Obtaining the Optimum Geometrical Parameters of a Refrigeration Helical Screw Compressor

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OBTAINING THE OPTIMUM GEOMETRICAL PARAMETERS OF A REFRIGERATION HELICAL SCREW COMPRESSOR

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Abstract

This paper presents a method for optimizing the geometrical parameters of a refrigeration twin-screw compressor. The method optimizes rotor profile and discharge port position etc., by making use of two efficient computer programs; the geometrical parameter calculation program and the performance prediction program. Two new parameters; relative blow hole area and relative contact line length, are introduced to compare the blow hole areas and contact line lengths of different rotor profiles. These new parameters, which are determined by profiles alone, have an important and direct influence on the performance of the compressor, so that they are suitable for used in an optimization process. A technique for reducing the blow hole area by manipulating its shape in accordance with a logical procedure is described. Calculation and test results are included and discussed in the paper.

INTRODUCTION

Altuough many factors influence the performance of a helical screw compressor, they may be grouped into two categories as follows:

- The operating parameters such as rotational speed, quantity of oil or liquid refrigerant injected etc..
- The geometrical parameters such as contact line length per lobe, blow hole area and discharge port position etc..

The increase of the quantity of liquid refrigerant injected always results in a slight decrease of the coefficient of performance but the use of the injection of liquid refrigerant eliminates the need for separate oil cooling. If the discharge temperature is lower than that permitted, the quantity of liquid refrigerant injected should be as small as possible. This quantity can be predicted by the computer program of the mathematical model for the working process. Increases of rotational speed and quantity of oil injected always result in a decrease of leakage through any leakage path the consequence of which is an increase in the volumetric efficiency. But on the other hand these increases will result in increases of fluid-dynamic losses, fluid and mechanical friction losses etc., so as to increase the energy consumption. For the rotational speed and quantity of oil injected there exist optimum values, between which there is a relationship. So far these optimum values have been obtained from tests or experience only, since there are no reliable theories of fluid-dynamics and friction yet, which can be used to develop a mathematical model to predict the increase of energy consumption.

The purpose of the procedure for the optimization of the geometrical parameters is to combine the smallest blow hole area, with the shortest contact line length per lobe and the correct discharge port position suitable for a given discharge pressure. This can be achived by the use of test results and two computer programs:

- The geometrical calculation program.
- The program simulating the working process.

For a given wrap angle and screw pitch, the blow hole area and contact line length per lobe are totally decided by the end profiles of the male and female rotors. The purposes of optimizing the geometrical parameters are to obtain small leakage areas and a better performance. The following two considerations should be kept in mind:

- The profiles should be streamlined.
- The cutter blades of the rotors should have ideal shapes.

In this paper some of the problems of optimization of geometrical parameters are discussed.

BLOW HOLE CONSTANT AND CONTACT LINE CONSTANT

It is obvious that the end profile shapes of rotors of a helical screw compressor have a great effect on the compressor's performance and reliability. A good profile will result in the compressor having small fluid- dynamic loses, good performance, high reliability and make the machining cost of its rotors low. A profile influences the performance of a helical screw compressor mainly through two important geometrical characteristics, that is, blow hole area and contact line length per lobe, which provide the two most important leakage paths from an enclosed cavity volume to the following cavity volume or to the suction chamber. Increased leakage results in decreases in both the volumetric and total efficiencies.

So that large and small compressors and compressors having different profiles may be compared on a rational basis, two relative values, relative blow hole area and relative contact line length per lobe, are introduced.

