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A STUDY ON THE REDUCTION OF TORQUE VARIATION IN SCROLL COMPRESSORS

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ABSTRACT

This paper introduces a way of reducing the torque variation in scroll compressors. In a scroll compressor where radial compliance device such as the slider bush is used for the flank sealing, the orbiting scroll member is given the freedom of movement in the radial direction. When such an orbiting scroll member orbits around the nonorbiting scroll member whose center is offset from that of the crankshaft, the distance from the crankshaft center to the point of loading is subject to variation, while constant orbiting radius can be maintained. If the variation of this loading distance is made opposite to that of the loading, then the amplitude of the torque variation can be made reduced. Illustrative calculation on a typical scroll compressor has been made to demonstrate this concept of reduction in the torque variation of a scroll compressor.

NOMENCLATURE

b : oldham ring key height
c : distance to orbiting scroll c.g
d : torque arm
F_1, F_2, F_3, F_4 : reaction forces at oldham ring
F_a, F_{tg}, F_{rg} : gas forces in the axial, tangential, and radial directions
F_{bl} : driving force of crank pin
F_c, F_{cbl} : centrifugal forces of orbiting scroll & slider bush
F_{sbt}, F_{sbr} : components of drive bearing force in tangential and radial directions
f_s : radial sealing force
h : wrap height
l : orbiting scroll hub length
m_{or} : oldham ring mass
O_1 : crankshaft center
O_2 : fixed scroll center
O_3 : orbiting scroll and bush center
O_4 : crank pin center
P : point of loading on the slider bush
r_a : orbiting radius
r_e : offset distance of fixed scroll
r_p : crank pin eccentricity
R_{ox}, R_{oy} : radii of oldham ring in x & y axes
r_{sb} : radius of the slider bush
y_r : oldham ring displacement in y axis
w_{bl} : distance from crank pin center to sliding surface
α_1 : offset angle of fixed scroll
γ : sliding surface inclination
δ : orbiting scroll end plate thickness
θ_c : crank angle
μ : friction coefficient

INTRODUCTION

One of the sources for the refrigerant compressor vibration is the crankshaft load fluctuation. It has been known that the crankshaft load fluctuation of a scroll compressor is far less than those of other types of compressors[1]. This distinctive characteristic of the scroll compressor results from smooth variation of the gas loading. In scroll compressors, several compression chambers are simultaneously involved in the gas compression process so that the gas compression rate is not so rapid as in other types of compressors.

In this study, we present a way of achieving further reduction in the crankshaft torque fluctuation for scroll compressors. For conventional scroll compressors, the fluctuation pattern of the crankshaft torque resembles that of the gas loading. Hence, if the crank arm can be made variable in the opposite sense to the variation of the gas loading, the torque fluctuation could be managed to be reduced. One way of generating variable crank arm is to displace the center of the fixed scroll from the axis of the crankshaft rotation with adoption of the radial compliance device for the orbiting scroll. While, in conventional scroll compressors, the main purpose of the application of the radial compliance is to promote the flank sealing between the fixed and orbiting scroll elements, primary role of the radial compliance in the present application is to satisfy the geometric constraint that constant orbiting radius should be maintained during normal operating condition, in addition to the radial sealing.

FORMULATION FOR DRIVING FORCE AND TORQUE

A slider bush is shown in Fig. 1 as the radial compliance device for the present analysis. The hub of the orbiting scroll is concentrically received by the bush so that the two elements have the same center, and the two elements can slide together along the flat sliding surface of the crank pin. Relative rotating motion between the orbiting scroll and the bush is made at the outer periphery of the bush as the crankshaft rotates. The orbiting scroll is actuated by the centrifugal force to maintain the flank contact with the fixed scroll wrap so as to yield constant orbiting radius.

Fig. 2 shows a diagram of the forces acting on the slider bush when the fixed scroll is off centered by (r_e, α_1) from the center of the crankshaft. The dashed line C represents the flat surface made on the crank pin. The inclination of this surface γ with respect to the radial direction is, in general, to control the radial sealing force between the two scroll elements. Then the slider bush can slide in the direction of O_3O_4 which is parallel to the line C, and its center, O_3, is determined by the geometric constraint such that the orbiting radius r_o be maintained for normal operation.

