. This was the case and is shown in Figs. 8(a) and (b).

THEORETICAL APPROACH AND RESULTS

The computer simulation was based on reference [3]. The only basic theoretical difference to earlier work is that the leakage holes were included in the effective flow area calculations. They were modeled as orifices in parallel, as shown in Fig. 9. The equivalent force area was also adjusted.

Theoretical results for the .035 inch and .053 inch holes in the discharge valve are shown in Fig. 10. The cylinder mass gain increases cylinder pressure, as measured previously. Qualitative and quantitative agreement with the experimental results is good.

When a .053 inch hole is in the suction valve, the predicted result of Fig. 11 indicates that cylinder pressures are relieved because of the net loss in cylinder mass. If the same size hole is used to represent leakage through by the cylinder, the same result is obtained since the model is an identical mathematical description of the suction failure case. Thus, there is presently some difficulty in distinguishing a failed suction valve from a leaky piston.

CONCLUSIONS

The approach for detecting valve failure or excessive piston leakage was to obtain the pressure signals for a failure free compressor and then compare them to a compressor with an induced failure. From the obtained information, valve failures cannot only be detected but also identified, with the exception that piston leakage and suction valve cracks produce very similar effects.

To help explain a few of the experimental results, a computer model to simulate the above mentioned failures was also created. It allowed not only the approximate prediction of the measured data, but also gave an insight of what parameters were crucial to the compressor's performance. Furthermore, one case whose measurements seemed wrong at first was later explained by the simulation. Naturally, a more detailed model would have lead to better agreement between the experimental and theoretical data, but enough was achieved by demonstrating even a simple model. The suction and discharge leakage failures could easily be identified from the cylinder pressure profile, i.e., a broader signal was obtained for discharge failures and a narrower signal was obtained for suction failures. This was confirmed by both the measured and calculated results. However, it was not possible to differentiate between suction valve failure and piston leakage, since both had a similar narrowed profile.
ACKNOWLEDGEMENT

The support of Whirlpool Corporation was appreciated.

REFERENCES AND FOOTNOTES

1. Now with TRW, Redondo Beach, California


Figure 1  Holes in Suction Reeds Simulating Partial Failure

Figure 2  Hot Gas Load Stand with Secondary Compressor

Figure 3  Comparison of Normal Discharge Valve with Failed Valve:
(a) 0.035 inch diameter hole
(b) 0.053 inch diameter hole
Figure 4  Comparison of Normal Discharge Valve with Failed Valve:
(a) 0.035 inch diameter hole
(b) 0.053 inch diameter hole

Figure 5  Comparison of Normal Suction Valve with Suction Valve Removed
Figure 6  Comparison of Normal Suction Valve with Failed Valve:
(a) 0.035 inch diameter hole 
(b) 0.053 inch diameter hole

Figure 7  Comparison of Normal Suction Valve with Failed Valve:
(a) 0.035 inch diameter hole 
(b) 0.053 inch diameter hole
Figure 8  Comparison of Normal Piston with Failed Piston Simulated by a 0.053 inch Diameter Hole
Theoretical Comparison for Normal Discharge Valve with Failed Valve:

(a) 0.035 inch diameter hole
(b) 0.033 inch diameter hole
APPENDIX A: Late Papers of the 1986 International Compressor Engineering Conference - at Purdue
ABSTRACT

Selection of shaft geometry is critical to the overall performance of a rotary compressor. Consideration is given to the development of rotary compressor shaft geometries to optimize efficiency and reliability.

The cylinder to piston dynamic clearance is of prime importance with respect to leakage losses. An analysis of bearing film thicknesses and shaft deflection was conducted to fully evaluate the impact of the clearance between cylinder and piston under operating conditions. Trade-offs between shaft stiffness and the impact on efficiency due to increasing component size is discussed.

The characteristic bearing edge loading found in rotary compressors is a major reliability concern. Examination of the causes of edge loading is made and design responses evaluated. Matching shaft stiffness to local bearing stiffness is critical.

Reliability and performance testing has confirmed the validity of the design approach selected.

DIMENSIONAL OPTIMIZATION

As rotary compressor technology becomes increasingly widespread, more is understood about the complex interrelationships between component geometries and factors such as efficiency, weight, machined surface area, bearing loads, vane P-V's, radiated noise, etc.
Each new rotary design represents an attempt to balance the various choices in order to most cost effectively meet the objectives established for the product.

To initially establish the base design of a rotary compressor, a computer model was created to develop and analyze alternate geometries for the various capacities under consideration. Figures 1.A and 1.B show schematics of the general construction of the rotary compressor. The model examined a number of cylinder bore to height ratios, as well as various bearing diameters. Based on previous investigations, limits were established for maximum vane P-V's and minimum rolling piston face widths $T_p$. A nondimensional loss function was established to facilitate evaluation of the leakage and heat transfer losses. The predicted response of the objective loss function to variation of cylinder bore and height while maintaining constant shaft diameters is shown in Figure 2 for an 18,000 btu/h compressor.

