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R410A ROTARY COMPRESSOR BEARING DESIGN CONSIDERATIONS

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ABSTRACT

In this paper we explore two design alternatives for a 12000 BTU/hr rolling piston compressor. Starting with an R22 compressor we will reduce the displacement of the compressor to the R410A displacement by either reducing the cylinder height or by increasing the roller diameter. The bearing loads are compared for these compressors.

INTRODUCTION

The planned phase-out of chlorinated refrigerants has left the air conditioning industry looking for an alternative to R22. Two candidates are R407C and R410A. The attractive feature of R407C, a 23/25/52 blend of R32, R125 and R134a, is that its operating pressures and volume flow rates are similar to R22, which will allow it to be used in the same systems. Because it is a “drop in replacement”, manufacturers of air conditioning equipment will not need to make any major hardware changes. For this reason R407C will probably be the refrigerant of choice at first. The attractive feature of R410A, a 50/50 blend of R32 and R125, is that its flow properties allow for smaller tubing, heat exchangers, and compressors. Therefore, the end product is potentially smaller. For this reason, R410A will probably appear first in window room air conditioners where size is an important consideration to the customer. The disadvantage to R410A is that operating pressures are significantly higher. Since pressure induced stresses in tubing is directly proportional to diameter, the smaller tubing used in R410A systems will theoretically compensate for the higher pressures. However the compressor design is significantly affected in three ways. First, the displacement must be reduced for a given capacity, second, the shell of the compressor must be strengthened for the higher system pressures, and third, the bearing loads may be higher, depending on the design and how the reduction in displacement is achieved. In general, bearing loads are a function of the suction to discharge pressure difference. The larger pressure difference in systems with R410A implies higher bearing loads.

ANALYSIS

In rotary compressors, there are the three journal bearings, known as the main bearing, the eccentric bearing, and the outboard bearing. There are also bearing surfaces associated with the vane. The two critical bearing surfaces on the vane are the vane slot and the vane tip. Of lesser importance are the bearing surfaces between the ends of the vane and the end plates that constrain the vane axially in the compressor. Theoretically these vane end bearings surfaces have no load on them and will not be discussed here.

Table 1 shows a comparison between an R22 and an R410A system operating at ARI rating point conditions. Note that the suction gas density with R410A is significantly higher then with R22. Ideally the capacity is directly related to the product of the suction gas density and the displacement of the compressors. Therefore the required displacement for the R410 compressor is reduced. Note also that the discharge to suction pressure difference has increased. This would cause higher bearing loads in the same machine. However, since the required displacement is reduced, there is an opportunity to mitigate the higher bearing loads, depending on the selection of cylinder height, cylinder bore, and roller diameter.

Computer simulation was used to predict bearing loads in two concept R410A rotary compressors under endurance test conditions. The results are compared with the same capacity R22 compressor in Table 2. All compressors use the same cylinder bore. The Concept 1 R410A compressor has the same roller outside
diameter (OD) as the R22 compressor and uses a reduced cylinder height to achieve the reduced displacement. The second compressor achieves the reduction in displacement by increasing the roller OD while maintaining the same cylinder height as the R22 version. In all cases the cylinder block has the same top view.

Vane Slot Loads

The portion of the vane extending into the cylinder has a net force acting on it that is the product of its area and the pressure difference between suction pressure, which acts on one side, and the compression volume pressure, which acts on the other. Both the extending area and the compression volume pressure (a.k.a. cylinder pressure) are a function of time. After vane bottom dead center (VBDC), the area of the vane extending into the cylinder is decreasing while the cylinder pressure is increasing. However the pressure is rising faster than the area is falling and the result is that the vane side pressure load rises until cylinder pressure reaches discharge pressure at which point it reaches a maximum.

