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REMOVING THE NON-RETURN VALVE FROM A REFRIGERATION SYSTEM WITH A TWIN SCREW COMPRESSOR

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ABSTRACT

The removal of the non-return valve from the refrigeration system with a twin screw compressor is realizable by fitting a non-reversing clutch to the prime mover to restrain the reverse rotation of the compressor following shut down. Utilizing a suite of computer programs developed by the authors, the influence of the non-return valve is examined; gas torques under normal running condition and on shut down are calculated and compared. The results suggest that the use of the non-reversing clutch results in a decrease in energy consumption and an increase in system COP. The shut down torque may have a peak value considerably higher than the normal driving torque, thus care must be taken with the choice or design of the clutch.

1 INTRODUCTION

Refrigeration systems driven by twin screw compressors are normally fitted with a non-return valve in the suction line of the compressor to prevent the gas in the high pressure side from flowing backwards through the compressor into the suction chamber following system shutdown. This reverse flow would cause the compressor and prime mover to run in the reverse direction at a dangerously high speed. However, the use of non-return valves has the disadvantage of causing extra gas pressure drop, resulting in an increase in energy consumption and a decrease in the coefficient of performance (COP) of the refrigeration system. In view of the huge number of refrigeration applications worldwide, even a small gain in plant efficiency is of significance and this is achievable with the removal of the non-return valve, especially from systems which have long periods of continuous running and infrequent shutdowns. Furthermore, malfunctions of non-return valves have been caused by contamination of the refrigerant. As a consequence, removing the valve results in an increase in system reliability. The purpose of the non-return valve in the high pressure side is to prevent the reverse flow of liquid refrigerant from the condenser into the oil separator following shutdown.

If the suction side non-return valve is removed, something must be done to prevent reverse rotation of the compressor and prime mover following shutdown. A practical approach is to fit a non-reversing clutch to the free end of the prime mover shaft. However, care must be taken with the design of the clutch since the shutdown torque caused by the high pressure gas filling the cavities of the compressor may reach a peak value that is considerably higher than the normal running torque.

Following the shutdown of a system without a suction non-return valve, gas will flow from the oil separator volume, which is of course at high pressure, backwards into the compressor and leak through various leakage paths to fill up the compressor cavities as shown in Fig. 1. A "quasi-steady-state" is reached when the mass flow rate into a cavity equals that out of the cavity. At this stage the pressures developed in the cavities attain their peak values and will remain unchanged as long as the pressures in the discharge and suction chambers remain constant. The pressure pattern developed in the cavities following shut down is not the same as that for normal running. Consequently, the gas torques resulting from this pressure pattern may be rather different from those due to normal running conditions, and the difference between them may vary with the running conditions. For safe operation, the non-reversing clutch must be designed to transmit the maximum gas torque following shutdown. The determination of the shut down torque is therefore of importance to the designers of such systems.

The most economic and powerful technique available for the determination of the static gas torque is computer simulation. Utilizing a suite of computer programs developed by the authors which are capable of simulating the compressor working processes and calculating the gas torques, the influence of the pressure drop across the non-return valve on the system COP and power consumption of a conventional single stage refrigeration system is examined; for both
normal running and shut down conditions the pressure distributions in the cavities of a real compressor are presented as an example, from which the normal running and shutdown torques are calculated and compared. The results obtained and the suggestions made may help the designers and operators to ensure the safe operation of a system with a non-reversing clutch used in preference to a non-return valve.

Fig. 1 Leakage paths in a twin screw compressor

2 COMPUTER PROGRAMS USED, COMPRESSOR SPECIFICATIONS AND RUNNING CONDITIONS

The following computer programs developed by the authors are used to do the calculations presented in this paper:

Profile Generation Program: This program is a profile library which can generate any profile within the SRM D definition. The profile generation program is the basis of all the research work. It calculates and outputs its calculated results into different data files, which are required by other programs for the calculation of geometrical characteristics, cutter blade shapes, etc.

