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STUDY ON THE RADIAL SEALING PRINCIPLES OF SCROLL FLUID COMPRESSORS WITH RADIAL COMPLIANT MECHANISM

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ABSTACT

In scroll compressors, various radial compliant mechanisms are widely employed to reduce the gas leakage between the orbiting and fixed scroll members, to lower configuration elements precision and to protect the scroll wrap when the pressure in the pockets is abnormally increased. But the investigations on the forces imposed on the radial compliant mechanism are quite complex and there are many factors influencing magnitude of the forces. In this paper: (1) The analysis models of forces exerted on the orbiting scroll and the radial compliant mechanism are set up while the ball coupling is employed as the anti-rotation mechanism of orbiting scrolls. A force — F_s *was put forward to judge the reliability of the radial compliant mechanism. (2) Two kinds of scroll compressors are analyzed, the top profile of one kind is not modified while the other is modified with symmetric arc curves, accordingly the gas forces in the pockets are calculated and their influences on the radial compliant mechanism are investigated.*

INTRODUCTION

The radial compliant mechanism is usually used to control the radial clearance to reduce the gas leakage. The orbiting radius can be varied to a certain extent according to the changes of pressure in the working pockets and working state.

When the Oldham ring anti-rotation mechanism cooperates with the radial compliant mechanism, the orbiting radius can change arbitrarily, if the practical orbiting radius is less than the theoretical one because the orbiting scroll movement loca is strictly confined, and thus the scroll wraps may have a tendency to contact with each other. However, when the ball coupling incorporated with the radial compliant mechanism used to keep orbiting scroll moving, the maximal orbiting radius will be restricted by the ball coupling and the radial clearance will always exists in practice.

In this paper, the mechanics of the ball coupling in cooperation with the eccentric-disc radial compliant mechanism are studied, and a force F_S, part of the interaction force between the steel-balls of the ball couplings and the orbiting scroll, is employed to judge whether the radial compliant mechanism in dysfunction.

The gas forces and operating conditions influence the working state of the radial compliant mechanism to some extent. The radial gas force is often ignored when studying the radial compliant mechanism, which is generally reliable. Nevertheless, if the modifications of the top profile with symmetric arc curves are adopted in the scroll compresses, the radial gas force can not be neglected because its magnitude is approximate to that of the tangential one. In this paper, F_S of two kinds of scroll compressors are calculated, with one modified with symmetric arc curves and the other kept unmodified, and the results show: it is reliable to ignore the radial gas force in the unmodified compressors; while it will lead error if the radial gas force is ignored in the modified compressors.

FORCES ON COMPLIANT MECHANISM

Fig.1 shows the schematic diagram of the radial compliant mechanism and the forces exerted on the radial compliant mechanism. O_s is the center of shaft (which is also to the center of the fixed scroll), O_b is the center of the eccentric-disc pin, O_p is the center of the crankpin, and O_m is center of the eccentric-disc

(also the center of the orbiting scroll). It is supposed that the distance between O_m and O_s is theoretical orbiting radius ρ when O_s is identical to O_b. The forces acted on the eccentric-disc are:

F_{dt}, F_{dr} — interaction forces between the orbiting scroll and the eccentric-plate; F_{pr}, F_{pt} — interaction forces between the crankpin and the eccentric-plate; F_{IB} — centrifugal forces of the balance weight connected with eccentric-disc. These forces result in the eccentric-disc rotation around the center of the crankpin and the center of the orbiting scroll O_m changes, so the orbiting radius could be enlarged as shown in Fig.1(b) or decreased as shown in Fig.1(c).

