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ABSTRACT

This paper introduces the theoretical study and practical applications of a computer-aided design (CAD) system for twin-screw compressors. The system includes three parts: the calculation of the geometrical parameters of the rotors; the simulation of the compressor working processes; and the optimization of the design parameters of the compressor. The system has been used in the design of several air and refrigerant twin-screw compressors. And the results are very satisfactory.

INTRODUCTION

The demand for twin-screw compressors is increasing because of their advantages such as compact construction, high reliability and low vibration. And they have to be designed to meet various usages such as severer operating conditions, longer service life and shorter period of product development. However, the conventional design method can't meet these demands very well.

In resent years, we have acquired some significant achievements in computer simulation for screw compressor working processes, new rotor profile design, and development of some special usage machines, etc. The key technologies have been grasped to optimize the contruction parameters, caculate forces acting on the rotors accurately, and match the operating parameters rationally. A complete CAD system for twin-screw compressors is established.

THE CAD SYSTEM

Fig. 1 shows the main construction and flow chart of the system. It is easy to be understood and modified because of the softwares in the system are independent relatively. The programs are all presented in FORTRAN lauguage and run on microcomputers.

Calculation of Geometrical Parameters

In Fig. 1, the length of contact line, area utilization coefficient, blow hole area, and cavity volume are all geometrical parameters of the rotors. The calculation of these parameters is the first step to simulate the compressor working processes and predict rotor profile performance preliminarily.

Before calculating the geometrical parameters, the male and female rotor's conjugating profile must be defined. In the CAD system, the profile is generated as follows: First, give some first or second order curved lines such as straight line, circle or ellipse, to be partial curves constituting one of the rotor's profile. Then, the curves conjugating with these curves on the other rotor can be obtained according to the mesh theory. After the basic constitution of the rotor profiles are decided, the profiles of different lobe combination and different profile parameters such as addendum radius, can be obtained easily with the CAD system .

The contact line seperates the cavity undergoing compression or discharge process and suction process. Therefore, its length has great effect on compressor performance, and calculating the contact line length accurately is very important for evaluating or optomizing new rotor profiles. Moreover, when calculating forces on the rotor, the contact line projection on different coordinate planes are also required to deal with contact region and no-contact region differently. In the CAD system, the equations of the two rotors helical surfaces is derived from their end profiles firstly. Then, the contact line equation is obtained by using the equations and the engagement conditions simultaneously. Solving this contact line equation, the coordinates of each point constituting the contact line can be obtained. Suming up the distances between the points, the contact line length can be gained. Projecting the points on appointed coordinate plane, the contact line projection on different coordinate plane can be obtained. Fig. 2 shows a typical results. It gives the contact line length in a function of male rotor rotation angle.

The blow hole area has great effect on compressor performance. Getting a deeper understanding of the blow hole is the very basis for evolving symetrical profile into a asymetrical one. Because the blow hole is a space curved triangle, the calculation of its area is very complicated. In the CAD system, a simple method is adopted to simplify the calculation. The blow hole is considered on a plane formed by the intersection line between the two bores of the cylinder housing and the highest point of the contact line. Therefore, it is considered as a plane curved triangle. Its three sides are intersection line of the cylinder bores, two intersection lines of the plane with the female and the male rotors' helical surfaces. The concrete steps to obtain the blow hole area are as follows: First, gain the equation of the plane mentioned above. Then, solve this equation and the two rotor helical surface equations simultaneously to gain the two curved side equations of the blow hole. Finally, use all these equations to do the plane integration, and the blow hole area can be obtained.

Area utilization coefficient indicates the effective utilization extent of compressor dimension. When the end profiles of the male and female rotor are obtained, area utilization coefficient can be gained by means of digital integral easily.

Discharge port area can affect compressore performance greatly, especially under high rotation speed and large pressure ratio. The CAD system regards the sum of the axial and radius discharge port area as a comparision criterion. Unfold graph of the radius port consists of two triangle, so its area can be obtained easily. And the axial port area can be gotten by means of plane digital integral method. The calculating results indicate that the effect of different profile on discharge port area is not great, but the lobe combination and the wrap angle affect this area more greatly.

Performance Simulation

On the basis of calculating the geometrical parameters mentioned above. After analyzing the processes taking place in the compressor cavity in detail, the mathematical model decribing the processes can be established.

To suction process, the velocity of gas flowing through the suction port is small because the suction port area is relatively large. So the flow resistance loss can be neglected in this process, and the process therefore is regarded as a constant pressure process with the pressure value as same as the suction pressure. However, the gas temperature in this process is higher than the suction temperature because some high temperature gas or oil-gas mixture leakaging from high pressure chamber to the cavity. A temperature compensation coefficient is adopted to define the gas temperature in the end of the suction process.

