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Optimisation Techniques Applied to the Design of a Refrigeration TiN Screw Compressor

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Some results from a procedure for optimising the basic geometrical parameters of rotor profile of a refrigeration twin screw compressor and the position of its discharge port were presented by the authors at the International Compressor Engineering Conference at Purdue, 1992 (1). Besides the basic parameters defining a profile and the discharge port position which influence the performance of a refrigeration twin screw compressor, the rotor definition parameters, such as the wrap angle and the length/diameter ratio, and the slide valve definition parameters also have a great influence on the performance of the compressor.

After the profile is chosen, in order to define the geometry of a pair of rotors completely the wrap angle of the male rotor and the length/diameter ratio of rotors must be decided. These two parameters, especially the wrap angle influence the performance, size and thus the manufacturing cost of the compressor considerably. For the given running conditions, the designer of the compressor should optimise these two parameters so as to design a compressor which has the best performance and a reasonable size. In this paper, the design is restricted to male and female rotors which have equal diameters.

Capacity control is needed to enable a refrigeration twin screw compressor to meet the wide range of load demands which are a feature of modern refrigeration plants. The most common method makes use of a slide valve which allows a measured quantity of the compressed cavity volume to "blow out" back to suction. However, when a compressor is at its full load condition, the slide valve reduces the discharge port area of the compressor, and increases the gas flow resistance through the discharge port so that the indicated efficiency decreases considerably. When at a partial loading condition, the adjusting characteristics of a slide valve, that is, the relationship between the indicated efficiency and the various capacities of the compressor, is defined by the slide stop. If a slide valve has a good adjusting characteristic, the compressor can obtain a high indicated efficiency over a wide range of load. The slide stop is the most important parameter for a slide valve and must be optimised.

In this paper the optimisation technique is developed by making use of programs developed by the authors and measured values derived from compressor tests. The discussion of the optimisation technique concentrates on the
wrap angle of the male rotor, length/diameter ratio and slide stop. The influence of the slide valve on the performance of a machine at the full load condition is also discussed and presented.

2 COMPUTER PROGRAMS USED, COMPRESSOR SPECIFICATIONS AND RUNNING CONDITIONS

The following computer programs developed by the authors are used for the analysis 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 developed by the authors for the compressor geometrical characteristics calculations, cutter blade calculations, etc.

Geometrical Characteristic Calculation Program This program (2) calculates all the required geometrical characteristics and parameters for both the working process simulation and design purposes. Many of the geometrical characteristics are expressed as a function of the male rotor rotation angle, especially those which are 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 (3) 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 the cavity volume, and also the derived parameters and efficiencies describing the behaviour of the compressor.

The specifications of the compressor used for the analysis in this paper are as follows:
- Rotor lobe profile—SRM D standard.
- Male and female rotor diameters (equal): 204mm.

The running conditions are:
- Refrigerant: R22.
- Speed of male (driven) rotor: 3000rev/min.
- Condensing temperature: 308.15K.
- Evaporating temperature: 263.15K.
- Superheat degrees: 30K.
- Oil injected.

3 ROTOR PARAMETERS AND COMPRESSOR PERFORMANCE

After the profile is chosen, the rotor geometry can be defined completely by the following two rotor parameters: the wrap angle of the male rotor and the length/diameter ratio of the rotors. The designers of the compressor should choose these two parameters carefully according to the given capacity and running conditions as they influence the compressor performance and size and thus manufacturing cost considerably.

Fig. 1 shows the relationship between the indicated and volumetric efficiencies and the wrap angle of the male rotor for the specified compressor and running conditions. For the range of wrap angles in Fig. 1, the length/diameter ratio remains constant, at 1.65. The measured and simulated efficiencies for the machine which has a wrap angle of 300° are listed in Table 1. It can be seen from Fig. 1 that the larger the wrap angle, the higher the indicated efficiency, but the lower the volumetric efficiency. These features are explained as follows: when the wrap angle increases, the sealing lines of the cavity increase much faster than its volume which results in greater relative leakage; the discharge port area also increases much faster than the cavity volume which results in a considerable reduction of discharge resistance. As a consequence, the volumetric efficiency decrease is accompanied by an indicated efficiency increase.
Table 1 The measured and simulated efficiencies

