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ABSTRACT

Rotational imbalance is one of the factors that affect performance and reliability of rotary compressor. The rotor system of the compressor is taken as a study object. Both the inertia force and the gas force are taken into account to build a mechanical model. The mechanical model is used to deduce the relation between the characteristics of balancer and rotational angle of the crankshaft. Residual imbalance function of the rotor system is constructed to optimize the parameters of the balance. Vibration experiment of a compressor is performed for validating the method.

1. INTRODUCTION

The eccentric part of a crankshaft induces a centrifugal inertia force that causes the imbalance issue of compressor. The traditional solving method is to add a pair of balancers in the upper and lower side of the rotor. But in the process of gas compression, the gas pressure in the chamber varies dramatically with the rotational angle of crankshaft, and the crankshaft bears a wide rage of gas force. As shown in figure 1, in some sectors of rotational angle, the gas force is less than the inertia force of eccentricity, but in other angle sectors, air force is much more. At the discharging angle, the air force is 10 times more than the eccentric inertia force of the crankshaft. Because of the imbalance impact of the gas force and the inertia force, crankshaft bears a huge dynamic reaction. This always results in an abrasion, and increases the distortion of the crankshaft, and worsens the vibration. As shown in figure 2, some part of the crankshaft has been frayed.

In order to study on the balance system of a rotary compressor, not only the inertia force of eccentricity but also the gas force is taken into account to build a mechanical model of the rotor system. The rotor system is constitutive of a crankshaft (including a roller), a motor rotor, and a pair of balancers, as shown in Figure 3. The mechanical model is used to deduce the relation between the characteristic of balance and rotational angle of the crankshaft. Residual imbalance function of the rotor system is constructed to optimize the parameters of the balancer. Vibration experiment of a compressor is performed for validating the method.

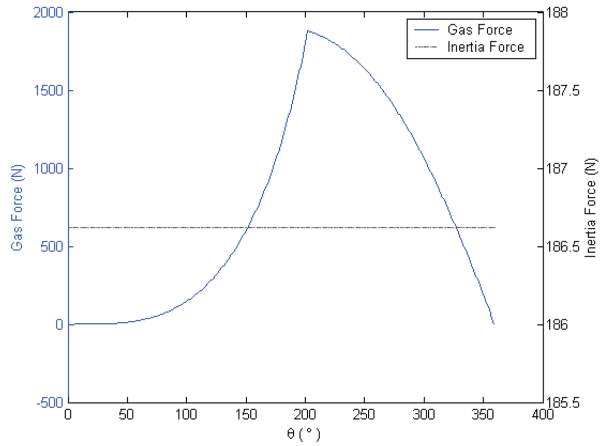

Figure 1 Both Gas Force and Inertia Force of Eccentric Mss
Vary with Rotational Angle

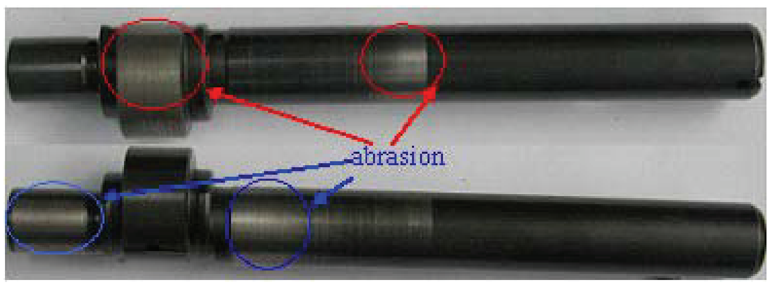

Figure 2 Frayed Part of Crankshaft

2. FORMULATIONS AND DISCUSSIONS

A fixed coordinates system XOY is fixed in the horizontal middle section of the cylinder, and its origin is set at the center of the cylinder. A moving coordinates system X_rOY_r is fixed in the crankshaft which rotates about the axis of the cylinder and its origin is aligned with the origin of the fixed coordinates system XOY.
To analyze the balance issue of rotor system, gas force F_g, inertia force F_t of eccentric mass, inertia force F_1 and F_2 of main and auxiliary balancer are taken into account, and other forces are ignored. The mechanics model is shown in Figure 3. Gas force works through the center of the eccentric part of the crankshaft. When gas force F_g is translated to the origin of the moving coordinates system X_rOY_r, an equivalent forces system, which including force F and torch M, will be obtained.
Assume the compressor angular velocity does not change, according to the mechanics principle, the following equations is built up in the moving coordinates system X_rOY_r:

$$F_g^{'}\sin\frac{\theta+\alpha}{2}+(F_2-F_1)\sin\beta=0 \tag{1}$$

$$F_g^{'}\cos\frac{\theta+\alpha}{2}+(F_2-F_1)\cos\beta+F_t=0 \tag{2}$$

$$F_1L_1=F_2L_2 \tag{3}$$

Where, refer to Figure 3, L1 and L2 are the distances from the mass centre of main and auxiliary balancer to XY plane, respectively. Refer to Figure 4, θ is the rotational angle of the crankshaft; β is an angle made with the axis of Y_r by the position vector for the mass centre of auxiliary balancer referred to the mass centre of main balancer, defined as a fixing angle of balancers; α is an angle made with the midline of slider by the line linking the center of arc of slider to the circle center of crankshaft eccentric part.