To compression process, after considering all leakage from each leakage channel, the basic equations, such as energy and mass conservation equations, are established for the cavity. Then, these equations are solved simultaneously by means of Runge-Kutta method, and the gas state parameters such as pressure, temperature and specific volume can be obtained in function of the rotor's rotation angle.

To discharge process the velocity of gas flow through the discharge port is large because the discharge port area is relatively small, especially during the earlier stage of this process. So the resistance loss is higher in discharge process. In the CAD system, at the earlier stage of this process, the flow is regarded as frictionless nozzle flow. And when the discharge port area is increased to a appointed value, it is considered as one-dimension steady flow to calculate its pressure loss. It must be pointed that mass leakage between the two adjacent cavities is also considered in this discharge process.

Fig. 3 shows the computer program chart written according to the mathematical model mentioned above. The input data include compressor construction parameters such as the rotor end profile parameters, operating conditions such as rotation speed and discharge pressure, and the gas properties such as adiabatic index. The output data include delivery volume, shaft power, volumatic and adiabatic efficiency, etc.

Fig. 4 shows the relation between pressure and volume. The solid line indicates the typical caculating results for processes actually taking place in the compressor. While the dash line gives out the theoritical results without considering leakage and discharge resistance. From Fig. 4 the following conclusion can be drawn: At the earlier stage of compression, the gas pressure of the cavities higher than that without considering leakage because, in this period, the gas leakaging from the higher pressure chamber to the cavity is larger than that from the cavity to thhe lower pressure chamber. Conversely, at the later stage of the compressiom, the the real process line approaches to the theoretical line gradually, and even has the trend of lower than that.

Analyzing the calculation results about compression and disccharge processes in Fig. 4, it is proven that the mathematical model established can describe the thermodynamical working processes correctly.

In order to calibrate some emperical coefficients in the model, examine the assumptions made in establishing the model, and verify the computer program, the performance data calculated by the program is compared with the data measured in the laboratory. The results are given out in Fig. 5. After the emperical coefficients are calibrated by the data measured under one operating condition, they can be used to calculate the compressor performance under other operating conditions. The results show that the error of the delivery volume calculated by the program is within 3 %, and the error of the shaft power is less than 5 %. So it is proven that the mathematical model and the computer program mentioned above are reliable and correct.

Optimization of the Design Parameters

With the correct mathematical model and computer program for the screw compressor, the effect of some major construction and operating parameters on the compressor performance can be analyzed on the computer, and the optimum combination of these parameters can also be selected to design a compressor with better performance or longer service life. This paper takes the discharge port position as a example to describe the function of the CAD system in optimizing the design parameters.

Different from the reciprocating piston compressor, the twin- screw compressor has no suction and discharge valve. The suction and discharge process are realized through suction and discharge port fited to the machine body. So it has a built-in volume and pressure ratio because of the intrinsic positions of the suction and discharge port. The suction port is always designed in the position where the cavity volume reaches the maximum value. So the compressor's built-in and pressure ratio is determined only by the discharge port position.

In view of the characteristics of twin-screw compressor mentioned above, its built-in pressure ratio can't always match well with the external pressure ratio determined by the suction and discharge pressure. So it is very important to determine the discharge port position carefully when designing a twin-screw compressor.

Theoretically, the built-in pressure ratio should equal to the external pressure ratio. Otherwise the compressor performance will be worse becuse the over-compression or under-compression can cause additional energy losses and increase power consumption. Pratically, at the earlier stage of the discharge process, the discharge port area is very small. The gas in the cavity can't flow into the discharge chamber timely. So the pressure of the gas in the cavity will increase continously and cause much more energy losses as shown in Fig.6 by line 1.

In order to improve the screw compressor performance, it is often adopted to design the discharge port with a early opening angle. That is let the discharge process begin before the cavity pressure reaches the discharge pressure as shown in Fig.6 by line 2. From Fig.6 we can also see that opening the discharge port early may cause high pressure gas in discharge chamber flow back into the cavity and cause some additional power consumption, but it can reduce over-compression loss greatly, and also result in a large discharge port which is helpful to reduce the flow resistance in the discharge process.

Of course, when the early opening angle exceeds a definite value, the compressor performance will be worse because the backflow loss is too much. Obviously, to make the compressor have the best performance there is an optimum early opening angle for discharge port. Fig. 7 shows the relation between compressor adiabatic efficiency and the early opening angle under different male rotor tip speed. From Fig.7 it can be seen that an optimum early opening angle does exist. The adiabatic efficiency of the compressor will reach maximum value with this optimum angle. However, the optimum angle is not distinct with different rotor tip speed. When the tip speed is higher, a bigger early opening angle is necessary to get larger discharge port area in order to reduce the flow resistence.