Geometrical Characteristic Calculation Program: This program calculates all the required geometrical characteristics and parameters for both the working process simulation and design purposes. All relevant geometrical characteristics are expressed as a function of the male rotor rotational angle, especially those which will be used in the working process simulation program. The start angle is the angle where the cavity volume equals zero.

Working Process Simulation Program: This program simulates working processes of a refrigeration twin screw compressor which may run under various conditions, such as partial loading, oil injection, liquid refrigerant injection, gaseous refrigerant superfeeding and different refrigerants etc. The program calculates the thermodynamic properties in the cavity as a function of the male rotor rotational angle or of the cavity volume. The volumetric and indicated isentropic efficiencies are also calculated.
Force Analysis Program: This program calculates the resultant gas forces and torques applied to male and female rotors and the axial and radial components of the bearing reactions caused by the gas forces. Factors such as axial gas forces acting on helical screw surfaces and on rotor ends are all taken into account. The results of calculated forces and torques are given as functions of male rotor angle of rotation.

The specifications of the compressor used for the calculations are as follows:
- Rotor lobe profile: SRM D standard
- Male rotor wrap angle: 263.6°
- Male and female rotor diameters: 204 mm
- Length/diameter ratio: 1.45

The supposed running conditions:
- Refrigerant: R22
- Condensing temperature: 35 °C
- Superheat degrees: 25 °C
- Male rotor driving speed: 3000 rpm
- Evaporating temperature: 0 °C
- Oil injected: Yes

3 THE INFLUENCE OF NON-RETURN VALVE ON COP OF THE SYSTEM

A single stage refrigeration system is assumed in this study as shown in Figure 2. For a range of pressure drops due to the non-return valve in the suction line the power consumption and the COP of the system have been computed and are given in Figures 3 and 4, respectively.
As shown in Fig. 3 the indicated power of the compressor increases linearly with the increase in the pressure drop across the non-return valve in the suction line. For a pressure drop of 0.20 bar, the indicated power increases by 0.84 kw, about 0.5 percent of that of the same system without the non-return valve.

![Graph showing the COP of the system vs the pressure drop across the non-return valve](image)

Fig. 4 The COP of the system vs the pressure drop across the non-return valve

The detrimental effect of the non-return valve in the suction line is also apparent in Figure 4, showing a linear decrease in the COP of the system with an increase in the pressure drop. Similarly, for a pressure drop of 0.20 bar, the COP of the system decreases by 5.0 percent compared with that of the same system after the removal of the non-return valve.

4 COMPARISON BETWEEN NORMAL RUNNING TORQUES AND SHUTDOWN TORQUES

For the compressor and operating conditions specified, the gas pressures in the cavities are obtained from the thermodynamic performance simulation program for both the normal running condition and the shut down situation. Both are shown in Figure 5 as functions of male rotor angle of rotation. The angle of zero degrees corresponds to the position where the cavity volume equals zero. As expected, a big difference exists between these two pressure distribution patterns. Unlike the smooth pressure-volume curve due to normal running, the pressure distribution in the cavities at the quasi-steady-state following shut down exhibits a step curve pattern. The shutdown pressure is higher than normal running pressure at many rotation angles of the male rotor. The possibility thus exists that the maximum shutdown torque is larger than that due to normal running.

![Graph showing pressures in cavities vs male rotor angle of rotation](image)

Fig. 5 Pressures in cavities vs male rotor angle of rotation

Figure 6 reveals the difference between the torques due to normal running and shut down. The friction torque of the compressor is not considered in this analysis. The zero-degree angle in this figure is the angle where the tip of the male rotor is located on the line connecting the male and female rotor centres in the suction end plane. The torque curves vary with a period of 90°, that is 360° divided by the number of the male rotor lobes. For the conditions described in
section 2, the peak value of the shut down torque is 8.3 percent higher than the peak value of the normal running torque, and 14.6 percent higher than the driving torque (taken as the average of the normal running torque here). The peak value of the shut down torque occurs at the male rotor rotational angle of 9°. Since the stop position of the rotors is absolutely random, a clutch chosen or designed according to the driving torque runs the risk of being broken.