It is clearly seen from Figure 2 that it is desirable to minimize the diameter, $2R_c$, and height, $H_c$, of the cylinder in order to minimize leakage and heat transfer losses. In addition to providing for improvement in efficiency, the trend towards the reduction in pump component size commensurate with the decrease in cylinder diameter and height would result in lower costs. As is examined later, further benefits accrue from reducing the dimensions of the pump assembly associated with reductions in shaft deflection and bearing eccentricities.

The opportunity for continuing the size reduction of pump componentry is finally checked by many factors. The most key of these is the greater vane extension required to achieve the necessary pump displacement as the cylinder height and diameter are decreased. The greater vane travel during each revolution and hence greater acceleration associated with the large extension configurations leads to vane P-V's not sustainable without a decrease in reliability. Figure 3 delineates the tracking of average vane P-V's with the various geometries evaluated. Limiting the height reduction in addition to the vane P-V is the available space for the suction inlet. The diameter shrinkage is limited additionally by the minimum rolling piston face width $T_p$ and minimum acceptable bearing diameters. A minimum piston face width is required to prevent excessive leakage to the suction chamber and the compression chamber (while still at intermediate pressure) from the
internal cavity of the piston which remains at discharge pressure. Minimum shaft dimensions are needed to provide adequate bearing surface for the support of internal loads and to achieve appropriate stiffness to prevent deflection acute enough to:

(a) allow contact with the bearing surfaces,
(b) allow contact between the piston and cylinder,
(c) allow contact between the motor rotor and stator.

Prototype Configuration

A first prototype, referred to herein as prototype A, was fabricated with conventional dimensions. Two successive prototypes, B and C, were built based on subsequent analysis and testing.

The three prototype cylinders considered are described in Table 1.

Table 1 Prototype Cylinder Dimensions

<table>
<thead>
<tr>
<th></th>
<th>Cyl Dia. (in)</th>
<th>Cyl Hgt. (in)</th>
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<tr>
<td>A</td>
<td>2.44</td>
<td>2.05</td>
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<tr>
<td>B</td>
<td>2.20</td>
<td>1.41</td>
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<tr>
<td>C</td>
<td>2.20</td>
<td>1.05</td>
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The three prototype eccentric shafts considered are described in Table 2.

Table 2 Prototype Eccentric Shaft Dimensions

<table>
<thead>
<tr>
<th></th>
<th>PEB Dia. (in)</th>
<th>ECC Dia. (in)</th>
<th>MEB Dia. (in)</th>
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<tr>
<td>A</td>
<td>1.00</td>
<td>1.51</td>
<td>1.00</td>
</tr>
<tr>
<td>B</td>
<td>0.86</td>
<td>1.36</td>
<td>0.94</td>
</tr>
<tr>
<td>C</td>
<td>0.86</td>
<td>1.36</td>
<td>0.94</td>
</tr>
</tbody>
</table>
The weights of the various pump assemblies, exclusive of motor and shell are given in Table 3.

Table 3 Prototype Pump Assembly Weights Less Shell and Motor

<table>
<thead>
<tr>
<th></th>
<th>Weight (lbs)</th>
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<tbody>
<tr>
<td>A</td>
<td>12.85</td>
</tr>
<tr>
<td>B</td>
<td>8.54</td>
</tr>
<tr>
<td>C</td>
<td>7.45</td>
</tr>
</tbody>
</table>

As can be seen, dramatic reductions in size and weight, and thus cost, can possibly be realized, but such reductions are not achievable without penalty. The impact of downsizing is considered with respect to shaft dimensions in particular.

ECCENTRIC SHAFT LOADS

The dynamic bearing loads for the pump end, motor end and eccentric bearings were calculated for several promising geometries. Initially, bearing loads were targeted to remain within the envelope found to be previously satisfactory for rotary compressors. For maximum enhancement of reliability it was desired to maintain the lowest bearing loads possible. The relationship among the forces involved is shown schematically in Figure 4. Derivations of the functions defining shaft loads for a rolling piston rotary compressor have appeared many times in the proceedings of this conference (Ref. [1,2,3]), and will not be repeated here. Incorporated into the bearing load analysis were the gas forces $F_{pc}$, vane tip forces $F_{pi}$ and $F_{pt}$, and the inertial force $F_{pi}$. The coefficients of friction for the vane tip and vane sides were as reported in Reference [2].

Since the forces acting on the eccentric represent the summation of all the forces generated internal to the pump mechanism, those forces are shown for prototypes A, B and C in Figure 5. Figure 5, is a polar plot depicting the eccentric loads at an operating condition of $P_s=55$ PSIG, $P_d=405$ PSIG. The frame of reference for the plot is rotating relative to the cylinder, but fixed relative to the eccentric journal.
The differences among the forces generated within the alternate configurations are dramatic. The peak loads for design C being only 40% of those for design A. The dominant geometric variable impacting total bearing shaft loads is the height of the cylinder. Of course, a force reduction independent of a commensurate reduction in the bearing unit loadings and/or a reduction in shaft flexure is of little value.