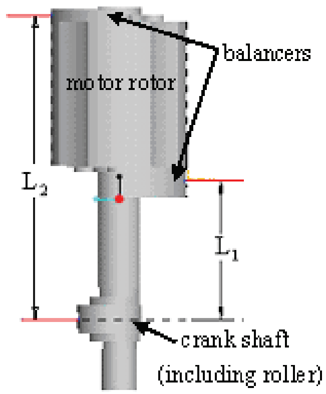

Figure 3 Rotor System

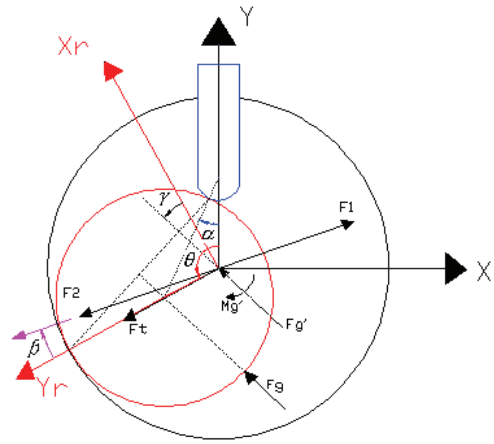

Figure 4 Mechanics Model

The angle α can be expressed as the following equation:

$$\alpha = \arcsin(\frac{\mathrm{e}\sin\theta}{\mathrm{r} + \mathrm{r_v}}) \tag{4}$$

Where, r is the outer radius of roller, and r_v is the radius of slider arc. If both r and r_v are constants, α is a function of θ.

Inertia force of main and auxiliary balancer is expressed as follows:

$$\mathrm{F_1} = m_1 r_1 \omega^2 \overset{def}{=} m_{r1}\omega^2 \tag{5}$$

$$\mathrm{F_2} = m_2 r_2 \omega^2 \overset{def}{=} m_{r2}\omega^2 \tag{6}$$

Where, ω is the angular velocity of the crankshaft in the fixed coordinates system XOY; m_{r1} and m_{r2}, determined by the product of balancer mass m_1 (or m_2) and centroid radius r_1 (or r_2) relative to the rotational axial, are defined as the mass moment of main and auxiliary balancer, respectively.

Combine equation (1) with equation (2), the fixing angle of balancers can be computed as follows:

$$\beta = \begin{cases} \mathrm{arctg}(\beta_0) & \beta_0 \geq 0 \\ \pi + \mathrm{arctg}(\beta_0) & \beta_0 < 0 \end{cases} \tag{7}$$

Where, β_0 is defined as follows:

$$\beta_0 = \frac{\mathrm{F_g'} \cos\frac{\theta + \alpha}{2}}{\mathrm{F_t} + \mathrm{F_g'} \sin\frac{\theta + \alpha}{2}} \tag{8}$$

Equation (7) indicates that the fixing angle of balancers can be determined by the gas force, the inertia force of eccentric mass and the rotational angle. Because the pump system of a compressor is always designed in advance, in terms of stationary angle velocity, the fixing angle of balancers only varies with the rotational angle. As shown in Figure 5, the fixing angle varies between 0°and 180° with the rotational angle of crankshaft when the compressor is at the speed of 60 Hz.
The moment of main and auxiliary balancer mass can be deduced from equation (1) combined with equations (3), (5) and (6) as follows:

$$m_{r1} = \frac{L_2 F_g^{'} \sin\frac{\theta+\alpha}{2}}{\omega^2 (L_2 - L_1)\sin\beta} \tag{9}$$

$$m_{r2} = \frac{L_1 F_g^{'} \sin\frac{\theta+\alpha}{2}}{\omega^2 (L_2 - L_1)\sin\beta} \tag{10}$$

Equations (9) and (10) indicate that both m_{r1} and m_{r2} will vary with the rotational angle of crankshaft if ω is fixed and L1 and L2 almost don't change, as shown in Figure 6. When the rotational angle of crankshaft equals to the discharge angle of the compressor, the mass moment of balancers reaches its maximum.

