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THE THEORETICAL ANALYSIS AND EXPERIMENTAL RESEARCH FOR A REFRIGERATING SCROLL COMPRESSOR

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ABSTRACT

This paper is a sum-up in experience by which authors developed a first prototype scroll compressor for air condition in China. Taking one pair of sealed fluid pockets which are composed of the orbiting and fixed scroll's as control volumes, the working process's thermodynamics simulation is made. It was founded on the first law of thermodynamics and the law of mass flow conservation, and non-ideal gas behavior of refrigerant. The calculated results are not only in agreement with the measured results obtained with the prototype scroll compressor, but also indicate the advanced quantity of this scroll compressor. A new point of view is that the discharge starting angle can be machined in the completed scroll walls in this paper. The acting force and its balance is discussed. Make use of a new structure of sealing circle, the force balance is realized easily.

INTRODUCTION

Comparing the other types of compressors, the inherent advantages of the scroll compressor are: the fixed built-in volume ratio making it valveless and more reliable, adaptable to variable speed, the large inhaling hole and nearly continuous inhaling making it have the less pressure drop, high volumetric efficiency, and the lower noise.

However, designing and manufacturing this machine is very difficult. In this paper, the basic relation describing the compression process is derived and it is founded on the first law of thermodynamics, at the same time the differences between scroll and other compressors were indicated. Because of the feature of the fixed volume ratio, the calculated pressure ratio should be agreement with the ratios under the experimental conditions so that useless power is low which influences the choice of structure parameters. A new point of view about the starting discharge angle is put forward. The axial force and its balance is discussed in this paper. Finally, the calculated and experimental results are plotted and show good agreement. The dominant influence of axial clearance to the volumetric efficiency is pointed out.
THE BASIC RELATION DESCRIBING THE WORKING PROCESS OF SCROLL COMPRESSOR

Working chambers of a scroll compressor are shown in Fig. 1. These chambers are three pairs of sealed fluid pockets which are composed of the orbiting and fixed scroll wraps when the involute angle \( \theta \) of scroll wrap is 1170°. If roll angle of shaft A is zero, the outermost pair of sealed pockets, symbol (I), represents the suction volume of the displacement volume of compressor. The intermediate sealed pockets, symbol (II), is being compressed. The (III) is discharge pressure chamber. When the \( \theta \) increases up to the starting discharge angle \( \theta^* \), (II) and (III) become one chamber - discharge pressure chamber. It should be taken into account that the sealed fluid pockets are only two then. When \( \theta \) continues to get large and is equal to 360°, crank shaft rolls one circle and fixed and orbiting scrolls back to the initial positions. Above is the case of every pair of pocket's volume changing as shaft rolls during 360°. If controlled volume varies from suction to discharge, the shaft's orbiting angle will be \( (360°+ \theta^*) \).

According to the above analysis, the modeling range of compression process will be:

\[
0^\circ \leq \theta \leq 360^\circ \\
\theta^* < \theta < 360^\circ
\]

Taking a pair of sealed fluid pockets as control volumes, and neglecting the gravitational and kinematic energies of working fluid, the following equation can be written:

\[
\frac{dp}{d\theta} = \left\{ \begin{align*}
\frac{\Delta h}{\Delta v} + \frac{\Delta h}{\Delta t} \frac{dp}{dv} + \frac{\Delta h}{\Delta t} \frac{dp}{dt} + \frac{1}{\Delta v} \frac{dm}{dv} \frac{1}{\Delta t} \frac{dm}{dt} \\
(\frac{\Delta h}{\Delta v} - \frac{\Delta v}{\Delta t} \frac{dp}{dv} - \frac{\Delta v}{\Delta t} \frac{dp}{dt}) \left( \frac{\Delta h}{\Delta v} - \frac{\Delta v}{\Delta t} \frac{dp}{dv} - \frac{\Delta v}{\Delta t} \frac{dp}{dt} \right) \right. \\
\left. \frac{\Delta h}{\Delta t} - \frac{\Delta v}{\Delta t} \frac{dp}{dv} - \frac{\Delta v}{\Delta t} \frac{dp}{dt} \right)
\right\} \\
0^\circ < \theta < 360^\circ
\]

The equation (2) was founded on the first law of thermodynamics and the law of mass conservation, and non-seal gas behavior of the refrigerant: the relation includes not only energy, mass, and heat transfer in compression process, but also the mass flow rate of the leakage from the control volumes.

Directly perceived through the senses of the relation (2), the difference between scroll and other displacement compressors can not be found, but their implications are not consistent. Scroll compressor has the continuous compression chamber which differences with others'. To solve the equation (2) applying only R-K method with initial value is efficient. It must be solved with another limited condition, this is:

\[
\begin{align*}
M_1(360^\circ) - M_2(0^\circ) & < E_1 \\
T_1(360^\circ) - T_2(0^\circ) & < E_2
\end{align*}
\]
As long as the condition is met, the end state of gas in the first compression chamber is just the initial state of gas in the second compression chamber.