The driving force F_{bi} from the crankshaft can be considered to be transmitted to the bush at a point P on the sliding surface of the bush. The distance from the crankshaft center O_1 to the driving force F_{bi} is denoted by d. The length of the crank arm d is affected by the sliding of the orbiting scroll, the range of which is proportional to the offset distance of the fixed scroll. μF_{bi} is the friction force at the sliding surface between the crank pin and the slider bush. F_{sbt} and F_{sbr} are the tangential and radial components of the reaction force from the orbiting scroll to the bush through the orbiting scroll hub, respectively. F_{cbi} is the centrifugal force of the bush. μF_{sbt} is the friction force due to the relative rotating motion between the orbiting scroll hub and the bush.

The forces and moment acting on the slider bush are balanced as in the equations (1)-(3).

$$\mu F_{sbt}+F_{sbr}+F_{cbl}(\cos\alpha_3-\beta_3)+F_{bl}\sin\alpha_3-\mu F_{bl}\cos\alpha_3=0 \tag{1}$$

$$F_{sbt}+F_{cbl}\sin(\alpha_3-\beta_3)-F_{bl}\cos\alpha_3-\mu F_{bl}\sin\alpha_3=0 \tag{2}$$

$$\mu r_{sb}F_{sbt}+(r_p\cos\gamma+r_{34}-d)F_{bl}-w_{bl}\mu F_{bl}=0 \tag{3}$$

From these equations, the driving force F_{bi}, the torque arm d, and the driving torque T are expressed by the equations, (4), (5), and (6), respectively.

$$F_{bl}=\frac{F_{sbt}+F_{cbl}\sin(\alpha_3-\beta_3)}{\cos\alpha_3+\mu\sin\alpha_3} \tag{4}$$

$$d=r_p\cos(\gamma)+r_{34}+\mu(r_{sb}F_{sbt}/F_{bl}-w_{bl}) \tag{5}$$

$$T=d\cdot F_{bi} \tag{6}$$

Here, α_3, β_3, and r_{34} are determined by given geometric parameters. The tangential component of the reaction force by the orbiting scroll, F_{sbt}, can not be obtained explicitly from the above equations. Instead, the relation between F_{sbt} and F_{sbr} is found as in the equation (7).

$$(1+\mu^2)F_{sbt}+(\cot\alpha_3+\mu)F_{sbr}+[(\cot\alpha_3+\mu)\cos(\alpha_3-\beta_3)+(1-\mu\tan\alpha_3)\sin(\alpha_3-\beta_3)]=0 \tag{7}$$

To obtain F_{sbt} and F_{sbr}, the motion of the orbiting scroll and the oldham ring also should be taken into consideration. The equations for the movement of these two members are given by the equation (8) in matrix form.

$$A \cdot F = B \tag{8}$$

where

$$A=\begin{bmatrix} a_{11} & a_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & -1 & a_{24} & a_{25} & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & a_{34} & a_{35} & 0 & 0 & 0 & 0 \\ a_{41} & 0 & 0 & a_{44} & a_{45} & 0 & 0 & 0 & F_a \\ 0 & a_{52} & a_{53} & a_{54} & a_{55} & 0 & 0 & -F_a & 0 \\ a_{61} & 0 & 0 & a_{64} & a_{65} & 0 & 0 & \mu_t F_a & 0 \\ 0 & 0 & 0 & -\mu_0 & -\mu_0 & -1 & 1 & 0 & 0 \\ 0 & 0 & 0 & -1 & 1 & -\mu_0 & -\mu_0 & 0 & 0 \\ 0 & 0 & 0 & R_{ox} & R_{ox} & R_{oy} & R_{oy} & 0 & 0 \end{bmatrix} \quad F=\begin{bmatrix} F_{sbt} \\ F_{sbr} \\ f_s \\ F_1 \\ F_2 \\ F_3 \\ F_4 \\ r_r \\ r_\theta \end{bmatrix} \quad B=\begin{bmatrix} F_{cbl} \\ -F_c+F_{rg} \\ F_{tg}+\mu F_a \\ (h/2+\delta)F_{tg} \\ (c+\delta)F_c-F_{rg}(h/2+\delta)-r_a/2\cdot F_a \\ -r_a/2F_{tg} \\ 0 \\ m_{or}\,\ddot{y}_r \\ 0 \end{bmatrix}$$

Forces in the matrix F are to be obtained. Elements in the matrices are referred to Ref.[2].