**DYNAMIC PISTON TO CYLINDER CLEARANCE**

The radial clearance between the rolling piston and cylinder wall represents a major potential leakage loss. Generally, therefore, it is favorable to reduce the nominal clearance as much as possible without a resultant collision between the piston and the cylinder. There are many practical limitations to this reduction, such as machining tolerances. The comments below focus on the dynamic behavior of the clearance as a result of bearing eccentricity, discussed in Ref. [5], and shaft deflection.

**Bearing Eccentricity Induced Displacement**

The output from the dynamic load analysis was used to determine the various oil film thicknesses and the shaft journal trajectories. Using Booker's mobility method for the solution of dynamically loaded bearings [6], with correction of the oil film calculation as suggested by Goenka [7], the minimum oil film thicknesses, \( H_0 \), were developed for the pump, motor and eccentric bearings. The calculations of film thicknesses were conducted presuming maximum centered radial bearing clearances of \( 0.0004"/0.0005" \) and compressor operating conditions of \( P_s = 55 \text{ PSIG}, \ P_d = 405 \text{ PSIG} \).

Radial displacements for the pump journal \( e_{pb} \), rolling piston \( e_{eb} \), and motor journal \( e_{mb} \) were then developed by multiplying the bearing eccentricities by the centered radial clearance. The compressors under study, per common practice, have larger motor bearings than pump bearings. Owing to the complexity of considering the tilting misalignment brought on by differential oil film thicknesses, the analysis was conducted presuming that the shaft was displaced evenly in both the pump and motor bearings, i.e., \( e_{pb} = e_{mb} \). As the pump bearing experiences a higher eccentricity, total shaft displacement was based on the pump journal trajectory.
Figure 6 illustrates the displacement of the center of the moveable member of a bearing system away from a concentric position with the stationary member (relative to a coordinate system fixed with respect to the concentric origin). In the case of the main bearings, the bearings are the fixed members while the journals are free to move within the bearing clearance. The rolling piston is the moveable member in the bearing system comprised of the piston and the eccentric shaft journal. Analysis of journal and piston motion within the bearing clearances does presume that all members involved are rigid bodies.

When displacements of the shaft and rolling piston are resolved along an axis normal to the tangent point between the rolling piston and the cylinder interior wall they are directly additive to the nominal clearance. This axis, X', is part of a rotating coordinate system fixed relative to the maximum eccentric point of the shaft.

The displacement of the piston clearance from a concentric trajectory due to the pump bearing eccentricity ranged from 41% to 54% of the displacement due to the eccentric bearing eccentricity. Shaft C realized a 30% reduction in pump bearing displacement over shaft A and a 10% reduction in eccentric bearing displacement.

Eccentric Shaft Deflection Induced Displacement

Once the displacement of the piston and shaft has been established, it becomes necessary to factor in the deflection of the shaft due to applied load in order to determine the total reduction in radial clearance between the piston and cylinder. An assumption was made that the shaft is supported at the mid-point of both the pump and motor end bearings.

The deflection was evaluated with a computer code written by one of the authors (Tomayko). It was determined, based on static testing with Prototype A, that there was excellent agreement between the results from the program and empirical data. The dominant feature affecting shaft deflection was found to be the length of the unsupported span. Shaft C was determined to afford a 68% reduction in deflection relative to Shaft A.

The trade-offs between increased shaft rigidity and component growth must be carefully weighed as was apparent from Figure 2. However, it should also be
realized that small changes in the diameters of critical sections could result in a dramatic increase in stiffness as the moment of inertia for circular sections increases proportionally to the fourth power of the radius. Therefore attention to the details of the configurations of transfer sections, grinding reliefs, and oil passages is of great value.

TOTAL VARIATION IN PISTON TO CYLINDER CLEARANCE

All of the components are now in place to determine the total dynamic variation in the piston to cylinder radial clearance. The total clearance deviation from the concentric clearance is given by summing the displacements of the pump journal, the piston, and the deflecting shaft:

\[ \Delta C = e_{pb} + e_{eb} + \delta \]  \hspace{1cm} (1)

Starting with a base case of concentric positioning between the bearings and the cylinder, the residual radial clearance at any crank angle is given by:

\[ C(\theta) = C_c - \Delta C(\theta) \]  \hspace{1cm} (2)

Obviously, a positive clearance change will result in a decreased final clearance while conversely, a negative change will increase the total clearance.

Clearance Trajectory Plots

Figures 7, 8, and 9 graphically illustrate the effect the impact of the individual contributors to clearance variation as well as the total deviation from a concentric trajectory for prototypes A, B, and C, respectively. The outermost circle in these trajectory plots represents the interior wall of the cylinder. Shown on a concentric origin with the cylinder wall is the trajectory that the minimum clearance point between piston and cylinder would take if all components were perfectly round, infinitely rigid and centered. This is nomenclatured as the concentric trajectory. The distance between the above described circles is proportional to the nominal radial clearance between the piston and the cylinder.

