Performance Analysis of a Sliding-Vane Rotary Compressor for a Household Refrigerator/Freezer

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PERFORMANCE ANALYSIS OF A SLIDING-VANE ROTARY COMPRESSOR FOR A HOUSEHOLD REFRIGERATOR/FREEZER

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ABSTRACT

A study of the power consumption and mass flow characteristics of a sliding vane rotary compressor for a household refrigerator/freezer is presented. The study is based on results from a computer simulation of the compressor mechanism which considers the participation of both refrigerant and oil in all fluid processes. Various modelling techniques used in the simulation are discussed. Several important simulation results are substantiated by direct comparison with experimental data from calorimeter tests for an instrumented compressor. The details available from the simulation results are used to infer causes for performance characteristics not readily apparent from the experimental data alone.

INTRODUCTION

The competitive nature of the refrigeration compressor industry encourages manufacturers to develop analytical tools for investigating product performance. Many of the analytical tools developed for the sliding-vane rotary compressor are based on work originated at Purdue University's Herrick Laboratories. Coates presented a detailed computer simulation model coupled to an optimization scheme in 1970 [1]. In 1977 Reed showed that refrigerant/oil miscibility can have a significant effect on capacity [2]. In 1978 Pandeya proposed improvements to several sliding and viscous friction loss models as well as certain flow loss models [3]. In 1982 Yee's analysis showed the significance of the re-expansion process as an energy loss [4].

Elements from the Purdue work have been combined with several analytical methods developed at Whirlpool to make a simulation capable of predicting the details of sliding vane rotary compressor behavior in arriving at accurate estimates of capacity and power consumption. The simulation has been used to study a production compressor design for which extensive operating test data was collected.

THE COMPRESSOR

The compressor studied in this analysis is a high-side hermetic sliding-vane rotary intended for use in a household refrigerator/freezer. The compressor operates with R12 refrigerant and commercially available napthenic mineral oil. The oil is stored in a sump at the bottom of the hermetic shell. The vertical-axis compressor mechanism and drive motor are suspended above the oil sump by springs attached to the shell.

The compressor mechanism consists of a rotor which carries two movable vanes in radially opposed slots within a round cylinder. The cylinder is located such that the rotor head is nearly tangent to the cylinder wall in the region between the suction and discharge ports. The discharge port is provided with a cylindrical reed valve and stop. The rotor head, vanes, cylinder, and discharge valve are placed between parallel end plates. The end plates form a gas-tight seal with the cylinder end surfaces. Clearance is provided to allow free rotation of the rotor and vanes. The rotor shaft is supported radially by a bearing boss formed as part of the lower end plate.

The motor consists of an armature pressed onto the rotor shaft below the bearing boss and a four-pole winding supported by the lower end plate. The rotor is provided with a hollow center along its
entire length. Attached to the bottom of the rotor shaft below the motor armature is a concentric pick-up tube which extends below the oil surface in the sump.

In operation the action of the vanes moving in the cylinder pulls refrigerant vapor from the low pressure side of the external system through a suction line which penetrates the shell and contacts directly to the mechanism. A cavity is provided in the mechanism which serves as a plenum for the cylinder suction ports.

The pressure in the shell forces oil and dissolved refrigerant liquid from the sump through the shaft to the bearings and rotor faces toward low pressure regions in the cylinder. The bearings are supplied with oil from the rotor shaft center by holes drilled through the shaft at right angles to its axis. Bearing grooves assure even oil distribution along the length of the bearing surfaces. The bases of the vane slots are supplied with axial oil flow from the upper shaft bearing and radial oil flow from the upper rotor face. The oil is available to lubricate, seal, and cool the vane/slot surfaces. The oil pressure established at the base of the vane slots and centrifugal action serve to force the vanes outward toward the cylinder wall.

As the extended vanes rotate, the rounded vane tips slide along the cylinder wall. Refrigerant vapor trapped ahead of the vanes is compressed and pushed with any oil present through the discharge port and past the discharge valve. The upper end plate contains interconnected cavities which act as a plenum for the discharge valve and a muffler for the discharge flow. The muffler is arranged to allow for the passage of liquid oil. The discharge flow leaves the muffler by a tube which goes through the shell to an external heat exchanger called a pre-cooler. After passing through the pre-cooler the flow returns to the compressor shell where the oil is allowed to drain back to the sump and the refrigerant vapor is allowed to flow to the external system through a second discharger tube positioned to minimize oil entrainment.

High-speed photography of an operating compressor equipped with a transparent upper end plate showed the presence of significant quantities of oil in the cylinder in the form of liquid, foam, and mist. Dense white mist was evident in the discharge and re-expansion flows. Foam was present at the trailing edge of the high-pressure vane and along the rotor periphery in the low-pressure suction region. Liquid and foam were seen in front of the vanes being pushed toward the discharge port.