CALCULATION RESULTS

Typical values of a 3 hp class scroll compressor were used for the input data to the equation (8). And the gas forces, Ftg, Frg, and Fa were calculated from the pressure distribution over each compression chambers by assuming adiabatic compression process. With these inputs, the equation (8) were solved to give various forces in the matrix F.

Fig. 3 shows the bush bearing forces, F_{sbt} and F_{sbr}, and the radial sealing force fs for the case of no offset and γ=0°. All the forces were normalized with mean tangential bush bearing force, F_{sbtm}. For conventional scroll compressors, where the center of the fixed scroll coincides with that of the crankshaft, the major reason for the torque variation is the variation of F_{sbt}, since the torque arm d does not vary with the crank angle as shown in Fig. 4.

To investigate the effects of the fixed scroll eccentricity on the torque variation, the center of the fixed scroll was displaced by r_e/r_a=0.06, and the offset orientation, or offset angle was varied from α_1= 0° to α_1 =315°. The orientation of the offset affects the phase of the torque arm variation (Fig.5). The change in the torque fluctuation caused by the offset orientation is shown in Fig. 6(a) and (b). The peak to peak amplitude of the torque fluctuation is found to increase as α_1 approaches 180°, and then decrease afterwards, being minimum at α_1=315°. The effects of the offset distance is shown in Fig. 7, where the eccentricity r_e was varied from r_e/r_a=0 to r_e/r_a=0.15 with the offset angle at α_1=315°. The amplitude of the torque fluctuation is found minimum at r_e/r_a=0.06.

The effects of the fixed scroll offset on the peak to peak level of the torque fluctuation are replotted in Fig. 8(a) and (b) for the offset angle and eccentricity, respectively. The best reduction in the torque fluctuation is found at r_e/r_a=0.06 and α_1 =335°. The torque fluctuation at this offset configuration is compared to that of conventional (no offset) in Fig. 9. While the peak to peak amplitude of torque variation is 17.6% for the conventional, it is reduced to nearly half of the value.

While an advantage of reducing the torque variation can be taken by offsetting the center of the fixed scroll, there occurs some drawback on the performance of the compressor. Fig. 10 shows that the variation in the radial sealing force f_s is amplified due to the fixed scroll offset. Changes in the drive surface inclination γ only shift the mean level of f_s with little effect on the amplitude of f_s variation, and the torque variation itself is not much influenced (Fig. 11(a) &(b)).

DISCUSSIONS

In a scroll compressor, the torque fluctuation is mainly caused by the variation of the tangential gas force. If the torque arm can be made inversely proportional to the force variation, the torque fluctuation could be suppressed. One way of accomplishing such a torque arm variation is to make the fixed scroll off-centered from the crankshaft center, while the orbiting scroll is sustained to orbit around the fixed scroll with constant orbiting radius by the aid of the radial compliance device.

The variation of the torque arm obtained in this way is sinusoidal, while that of the driving force is not so. Therefore, perfect cancellation of the torque variation can not be expected, but at least one of the major peaks of the torque variation can be effectively eliminated. For instance, as shown in Fig. 9, peak at θ_c =45° was eliminated by having the offset angle at α_1=315°, that is, providing the longest torque arm for the minimum torque(the crank angle θ_c is defined to be positive for the clockwise rotation, and the offset angle α_1 positive for counter-clockwise rotation). In addition to this, another maximum suppression of the torque fluctuation was obtained at the angle opposite to the offset orientation, where the torque arm was made minimum.