Trajectory excursions to the outside of the concentric line represent decreases in radial clearance. Excursions inside the concentric line represent increases in radial clearance. The deviations from the concentric path for the various trajectories
represented are scaled to the total clearance. In the case of Figures 7, 8 and 9, the nominal clearance is identical, therefore, the figures are all proportional.

It can be clearly seen that the impact on clearance can be ranked in descending order as follows:

(a) displacement of the rolling piston,
(b) displacement of the pump journal,
(c) deflection of the eccentric shaft.

Figure 10 compares the trajectories of $C(\theta)$ resulting from the combined displacements of bearings and shaft deflection for prototypes A, B and C. The differences in shaft dimensions result in significant variations in the clearance trajectory. The maximum $C(\theta)$ for prototype C is 17% less than for prototype A, while the minimum $C(\theta)$ is 24% greater for C than for A. The total variation in $C(\theta)$ during one revolution is 19% less for configuration C than for configuration A.

A tighter trajectory of the piston allows for the selection of reduced nominal clearance which can enhance efficiency through the reduction of leakage.

The trajectory plots have value in quickly illustrating the subrevolution variation in piston to cylinder radial clearance enabling the designer to make the most appropriate choice in deciding on the magnitude and direction of the eccentric assembly of the bearings to the cylinder.

**EDGE LOADING**

The phenomenon of bearing edge loading in rotary compressors has been well reported in the literature [8]. One widely employed method of eliminating the problem has been to relieve the edge of the bearing as illustrated in Figure 11. This is done to allow for enough elasticity in the bearing such that it tilts slightly under load assuming the same attitude as the shaft. Thus, a uniform thickness oil film can be maintained over the entire axial length of the bearing eliminating the edge loading problems. The disadvantage of using conventional machining techniques to fabricate this feature is the required width of the relief groove. The groove as it breaches the sealing face of the piston causes a constant high to low side leak. A partial intrusion of the groove across the sealing face reduces the effectivity of the seal if the residual width of the seal, $T_p$, falls below an experi-
mentally determined minimum. The net effect of placing a groove on the seal side of the bearing is to necessitate an increase in the outside diameter of the rolling piston. In turn the cylinder bore must be increased. These changes tend toward lower efficiency as shown previously in Figure 2.

A very narrow groove would minimize the trend towards growth of the other components, however, it is difficult to fabricate on a production basis. A first prototype was fabricated with no groove and experienced bearing distress after a short period under load. The addition of the narrow groove shown in Figure 12 alleviated the bearing distress problem. Of course it represented a low level of manufacturability.

It was postulated that the bearing nearly behaves as a plate with a reinforced center hole and that the main effect of the slot is to reduce the effective plate thickness. Experiment determined that the reduction in stiffness that the grooved bearing represented over the ungrooved plate was 28%. Thus, presuming that global flexibility of the bearing was critical, an alternate design bearing with the same stiffness as the grooved bearing was tested. The key feature needed for the alternate design was ease of manufacture. Proposed was a wider groove on the opposite cylinder side of the bearing as shown in Figure 13. The depth of the groove was chosen to achieve the same flexibility as the bearing in Figure 12. The width was chosen to give an L/D ratio of the groove allowing for conventional manufacturing processes such as high speed turning or net shape forming through the use of powdered metal.

Testing of bearings with the opposite cylinder side groove has proven eminently successful with no sacrifice in efficiency.

CONCLUSIONS

An evaluation was made of the sensitivity of the rotary compressor geometry to dimensional variation. Generally, it was determined that benefits result from maintaining the smallest pump size compatible with other limits.

The response of the bearing loads and shaft deflections to the reduced pump envelope was found to be favorable.
To visualize the dynamic piston to cylinder clearance, trajectory plots were generated which can aid in the appropriate selection of nominal clearance and assembly offset.