This pressure force acting on the vane extension is balanced by a reaction force and moment from the vane slot. For design purposes it is useful to think of these reactions as an equivalent set of forces acting at the top and bottom of the vane slot bearing. These reaction forces are also a function of the distance between the two points at which they act, and this also varies with crank angle. For the purposes of this paper the vane slot force closest to the roller will be referred to as the lower vane slot force. The lower vane slot force is always larger than the upper, and it is cited here for comparison purposes. Figure 1 shows cylinder pressure and lower vane slot load as a function of crank angle for the R22 compressor. Figure 2 shows lower vane slot load for the two R410A compressors. As can be seen, the peak value is about the same for all compressors. Note however that the load is conceptually applied over the height of the cylinder, so the unit load (i.e. force per unit length) is more important than the magnitude of the load itself. The unit load is much higher for the short cylinder height Concept 1 compressor.

Vane Tip Loading

Under normal operating conditions the vane tip is in contact with the roller. In a high side compressor (i.e. discharge pressure in the housing) the contact force between the tip and roller is primarily due to discharge pressure acting on the back of the vane. This is reduced somewhat by the pressure distribution on the tip of the vane. Suction pressure acts over that portion of the vane tip on the suction side of the vane-roller contact point, and cylinder pressure acts over the portion of the vane tip on the discharge side. Again the cylinder pressure varies with time, as does the position of the vane tip in contact with the roller. Cylinder pressure and vane tip normal load is plotted against crank angle in Figure 3. Vane tip load is plotted for the two concept R410A compressors in Figure 4. Peak vane tip normal load happens about 90 degrees after vane top dead center (VTDC), though its value is relatively constant between 0 and 120 degrees. This can be explained as follows. The component of the tip load in the direction of the vane centerline, call this the x-direction, peaks out at VTDC because there is only suction pressure on both sides of the vane-roller contact point. This continues until about 40 degrees after VTDC when the suction port is sealed off by the cylinder-roller contact point. But even then the cylinder pressure is changing very slowly. Thus the x-component of the tip load remains fairly constant. However, because of the vane tip radius, the angle of the normal vector at the contact point on the vane tip increases until about 90 degrees after VTDC. This creates a y-component of the tip force vector. This is what causes the slight rise in vane tip load magnitude after VTDC. Inertia forces are relatively small under endurance test conditions.

As shown in Table 2, the vane tip load is significantly higher for the Concept 2 R410A compressor. However after dividing by the cylinder height, the unit load is about the same for the two R410A compressors. Unfortunately the unit load is significantly higher for both R410A models.

Eccentric Bearing Loads

Peak eccentric bearing load is also a strong function of the high side to low side pressure difference. Again, inertia and vane spring forces are relatively small, and in any case will not vary with system pressures. The cylinder to suction pressure difference acts on the rectangular area formed by the cylinder
height and the line between the two contact points (vane-roller and cylinder-roller). This area varies with time, as does the cylinder. Figure 5 shows cylinder pressure and eccentric bearing load as a function of crank angle for the R22 compressor. Figure 6 shows the eccentric bearing load for the two R410A compressors. For the same mechanism charging with R410A would lead to significantly higher bearing loads. However, for a given capacity, the required displacement with R410A is .71 times that required for R22. If this reduction in capacity were taken by reducing the cylinder height to .71 times the R22 compressor cylinder height, then the peak eccentric bearing load would be reduced proportionately which almost puts it back to the original R22 compressor value.

Main and Outboard Bearing Loads

Except for some minor inertia effects, the main and outboard bearing loads follow directly from the eccentric bearing loads. Because the first concept compressor has the smaller eccentric bearing load, the Concept 1 main and outboard bearings have the smaller loads of the two R410 compressors.

CONCLUSIONS

From the above theoretical analyses we can draw the following conclusions:

1. Reducing cylinder height can reduce journal bearing loads. The higher pressure difference from using R410A does not necessarily mean higher bearing loads if the lower displacement needed for R410A is achieved solely by reducing cylinder height, as in the Concept 1 compressor. Maintaining journal bearing loads at R22 levels means that bearing dimensions, in other words diameters, lengths, and clearances, do not need to change. Furthermore, crankshaft deflection will not increase, so increasing the structural stiffness of the crankshaft and main bearing will not be required.

2. Conversely, increasing roller diameter can reduce unit vane slot loads. Since vane slot lubrication is less understood than journal bearing lubrication, one can argue that maintaining R22 levels by increasing roller diameter and maintaining cylinder height, as in the Concept 2 compressor, is the way to go. Increasing the bearing lengths or diameters can compensate for the increased journal bearing loads.