<table>
<thead>
<tr>
<th>Simulated volumetric efficiency (%)</th>
<th>Measured volumetric efficiency (%)</th>
<th>Simulated indicated efficiency (%)</th>
<th>Measured total efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>91.6</td>
<td>91.0-91.8</td>
<td>81.0</td>
<td>72.4</td>
</tr>
</tbody>
</table>

The highest volumetric efficiency, 92.1 percent, can be obtained when the wrap angle equals about 240°. The highest indicated efficiency, 83.3 percent, can be obtained when the wrap angle is about 380°. It is a pity that the wrap angles corresponding the highest volumetric and indicated efficiencies do not coincide together. In general a “reasonable” wrap angle should be chosen between the wrap angles corresponding the highest volumetric and indicated efficiencies to make sure that the compressor has relatively high values of both efficiencies. When the wrap angle is being chosen, the relationship between the wrap angle and the overlap constant of the compressor must be considered (see Fig. 2). If a wrap angle larger than the optimum is chosen, the compressor will have an increased size for the given real capacity due to both the low volumetric efficiency and small overlap constant. This means an increase in manufacturing cost. For the compressor specified in this paper, a wrap angle about 300° is a suitable choice.

![Fig. 1 Efficiencies vs the wrap angle of the male rotor](image)

![Fig. 2 The overlap constant vs the wrap angle of the male rotor](image)

The influence of the length/diameter ratio of rotors on the compressor performance is not as great as that of the wrap angle (see Fig. 3). Both the volumetric and indicated efficiencies increase slowly as the length/diameter ratio increases. This is due to the relatively reduced sealing line length for the cavity. From the point of view of improving the compressor performance, a large length/diameter ratio should be chosen, but from the point of view of reducing the dynamic distortion of rotors, a small one should be chosen. If the object is to obtain the smallest bearing forces, an optimum length/diameter ratio must exist. This is beyond the scope of this paper, and the authors will publish on this topic elsewhere.

4 WITH AND WITHOUT SLIDE VALVE

In a refrigeration compressor the slide valve is applied to adjust capacity. The volume ratio for the definition of the axial discharge port is much higher than that for the radial discharge port. Usually the volume ratio for the radial discharge port can be chosen optimally according the running conditions of the compressor (1).

Table 2 The measured and simulated efficiencies

<table>
<thead>
<tr>
<th>Volume ratio</th>
<th>Simulated volumetric efficiency (%)</th>
<th>Measured volumetric efficiency (%)</th>
<th>Simulated indicated efficiency (%)</th>
<th>Measured total efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.6</td>
<td>91.4</td>
<td>89.0-92.2</td>
<td>87.2</td>
<td>75.0</td>
</tr>
<tr>
<td>3.6</td>
<td>91.6</td>
<td>91.0-91.8</td>
<td>81.0</td>
<td>72.4</td>
</tr>
</tbody>
</table>
A slide valve can make a twin screw compressor run with a high indicated efficiency during partial loading of the compressor, but at the full load condition, it reduces the compressor indicated efficiency. Fig. 4 shows the relationship between the indicated efficiency and the volume ratio of the radial discharge port for the specified compressor and running conditions. For the compressor with a slide valve, the volume ratio of the axial discharge port is 5.0, and for the compressor without a slide valve the volume ratio of the axial discharge port has the same value as the radial discharge port. The tested and simulated performance for the compressor with a slide valve is listed in Table 2.

Over a wide range of volume ratio for the radial discharge port, the indicated efficiency for the compressor without a slide valve is higher or much higher than that for the compressor with a slide valve. The highest indicated efficiency for the compressor with a slide valve, 87.3 percent, is obtained when the volume ratio equals 2.75. However, the highest indicated efficiency for the compressor without a slide valve, 88.1 percent, appears when the volume ratio is equal to 2.9. For the specified running conditions, the indicated efficiency of the compressor without a slide valve is 4.76 percent points higher than that of the compressor with a slide valve. This is due to the following reason: the slide valve in the compressor reduces the discharge port area and increases the resistance to gas flow through the discharge port, thus increasing the indicated power considerably. Fig. 5 shows the relationship between the Mach numbers of gas flow through the discharge ports and the rotation angle of the male rotor for the compressors with and without slide valves. Fig. 6 shows the $p-V$ diagrams. The larger the discharge Mach number, the larger is the indicated power and the lower the indicated efficiency.