The characteristic edge loading found in rotary compressors was examined and alternative solutions analyzed. A satisfactory solution has been identified and verified.
### LIST OF SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>C(θ)</td>
<td>Clearance between rolling piston and cylinder at shaft angle θ</td>
</tr>
<tr>
<td>E</td>
<td>Shaft eccentricity</td>
</tr>
<tr>
<td>e_pb</td>
<td>Displacement from center of the pump journal</td>
</tr>
<tr>
<td>e_mb</td>
<td>Displacement from center of the motor journal</td>
</tr>
<tr>
<td>e_db</td>
<td>Displacement from center of the rolling piston</td>
</tr>
<tr>
<td>F_pc</td>
<td>Compression chamber gas force</td>
</tr>
<tr>
<td>F_pi</td>
<td>Rotating component inertial force</td>
</tr>
<tr>
<td>F_pn</td>
<td>Vane formal force</td>
</tr>
<tr>
<td>F_pt</td>
<td>Vane tangential force</td>
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<td>F_vd</td>
<td>Vane gas force</td>
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<td>Vane inertial force</td>
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<tr>
<td>F_vs</td>
<td>Vane spring force</td>
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<tr>
<td>F_vsfl,2</td>
<td>Vane slot frictional forces</td>
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<tr>
<td>F_vsll,2</td>
<td>Vane slot reaction forces</td>
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<tr>
<td>H_c</td>
<td>Cylinder height</td>
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<tr>
<td>H_o</td>
<td>Minimum bearing oil film thickness</td>
</tr>
<tr>
<td>L_g</td>
<td>Bearing relief groove depth</td>
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<tr>
<td>P_d</td>
<td>Discharge pressure</td>
</tr>
<tr>
<td>P_s</td>
<td>Suction pressure</td>
</tr>
<tr>
<td>R_c</td>
<td>Cylinder radius</td>
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<tr>
<td>T_g</td>
<td>Bearing relief groove width</td>
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<tr>
<td>T_p</td>
<td>Rolling piston sealing face width</td>
</tr>
<tr>
<td>w</td>
<td>Angular velocity of eccentric shaft</td>
</tr>
<tr>
<td>wp</td>
<td>Angular velocity of rolling piston</td>
</tr>
<tr>
<td>δ</td>
<td>Deflection of eccentric shaft</td>
</tr>
<tr>
<td>θ</td>
<td>Shaft angle</td>
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565
REFERENCES


FIGURE 1A
SCHEMATIC OF A ROLLING PISTON COMPRESSOR SECTION PERPENDICULAR TO THE ECCENTRIC SHAFT.

FIGURE 1B
SCHEMATIC OF A ROLLING PISTON COMPRESSOR SECTION PARALLEL TO THE ECCENTRIC SHAFT.

FIGURE 2
NON-DIMENSIONAL LOSS FUNCTION VS. CYLINDER HEIGHT AND DIAMETER FOR AN 18,000 BTU/H COMPRESSOR.

FIGURE 3
AVERAGE VANE P-V FOR VARIOUS 18,000 BTU/H GEOMETRIES.
FIGURE 4
Schematic of the forces involved in generating the bearing load.

FIGURE 5
Polar plot of the eccentric bearing loads at \( P_0 = 55 \) psig, \( P_d = 455 \) psig for prototypes A, B, and C. The coordinate system is rotating relative to the cylinder, fixed relative to the eccentric journal.

FIGURE 6
Schematic of the displacement of the bearing journals and rolling piston due to eccentric journal loads are in lbs.

FIGURE 7
Trajectory of C.O.G., the clearance between the rolling piston and the cylinder wall through one revolution for prototype A.

- Shaft displacement trajectory
- Piston displacement trajectory
- Shaft deflection trajectory
- Combined displacement trajectory
Figure 8
Trajectory of C(θ), the clearance between the rolling piston and the cylinder wall through one revolution for prototype B.

- Shaft displacement trajectory
- Piston displacement trajectory
- Shaft deflection trajectory
- Combined displacement trajectory

Figure 9
Trajectory of C(θ), the clearance between the rolling piston and the cylinder wall through one revolution for prototype C.

- Shaft displacement trajectory
- Piston displacement trajectory
- Shaft deflection trajectory
- Combined displacement trajectory

Figure 10
Comparison of the trajectories of C(θ) resulting from the combined displacements of bearings and shaft deflection, for prototypes A, B, and C.

Figure 11
Cross section of bearing incorporating a wide groove on sealing face to allow for alignment between the bearing and the shaft.
FIGURE 12
CROSS SECTION OF BEARING INCORPORATING A NARROW GROOVE ON THE SEALING FACE.

FIGURE 13
CROSS SECTION OF BEARING INCORPORATING A WIDE GROOVE ON THE NON SEALING FACE.
THE INFLUENCE OF CYLINDER WALL PORTS ON THE PERFORMANCE OF A REFRIGERANT COMPRESSOR

Jin Min and Miao Dao-Ping
Shanghai Institute of Mechanical Engineering
Chen Fu-Chang
Shanghai Refrigerating Machinery Works

ABSTRACT

This investigation studied the influence of cylinder wall ports on the performance of a refrigerant compressor by means of computer simulation and experiment. It proved that wall ports in the cylinder liner are useful in increasing the cooling capacity and COP of a refrigerant compressor when they are correctly designed. To give full play to the charging effect of the wall ports, valve lift and spring characteristics should be adjusted properly in coordination with the wall port design. With the increase of compression ratio, wall ports' benefit becomes still more remarkable. So, provision of wall ports are more desirable in case of compressor working under lower evaporating temperature.

INTRODUCTION

The idea of using proper cylinder wall ports around the cylinder liner at a position near the B.D.C. of the piston stroke for compressor performance improvement was patented in the late 30s [1], but had not been put into practice until 70s when S.A. Parker had made an attempt to test it on an air-conditioning compressor, resulting in a volumetric efficiency increase less than 1% [2]. Later, D.Squarer and others tried by means of compressor computer modeling to evaluate the possible performance improvement of a heat pump compressor through several potential compressor design modifications, including a provision of cylinder wall inlet ports. He concluded that a marked improvement in efficiency was clearly evident only at a temperature below -29°C [3], which has not been ascertained by experiment.

