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ABSTRACT

Based on thermodynamic and mechanical models, the axial and tangential forces acting on the orbiting scroll and their balance methods are discussed in this paper. Besides, a detailed analysis of orbiting mechanism of three pins is carried out too.

INTRODUCTION

The advantageous characteristics of scroll compressor have been proved in theory and experiment [1] [3] in recent years. Remarking the main influences of the axial and tangential forces acting on orbiting scroll on the performance and reliability, this paper puts forward a new balancing method and compares it with the balance means used. The mathematical model and computer program of scroll compressor working process [2] are applied to the calculation of this paper. A prototype of scroll for air condition has been developed and tested by authors. The calculated results and experimental results are consistent

THE AXIAL AND TANGENTIAL GAS FORCES ACTING ON ORBITING SCROLL

With crank shaft rotation, the center of orbiting scroll is pushed around the center of fixed scroll or crank shaft. As the gas in the sealed volumetric chambers, which are composed of the orbiting and fixed scroll, is compressed, the gas pressure increases and the orbiting scroll is forced. With the thermodyamic mathmatic model [2] and the mechanical model, the gas force exerting model is shown in Fig.1. In Fig.1 a), because the pushing force $F_{N\tau}$ of crank shaft and the tangential gas force $F_{g\tau}$ are not acted on the same point, thus makes the orbiting scroll rotate in itsself, the normal compression chambers are interfered, self-rotation moment is given by

$$M_{r\tau} = F_{g\tau} * r/2 \qquad (1)$$

In terms of Fig.1 b), the axial gas force F_{gt} acting on the orbiting scroll makes the orbiting scroll departure the fixed scroll, increasing the axial clearance C_t, and resulting in apparent decrease of volumetric efficiency.

BALANCING MECHANISM OF AXIAL GAS FORCE

The axial gas force acting on orbiting scroll should be balanced so that the normal gas compression can be completed.Balancing gas force can be accomplished by intermediate pressure chamber or with discharge gas pressure.

Intermediate Pressure Chamber

The intermediate pressure chamber used for balancing the axial gas force acting on the orbiting scroll is shown in Fig.2.It is established on the back of orbiting scroll and connected to the intermediate sealed compression chamber through a small apeture on the orbiting scroll plate. In this case, the gas pressures in the intermediate compression chamber and the intermediate balancing chamber are equal and change with the rotation angle θ of crank shaft.

The balancing equation of axial gas force is represented by:

$$F_{rt} = P(\theta) * A1$$

$$F_{rt} - F_{gt} > 0. \qquad (2)$$

or

$$F_{rt} = 1.05 \sim 1.1\ F_{gt}$$

When orbiting scroll moves, the gas in the intermediate chamber leaks to the suction pressure side and the seal of this kind of leakage is more difficult.

The Gas In Discharge Pressure

The axial gas force balance is also completed by using the balancing mechanism of gas discharge pressure, which is consisted of a sealing ring and put in the frame, as shown in Fig.3 a), Fig.3b) shows the relation of forces. The balance relation of force is :

$$F_{rt} = F_s + \pi\ d * t * p_d$$

$$F_{rt} - F_{gt} > 0. \qquad (3)$$

Owing to the oil in discharge pressure being exerted directly on the sealing ring, the gas leakage through the sealing ring will be very little. It should be pointed out that the liquid collision and overload of scroll compressor need a special adjusting mechanism of orbiting radius.

BALANCING MECHANISM OF SELF-ROTATION MOVEMENT

The three pins mechanism used to prevent the orbiting scroll from self-rotation is showen in Fig.4. The one end of pins is put to the orbiting scroll or frame, the other are put into the holes in the frame or the orbiting scroll. The balancing equtions of forces acting on the orbiting scroll are represented as follows:

In X axis,

$$I_x + F_r * \cos(\theta) + (f_1+f_2+f_3) * \sin(\theta) - (F_1+F_2+F_3) * \cos(\theta) = 0. \quad (4)$$

In Y axis,

$$I_y + F_r * \sin(\theta) + (f_1+f_2+f_3) * \cos(\theta) - (F_1+F_2+F_3) * \sin(\theta) = 0. \quad (5)$$

The movement about point O is

$$-F_{\varepsilon}\tau * r/2 - F_1 * R * \sin(120^{\circ}-\theta) - f_1 * [R * \cos(120^{\circ}-\theta - D/2) + F_2 * R\sin(\theta)$$

$$+ f_2 * (R * \cos(\theta) - D/2) + F_3 * R * \sin(60^{\circ}-\theta) - f_3 * [R * \cos(60^{\circ}-\theta) + D/2] = 0. \quad (6)$$

EXPERIMENT AND RESULLTS ANALYSIS

The thermodyamic model and computer program of scroll compressor working process [2] are directly applied to the calculation of computer.

A prototype of scroll compressor for air condition has been developed by authors, which is consisted of the anti-self-rotation mechanism using three pins and the axial gas force balancing mechanism using oil in discharge pressure in a sealing ring. The refrigeration quantity of compressor is 10,000 Kcal/h.