The peak to peak fluctuation of the driving force is about 17.6% for the present example, but the best suppression of the torque fluctuation has been obtained at r_e/r_a=0.06, rather than r_e/r_a=0.088. This is because r_{34} in the expression of d is not directly related to the offset amount r_e, but determined by trigonometric relation with involved geometric parameters.

CONCLUSION

It has been demonstrated that by the application of the variable torque arm to a scroll compressor, the fluctuation amplitude of the crankshaft torque could be reduced by as much as about 50%. The most effective reduction in the torque fluctuation has been achieved when the orientation of the fixed scroll is made so as to yield a maximum torque arm at the crank angle where the driving force is minimum. Optimum offset distance of the fixed scroll depends on the amplitude of the load fluctuation.

One reverse effect of the variable torque arm on the compressor performance is that it magnifies the amplitude of the radial sealing force variation. So, for the realization of this concept of torque fluctuation reduction, it may be necessary to make some compromise between the two opposite effects brought by the application of the variable torque arm .

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1. Ishii, N., et al, "A study on dynamic behavior of a scroll compressor," Proc. of Inter. Comp. Eng. Conf. at Purdue, Vol. 3, pp.901-916, Aug., 1986.
2. Kim, H.J., et al, "A study of the radial sealing force control between the scroll wraps in scroll compressors," Proc. of Air Conditioning and Refrigeration Engineering, SAREK, pp.341-347, Jun., 1995.