With the purpose of clarifying the actual influence of wall ports on the compressor performance at different test conditions, displaying the changes in cylinder processes and valve behaviour produced by the disposal of wall inlet ports and searching for optimal correlation between port size, location and valve design parameters, computer modeling technique combined with testing on an actual compressor has been used successfully and achieved good
results. Theoretical analysis and experiment demonstrate that a noticeable gain in refrigerating capacity of compressor accompanied with some increase of COP can be expected only when the suction valve design parameters are properly determined to adapt the design of wall ports.

The investigation was conducted with a domestic-made six cylinder compressor for R-12 having a rated capacity of 73.26 Kw under standard condition (evap. temp. -15°C, cond. temp. 30°C, return gas temp. 15°C, subcooling 5°C) with 100 mm bore diameter, 70 mm stroke and 1440 rpm. The compressor valve system is of ring-plate type (Fig. 1) with intake holes uniformly disposed on the periphery around the upper flange of the cylinder liner. The determination of design parameters like valve lift and spring characteristics have been elaborately studied and checked up by experiment so that the compressor was featured with satisfactory characteristics.

**COMPRESSOR PROCESSES COMPUTER MODELING**

The simulation model describing the processes taking place in the cylinder is based on the First Law of thermodynamics applied to well defined control volumes within the compressor. It contains all necessary details to permit true simulation of the internal compressor modification related specifically to wall port and valve design, and is easily manipulatable, rapid and inexpensive to run. The present model comprises a set of coupled differential and algebraic equations, such as energy conservation equation, mass flow equation, valve dynamic equation, and refrigerant equation etc. which were solved digitally by iteration with a step size of one crank angle degree. In the model the gas force coefficient was assumed to be independent of valve lift and the valve flow coefficient was represented by a bi-linear characteristic. The wall port flow coefficient was considered as an empirical constant which had been generally employed in the scavenging port calculation in two-stroke internal combustion engine.

Computer modeling was carried out to study the effect of the following design changes:
1) Change of wall inlet port number, size and location.
2) Change of suction valve lift.
3) Change of valve spring characteristic.

**Change of wall inlet port size, number and location.**

The effect of wall ports communicating with the surrounding suction chamber consists in that they allow an additional flow area for suction vapour when uncovered by the piston top edge and hence, a larger quantity of refrigerant can be drawn into the cylinder if the timing and port size are correctly designed. Otherwise, an adverse effect occurs.

In compressor simulation, two sets of port data were calculated. One was 50 smaller holes and the other 20 larger holes. The 50 holes had a maximum opening area 18% larger than that of the latter. Fig. 2 and 3 give the simulation results which show the
cooling capacity $Q_0$ and the indicated power consumption $P_{01} \text{ vs.}$ the location of the wall ports $H$ (distance from the top edge of the piston to the top of inlet port expressed in percentage of the piston stroke, Fig. 1) under standard test condition and low temperature condition (evap. temp. $-25^\circ C$, cond. temp. $30^\circ C$, return gas temp $5^\circ C$, subcooling $5^\circ C$) respectively with different number of holes.

One can see clearly from the plots that each curve has a maximized capacity at a port location near $H = (96-97)\%$ which is almost the same value as indicated in reference [2]. However, it must be noticed that the port location $H$ in reference [2] was defined as the distance from the top of ring at T.D.C. to top of wall ports in percentage of piston stroke. The figures show that the compressor with 20 holes in the cylinder liner produces 1.2% and 1.9% more cold under standard condition and low temperature condition respectively than without wall port provision, which are higher than the 0.86% capacity gain under high temperature condition (evap. temp. 7.2°C, cond. temp. 51.6°C, return gas temp, 18.3°C) [2]. Simulation also shows that the port location is optimal when a period of back flow from cylinder to the suction chamber lasts some (10-15) crank angle degrees before the complete closure of wall ports (see Table 1). This results in a higher cylinder pressure over the suction pressure at the moment when the piston has just fully covered the wall ports. But, moving the wall ports up further toward the T.D.C. as shown in Fig. 2 and 3 causes excessive refrigerant backflow and therefore, capacity reduction.

Fig. 2 and 3 indicate also that cylinder with 50 holes produces more cooling capacity and has a higher $COP_{1}$ than that with 20 holes.

Change of Suction Valve Lift

The wall ports function cooperatively with the suction valve during the late period of the suction process. Table 1 lists the cylinder charge through wall ports per cycle. In spite of only a few percent of the total cylinder charge it is an important and innegligible part of the charging effect.

In spite of the elaborate valve system design and satisfactory operation of the original compressor, it is apparent that the valve behavior has to be changed with the provision of wall ports. So, there exists the need for better coordination between port and valve design.