The results of theoretical calculation and experimental test are shown in Fig.6 and Fig.7. Fig.6 represents the relation between EER and A_t, Fig.7 represents the relation between EER and d_1.

Under the fixed testing conditions that include the compressor working condition and the surrounding condition, because F_{rt} varies with the change of sealing ring diameter, i.e.the acting area of oil against the orbiting scroll, the input power or the volumetric efficiency of compressor will change at the same time. According to this, at a certain A_t*, there will be a maximum EER,because when A_t is greater than A_t*, input power of compressor increases; while the A_t is less than A_t*, the axial clearance enlarges,and the volumetric efficiency decreases. The above two cases all makes the EER of scroll copressor decrease.

In Fig.7, there is also a maximum EER which is acquired at a certain d_1*,the cause is that when d_1 is greater than d_1*,the friction between tangential walls of orbiting and fixed scroll increase, the input power of compressor increases and the EER decreases; when d_1 is less than d_1*, the orbiting scroll has a little self-rotation, the tangential gas leakage of scroll compressor enhances,and the volumetric efficiency and the EER decreases.

CONCLUSIONS

1. Axial and tangential gas forces acting on the orbiting scroll have been analysed. These two forces will influence efficiency and reliability of the compressor, it must be balanced perfectly.

2. Using discharge pressure in a sealing ring is a easy way to balance the axial force and a optimizing diameter of the ring is found by calculation and experiment.

3. The self-rotation of orbiting scroll may be prevented by the pin mechanism and the analysis of the acting forces on the pins has been carried out.

SYMBOLS

A_1---theacting area of intermediate pressure chamber on the orbiting scroll;

A_t---the acting area of oil in the sealing ring;

C_t---axial clearance;

d ---the intermediate diameter of sealing;

d_1---pin diameter;

D ---pin hole diameter ($D=d1 + 2 * r$);

EER---energy efficiency ratio;

f_1, f_2, f_3---friction force between orbiting scroll and pins;

F_1, F_2, F_3---the acting forces between pins and orbiting scroll;

F_{gt}---axial gas force acting on orbiting scroll;

$F_{g\tau}$ ---tangential gas force acting on orbiting scroll;

FN_τ ---pushed force of crank shaft;

F_r---radial gas force acting on orbiting scroll;

F_{rt}---pushed force of axial direction;

F_s---spring force;

I_x, I_y---inertial force of orbiting scroll;

M_s---mass of orbiting scroll;

$M_{r\tau}$---self-rotation moment of orbiting scroll;

O---center of orbiting scroll;

O_1---center of fixed scroll or crank shaft;

p_s---suction pressure;

p_d---discharge pressure;

$P(\theta)$---gas pressure in intermediate pressure chamber;

r----orbiting radius;

R----distance from the center of orbiting to the center of pin hole;

t----width of sealing ring;

θ---orbiting angle;

ω---angular velocity of shaft;

ω_1---angular velocity of orbiting scroll self-rotation;

μ_0---coefficient of sliding friction.

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[3]. James W.Bush,"SCROLL COMPRESSOR DESIGN CRITERIA FOR RESIDENTIAL AIR CONDITIONING AND HEAT PUMP APPLICATIONS",Proc. of 1988(PURDUE)

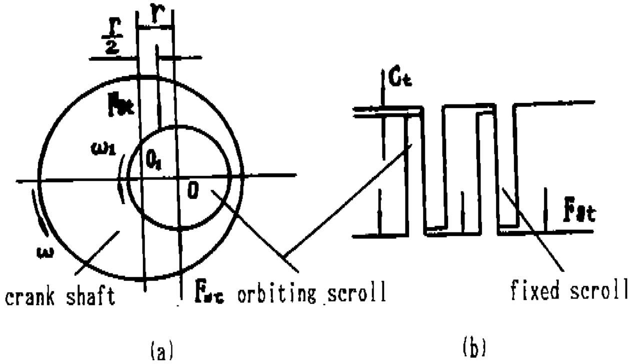

Fig.1 Gas forces acting on orbiting scroll

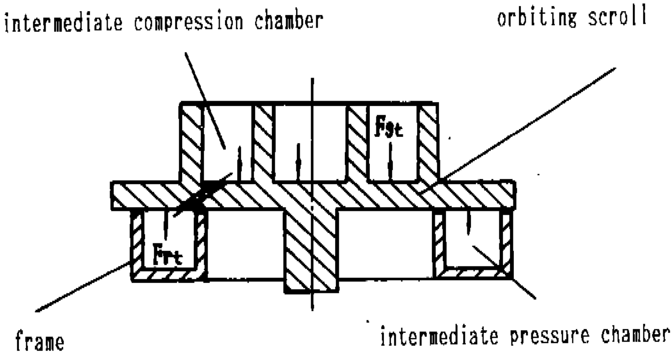

Fig.2 Balancing mechanism of intermediate pressure chamber

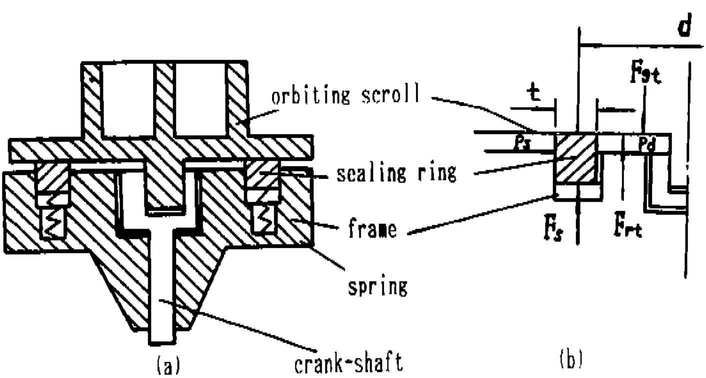

Fig.3 Balancing mechanism of axial gas force .