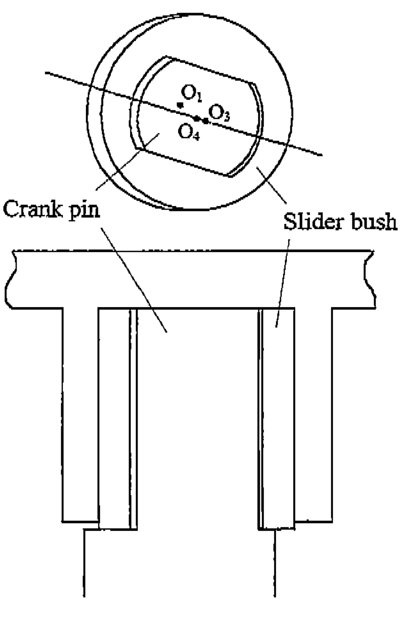

Fig.1 Slider bush mounted on crank pin

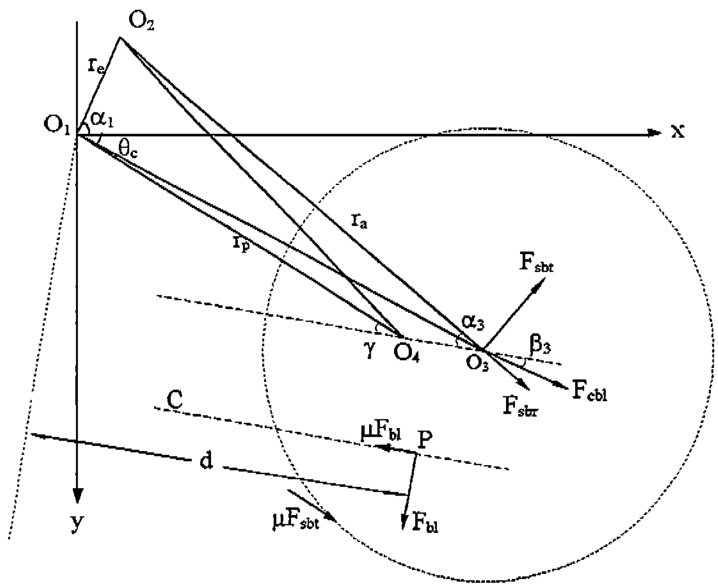

Fig.2 Force diagram on slider bush

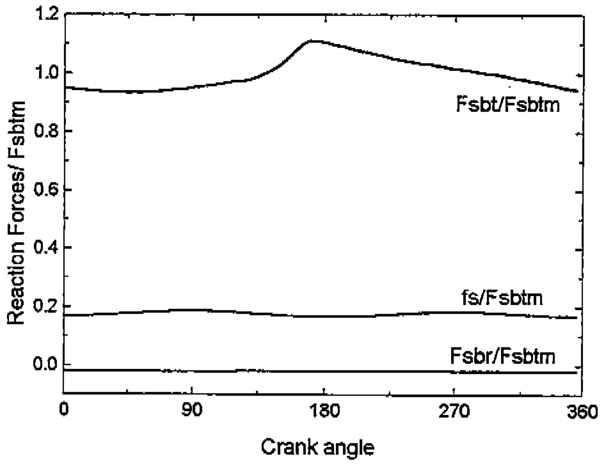

Fig.3 Drive bearing forces and radial sealing force

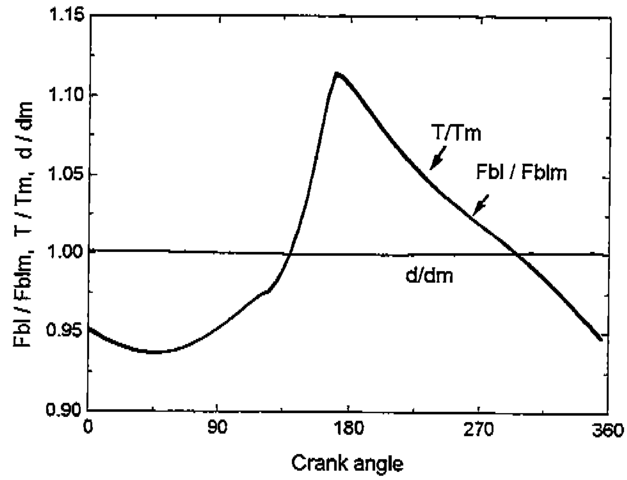

Fig.4 Torque variation of conventional scroll

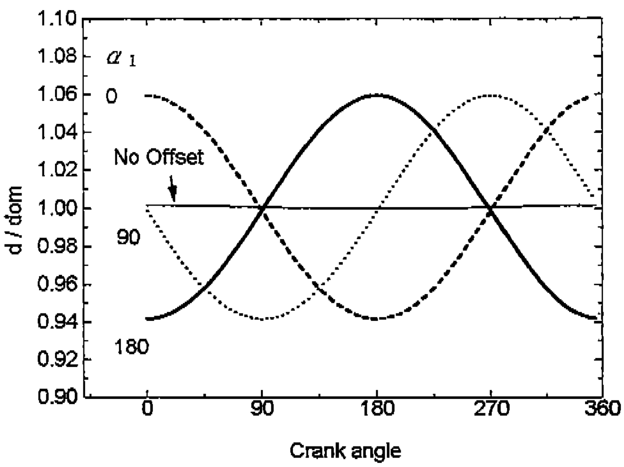

Fig.5 Effect of offset angle on variation of torque arm(α_1=0°~180° at r_e/r_a=0.06)

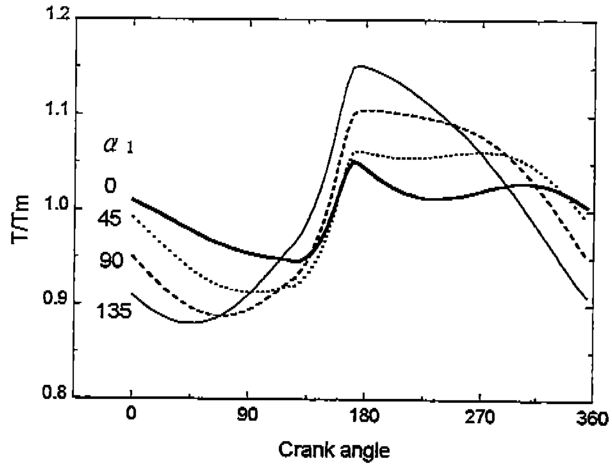

Fig.6(a) Effect of offset angle on torque fluctuation(α_1=0°~135° at r_c/r_a=0.06)