The other models for determining the compression process are made:

1. The leak flow from axial clearance was assumed to be the leakage of oil with refrigerant when the clearance is small.

2. The leak flow from radial (flank) clearance or from the axial clearance when the clearance is large are considered as compressible fluid flowing with friction through a convergent nozzle.

3. Takes freon-22 as refrigerant.

**MAKING THE STARTING DISCHARGE ROLL ANGLE IN THE SHAPED SCROLL WALLS**

The starting discharge roll angle is the roll angle at the contact point of scroll walls when the discharge process commences. It is determined by the interaction between the involute spiral and cutter, and under the influence of the starting involute angle of scroll wrap. But varying the starting involute angle $\alpha$ is varying the structure parameters, because:

$$t = 2\alpha \alpha$$  \hspace{1cm} (4)

Otherwise, the built-in volume ratio is decided by following relation:

$$\rho = \frac{2N-1}{3-\delta^2/A}$$  \hspace{1cm} (5)

Generally speaking, the sealed fluid pocket number $N$ is an integer chosen in a small range. To meet the need of the built-in volume ratio, the $\alpha$ is considered to be a known number. In terms of relation (3), the two important structure parameters (wrap thickness $t$ and basic circle's radius $a$) exist a direct relation, which is not our willingness, especially in the optimizing design.

The method of resolving this question is: when starting involute angle of scroll $\alpha$ is large and structure parameters is ideal, or the scroll wrap is shaped with the cutter whose diameter is less than (p-t), the starting discharge roll angle may be machined with the cutter whose diameter is less than (p-t) after the cutter center position is set again, the results are: 0 make starting involute angle of scroll expanded; the starting involute angle will be not limited by the compression ratio.

**AXIAL FORCE AND ITS BALANCE**

The method taken to prevent the orbiting scroll from being separated from the fixed scroll under the action of
axial force are: utilizing the intermediate chamber or the unity of it and discharge pressure district; applying thrust bearing between the frame and orbiting scroll.

In the prototype of scroll compressor developed by authors, the compressed gas enters the shell first, circulates the motor and inhales heat from it, finally goes out from the small hole in the shell. Fig. 2 shows the structure of axial force balance. The gas in intermediate chamber in which there is a oldham coupling goes in and out the intermediate compression chamber through a small aperture located in the orbiting scroll. A sealing circle is installed between intermediate pressure chamber and high pressure district.

DISCUSSION OF RESULTS

the curves concerning the volumetric efficiency are expressed in Fig. 3. The curves show axial clearance's influence to volumetric efficiency is larger than the radial clearance. The larger the clearance, the faster the volumetric efficiency falls. The reasons for this are as follows: on the one hand, the leakage length of axial clearance is longer than that of radial clearance; on the other hand, as the clearance increases the mixture ratio of oil and refrigerant varies and the mass ratio of refrigerant gets gradually large. The calculated and experimental results are shown in Fig. 4, the plot show their coincidence.

CONCLUSIONS

1. The relation describing the working process of scroll compressor is agreement with that of other displacement compressors in shape, but the conditions solving the relation are different.

2. The leakage through the axial clearance is dominant and influences the volumetric efficiency to a greater degree.

3. The new point of view that starting discharge roll angle is machined in the shaped scroll wrap makes the starting involute angle of scroll variable, and is useful to the optimizing of structure parameters.

4. The effective sealing circle between high and intermediate pressure districts is important in the calculation of axial force and its balance.

NOMENCLATURE

a --- Radius of basic circle of the involute spiral
EER --- Energy efficiency ratio
E1 ---- Iterative precision for mass
E2 ---- Iterative precision for temperature
h --- Enthalpy of gas in the sealed pocket
h1 --- Enthalpy into the sealed pocket
m1 --- Mass into the sealed pocket
m2 --- Mass of the second compression chamber
N --- Number of sealed fluidic pockets
p --- Pressure or wrap pitch
q --- Heat transferred to compression chamber
r --- Wrap thickness
T --- Temperature
T1 --- Temperature of the first compression chamber
T2 --- Temperature of the second compression chamber
T0 --- Evaporation temperature of refrigerant
Tk --- Condensation temperature of refrigerant
V --- Volume of compression chamber
Vc --- Displacement volume of compressor
\( \rho \) --- Build-in volume ratio
\( \pi \) --- 3.1415926...
\( \alpha \) --- Starting involute angle of scroll
\( \psi \) --- Roll angle of shaft
\( \gamma \) --- Roll angle at the discharge starting position

REFERENCES

Fig. 1 Compression Chambers of Scroll Compressor

Fig. 2 The Basic Structure of the Prototype Scroll Compressor
Fig. 3 The Influence of Axial and Radial Clearance on Volumetric Efficiency
Fig. 4 Pressure Ratio Effects on Compressor Performance