To other construction parameters and operating parameters such as the ratio of rotor length to its diameter, wrap angle and rotor tip speed, their optimum value all exist for a twin-screw compressor like the discharge port angle. With the CAD system mentioned above, the optimums can be obtained by means of the method used to analyze the early opening angle of the discharge port.

Moreover, besides the major computer programs mentioned above, some other contents related to screw compressor design are also included in the CAD system. The calculation of the forces acting on the rotors and tool contour for cutting the rotors are introduced briefly here.

In view of twin-screw compressor design, especially miniature and small size twin-screw compressor design, selecting or designing of the bearings is one of the key factors to determine the machine whether successful or not. While calculating the forces acting on the rotors is the basis for selecting or designing the bearings. Besides calculation of the forces is also necessary for new profile design. That is because , to asymetrical profile, the torque acting on the femmale rotor produced by the gas axial force is a driving torque. And with the rotor rotation angle changing, this torque changes its magnitage periodically. In new profile design , it is necessary to let the minimum value of this torque still be larger than the torque acting on the female rotor produced by the friction force. Otherwise the direction of the sum torque acting on the female rotor will change periodically when the male rotor is drived by the female. That can result in the two rotors strike each other, and demage the oil film between the rotors. So the compressor performance will descend because the leakage is more. In the CAD system , the axial force is dealed with end axial force and gas axial force (only exists in the contact region) differently. To the radial force, firstly seek the forces on the contact region and non-contact region distinctly, then sum them to obtain the overall radial force.

In the CAD system, there are also some programs to calculate the contours of the discal cutter for machining the rotors. It deserves to be pointed that the CAD system can not only calculate the contours of the cutter with zero degree front angle, but also can calculate that with non-zero degree front angle or with a helical oblique angle. So the machining performance of the cutter can be improved.

APPLICATIONS OF THE CAD SYSTEM

Compressor Design

The CAD system can be used to do some conventional calculations quickly and accurately, such as the contours of the cutter for machining the rotors, the bearing load and the discharge port positions etc. Moreover, it can be used to do ' numerical test ' on computer to analyze the effect of each design parameter on compressor performance, and then obtain the optimium values of these parameters.

This CAD system has ever been used to determine the construction and operating parameters for several air and refrigerant twin-screw compressors. Tab. 1 shows the major parameters and performance of a small sized single stage air cooling screw compressor. The development period of this machine is very short(about one year) because the CAD system was used efficiently.

Development of New Rotor Profile

When a new profile is designed, the parameters can be determined initially by using the programs calculating the rotors' geometrical characterics in the CAD system. Based on this, the profile parameters such as the lobe combination and the addendum radius, can be optimized further with the compressor performance simulation program. The related contents and achievements have been reported in reference [4].

Morever, it is deserved to be pointed out that the CAD system can not only be used to design twin-screw compressors, but also to design twin-screw expanders or vaccum pumps. With being revised simply, the system can be used to design other meshed-rotor machines such as Roots blower.

CONCLUSION

With many years effort, a specialized CAD system for twin-screw compressors has been developed. This system can be used to do the design calculations of the screw compressors. And it can make the construction and operating parameters match well and let the compressor performance reaches optimum value. Some types of twin-screw compressor designed with this CAD system have ever been put into mass production. Practical applications indicates the performances of these compressors are very satisfactory.

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Tab. 1a Parameters of a single stage screw compressor

rotor diameter mm		wrap angle		lobe number		rotor length mm	discharge early opening angle
male	female	male	female	male	female		
100	96	325°	217°	4	6	130	7°

Tab. 1b Performance of a single stage compressor

male rotor speed rpm	suction pressure MPa	discharge pressure MPa	delivery volume m^3/min	shaft power kw	compressor unit wight kg	noise dB (A)	discharge temperature
6000	0.1	1.1	3.02	27	170	81	83.5°