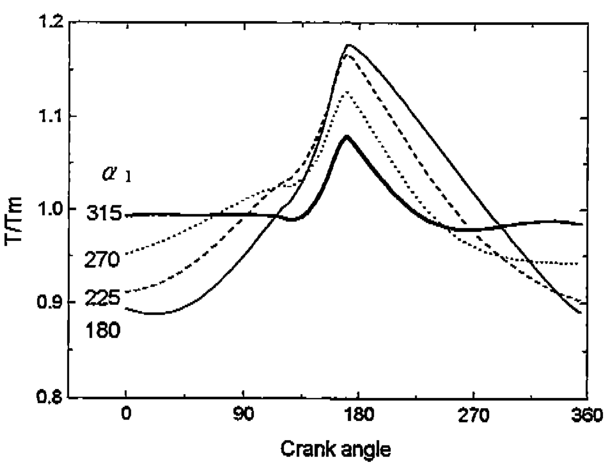

Fig.6(b) Effect of offset angle on torque fluctuation(α_1=180°~315° at r_e/r_a=0.06)

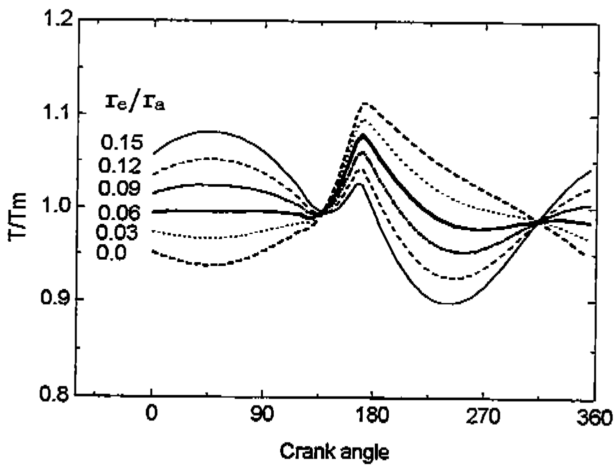

Fig.7 Effect of offset distance on torque fluctuation(r_e/r_a=0~0.15 at α_1=315°)