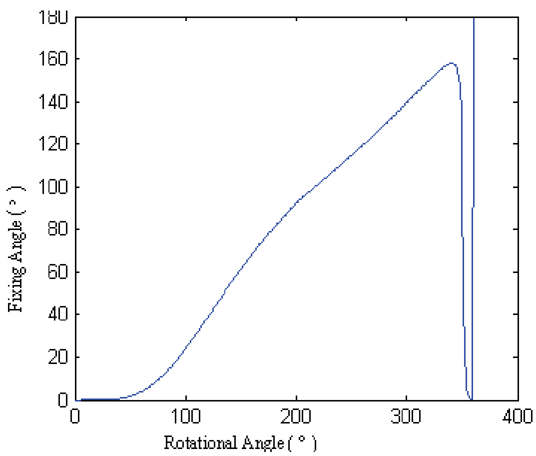

Figure 5 Fixing Angle Varies with Rotational Angle

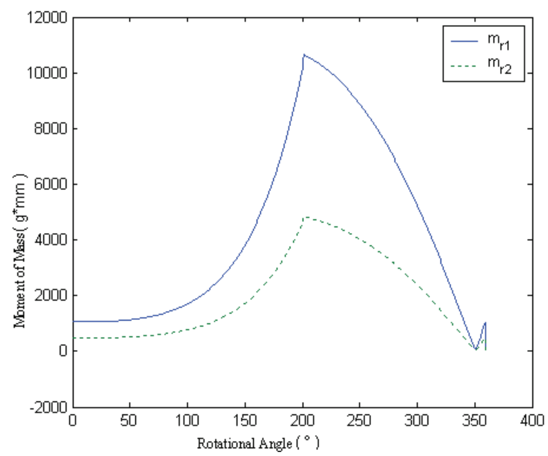

Figure 6 Moment of Mass Varies with Rotational Angle

To ensure the absolute balance of the rotor system, both the fixing angle and the mass moment of balancers vary with the rotational angle of crankshaft. However, In practical product, the fixing angle and the mass moment of the balancers are fixed. To ensure there's a good balance of the rotor system, it's necessary to choose an appropriate fixing angle and moment of mass.
If β and the mass moment of the balancers are fixed, then in moving coordinates system X_rOY_r, the resultant F_{tg} of the gas force F and the eccentric inertia force of crankshaft Ft combines with the inertia force of balancers to be a resultant force F, as shown in Figure 7. Theoretically, to ensure the absolute balance of the rotor system, the resultant force F is zero and the moment of the system of forces is zero also. To evaluate these two items, a residual imbalance function of a rotor system is defined as follows:

$$J = \int_0^{2\pi} \left[(F)^2 L_2^2 + \left[F_{tg} L_2 \cos\eta - F_1 (\mathrm{L}_2 - \mathrm{L}_1) \right]^2 + (F_{tg} L_2 \sin\eta)^2 \right] d\theta \tag{11}$$

Where, η is an angle made by the resultant force F with the position vector for the mass centre of auxiliary balancer referred to the mass centre of main balancer. This angle is a function of the rotational angle θ and the fixing angle β.

In the condition of fixed speed, substitute equations (1), (2) and (3) into equation (8), J can be an object function in terms of β and m_{r1}.

$$J = f(\beta, m_{r1}) \tag{12}$$

Now, the problem of balance system of rotary compressor is looked upon a problem of optimization design. Both a fixing angle and a moment of mass will be found, to minimize the residual imbalance.

