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The Effect of Flashing Refrigerant on Mechanical Shaft Seal Face Temperatures

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THE EFFECT OF FLASHING REFRIGERANT ON MECHANICAL SHAFT SEAL FACE TEMPERATURES

Jointly submitted by:


ABSTRACT

This paper will report on a series of tests designed and conducted to evaluate the effect of flashing refrigerant on mechanical shaft seal face temperatures. This testing is important because seal reliability and performance are very closely related to seal face temperature. Field experience has shown that when face temperature is allowed to climb above a critical level, refrigerant and oil solutions lose important lubricating properties. This degradation of the lubricant can result in damage to the seal face tribological pairs and thereby cause excessive leakage. It is known that most compressor lubrication systems deliver oil containing a certain percentage of dissolved refrigerant to the seal chamber. Depending upon the pressure and temperature differentials in any system the potential exists for this refrigerant to flash out of the oil as it enters the seal chamber. As a result, a foaming mixture often exists in the seal chamber and this oil/gas foam is presented to the seal faces for lubrication. It was the authors' hypothesis that seal faces run hotter in foaming lubricants than in a liquid oil condition. A discussion of the test results and some general conclusions are offered in the closing section of the paper.

INTRODUCTION

Carbon graphite versus a metallic face pairing has been almost universally employed in shaft seals of open refrigeration compressors and has served the industry well for many years. Excessive seal leakage due to blistering of the carbon graphite surface was limited to the occasional incident and the resulting oil and gas leakage was treated by simply replacing the seal assembly. In recent years, however, blistering has become a major issue and the most common solution has been to use non-absorbent hard face pairings such as Silicon Carbide. While this eliminated blistering, the average seal life and performance was still not acceptable to the refrigeration industry. The change from a forgiving face combination to hard faces led to a focus on the seal environment to improve performance.

Examination of the hard faced seals, removed from compressors, frequently revealed evidence of marginal lubrication at the seal faces. For acceptable life and performance, mechanical seal faces must run on a thin liquid film of lubricant. Much has been written about the selection of the optimum seal design and tribological face pair which will maintain a stable and reliable running film. (See references 1, 2, 4, 5) This paper however focuses on the effect that the seal environment has on the stability of the lubricating film.

Two important assumptions are made. The first is that thin films are more stable at lower temperatures and the second that stable films will result in lower face temperatures. Supporting evidence comes from bearing reliability studies as well as from the experience of mechanical seal users and suppliers. (See reference 3) Lubricating oils, which provide the face film, do not function properly above certain critical temperatures; consequently, a cooler running seal is more reliable.

Based on the assumptions above, a temperature study was conducted to identify the best environmental conditions for optimal seal performance. A test rig was designed to study key system variables and their effects on oil condition at the seal interface. These conditions were then correlated in real-time to the operating seal face temperatures. The independent variables were inlet oil temperature, oil flow rate, inlet oil pressure and seal chamber pressure. Direct observations also indicated temperature reductions could be achieved by modifying the oil distribution to the faces by using a multi-port injector. (See Figure 1)
The compressor manufacturer performed the experiments on a test rig that was hard piped to a working R22 chiller system. The oil supply line to the test chamber was connected to the compressor oil sump in parallel with the actual compressor lubrication system. All of the tubing, fittings and filters employed in the chiller system were replicated in the oil supply to the test chamber. The chiller was then operated during the tests in order to supply oil to the test rig. The test seal itself was mounted in the test apparatus on a rotating shaft, driven independently but at the same speed as the compressor, and run under conditions which imitated actual compressor operating parameters. In this manner the live chiller imposed its operating conditions on the test rig. This is an important element of the test as these fluid mixtures and environmental conditions are very difficult to duplicate in an ordinary laboratory test stand. The seal chamber, including the seal interface area, was fully visible during the tests as the rig was designed to include several large glass viewing windows. (See Figures 2 & 5)
A 2 3/4" (60mm) shaft diameter edge-welded metal bellows shaft seal was used in the testing. The stationary face of the seal was fitted with three thermocouples mounted in drilled holes in the back of the silicon carbide mating ring. This arrangement allowed temperature measurements to be taken within .100" (2.5 mm) of the seal interface. Both seal faces were of silicon carbide material. In a number of tests a separate multi-port injector ring was installed in the test chamber. The face temperatures obtained when using the injector were measured and compared with those obtained with the standard oil delivery system.