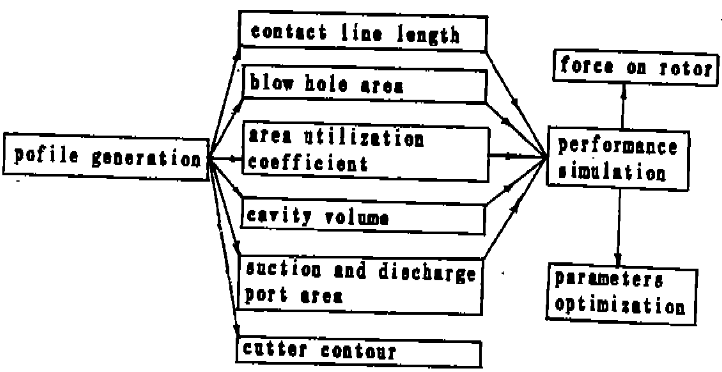

Fig. 1 Construction of the CAD system

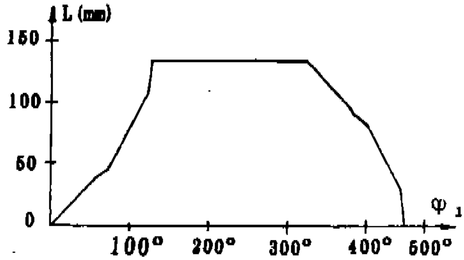

Fig. 2 Contact line length versus male rotor rotation angle

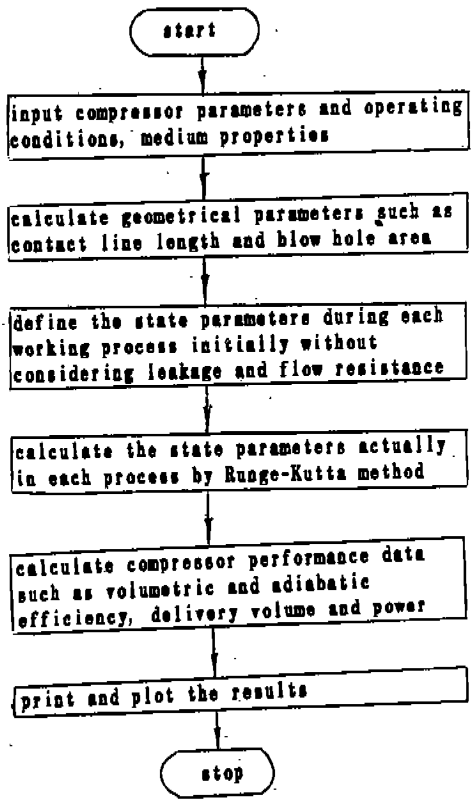

Fig. 3 Chart of the performance simulation program

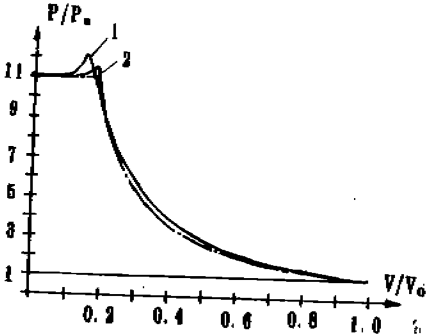

Fig. 6 Performance affected by discharge port position

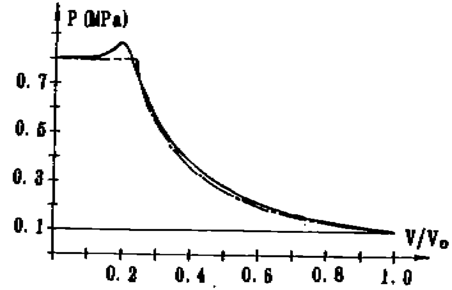

Fig. 4 Relation between pressure and volume

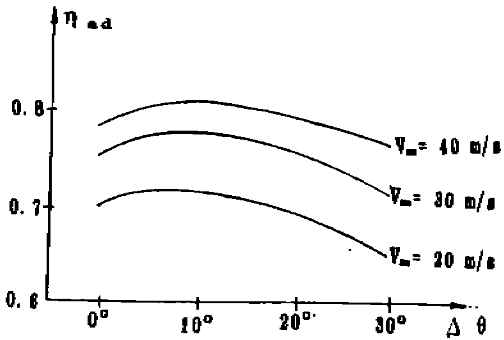

Fig. 7 Adiabatic efficiency versus early opening angle under different male rotor tip speed

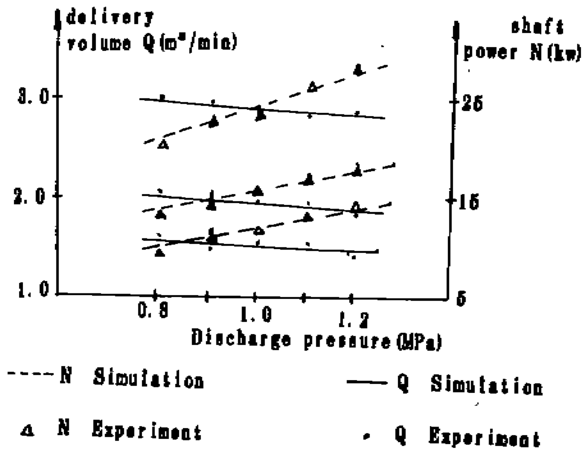

Fig. 5 Comparsion between simulation and experiment