![Graph showing torques due to gas pressure](image)

Fig. 6 Gas torques under full load condition vs male rotor angle of rotation

It should be noted that the difference between the normal running torque and the shut down torque will vary with the operating conditions. As an example, five different evaporating conditions are used in this study and the results are tabulated in Table 1, in which only peak values of the torques and the corresponding rotor positions are given. It can be seen that the percentage by which the shut down torque exceeds the driving torque increases with the decrease in evaporating temperature. Since the discharge pressures are same for all these cases, it can be concluded that the extent to which the maximum shut down torque exceeds the driving torque increases with the increase in the pressure ratio of the compressor. In contrast, the change in evaporating temperature has little influence on the angular position of the peak value of the shut down torque.

Table 1 The influence of evaporating temperature on the maximum shutdown torque

<table>
<thead>
<tr>
<th>Evaporating temperature (°C)</th>
<th>Shutdown torque (N.m)</th>
<th>Driving torque (N.m)</th>
<th>Percentage by which the shutdown torque exceeds the driving torque</th>
<th>Corresponding rotor position (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>650.61</td>
<td>567.90</td>
<td>14.56</td>
<td>9</td>
</tr>
<tr>
<td>-5</td>
<td>718.58</td>
<td>580.65</td>
<td>23.75</td>
<td>9</td>
</tr>
<tr>
<td>-10</td>
<td>780.32</td>
<td>591.81</td>
<td>31.85</td>
<td>9</td>
</tr>
<tr>
<td>-15</td>
<td>835.04</td>
<td>600.51</td>
<td>39.06</td>
<td>7</td>
</tr>
<tr>
<td>-20</td>
<td>883.32</td>
<td>607.50</td>
<td>45.40</td>
<td>7</td>
</tr>
</tbody>
</table>

5 SUGGESTIONS FOR SHUTTING THE SYSTEM DOWN SAFELY

From the above discussion, it is clear that the non-reversing clutch in a refrigeration system should be chosen or designed according to the maximum shutdown torque, which guarantees a safe shut down of the system whether the shutdown is intentional or unintentional.

However, if the clutch is chosen or designed according to the driving torque due to the lack of the computing programs or measurement data, making use of the slide valve fitted in the compressor will help to achieve a safe shut down of the system. Fully opening the slide valve alters the pressure pattern in the cavities of the compressor and results in a large reduction of shutdown gas torque. Fig. 7 shows the shutdown torque calculation results for the specified compressor under partial load conditions, from which it can be seen that the shutdown torque under partial load condition is much
lower than the driving torque under full load condition. It is thus suggested that the slide valve should be fully opened prior to the shutdown of the system, which would greatly reduce the gas torque applied to the clutch following shut down. However, it must be pointed out that this approach does not relieve the risk of the clutch breaking following an unintentionally shut down of a system running under full load condition.

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Fig. 7 Gas torques under partial load condition vs male rotor angle of rotation

6 CONCLUSIONS

The following conclusions can be drawn from this study:

• The COP of a refrigeration system with a twin screw compressor can be enhanced and the energy consumption of the compressor can be reduced by removing the non-return valve from the system. For the specified compressor and running condition, a 5.0% increase in COP and a 0.5% decrease in indicated power can be achieved after the removal of a non-return valve in the suction line which causes a gas pressure drop of 0.20 bar.

• A non-reversing clutch can be used instead of a non-return valve to prevent the reverse rotation of the compressor and prime mover. Since the peak value of the shutdown torque may be considerably larger than the driving torque and the stop position of the rotors is absolutely random, the clutch must be chosen or designed according to the maximum shutdown torque which varies with running conditions.

• In the case of clutches designed according to the driving torque due to the lack of appropriate facilities for calculating static and running torques, it is suggested that the compressor should be unloaded via its slide valve prior to the shutdown of the system. This will ensure that the maximum shutdown torque is smaller than that under the full load running condition.

ACKNOWLEDGEMENTS

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REFERENCES

