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Dynamics Analysis of a Rolling Rotor Compressor Considering the Electromagnetic Force

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Abstract: Rolling rotor compressor is a commonly used compressor in air conditioning systems, and studying its internal dynamic characteristics plays an important role in the vibration and noise of air conditioning systems. Due to the integration of the motor rotor and shaft inside the compressor, the influence of electromagnetic force needs to be considered when analyzing the dynamics of the compressor. This paper studies the rotor dynamics characteristics of a rolling rotor compressor, considering not only the bearing force and gas force inside the cylinder, but also the electromagnetic force of the motor rotor. Firstly, the gas force, electromagnetic force, and bearing force on the compressor shaft are analyzed; Secondly, a two degree of freedom rotor bearing dynamic model of the compressor shaft is established, and the dynamic characteristics of the shaft are obtained by solving the model though the Runge-Kutta method; Finally, the vibration characteristics of the compressor shaft are analyzed in both the time and frequency domains. The results indicate that the influence of electromagnetic force cannot be ignored, and the coupling effect between electromagnetic force and bearing force can be seen.

Keywords: Rolling rotor compressor; Electromagnetic force; Dynamic characteristics; Coupling effect.

1. Introduction

With the rapid development of urban modernization, air conditioning systems have been widely used, and people's demand for air conditioning comfort is also increasing. One important indicator is the vibration and noise level of air conditioning. As the core component of the air conditioning system, the vibration of the compressor determines the vibration and noise performance of the entire air conditioning system. Rolling rotor compressors are widely used in air conditioning systems due to the small size, stable performance, and simple structure [1]. Therefore, studying the dynamic characteristics of rolling rotor compressors is important for improving the performance of air conditioning systems.

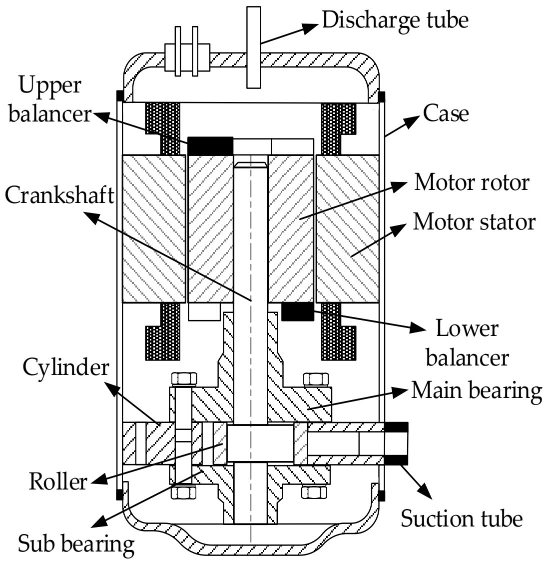

Fig. 1 Structures of a rotary compressor

The simplified structure diagram of a rolling rotor compressor is shown in Figure 1. The rotating shaft in the middle of the compressor and the motor rotor are integrated, and the compressor is driven by the motor. As the shaft rotates, the roller rotates inside the cylinder, while the refrigerant enters the suction chamber of the cylinder through the suction tube. The refrigerant is released from the compression case through the compression of the roller and enters the pipeline through the discharge tube. Due to the compression of gas inside the cylinder, the shaft will be subjected to gas force in the cylinder section, causing the shaft to deviate from the axis position and causing vibration and noise of the compressor. Lubrication, as an important factor in the operation of compressors, plays an important role in the dynamic characteristics of compressors. Some scholars have studied the influence of bearing lubrication on the dynamic characteristics of compressors [2, 3]. In addition, researchers have also studied the influence of relevant parameters on the dynamic characteristics of the compressor shaft using numerical methods through force analysis [4-6].

However, the above related studies have ignored the influence of the motor part. As the rotor and shaft of the motor are integrated, the dynamic behavior of the shaft will inevitably lead to uneven air gap between the rotor and stator of the motor, which will generate electromagnetic force in the motor part [7]. When the air gap between the rotor and stator of the motor is uneven, the effect of electromagnetic force cannot be ignored. In the study of motor vibration, some references have conducted detailed analysis and explanation of the influence of electromagnetic force [8-10]. The main purpose of this paper is to study the dynamic characteristics of the compressor shaft when considering electromagnetic force. By establishing a dynamic model of the shaft, analyzing the forces acting on each component, and using numerical methods to analyze the dynamic characteristics of the shaft, a certain theoretical basis is provided for the overall vibration and noise analysis of the compressor.

2. Force analysis of Crankshaft in Compressor

This paper mainly studies the transverse dynamics of the crankshaft which means only the effect of transverse load is considered. The main loads on the crankshaft are the gas force inside cylinder (Fc), the electromagnetic force on motor rotor (Fump), the oil film force on upper and lower bearings (Fmj, Fsj) and the centrifugal force on unbalanced block and crank (Fe, Fb1, Fb2).