Another compressor, fitted with cylinder pressure transducers and thermocouple probes at key points, was tested on a calorimeter for a wide range of suction and discharge conditions. The motor for this compressor was dynamometer tested to establish performance. Cylinder pressure traces were generated at each test condition which allowed the compression power to be evaluated. Combining this result with the motor dynamometer data allowed the friction power to be calculated. Oil in the discharge flow was measured using a special separator with means for determining dissolved refrigerant concentration. The refrigerant concentration for the sump oil was also measured. Temperatures were measured at several points around the compressor including the oil sump, motor winding, bearings, upper rotor face, suction plenum, and discharge plenum.

THE SIMULATION

The computer simulation developed to predict the behavior of the sliding-vane rotary compressor consists of approximately 4600 lines of Fortran programming arranged in 24 subroutines. Execution requires about 750 K bytes of memory and 4.5 minutes of CPU time when using an IBM 370-168. Each run requires a set of input data which specifies various details of the compressor design and several mechanism operating temperatures. Temperature values based on test results are used to avoid the need for complex heat transfer analysis.

Geometry models similar to those reported by Coates [1] and Yee [4] are used to define the sizes of the cylinder compartments and the motion of the vanes as a function of rotor angle. The vane tips are assumed to be in line contact with the cylinder wall. Constant clearances specified at input are assumed to exist between the rotor head and cylinder wall at the point where the rotor nearly touches the cylinder, between vane edges and end plates, and between rotor faces and end plates. Particular attention is given to the geometry changes which occur when a vane passes through the discharge port region so that the volume of the region supplying flow during re-expansion and the area of the orifice through which re-expansion occurs are well defined.

The simulation treats the cylinder compartments defined by the geometry calculations as a set of control volumes.
linked together by several fluid flow passages. The control volumes are assumed to be filled with refrigerant vapor and a liquid solution of refrigerant and oil. Oil vapor is neglected. Real refrigerant property subroutines are used to determine values for the refrigerant thermophysical properties. Data supplied by the oil manufacturer are used to establish pure oil properties as a function of temperature, refrigerant/oil solution concentration as a function of temperature and pressure, and refrigerant/oil solution viscosity as a function of temperature and solution concentration. Values for the specific volume and specific enthalpy of a refrigerant/oil solution are found by combining the properties of pure oil and refrigerant liquid at the solution temperature in proportion to their mass fractions in the solution.

Each control volume is assumed to vary in size according to the geometry calculations. Frictional energy added to a control volume is assumed to be offset by heat transfer away from the control volume so that their net effect is negligible. The conditions within a control volume are assumed to vary quasi-statically. The refrigerant and oil within a control volume are assumed to be in miscible equilibrium. The properties of the fluid which enter or leave a control volume through a linking flow passage are determined by conditions in the control volume supplying the flow. All flow processes are assumed to be quasi-steady. The suction and discharge plenum pressures are assumed to be constant. It is assumed that the temperature in the suction control volume at the end of the suction process is known from experimental results.

Starting with initial estimates for the temperature, pressure, oil mass, and refrigerant mass in each control volume, the mass and energy changes of the control volumes are computed for an incremental change in rotor angle using a reiterative forward finite difference technique. At the end of each increment the temperature and pressure for a control volume are found for which the fluid properties of the contents satisfy the mass, energy, and volume requirements imposed on the control volume.

The clearance between the rotor and cylinder at the near-tangent point and the clearance between each extended vane edge and end plate form linking flow passages between control volumes. The passages are assumed to be filled with an incompressible viscous liquid refrigerant/oil solution whose properties are determined by instantaneous conditions in the supplying control volume. A velocity profile is estimated for the fluid in the clearance space which accounts for the relative motion of the clearance surfaces and the instantaneous pressure difference across the clearance path. The properties of the fluid and the velocity profile in the clearance are used to compute incremental values for mass and energy transfer through the clearance.

The clearance between the rotor head and the end plates in combination with the upper bearing clearance provide a complex passage for the flow of refrigerant and oil supplied by the sump. The rotor head and bearing passages are treated as a network of parallel-plate flow elements which connect the sump to the base of the vanes and the control volumes at the rotor periphery. Each flow element is assumed to be filled with incompressible, viscous fluid whose properties are determined by conditions at the inlet to the flow element. Incremental flow through each element is estimated using the instantaneous pressure difference across the element and the instantaneous shape of the element. The rotor/bearing model establishes the flow of refrigerant and oil from the sump to the cylinder and the pressure at the base of the vanes.