We tried to decrease the suction valve lift. This was only because in valve system shown in Fig. 1, the size of the cylinder clearance volume is closely related with the suction valve lift. Suction valve lift depression reduces the clearance volume but at the meanwhile it decreases the suction flow area as well. The adverse effect of the latter can be compensated for with the presence of wall ports. Computer simulation results for port location $H = 96.5\%$ are plot in Fig. 4. Just as what we have expected with the reduction of suction valve lift beginning from 2 mm, the capacity keeps on increasing until a 1.5 mm lift where a maximum value is reached. Reduction of suction valve lift to 1.5 mm ensures a capacity increase of 1% and 5.5% under standard and low temperature.
conditions respectively in relation to suction valve lift 2 mm un-
changed, i.e. 2.3% and 7.5% in comparison with the case without
wall ports (see Table 1). The COPI decreases under standard condi-
tion and increases under low temperature condition as the suction
valve lift is reduced to 1.5 mm (Fig.4).

It is also shown in Table 1 that the cylinder charge through
wall ports has increased from 5.7% to 7.7% of total cylinder charge
at standard condition and from 4.9% to 9.1% at low temperature
condition.

Change of Valve Spring Characteristics

Three valve systems with three different spring characteristics—two springs with constant stiffness 1.47 and 2.06 KN/m, the
third one which we actually tested and had a variable stiffness
proportional to the spring deflection—have been simulated. Fig.5
shows one of those simulation results under standard condition with
optimal port location H = 96.5% and reduced suction valve lift 1.5
mm. It can be seen from the plot that compressor using springs
with constant stiffness 1.47 KN/m has the highest output and COPI
while springs with constant stiffness 2.06 KN/m give the smallest
capacity and middle COPI.

EXPERIMENTAL VERIFICATION OF THE COMPRESSOR
MODELING AND RESULTS ANALYSIS

Experimental verification has been carried out on a hot gas
bypass loop test stand. During the test period only two of the six
cylinders were actually working and the other four cylinders were
unloaded by removing away all their valve elements and isolating
them from the working ones. That is the reason why we have taken
account of compressor power consumption and COP by indicated para-
eters in our investigation. Cylinder liners only with 20 larger
holes located at H = 99%, 97.69, 96.5% and 95.33% respectively were
tested under both standard and low temperature conditions. In add-
tion to the original suction valve lift 2 mm, the optimal valve
1.5 mm determined in simulation analysis has also been checked up
by experiment while the existing spring with variable stiffness re-
mained unchanged through out the entire test period.

Simultaneous recording of cylinder pressure, suction and dis-
charge pressures, motion of the valve plate and B.D.C. were done
in the experiment. For determining the valve lift inductive valve
plate displacement transducers have been used. The cylinder pres-
sure transducer, suction and discharge pressure transducer were of
strain gauge type. A typical pressure and valve motion vs. crank
angle oscilloscope trace is illustrated in Fig.6 which shows fair
match of the theoretical and experimental results. The valve mo-
tions, however, are less steep than the computed ones. This might
be considered as a result of the non-linear spring characteristics.

Actual cooling capacity, POWI and COPI of the compressor with
2 mm suction valve lift at different test conditions and their com-
parison with simulation results (solid curves) have been plot on
relevant diagrams by Δ and ⋄ (Fig.4 and 7).
The accuracy of compressor simulation is within acceptable limits. Accuracy for predicting cooling capacity is within 1% over the entire test range. The worst error for predicting POWI is about 4% occurring at low temperature condition.

Analysis on the recorded pressure indicating diagrams of the suction process (Fig. 8) reveals the actual influence of the wall ports and suction valve lift depression on the thermal process occurring in the suction period. Optimal arrangement of the wall ports at $H = 96.5\%$ with the original suction valve lift allows more captured cylinder charge described by the higher cylinder pressure at the closing moment of the wall ports. and the suction valve lift reduction brings about a noticeable cylinder pressure depression owing to the reduced suction flow area from the beginning of the suction process until the opening of the wall. Once the ports open, the increased pressure difference between the cylinder and suction chamber pushes the refrigerant rushing into the cylinder and causing a more steep pressure rise. This raises further the cylinder pressure at the closing moment of the ports and therefore, increases the cylinder charge although there exists a longer duration of backflow.

CONCLUSION

The results of this investigation can be summarized as follows.

1) The agreement of the theoretical and experimental results proves the compressor computer model a successful one for wall port design.

2) Cylinder wall ports are useful in increasing the cooling capacity and COP of a refrigerant compressor if their size and location are reasonably determined.

3) In order to give full play to the charging effect of the cylinder wall ports the valve lift and spring characteristics should be adjusted properly to coordinate the port design.

4) With the increase of compression ratio wall ports' benefit becomes still more remarkable.

5) In our experimental investigation, with 20 holes at $H = 96.5\%$ the compressor performance improvement against the unmodified one are listed below.

<table>
<thead>
<tr>
<th>Running condition</th>
<th>Standard</th>
<th>Low temp.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suct. valv. lift mm</td>
<td>2 1.5</td>
<td>2 1.5</td>
</tr>
<tr>
<td>Capacity gain %</td>
<td>2 4.5</td>
<td>3 7</td>
</tr>
<tr>
<td>COPI increase %</td>
<td>2 0</td>
<td>3 6</td>
</tr>
</tbody>
</table>

6) More effective wall port flow area at an optimal $H$, if possible, is desirable for better improvement.