Objective: Min J

Design variables: $\beta \in [\beta_{\min}, \beta_{\max}]$

$\mathrm{m_{r1}} \in [\mathrm{m_{r1min}}, \beta_{\mathrm{r1max}}]$

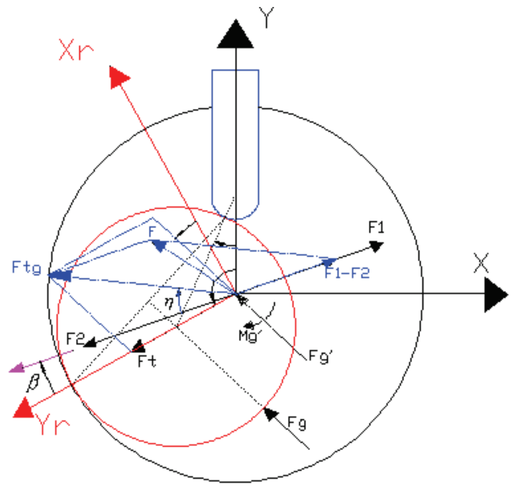

Figure 7 Resultant of Forces

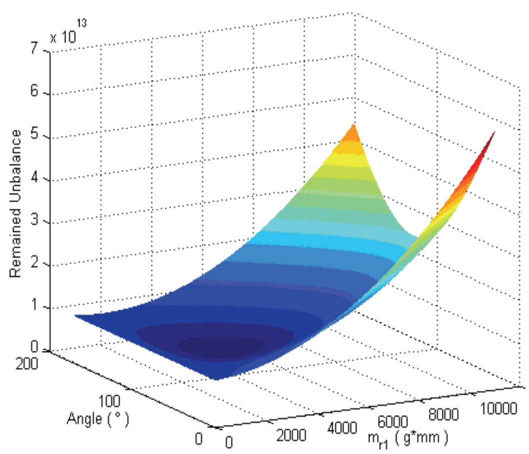

Figure 8 residual imbalance

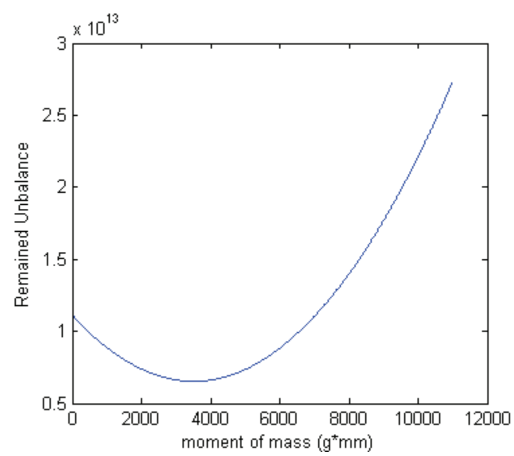

Figure 9 Residual imbalance of fixing angle 107

According to the above analysis, if the fixing angle ranges between 0° and 180° and the mass moment of main balancer ranges between 0 and $11000gmm$, residual imbalance at speed of 60 Hz which varies with both the fixing angle and the mass moment of main balancer is shown in Figure 8. At an arbitrary moment of mass, the residual imbalance at 170° fixing angle is smaller than at other fixing angle. Residual imbalance at 170° fixing angle varying with a mass moment of main balancer is shown in Figure 9. It looks like a parabola. At the fixing angle of 170° and the mass moment of 3531 gmm , residual imbalance of the rotor system is the minimum.

3. EXPERIMENT RESULTS

Consider the space and cost of the actual product, a pair of appropriate balancers is designed. One compressor with 0° fixing angle of balancers is labeled no.1, the other with 107° fixing angle is labeled no.2. A vibration experiment on these compressors is done. Accelerometers are mounted on four test points, as shown in Figure 10. Accelerations in direction x and direction z at each point are measured, as shown in Table 1.
Acceleration of x direction at each point on No 2 compressor shell is less than that of No 1 compressor, and the vibration level of the accumulator of No 2 compressor is less than that of No 1 compressor. The method developed in this paper can improve the tangential vibration of the compressor and the vibration of the accumulator.
On the other hand, Acceleration of z direction at each point on No 2 compressor shell is larger than that of No 1 compressors. The results indicate that the fixing angle worsens the radial vibration of compressor.

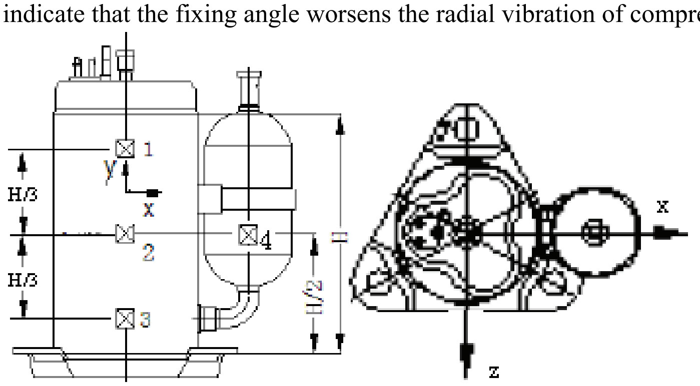

Figure 10 Measurement Locations and Directions

Table 1 test data of acceleration

Compressor	Fixing Angle	Acceleration m/s²							
		1		2		3		4	
		x	z	x	z	x	z	x	z
No 1	0°	11.2	5.51	12.6	4.36	12.5	4.12	24	25.9
No 2	107°	7.27	5.82	3.45	9.97	12.4	15.7	11.5	22.5

3. CONCLUSIONS

A method of considering not only the inertia force but also the gas force is proposed to solve the problem of the balance system of a rotary compressor. The results of the theoretical analysis show:
(1). To ensure the absolute balance of the rotor system, the fixing angle of the balancers varies between 0°and 180° with the rotational angle of crankshaft ranging between 0°and 360°. In addition to, the mass moment of balancers also varies with rotational angle, and it reaches its maximum when the rotational angle reaches the discharge angle of a compressor.
(2).At any moment of mass and any rotary speed, there is an fixing angle. The residual imbalance at this angle is smallest.

Vibration measurement test is carried out to validate the method. The results of this experiment show: The method will be used to improve the tangential vibration of a compressor shell and the vibration of an accumulator, but in the same time it will worsens the radial vibration of compressor shell.
It is necessary to study the problem of balance system of a rotary compressor considering both the gas force and the inertia force further.

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