Flow through the throat of the discharge port region is represented by a simple orifice model. The flowing fluid is treated as a homogenized incompressible mixture of refrigerant vapor and liquid refrigerant/oil solution whose properties are determined by the temperature, pressure, vapor mass, and liquid mass in the supplying control volume. Specific volume and enthalpy for the fluid are calculated by mass weighting the properties of the vapor and liquid. The orifice area is constant except during re-expansion when the passage of the vane through the discharge port region causes the shape of the orifice to vary.

Flow through the discharge port and past the discharge valve is assumed to be incompressible and irreversible such that velocity head is lost to fluid friction and cannot be converted to static pressure. The valve is treated as a round disc which remains parallel to the valve seat and whose axis aligns with the axis of the discharge port. The distance from valve to seat varies according to valve motion calculations. The discharge flow model determines the forces acting on the valve in addition to mass and energy flow through the discharge port.
The discharge valve is modeled as a system of a four lumped-mass cantilever beams connected end-to-end in a way which approximates the valve's cylindrical shape. The equations of motion for the mass nodes account for discharge flow, inertia, fluid damping, material damping, and deflection forces. Valve motion is assumed to be planar. Provision is made for the constraints imposed by the seat and stop. The equations of motion are solved using a third-order Runge-Kutta integrator and a Gauss elimination equation solver.

Because of the mechanism geometry, all the cylinder processes repeat themselves each 180 degrees of rotor rotation. The simulation proceeds in small increments through 180 degrees of rotor rotation from a pre-selected starting point which corresponds to the position of the rotor when the large suction compartment just completes its suction process and no longer communicates directly with the suction port. Four 180-degree iterations are usually required to achieve reasonable convergence for all variables. It was found that .1 degree increments of rotor rotation provide a suitable time base large enough to avoid significant computer round-off error and small enough to eliminate numerical instability in making the control volume and flow model calculations.

After establishing the transient conditions for the thermal and flow processes in the mechanism, the average mass flow rates of refrigerant and oil for each flow path are calculated by integrating stored incremental results. The net mass flow rate to the external system is calculated by combining integrated results from the discharge valve, rotor face, and lower shaft bearing flow models. Calculations for the suction control volume are made using integrated results from the rotor face, vane edge, and minimum rotor clearance flow models to establish the leakage and superheat capacity effects for each flow path.

The average power required to drive the compressor mechanism is calculated in friction and compression components. Friction power estimates are made for the bearing, rotor faces, minimum rotor clearance, and vanes. Instantaneous radial bearing loads are computed using the cylinder pressure histories. A technique proposed by Raimondi and Boyd [5] is used to estimate viscous bearing friction assuming that the radial loads are quasi-static. Viscous friction power estimates are also made for the rotor faces, minimum rotor clearance, and extended vane edges. Each estimate accounts for the variation in oil viscosity due to temperature and refrigerant concentration. Sliding friction power estimates are made for the vane tips and the rotor slots after calculating the instantaneous pressure, inertia, and friction forces applied to the vanes.

Total compression power is calculated using cylinder pressure and volume histories. An estimate of the contribution to the total due to recompression of recycled refrigerant is made for each leakage path and the re-expansion process. Each estimate assumes that the mass, both refrigerant and oil, which flows through a particular path during a complete cycle occupies imaginary control volumes on the supplying and receiving sides for which cylindrical pressure and volume histories can be found.

Program execution concludes with an itemized report which summarizes the effects on performance due to the mass flow and power consumption associated with the various mechanism processes modeled. Predicted values for total power, net capacity, and coefficient of performance are included. Output options are available to print incremental values for several important variables as a function of rotor angle.

THE COMPARISON

Experimental results from the test compressor were obtained at ten different calorimeter conditions. Using temperature data from the tests along with bench-measured clearance values for the compressor, the simulation was run for the same conditions. Table 1 compares simulation results to experimental results for four of the ten calorimeter conditions. The four points selected are at the limits of the ranges used for suction and discharge pressures. The average error figures in Table 1 are based on all ten test points.

Table 1 shows that agreement between simulation and experiment is reasonably close over a wide range of operating conditions. Trends for friction and compression power components are similar. Mass flow rates for refrigeration and oil show good correlation.
### TABLE 1
SIMULATION AND EXPERIMENTAL RESULTS FOR COMPRESSOR AT FOUR TEST CONDITIONS
(Simulation/Experimental)