It is quite common for two or more twin screw compressors to be used in one refrigeration plant. In order to reduce the total energy consumption of the compressors, the authors suggest that in such a situation some of the compressors should be without slide valves. They should have the same volume ratios for the radial and axial discharge ports, and always run under the full loading conditions. The numbers can be decided according to the minimum refrigeration capacity of the plant in a complete loading cycle of one year say.
5 SLIDE STOP AND ADJUSTING CHARACTERISTICS OF A SLIDE VALVE

The most important geometrical parameter describing a slide valve is the slide stop ($L_{stop}$ in Fig. 7). After the volume ratios for the axial and radial discharge ports are chosen, a slide stop must be chosen which limits the adjusting characteristics of the slide valve. An important adjusting characteristic is the relationship between the indicated efficiency of the compressor and its partial loading capacity. A good adjusting characteristic should be to have over a wide range of partial loading capacity a high indicated efficiency. Fig. 8 shows a choice of three adjusting characteristics for the specified compressor and running conditions. The compressor has 3.6 and 5.0 as its volume ratios for the radial and axial discharge ports respectively. As shown in Fig. 8, different slide stops result in different adjusting characteristics. When the slide stop equals 111mm in length, the adjusting characteristic is at its best (solid line in Fig. 8). Fig. 9 shows the relationship between the measured and simulated volumetric efficiencies and the partial loading capacity for the same running conditions and compressor, the slide stop of which is 111mm in length.

Fig. 7 Stop and position parameters of a slide valve

Fig. 8 Adjusting characteristics of different slide stops

Fig. 9 Simulated and tested volumetric efficiency vs capacity

As mentioned in the last section, for the specified running conditions the compressor with a slide valve has its highest indicated efficiency when the volume ratio of the radial discharge port equals 2.75. Fig. 10 shows the adjusting characteristics of the compressor, the volume ratio of the radial discharge port of which is 2.75. When the slide stop equals 76mm, the slide valve obtains its best adjusting characteristic (solid line in Fig. 10). For the two volume ratios, the two optimally adjusted characteristics are different, and their comparison is shown in Fig. 11. Over a wide range of partial loading capacity, the compressor with 2.75 as the volume ratio for its radial discharge port has
a considerably higher indicated efficiency. This means that when a designer designs a twin screw compressor, he or she should decide the optimum volume ratio for the given running conditions and for full load, and then choose a suitable slide stop to ensure that the slide valve has the optimum adjusting characteristics.

6 CONCLUSIONS

1. Of the two parameters which define rotor geometry, the wrap angle and length/diameter ratio, the wrap angle has a greater influence on the performance and size of the compressor and thus its manufacturing cost. The choice of the wrap angle is based on the following three factors: reasonable indicated and volumetric efficiencies and compressor size.

2. A slide valve ensures that the compressor obtains a relatively high indicated efficiency on the partial loading condition, but for full load it reduces the compressor performance considerably. In order to reduce the energy consumption of the compressors, the authors suggest that in the situation in which two or more twin screw compressors are applied in one refrigeration plant one or more of the compressors should be without slide valves. They should have the same volume ratios for the radial and axial discharge ports, and always run under the full load conditions. The compressor numbers, with and without slide valves, can be decided according to the minimum refrigeration capacity of the plant in a complete load cycle of (say) one year.

3. If the running conditions and the volume ratios are fixed, a careful choice of the slide stop can result in a good optimum adjusting characteristic of the slide valve. When designing a twin screw compressor with a slide valve, the optimum volume ratio on the full loading condition can be decided first, and then a suitable slide stop should be chosen to ensure that the slide valve has the optimum adjusting characteristics.

REFERENCES


2 Tang, Yan and Fleming, John S. Computer Aided Geometrical Analysis of the Geometrical Characteristics of a Helical Screw Compressor. International Compressor Engineering Conference at Xi’an, Xi’an Jiaotong University, Xi’an, China, 1993.