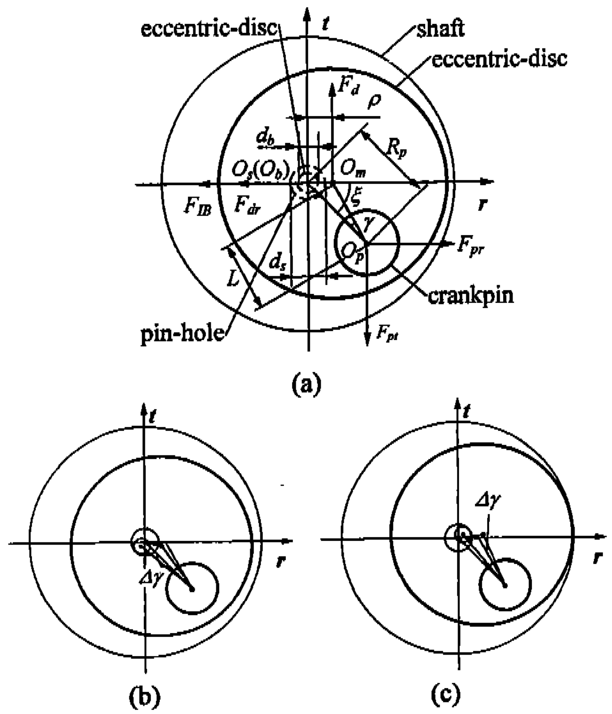


Fig.1 schematic diagram of compliant mechanism

No matter whether the rotating direction of the eccentric-disc is clockwise or counterclockwise, the maximum rotation angle is

$$\Delta\gamma_{max} = 2 \cdot \arcsin \frac{d_s - d_b}{4R_p} \qquad (1)$$

where, d_s is the diameter of the pin-hole on the shaft; d_b is the diameter of the eccentric-disc pin; R_p is the radius of the crankpin.

Fig.1(b) shows the position of eccentric-disc at the case of the maximal clockwise rotation angle and the maximal orbiting radius is obtained

$$\rho_{max} = \sqrt{R_p^2 + L^2 - 2 \cdot R_p \cdot L \cdot \cos(\gamma + \Delta\gamma)} \qquad (2)$$

On the Contrary, the minimal orbiting radius is obtained by the counterclockwise rotation of the eccentric-plate as shown in Fig.1(c)

$$\rho_{max} = \sqrt{R_p^2 + L^2 - 2 \cdot R_p \cdot L \cdot \cos(\gamma - \Delta\gamma)} \qquad (3)$$

BALL COUPLING

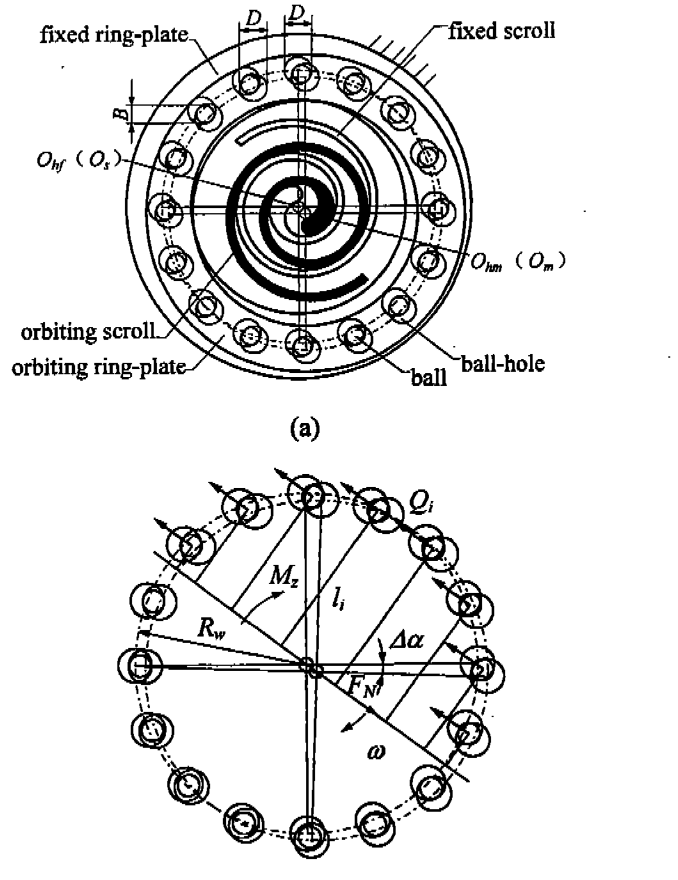


Fig.2 schematic diagram of ball coupling

The ball coupling is such a anti-rotation device that is capable of supporting the axial gas force through the balls. The schematic structure is shown in Fig.2(a), the device is composed of two pieces of ring-plates and some steel-balls, and there are some ball-holes in the ring-plates to hold the steel-balls.