The blow hole is a small triangular-shaped area formed by the housing cusp and the male and female rotor tips. The blow hole area is computed by locating the two points where the male and female rotors intersect the housing cusp, and the third point which is one of the nearby male-female contact points. These three points form a plane, which intersects the male and female rotor tips to obtain the two curved sides of a curvilinear triangle. The area of the curvilinear triangle is calculated by a numerical procedure. In the geometrical calculation program a numerical scheme automatically scans all the contact points in the cusp's vicinity, calculates the two curved sides and the areas, and then identifies the minimum area. This minimum area should be projected to a plane normal to the female rotor tip helix, which has the same tangent as the gas leakage streamline, to yield the final blow hole area. The relative blow hole area is defined as the blow hole area, which is projected to the longitudinal section along the direction of female rotor tip helix, divided by the cross-sectional areas between two lobes of the male rotor and of the female rotor, which are projected to the longitudinal sections along the direction of helix on the pitch circle:

$$A_r = \frac{A_b/cos(\beta_{ftip})}{(A_{01} + A_{02})/tg(\beta_{pitch})} \tag{1}$$

where,

A_r Relative blow hole area.

A_b Blow hole area, $[m^2]$.

A_{01} Cross-sectional area between two lobes of the male rotor, $[m^2]$.

A_{02} Cross-sectional area between two lobes of the female rotor, $[m^2]$.

β_{ftip} Screw angle at the tip of female rotor, $[rad]$.

β_{pitch} Screw angle on the pitch circle, $[rad]$.

The relative contact line length per lobe is defined as the contact line length per lobe, which is projected to the cross section, that is, the length of contact line per lobe in $X - Y$ plane, divided by the distance between the centers of the male and female rotors:

$$L_r = \frac{L_c}{D_c} \tag{2}$$

where,

L_r Relative contact line length per lobe.

L_c Length of contact line per lobe in the $X-Y$ plane, $[m]$.

D_c Distance between the centers of the male and female rotors, $[m]$.

These relative values are dimensionless constants which depend only on the profile or the parameters that define the profile and are independent of compressor size, wrap angles and lengths of rotors, so they can be called blow hole constant and contact line constant respectively. Their values for the profile shown in Figure 1 are 0.00731301 and 1.290618 respectively. Since for a given profile the blow hole and contact line constants are determined uniquely, they can be used as a basis for comparing different profiles. The optimization of a profile consists of determining the basic parameters which define the profile, so as to produce the smallest blow hole and contact line constants consistant with a streamlined shape and good practical cutter blade shapes.

A METHOD OF REDUCING THE BLOW HOLE AREA

A few basic parameters are used to defined the shape of a profile. If these basic parameters are changed, the shape of profile changes, as do the blow hole and contact line constants.

The blow hole is a small roughly triangular-shaped area formed by the housing cusp and male and female rotor tips. A common method used to reduce its area, is to reduce its height. But a larger height of blow hole reduces the relative sliding velocity between the rotors and with it, the wear. It also contributes to increasing the maximum cavity volume and as a consequence, to the production of a smaller compressor. Another method of reducing the blow hole constant is to change the shapes of two curved sides of the curvilinear triangle. Tip profile parameters can be selected to make the curvilinear triangle narrower and sharper, so that the blow hole constant will be reduced considerably.

Figure 2 shows a comparison of two blow holes in the $Y-Z$ plane. These blow holes have the same height and the same bottom width. From the original profile choice to the improved profile, the blow hole constant is reduced by thirty two per cent, from 0.00753 to 0.00514 , and the contact line constant remains unchanged. Due to the reduction of blow hole constant the indicated efficiency and volumetric efficiency are increased by 0.933 per cent and 0.313 per cent respectively for the compressor, the center distance of which is $160mm$ and the operating conditions are: refrigerant, $R22$; rotation speed, $3000rpm$; evaporating temperature, -10°; condensing temperature,25°; suction superheat, 30°. The blow hole area has a much greater influence on the indicated efficiency than on the volumetric efficiency. The program predicting thermodynamic performance [2] is used to obtain these results.

It should be mentioned that in order to reduce the blow hole constant the other basic parameters can be optimized besides those which define the curved sides of the blow hole. For example, for the above original profile if the other basic parameters are optimized, the blow hole constant will be reduced to 0.00426 and the contact line constant will be reduced from 1.306 to 1.198, and the indicated and volumetric efficiencies will be increased by 1.16 per cent and 0.214 per cent respectively.