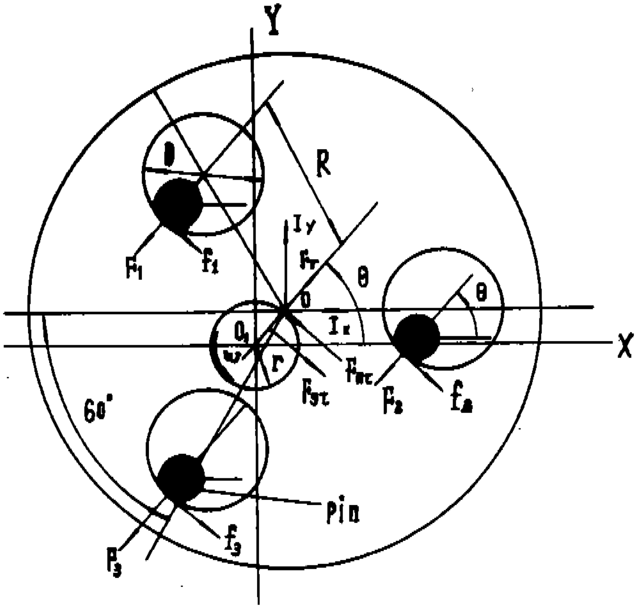

Fig.4 Mechanism of preventing orbiting scroll from self-rotation

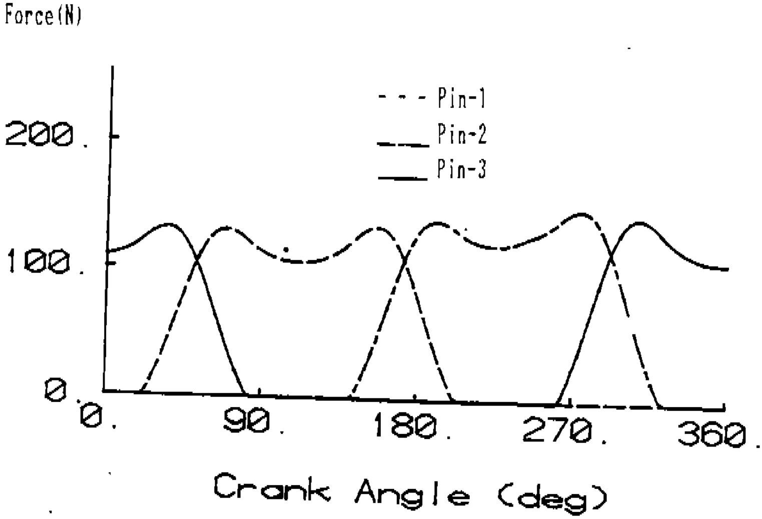

Fig.5 Force acting on 3--pins

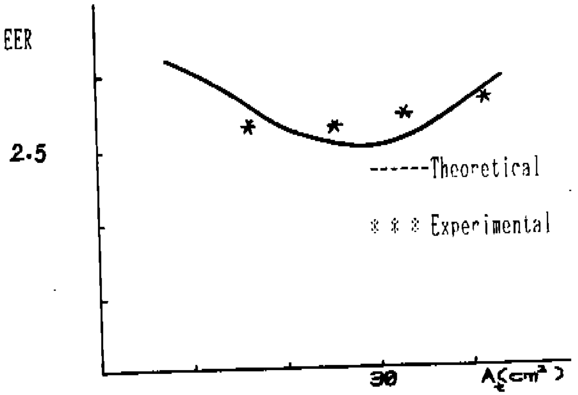

Fig.6 EER / sealing ring area diagram

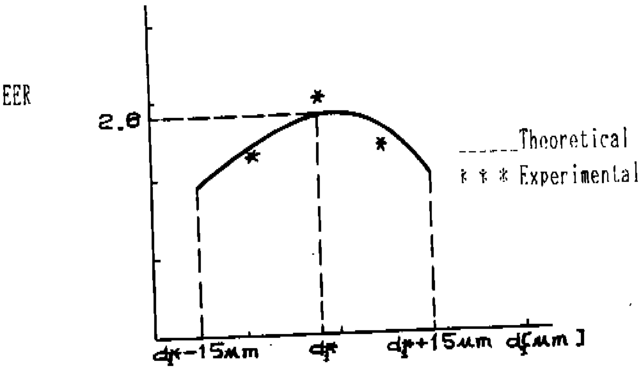

Fig.7 EER / pin diameter diagram