The numbers and the diameter of the steel-balls are the same as those of the ball-holes. One ring-plate is sticky mounted with the fixed scroll while the other with the orbiting scroll. The steel-balls are placed between the fixed and the orbiting ring-plate. Apparently, the following equation can be derived to meet working.

$$B = D - \rho \tag{4}$$

The self-rotation moment M_z shown in Fig.2(b) are acted on the orbiting scroll, which tends to make the orbiting scroll rotate. Supposed that self-rotation moment M_z makes the orbiting scroll change with $\Delta\alpha$ and in the meantime the ring-plates are regarded as rigid body, the elastic deformation of some of the steel-balls will occur.

The center coordinate of the ball-holes on the fixed ring-plate is

$$\begin{cases} x_{fi} = R_w \cdot \cos[(i-1)\Delta\psi] \\ y_{fi} = R_w \cdot \sin[(i-1)\Delta\psi] \end{cases} \tag{5}$$

The center coordinate of the ball-holes on the orbiting ring-plate changes as

$$\begin{cases} x_{mi} = R_w \cdot \cos[(i-1)\Delta\psi - \Delta\alpha] + \rho \cdot \cos(-\theta) \\ y_{mi} = R_w \cdot \sin[(i-1)\Delta\psi - \Delta\alpha] + \rho \cdot \sin(-\theta) \end{cases} \tag{6}$$

Where, $\Delta\psi$ is the phase difference angle of the adjacent hole-balls.

The deformation of the steel-balls is

$$\varepsilon_i = \sqrt{(x_{mi} - x_{fi})^2 + (y_{mi} - y_{fi})^2} - \rho \tag{7}$$

$i = 1,2,\cdots,N$ in equation(5)(6)(7).

Because the deformation of the steel-balls is very little, in other words, $\Delta\alpha$ is very small, the directions of Q_i are parallel to each other and also parallel to the direction of radial gas force. The normal length between the center of the orbiting scroll and the point at which the forces Q_i acted is

$$l_i = \left|R_w \cdot \sin[(i-1)\Delta\psi + \theta]\right| \tag{8}$$

Apparently, there is

$$\sum Q_i \cdot l_i = M_z \tag{9}$$

Where, M_z is the self-rotation moment.

It is quite complex to calculate the values of Q_i accurately at the deformation of ε_i, because the ε_i are quite small, then it can be simplified as

$$Q_i \propto \varepsilon_i \tag{10}$$

Therefore, the composition of forces — Q_i is

$$Q = \sum Q_i \tag{11}$$

On the other hand, to make the orbiting scroll move accurately, the forces on the orbiting scroll must be balanced out, so there exists a force F_N acted on the orbiting scroll by the eccentric-disc, $F_N = -Q$.

FORCE EQUATION OF RADIAL COMPLIANT MECHANISM AND ORBITING SCROLL

Generally, the distance between the original center of the fixed and orbiting scroll is the theoretical orbiting radius when the center of the eccentric-disc pin is identical to that of the shaft-pin hole. In this case, the eccentric-disc pin does not contact with the inner wall of the shaft-pin hole. The forces acted on the orbiting scroll are shown in Fig.3. It is prescribed that the positive direction of the r and t axis is also the positive direction of the forces.

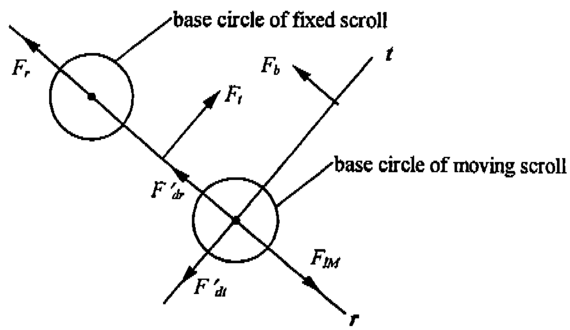


Fig.3 Forces on orbiting scroll

The force equations of the eccentric-disc are (as shown in Fig.1(a))

t direction

$$F_{dt} + F_{pt} = 0 \tag{12}$$

r direction

$$F_{dr} + F_{pr} + F_{IB} = 0 \tag{13}$$

The moment equation of the eccentric-disc are

$$F_{dt} \cdot (R_p \cdot \cos\xi - \rho) + (F_{dr} + F_{IB}) \cdot R_p \cdot \sin\xi = 0 \tag{14}$$

The forces equations of the orbiting scroll are as follows (as shown in Fig 3)

t direction

$$F_t + F'_{dt} = 0 \tag{15}$$

r direction

$$F_r + F_{IM} + F_b + F'_{dr} = 0 \tag{16}$$

Where, F_r — radial gas force; F_t — tangential gas force; F_{IM} — centrifugal forces of the orbiting scroll member; F_b — interaction force between the steel-balls and the orbiting scroll; F'_{dr}, F'_{dt} — interaction force between the eccentric-plate and the orbiting scroll (as shown in Fig.3).