All of the key process conditions in the testing could be varied within finite limits. The seal chamber inlet oil temperature was adjusted by piping the supply through a small water-cooled heat exchanger. Inlet oil pressure was manipulated by directing the flow through one of two different sized orifices. The chamber's internal pressure and oil flow rate were adjustable via a manual outlet throttle valve. The flow rate of the oil/refrigerant solution was measured with an in-line flowmeter mounted up-stream of the test rig. Additional sight glasses were also installed in other key locations to allow visual inspection of the fluid throughout the system.

The test procedure consisted of creating a series of different conditions (via the live chiller) in the seal chamber rig while recording the stationary face temperature. By altering the independently variable parameters, as indicated above under Introduction, the oil/refrigerant solution in the test chamber could be induced to change state from full liquid to a flashing or total foam condition and back again to a liquid. (See Figure 3) A clear relationship was established between the state of the oil/refrigerant solution and the seal face temperature. As with many experiments of this sort, the strongest influences on face temperature could be reduced to just a few important variables. The tests demonstrated that these included the inlet oil temperature and the oil flow rate through the seal chamber.

The following tests were conducted with the following conditions:
Condenser pressure or Oil Supply Pressure: 160 psig (10.9 bar g)
Evaporator pressure or Drain Pressure: 75 psig (5.1 bar g)
Inlet oil temperature was 85 °F (29 °C)
The oil was a Polyol ester and the refrigerant was R22.
## RESULTS

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Injection Fitted</th>
<th>Orifice Inlet</th>
<th>Orifice Outlet</th>
<th>Flow Rate Chamber Pressure Face Temp. Conditions in Chamber</th>
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<tbody>
<tr>
<td>1</td>
<td>Yes</td>
<td>L</td>
<td>S</td>
<td>4.3 Usgpm 16.3 l/min 125 psi 8.5 g 267 °F 131 °C Corona</td>
</tr>
<tr>
<td>2</td>
<td>Yes</td>
<td>L</td>
<td>L</td>
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</tr>
<tr>
<td>3</td>
<td>Yes</td>
<td>L</td>
<td>L</td>
<td>3.0 Usgpm 11.4 l/min 135 psi 9.2 g 275 °F 135 °C Corona</td>
</tr>
<tr>
<td>4</td>
<td>Yes</td>
<td>L</td>
<td>S</td>
<td>2.2 Usgpm 8.4 l/min 138 psi 9.4 g 287 °F 142 °C Corona</td>
</tr>
<tr>
<td>5</td>
<td>Yes</td>
<td>S</td>
<td>S</td>
<td>1.9 Usgpm 7.2 l/min 129 psi 8.8 g 291 °F 144 °C Foam</td>
</tr>
<tr>
<td>6</td>
<td>Yes</td>
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<td>L</td>
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<tr>
<td>7</td>
<td>No</td>
<td>L</td>
<td>S</td>
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</tr>
<tr>
<td>8</td>
<td>No</td>
<td>L</td>
<td>S</td>
<td>4.9 Usgpm 18.6 l/min 125 psi 8.5 g 302 °F 150 °C Corona</td>
</tr>
<tr>
<td>9</td>
<td>No</td>
<td>L</td>
<td>L</td>
<td>3.4 Usgpm 12.9 l/min 127 psi 8.6 g 304 °F 151 °C Corona</td>
</tr>
<tr>
<td>10</td>
<td>No</td>
<td>L</td>
<td>L</td>
<td>4.7 Usgpm 17.9 l/min 119 psi 8.1 g 304 °F 151 °C Corona</td>
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<tr>
<td>11</td>
<td>Yes</td>
<td>S</td>
<td>S</td>
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<tr>
<td>12</td>
<td>No</td>
<td>S</td>
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<tr>
<td>13</td>
<td>No</td>
<td>S</td>
<td>S</td>
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<tr>
<td>14</td>
<td>No</td>
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<tr>
<td>15</td>
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</tr>
</tbody>
</table>