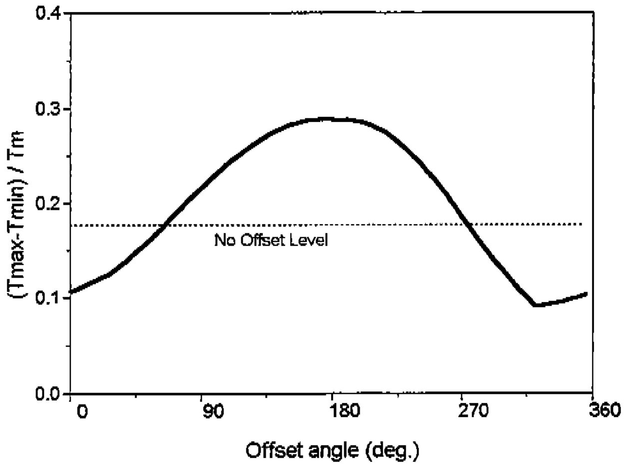

Fig.8(a) Effect of offset angle on torque fluctuation amplitude

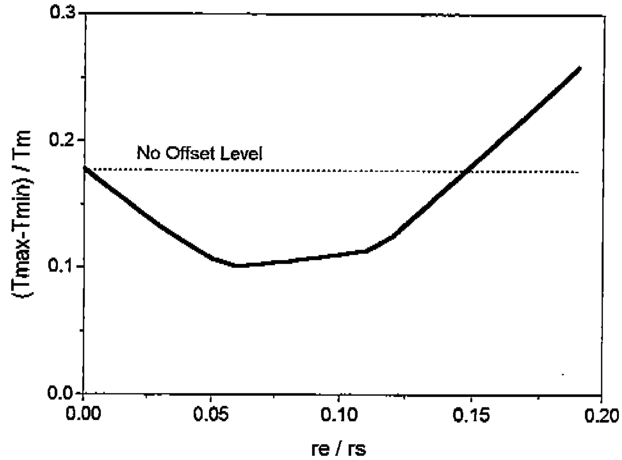

Fig.8(b) Effect of offset distance on torque fluctuation amplitude

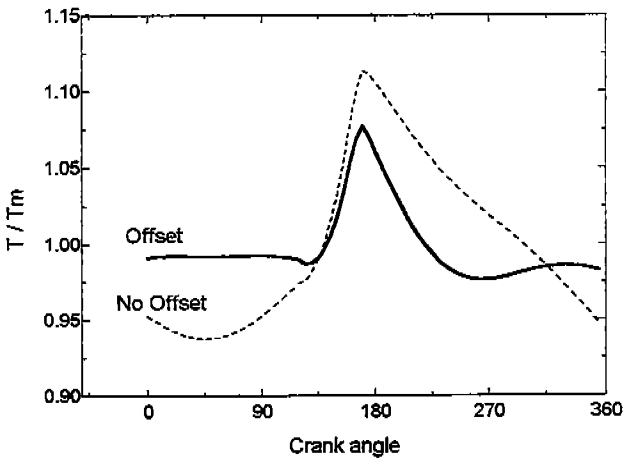

Fig.9 Comparison of torque fluctuations between offset and conventional scrolls

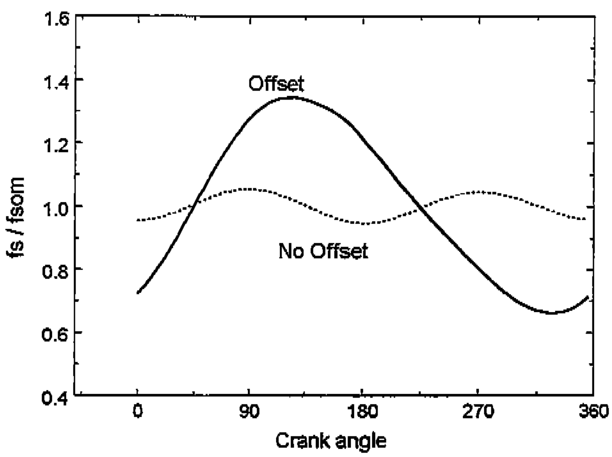

Fig.10 Effect of fixed scroll offset on radial sealing force

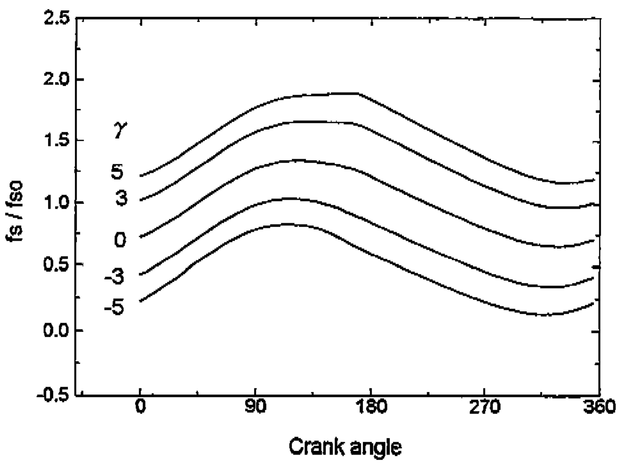

Fig.11(a) Effect of sliding surface inclination γ on radial sealing force variation(γ =-5°~5°)

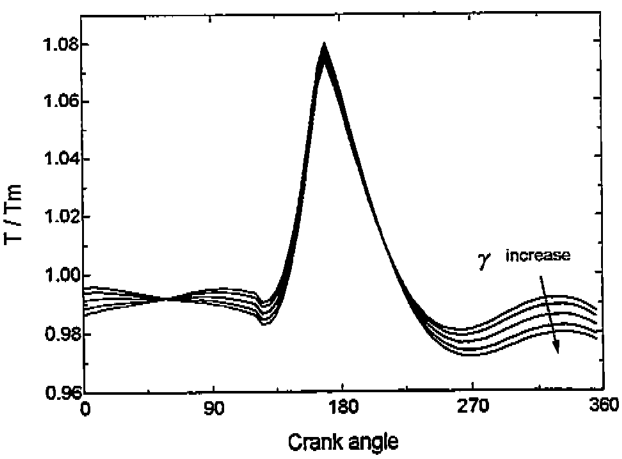

Fig.11(b) Effect of sliding surface inclination γ on torque variation