The interaction force — F_b can be decomposed of two components, one is Q, which produce converse moment to balance the self-rotation moment M_z, the other is named as F_S, which balances the radial forces, because the steel-balls is symmetrical, the components of F_S — F_{si} are equivalent, $F_{Si} = F_S / N$. Likewise, the interaction force F'_{dr} is comprised of F'_{dr} and F_N. Equation (16) can be translated into

$$F_r + F_{IM} + (F_S + Q) + (F''_{dr} + F_N) = 0 \quad (17)$$

Furthermore

$$F_{dr} = -F'_{dr} \qquad F_{dt} = -F'_{dt} \quad (18)$$

The moment equation can be obtained form equation (13) (14) (17) (18)

$$F_t \cdot (R_p \cdot \cos\xi - \rho) + (F_r + F_{IP} + F_{IM} + F_b) \cdot R_p \cdot \sin\xi$$
$$=$$
$$F_t \cdot (R_p \cdot \cos\xi - \rho) + (F_r + F_{IB} + F_{IM} + Q + F_S) \cdot R_p \cdot \sin\xi$$
$$= 0 \quad (19)$$

In equation(19), the forces F_{IB}, F_{IM}, F_t and F_r are determined by operating conditions, the rotation angle θ and the structure parameters such as R_p, R_b, m_M, m_B, ξ, ρ and et al. Interaction force Q is related to tangential gas force F_t and orbiting radius ρ. Only interaction force F_S is an unknown quantity and thus can be determined. It must be noticed that F_S is the force caused by elastic deformation and occurs only the direction of F_S is negative, which means that the eccentric-plate can not rotate clockwise under the action of the ring-plates and the steel-balls and the machine can operate normally; If the value of F_S is positive, it implies that the eccentric-disc can rotate clockwise and radial sealing is not available, in this case, the leakage increases and the compressor could be degraded. So the interaction force F_S should be regarded as a criteria to judge the operating state of the radial compliant mechanism.

During the design of the radial compliant mechanism, it must be ensured that the direction of F_S is negative and the magnitude of F_S is enough to keep the radial sealing under any possible operating conditions and arbitrary rotating angles through choosing reasonable structure parameters such as R_p, ξ, m_b and et al.

The criterion to choice the feasible F_S is: (1) The direction of F_S is negative under arbitrary rotating angles, otherwise, the eccentric-disc will rotate clockwise and the radial clearance will increase, which leads the increment of the radial leakage; (2) If the magnitude of F_S is very large, it will result in much more friction loss between the steel-balls and the ring-plates. On the other hand, the radial clearance can not be increased in time when the pressure in the working chambers increase abnormally.

INFLUENCES ON COMPLIANT MECHANISM

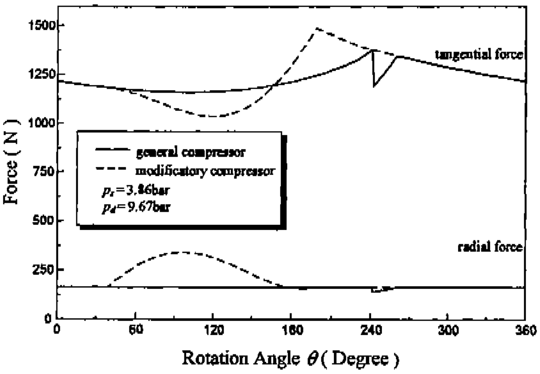
There are many parameters to impact the moving state of the radial compliant mechanism, which complicates the design of the radial compliant mechanism.