### Legend
- **Corona**: Corona visible in otherwise liquid oil.
- **Foam**: Seal chamber completely filled with oil/refrigerant foam.

### Configuration
- **Yes**: Multi-port injector fitted
- **No**: Multi-port injector not fitted

### Inlet Orifice Diameter
- **S** = 0.156 in. (3.9 mm.)
- **L** = 0.375 in. (9.5 mm.)

### Outlet Orifice Diameter
- **S** = 0.094 in. (2.4 mm.)
- **L** = 0.125 in. (3.2 mm.)
DISCUSSION OF RESULTS

Some of the results did not support one of our initial assumptions; namely, that lower face temperature would be recorded when keeping the oil/refrigerant solution free from flashing and foam. The seal chamber temperature and pressure readings clearly placed the oil quality in the fully liquid regime. At similar oil flow rate conditions there was little difference in face temperature recorded when the lubricant was in the fully liquid or total foam state.

However, visual evidence revealed the development of a corona of gas at the periphery of the seal interface even when the rest of the seal chamber clearly contained oil in a liquid state. (See Figure 4) The corona was not visible at start up but only when the seal faces reached a critical operating temperature. This condition was shown to occur above very specific temperatures which were themselves dependent on the particular conditions in the test chamber. It is very doubtful if this would have been fully appreciated without the visual capability of the test rig. This observation resulted in seal design modifications that have proven to be effective in significantly lowering seal face temperatures and consequently improving overall seal performance.

Figure 4  Face Corona

CONCLUSIONS

1. Even when oil lubricating conditions may appear ideal in the seal chamber, the conditions very close to or at the periphery of the seal faces themselves may not be ideal. Pressure and temperature measurements, of themselves, in or near the seal chamber do not reliably predict whether boundary layer conditions exist at the periphery of the seal faces of a rotating shaft seal.

2. Direct visual observations are a valuable diagnostic tool when attempting to optimize the oil flow and seal chamber configuration. As each compressor design gives rise to unique flow patterns, it is a very cost effective method of confirming that design changes work before expensive field trials are started.

3. The seal environment can be improved with the use of a multi-port injection oil delivery system. Temperature reductions of 60 degrees F (15.6 C) were measured. Recent work is indicating that the size and configuration of the porting can also play an important part in reducing the seal face temperature.
4. The effect of a correctly designed multi-port injector is to break up the corona at the seal interface and allow liquid to reach the faces for lubrication.

5. Refrigerant flashing in the seal chamber does not always increase temperature at the faces. The major influences on face temperature are oil flow rate and inlet oil temperatures. In general, the lower the inlet temperature and the higher the flow rate, the lower will be the face temperature with a corresponding improvement in seal performance.

6. From the growing body of evidence from the field both in the USA and Europe, we believe that the principles established in this paper are likely to be applicable to other oil/refrigerant solutions, and particularly those where synthetic oils of any type are involved. Further testing would be needed to confirm this belief but improving the seal environment will always be repaid by improved seal performance.

Figure 5
Test Rig Schematic

REFERENCES & ACKNOWLEDGMENTS


6. Thanks to Richard Morse of York International for his help in photographing the test rig and for his enthusiastic encouragement in the project. Also, thanks to Midnight Graphics, York, PA for the digital modeling.