3. Unit vane tip loads are going to go up in either case. Further research in vane tip tribology and extensive endurance testing with alternative materials and lubricants will be required.

Table 1: R410A vs. R22 ARI Rating Point Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>R22</th>
<th>R410A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporating temperature = 45 F</td>
<td>90.761 psia</td>
<td>144.46 psia (dew point)</td>
</tr>
<tr>
<td>Condensing temperature = 130 F</td>
<td>311.58 psia</td>
<td>490.11 psia (dew point)</td>
</tr>
<tr>
<td>Return gas temperature = 65 F</td>
<td>68.06 BTU/lbm</td>
<td>77.397 BTU/lbm</td>
</tr>
<tr>
<td>Temperature of Liquid entering expansion valve = 115 F</td>
<td>176.31 lbf/m</td>
<td>148.05 lbf/m</td>
</tr>
<tr>
<td>Enthalpy increase across expansion valve and evaporator</td>
<td>178.05 lbf/m</td>
<td>176.31 lbf/m</td>
</tr>
<tr>
<td>Mass Flow Rate required for 12000 BTU/hr Capacity</td>
<td>1.5602 lbf/m</td>
<td>2.2132 lbf/m</td>
</tr>
<tr>
<td>Suction gas density</td>
<td>113.0 cu.ft/hr</td>
<td>80.449 cu.ft/hr</td>
</tr>
<tr>
<td>Volume flow rate of suction gas for 12000 BTU/hr</td>
<td>0.9433 cu.in</td>
<td>0.6716 cu.in</td>
</tr>
<tr>
<td>Ideal displacement for 12000 BTU/hr at 3450 RPM</td>
<td>8.3E-6</td>
<td>9.0E-6</td>
</tr>
<tr>
<td>Suction gas viscosity</td>
<td>10.8E-6</td>
<td>12.1E-6</td>
</tr>
<tr>
<td>Discharge gas viscosity (200 F)</td>
<td>0.9433 cu.in</td>
<td>0.6716 cu.in</td>
</tr>
<tr>
<td>R410A displacement / R22 displacement</td>
<td>90.761 psia</td>
<td>144.46 psia (dew point)</td>
</tr>
</tbody>
</table>
Table 2: Endurance Test Bearing Load Comparison
Evaporator Temperature = 20 F
Condenser Temperature = 155 F

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Cylinder Height (inches)</th>
<th>Roller OD (inches)</th>
<th>Displacement (cu.in)</th>
<th>Lower Vane Slot Load (lbf)</th>
<th>Unit Lower Vane Slot Load (lbf/in)</th>
<th>Vane Tip Normal Load (lbf)</th>
<th>Unit Vane Tip Normal Load (lbf/in)</th>
<th>Peak Ecc Brg Load (lbf)</th>
<th>Unit Peak Ecc Brg Load (lbf/in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Production</td>
<td>R22</td>
<td>0.985</td>
<td>1.3646</td>
<td>0.985</td>
<td>157</td>
<td>159</td>
<td>53</td>
<td>54</td>
<td>447</td>
</tr>
<tr>
<td>Concept 1</td>
<td>R410A</td>
<td>0.718</td>
<td>1.3646</td>
<td>0.717</td>
<td>172</td>
<td>240</td>
<td>58</td>
<td>80</td>
<td>493</td>
</tr>
<tr>
<td>Concept 2</td>
<td>R410A</td>
<td>0.985</td>
<td>1.4855</td>
<td>0.717</td>
<td>158</td>
<td>160</td>
<td>78</td>
<td>79</td>
<td>722</td>
</tr>
</tbody>
</table>

Figure 1: Cylinder Pressure and Vane Slot Loading for the R22 Model

Figure 2: Vane Slot Loading for Concept 1 and Concept 2 R410A Models
Figure 3: Cylinder Pressure and Vane Tip Loading for R22 Model

Figure 4: Vane Tip Loading for Concept 1 and Concept 2 R410A Models

Figure 6: Cylinder Pressure and Eccentric Bearing Load for R22 Models

Figure 6: Eccentric Bearing Load for Concept 1 and Concept 2 R410A Models