COMPRESSION START BLOW HOLE

The introduction of the unsymmetrical profile improved performance considerably since the blow holes were reduced greatly. On the other hand the unsymmetrical profile resulted in another large blow hole, which exits during the early stage of the compression process. Figure 3 shows such a blow hole, which may be called the compression start blow hole. In Figure 3 the volume A is part of the cavity volume being compressed, that is, the pressure in it is higher than suction pressure, but it still connects to the suction pressure through the compression start blow hole. Figure 4 shows the relationship between its area and the rotation angle of the male rotor, the wrap angle of which is 300°.

Although the compression start blow hole only exits on the early stage of the compression process where the pressure in the cavity volume is low, it has a noticeable influence on the performance of the compressor since its area is ten to twenty times as large as the area of the blow hole along the other housing cusp, and the larger the wrap angle, the larger the influence. Figure 5 shows the relationship between the leakage (percentage of theoretical capacity) through the compression start blow hole and the wrap angle of the male rotor, where the operating conditions are the same as those mentioned above. For the same operating conditions (with the exception of the evaporating temperature) Figure 6 shows the relationship between the leakage and the pressure ratio.

In order to design an efficient screw compressor attention should be paid to the compression start blow hole especially in the case where the compressor will be used in a refrigeration system with an economizer. The choices of both the superfeed port position and the wrap angle should be made with care, otherwise the leakage may be increased.

DETERMINATION OF POSITION OF DISCHARGE PORT

The position of the discharge port of a helical screw compressor is determined according to the built-in volume ratio, which is the ratio between the real maximum cavity volume and the discharge cavity volume at the instant at which it connects to the discharge chamber of the compressor. For a given compressor the built-in volume ratio is fixed by the basic geometry. Corresponding to the built-in volume ratio there exists a so-called built-in pressure ratio or internal pressure ratio, which is the ratio between the pressure in the discharge cavity volume and the pressure in the suction chamber. For a given compressor the internal pressure ratio is not determined by its operating conditions. Besides the internal pressure ratio for a given operating condition there exists an external pressure ratio or nominal pressure ratio, which is the ratio between the pressure in the discharge chamber and the pressure in the suction chamber. The internal and external pressure ratios may be equal or not, as determined by the operating conditions.

Theoretically, in order to get the highest indicated efficiency the internal pressure in the discharge cavity volume, represented by P_i, should be equal to the nominal discharge pressure in the discharge chamber, represented by P_d. That is it appears that to eliminate wasteful energy consumption the internal pressure ratio should be equal to the external pressure ratio, but for a real working process the highest indicated efficiency can not be obtained when P_i equals P_d. Figure 7 shows the relationships between the indicated and total efficiencies and the ratio of P_d to P_i. In Figure 7 the highest efficiencies do not accur at the position where P_d/P_i equals one, but where the ratio equals about 1.25. When P_d/P_i is less than one the efficiencies decreases rapidly along with the reduction of the ratio, but when P_d/P_i is larger than one high efficiencies can be obtained over a range of values of the ratio.

The four $P-V$ diagrams in Figure 8 explain why the above phenomenon happens. All four diagrams have the same nominal pressure ratios, but different internal pressure ratios. For the diagram B P_i is less than P_d. After the cavity volume connects to the discharge chamber the refrigerant vapour will flow back into the cavity volume, but since the discharge port area is small at the beginning the back-flow rate is small. At this stage the cavity volume is reducing quickly, so the pressure in it quickly becomes larger than the nominal discharge pressure, and the discharge process begins. When normal forward flow begins the discharge port area is large so that the discharge process has a small flow resistance. This is why the diagram B encloses the smallest area, that is, the smallest indicated work, in the four diagrams, even smaller than the diagram C for $P_i = P_d$, which begins its discharge process immediately the cavity volume connects to the discharge chamber and as a consequence meets a larger flow loss since both the discharge port area is very small in the beginning and the cavity volume continues to reduce rapidly, which results in a larger energy consumption than for $P_i < P_d$. If P_i is larger than P_d more energy consumption is needed (the digram D in Figure 8). The diagram A shows that if P_i is much less than P_d the energy consumption is also increased.