The scroll compressors studied in this paper are used for automobile air-conditioning system, the operating conditions changes dramatically (including rotation speed, suction and discharge pressure); The top profile of one widely used scroll compressor is machined directly with milling cutter (in this paper, this kind of scroll compressor is named as general scroll compressor) and the other adopts the method of modification with symmetric arc curves for the sake of reducing discharge dead volume and noise (in this paper, this kind of compressor is named as modificatory scroll compressor); The suction volume and the inner pressure-ratio of these two kinds of scroll compressors are equal to each other; Discharge valves are adopted in the both kinds of the scroll compressors. Consequently, the gas forces such as the radial gas force and the tangential gas force in the modificatory scroll compressor are quite different from these of the general scroll compressors.

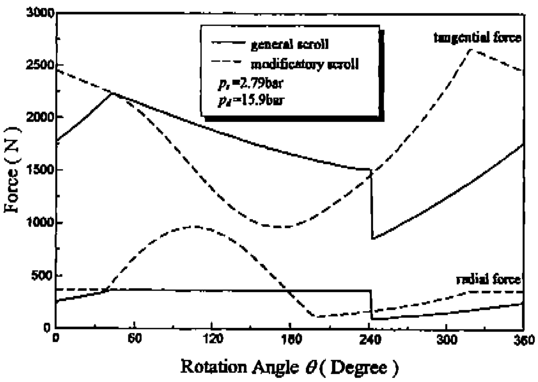
It is well known, the magnitude of the radial gas force is much less than that of the tangential gas force in the general scroll compressors. However, the magnitude of the radial gas force approaches to that of the tangential one in the modificatory scroll compressors.

On the assumption that: the compression process can be simplified as an adiabatic and no-leakage process, the valve opens immediately if the pressure in the innermost chamber is more than the discharge chamber and vice versa.

The tangential and radial gas forces at different conditions are shown in Fig. 4. The inner pressure-ratio is equal to the outer pressure-ratio in Fig.4(a) and the inner pressure-ratio is more than the outer pressure-ratio in Fig.4(b). The solid line represents the gas forces of the general compressors while the dashed line represents the gas forces of the modificatory scroll compressors. Apparently, the radial gas force can not keep constant and a peak appears in modificatory scroll compressors.



(a) inner pressure-ratio = outer pressure-ratio



(b) inner pressure-ratio = outer pressure-ratio

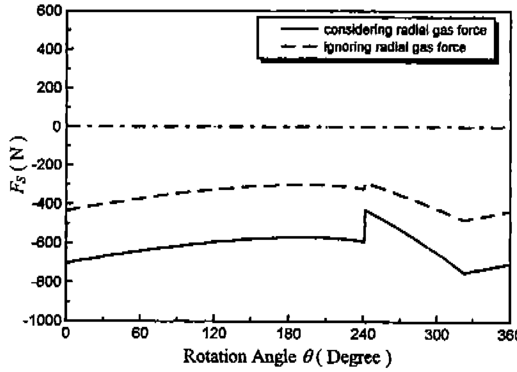
Fig.4 radial and tangential gas force

The outer pressure-ratio may be more than the inner pressure-ratio, therefore, the swings of the radial and tangential gas forces change more acutely as shown in Fig 4(b). It could be seen that the magnitude of the radial gas force could reach the order of the tangential force. So it is not reasonable to ignore the effect of the radial gas force on the radial compliant mechanism.

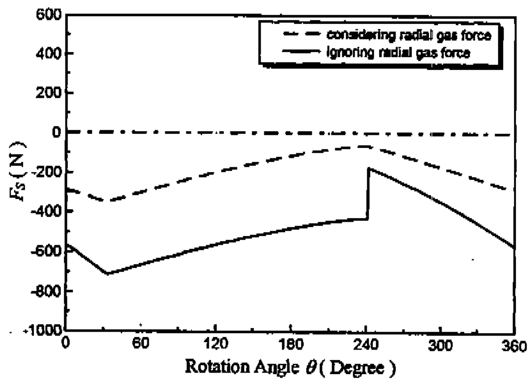
An example is presented to illuminate the effects of the radial gas force on the radial compliant mechanism in the modificatory and general scroll compressors. Two conditions are chosen as following (Working fluid used in the system is R12):

Condition1: n=2000 r/min; suction pressure is 3.88bar (evaporation temperature is 7.2 ℃); discharge pressure is 13.48bar (condensation temperature is 54.4 ℃).