ACKNOWLEDGEMENT

The authors wish to thank Zheng Xian-Dong and Cui Chuan-Liang,
Shanghai refrigeration Machinery Works, for their help and corporation in the experiment.

REFERENCE

5. J.F. Hamilton, 'Extensions of Mathematical Modeling of Positive Displacement Type Compressor', Ray W. Herrick Laboratories, Purdue University.
## TABLE 1

**COMPRESSOR SIMULATION RESULTS**
(20 holes, valve springs with variable stiffness, 2 working cylinders)

<table>
<thead>
<tr>
<th>Running condition</th>
<th>Standard condition</th>
<th>Low temp. condition</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>H (%)</strong></td>
<td>100 96.5 93 96.5</td>
<td>100 96.5 94 96.5</td>
</tr>
<tr>
<td><strong>Suct. valve lift, mm</strong></td>
<td>2 2 2 1.5</td>
<td>2 2 2 1.5</td>
</tr>
<tr>
<td><strong>Ref. capacity Kw</strong></td>
<td>26.6 27.0 26.8 27.2</td>
<td>15.4 15.7 15.6 16.5</td>
</tr>
<tr>
<td><strong>Vol. efficiency</strong></td>
<td>.75 .76 .754 .777</td>
<td>.713 .724 .72 .74</td>
</tr>
<tr>
<td><strong>Total cyl. charge x10^2 Kg/cycle</strong></td>
<td>.132 .134 .133 .137</td>
<td>.085 .087 .086 .092</td>
</tr>
<tr>
<td><strong>Wall port charge x10^2 Kg/cycle</strong></td>
<td>0 .077 .057 .106</td>
<td>0 .043 .07 .084</td>
</tr>
<tr>
<td><strong>Indicated power Kw</strong></td>
<td>7.60 7.63 7.50 7.94</td>
<td>5.93 5.99 5.91 6.16</td>
</tr>
<tr>
<td><strong>COPI Kw/Kw</strong></td>
<td>3.51 3.54 3.57 3.43</td>
<td>3.07 3.04 3.07 3.12</td>
</tr>
<tr>
<td><strong>Suction valve:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>open °</td>
<td>38 38 38 36</td>
<td>48 48 48 48</td>
</tr>
<tr>
<td>close °</td>
<td>202 200 197 189</td>
<td>200 195 189 204</td>
</tr>
<tr>
<td><strong>Discharge valve:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>open °</td>
<td>305 305 305 305</td>
<td>317 317 317 317</td>
</tr>
<tr>
<td>close °</td>
<td>5 5 5 4</td>
<td>5 6 5 6</td>
</tr>
<tr>
<td><strong>Max. impact vel.m/s</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Suct. valve:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>toward seat</td>
<td>1.69 1.69 1.64 1.61</td>
<td>1.48 1.57 1.57 1.52</td>
</tr>
<tr>
<td>toward stop</td>
<td>3.56 3.99 3.60 2.97</td>
<td>2.97 3.26 3.01 2.51</td>
</tr>
<tr>
<td>Disch. valve:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>toward seat</td>
<td>1.35 1.35 1.59 1.38</td>
<td>1.34 1.69 1.53 1.56</td>
</tr>
<tr>
<td>toward stop</td>
<td>5.01 5.25 4.43 4.96</td>
<td>4.87 5.20 4.74 4.23</td>
</tr>
<tr>
<td><strong>Wall inlet ports</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>open angle °</td>
<td>157 147 157</td>
<td>157 149 157</td>
</tr>
<tr>
<td>close angle °</td>
<td>203 213 203</td>
<td>203 211 203</td>
</tr>
<tr>
<td>back flow °</td>
<td>188- 180- 188-</td>
<td>190- 183- 202-</td>
</tr>
<tr>
<td></td>
<td>203 213 203</td>
<td>203 211 203</td>
</tr>
</tbody>
</table>

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Fig 1. Section view of the suction valve and wall inlet ports of the compressor

Fig 2. Effect of wallport area and location on QO, POWI and COPI

--- 20 holes ------ 50 holes
Fig 3. Effect of wall port area and location on QO, POWI and COPI

- 20 holes ——— 50 holes

Fig 4. Effect of suction valve lift reduction on capacity ratio and COPI ratio

- Standard condition ——— Low temp. condition
Fig 5. Effect of valve spring characteristics change on QO, POWI and COPI at standard condition

--- variable stiffness ---- spring constant 1.47 KN/m
--- spring constant 2.06 KN/m

Fig 6. Pressures and valve plate motions vs. crank angle diagrams at standard condition
Fig 7. Actual QO, POWI COPI of the compressor plot against the simulation results

Fig 8. recorded cylinder and suction chamber pressure vs. crank angle diagrams of the suction process at standard condition