For a chosen operating condition the optimum position of the discharge port can be determined by the geometrical calculation program and the working process modelling program very easily so that the highest indicated or total efficiency is obtained. Since a refrigeration screw compressor will operate under various conditions it is difficult to give the optimum position of the discharge port. It is reasonable to design or choose a compressor according to the minimum nominal pressure ratio of the various conditions under which the compressor will operate.

CONCLUSIONS

1. The introduction of two dimensionless constants, the blow hole constant and contact line constant, which are determined uniquely by profiles and are independent of compressor size, wrap angles and lengths of rotors, makes it possible and reasonable to compare the blow hole areas and contact line lengths of different profiles. They can be used to optimize a profile.
2. A very efficient method, which is used to reduce the blow hole area, is to change the shapes of its two curve sides, and to make it as sharp and narrow as possible. This method is useful for the optimization of a profile or the generation of a new profile.
3. Since helical screw compressors have been very highly developed and are a "mature technology",to continue to improve their performance attention should be paid to the compression start blow hole area. When they are used in economizer refrigeration systems the selection of wrap angle should be made carefully.
4. For a real working process the highest indicated or total efficiency always is obtained when P_i is less than P_d. A dicharge port position should be chosen to make the internal pressure ratio less than the external.

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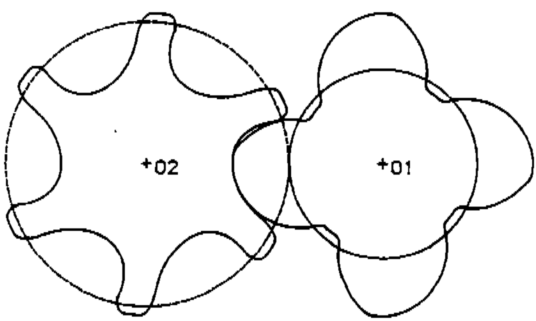

Figure 1 An example of profile

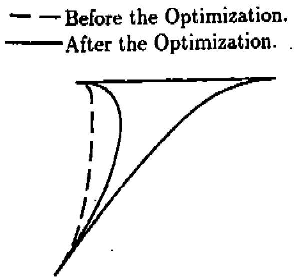

Figure 2 Blow hole area reduction

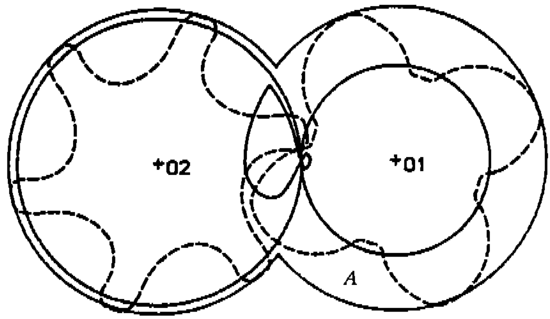

Figure 3 View of the compression start blow hole at the discharge end of compressor

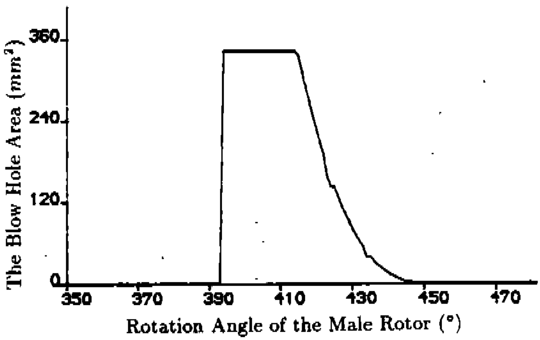

Figure 4 The compression start blow hole area vs the rotation angle of male rotor