Condition2: n=4000 r/min; suction pressure is 2.99bar (evaporation temperature is −1 ℃); discharge pressure is 15.94bar (condensation temperature is 62 ℃).



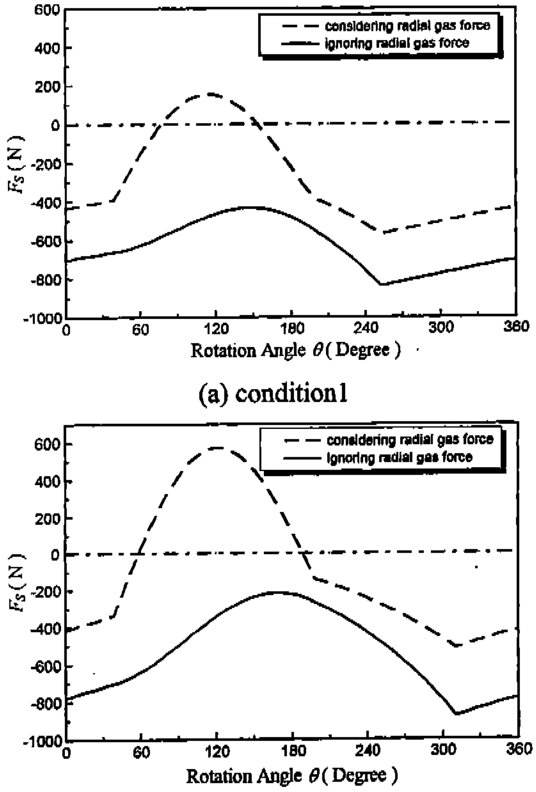
(a) condition 1



(b) condition 2

Fig.5 F_S in the general compressors

Fig.5 shows the calculation results of the general scroll compressors under operating condition1 (Fig.5(a)) and condition2 (Fig.5(b)). The solid line represents the case ignoring the radial gas force thoroughly while dashed line represents the case considering the effect of the radial gas force. It was discussed in section 3 (*Force Equation of Radial Compliant Mechanism and Orbiting Scroll*) that F_S is the criterion to judge the reliability of the radial compliant mechanism. It could be seen in Fig.5 that F_S is able to satisfy the request (the value of F_S must be less than zero) under condition1 and 2. Therefore, the radial gas force can be ignored in the general scroll compressors, which only leads the difference of the magnitude of F_S.



(b) condition 2

Fig.6 F_S in the modificatory compressors

Fig.6 shows the calculation results of the modificatory scroll compressors under condition 1 (Fig.6(a)) and condition 2 (Fig.6(b)). Apparently, It will bring remarkable error if the effect of the radial gas force is ignored.

To design the experimental scroll compressor modified with symmetric arc curves, the structure parameters are first selected by ignoring the radial gas force, and during the experimental process, it has been found that the discharge temperature is very high and the volume efficiency is quite low (< 60%) when the scroll compressor operates under condition 2, which implies the leakage is very serious. Consequently, the radial gas force is considered and F_S is calculated. According to the results, the structure parameters of this scroll compressor are chosen over again, which improve the discharge temperature and the volume efficiency distinctly (the volume efficiency > 80%).

Because of the action of the lubrication, the friction forces are much less than the other forces, it is reasonable to neglect the action of the friction forces.

CONCLUSION

(1) The forces on the radial compliant mechanism and the orbiting scroll member are analyzed in details the conjunct action of the ball-coupling and the eccentric-disc radial compliant mechanism. The moments and forces equations are set up to analyze the characteristic of the radial compliant mechanism.

(2) An interaction force between the steel-balls of the balls coupling and the orbiting scroll — F_S is supposed to judge the operating state of the radial compliant mechanism.

(3) The difference of the tangential and radial gas forces are discussed in two kinds of scroll compressors, the top profile of one kind of scroll compressors is modified with symmetric arc curves while the other is kept unmodified.

(4) The influences of the radial gas force and operating conditions on the radial compliant mechanism are discussed in this paper, from which the conclusion is obtained that the influence of the radial gas force can be ignored in the unmodified scroll compressors while that should be considered in the modified ones (Especially, the radial gas force cannot be ignored if the modification scroll compressors operate under high rotation speed and high pressure-ratio).

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