<table>
<thead>
<tr>
<th></th>
<th>Simulation</th>
<th>Experiment</th>
<th>Average</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{suc}$, psia</td>
<td>15.25</td>
<td>23.92</td>
<td>15.25</td>
<td>23.84</td>
</tr>
<tr>
<td>$P_{dis}$, psia</td>
<td>195.7</td>
<td>195.7</td>
<td>154.7</td>
<td>154.7</td>
</tr>
<tr>
<td>$Q$, Btu/hr</td>
<td>504/505</td>
<td>884/875</td>
<td>619/611</td>
<td>992/989</td>
</tr>
<tr>
<td>$\dot{m}_{oil}$, lbm/hr</td>
<td>5.76/5.95</td>
<td>5.30/5.51</td>
<td>3.23/3.51</td>
<td>2.91/2.61</td>
</tr>
<tr>
<td>$\dot{w}_{tot}$, watts</td>
<td>201/200</td>
<td>226/230</td>
<td>174/170</td>
<td>197/194</td>
</tr>
<tr>
<td>$\dot{w}_{pv}$, watts</td>
<td>107/110</td>
<td>127/130</td>
<td>92/92</td>
<td>111/106</td>
</tr>
<tr>
<td>$\dot{w}_{frc}$, watts</td>
<td>44/40</td>
<td>42/42</td>
<td>37/34</td>
<td>35/38</td>
</tr>
<tr>
<td>$\dot{w}_{ml}$, watts</td>
<td>50/50</td>
<td>57/58</td>
<td>45/44</td>
<td>51/50</td>
</tr>
<tr>
<td>C.O.P.</td>
<td>2.51/2.54</td>
<td>3.91/3.80</td>
<td>3.56/3.59</td>
<td>5.04/5.10</td>
</tr>
</tbody>
</table>

### TABLE 2
SIMULATION RESULTS FOR COMPRESSOR PERFORMANCE AT AHAM CONDITIONS
(-15°F Evaporator, 112°F Condenser, 90°F Liquid, 90°F Suction, 90°F Ambient)

<table>
<thead>
<tr>
<th>Capacity Effect, Btu/HR</th>
<th>Power Consumption Effect, Watts</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leaks</td>
<td>Friction</td>
</tr>
<tr>
<td>Superheat</td>
<td>Compression</td>
</tr>
<tr>
<td></td>
<td>Rotor</td>
</tr>
<tr>
<td>-37.2</td>
<td>3.3</td>
</tr>
<tr>
<td>1.9</td>
<td>33.3</td>
</tr>
<tr>
<td>-8.7</td>
<td>0</td>
</tr>
<tr>
<td>-47.8</td>
<td>36.6</td>
</tr>
<tr>
<td>-164.4</td>
<td>24.9</td>
</tr>
<tr>
<td>TOTAL</td>
<td>61.5</td>
</tr>
</tbody>
</table>

| 866.0 | Ideal Capacity |
| -164.4 | Capacity Effects |
| 701.6 | Net Capacity, Btu/hr. |

71.1 | Ideal Power for Net Capacity
61.5 | Friction and Compression Losses
46.6 | Motor Loss @ 74% Efficiency
179.2 | Total Power, Watts
THE EXTENDED ANALYSIS

Since the simulation and experimental results tend to approximate each other, the significance of several of the compressor's design features can be inferred with some confidence from the details provided by the simulation.

Table 2 presents a summary of simulation results for the compressor's performance at the AHAM test point. For this analysis the mechanism is divided into three major components: the rotor, vanes, and cylinder. The effects due to various mechanism processes are grouped by component. Rotor face and bearing effects are included with the rotor. Vane tip, slot, and edge effects are included with the vanes. Minimum rotor clearance, reexpansion, discharge flow, and suction path superheat effects are included with the cylinder.

Table 2 shows that the rotor design affects both capacity and power consumption. The flow of refrigerant and oil from the sump accounts for a large capacity loss in direct leakage and associated superheat. Although friction power is small, the compression power required to recirculate the sump flow is substantial.

Table 2 shows that the vanes have a significant effect on power consumption due to friction. The major portion of vane friction power is associated with the tips which slide along the cylinder wall. It should be noted that the cylinder and rotor participate in causing vane friction since the cylinder surface affects the sliding friction coefficient and the rotor/bearing design affects the fluid pressure at the base of the vanes. The relatively small capacity effects linked to the vanes are due to leakage past the vane edges.

Table 2 shows that the cylinder has an important effect on capacity. The capacity loss is caused by leakage past the minimum rotor clearance and heating of the refrigerant in the suction path. The cylinder compression power is the result of the re-expansion and over-compression processes.

CONCLUSION

A new computer simulation has been applied to the performance analysis of a sliding-vane rotary refrigeration compressor. The simulation attempts to describe the behavior of the compressor mechanism using mathematical models for a number of mechanical, thermal, and fluid flow processes presumed to occur during operation. Simulation results for net capacity, total power by component, and oil flow have been verified by experimental data. Inferences drawn from simulation details show that the participation of oil has significant effects on capacity and power consumption, the rotor/bearing design plays a fundamental role in determining oil flow and vane friction, and superheat capacity losses are caused by both sensible heat transfer and leakage flow mixing.

REFERENCES