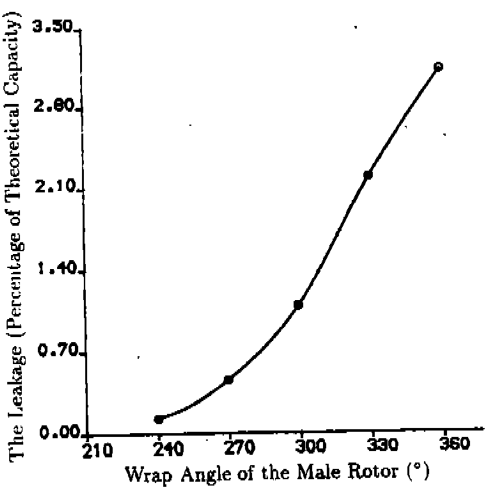

Figure 5 The leakage through the compression start blow hole vs the wrap angle of male rotor

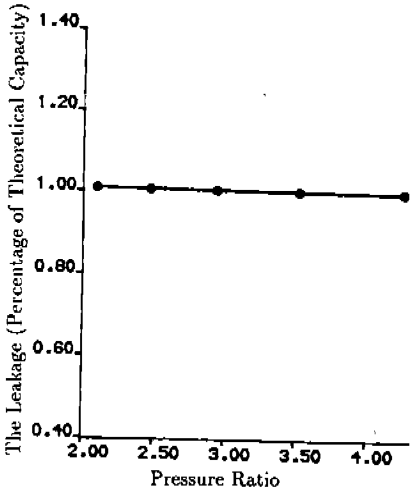

Figure 6 The leakage through the compression start blow hole vs the pressure ratio

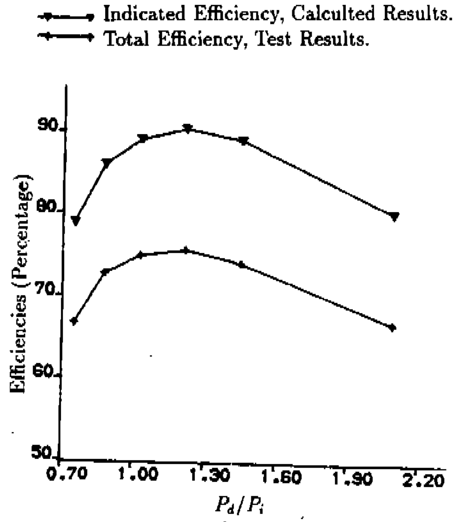

Figure 7 The total and indicated efficiencies vs P_d/P_t

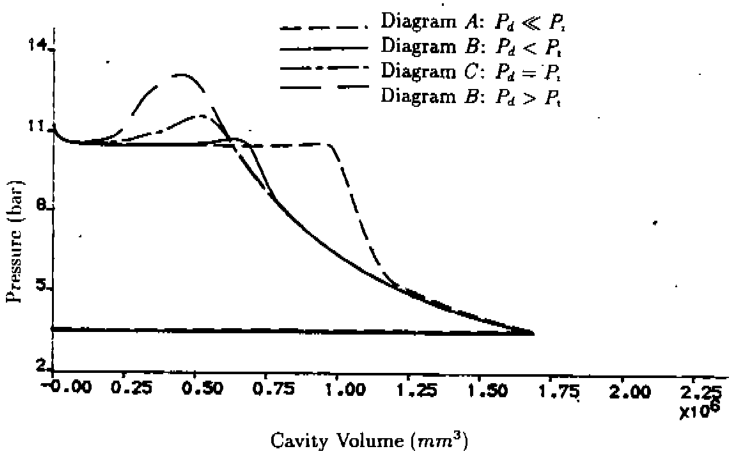

Figure 8 $P-V$ diagrams for different P_d/P_i P_d/P_i ratios