2.1. Gas Force inside Cylinder

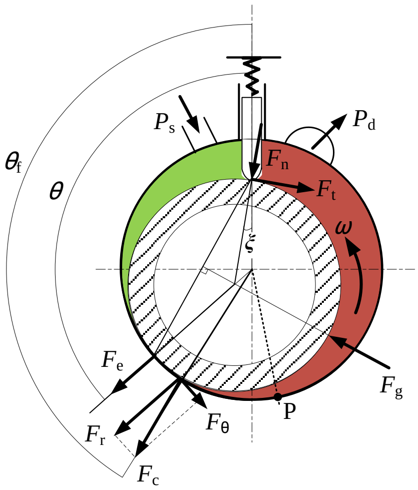

Fig. 2 Force analysis of the internal cross section of the cylinder

The internal cross section of the cylinder is shown in Figure 2. The main forces inside the cylinder are: contact force at the tip of the slide (Fn, Ft), gas compression force (Fg) and centrifugal force of the crank (Fe), where the contact force at the tip of the slide can be calculated by performing a force analysis on the slide. The gas force can be expressed as [5]:

$$F_g = 2R_{ro}\sin\left(\frac{\theta+\xi}{2}\right)H_c(P_c - P_s) \tag{1}$$

$$F_r = F_g\cos\left(\frac{\theta+\xi}{2}\right) + F_e - F_n\cos(\theta+\xi) + F_t\sin(\theta+\xi) \tag{2}$$

$$F_\theta = -F_g\sin\left(\frac{\theta+\xi}{2}\right) + F_n\sin(\theta+\xi) - F_t\cos(\theta+\xi) \tag{3}$$

$$F_c = \sqrt{F_r^{\,2} + F_\theta^{\,2}} \tag{4}$$

$$\theta_f = \mathrm{tg}^{-1}\frac{F_\theta}{F_r} + \theta \tag{5}$$

where Ps denotes the pressure in the suction chamber, Pc denotes the pressure in the compression chamber, and Pd denotes the pressure at the exhaust port during exhaust. It can be seen that the gas compression force mainly depends on the pressure inside the cylinder. According to the working principle of the rotor compressor, the pressure change inside the cylinder can be expressed as:

$$P_c = \begin{cases} P_s & 0 \le \theta < \theta_s \\ P_s\left(\dfrac{V_s}{V_c}\right)^n & \theta_s \le \theta < \theta_p \\ P_d + (P_{d0} - P_d)\left(\dfrac{\theta_d - \theta}{\theta_d - \theta_p}\right) & \theta_p \le \theta < \theta_d \\ P_s + (P_d - P_s)\left(\dfrac{2\pi - \theta}{2\pi - \theta_d}\right) & \theta_d \le \theta < 2\pi \end{cases} \tag{6}$$

where Vs denotes the volume of the suction chamber, Vc denotes the volume of the compression chamber, and n is the adiabatic compression index associated with the refrigerant (1.194 in this paper). Pd0 denotes the maximum pressure at which the exhaust valve piece opens, which is usually 0.06 MPa greater than the exhaust pressure Pd. θs denotes the angle of the suction port, θp denotes the angle at which the exhaust valve opens, and θd denotes the angle of the exhaust port.

2.2. Electromagnetic Force on Motor Rotor

The type of motor used in rolling rotor compressor is permanent magnet synchronous motor, there is a certain air gap between the motor rotor and motor stator. According to Maxwell's electromagnetic induction principle, the motor rotor will generate electromagnetic stress during rotation by the influence of electromagnetic field. When the air gap distribution between the rotor and stator is uniform, the electromagnetic stress on the rotor surface will balance each other. However, when the air gap distribution is not uniform, the distribution of electromagnetic stress is also not uniform, which will generate an electromagnetic combined force that is electromagnetic force. This electromagnetic force can be derived by integrating the electromagnetic stress on the rotor surface, expressed as follows:

$$\begin{cases} F_{UMPx} = R_r L\displaystyle\int_0^{2\pi}\sigma_r(\alpha,t)\cos\alpha \, d\alpha \\ F_{UMPy} = R_r L\displaystyle\int_0^{2\pi}\sigma_r(\alpha,t)\sin\alpha \, d\alpha \end{cases} \tag{7}$$

where σr denotes the electromagnetic stress on the surface of the motor rotor, which can be expressed by Maxwell's stress tensor method as:

$$\sigma_r(\alpha,t) = \frac{B_r^2 - B_t^2}{2\mu_0} \approx \frac{B_r^2}{2\mu_0} \tag{8}$$

where Br denotes the radial component of the air-gap magnetic field and Bt denotes the circumferential component of the air-gap magnetic field, and the study shows that the air-gap magnetic field depends mainly on the radial component Br, so its circumferential component Bt can be neglected. When the air gap distribution between the rotor and the stator is uniform, the radial component of the air gap magnetic field can be expressed as [7]:

$$B_r(\alpha,t) = \lambda(\alpha)\cdot\left[\sum_{n=1,3,5...}^{\infty} B_{mr}\cos(np\alpha - n\omega_e t) + \sum_{v=1,2,3...}^{\infty} B_{ar}\cos(pv\alpha \mp \omega_e t)\right] \tag{9}$$

where λ(α) denotes the relative air gap permeability, and Bmr and Bar denote the amplitude of the magnetic field generated by the permanent magnet and armature winding, respectively.

Since the crankshaft and the motor rotor are integrated, the vibration of the crankshaft will lead to uneven air gap distribution, and the air gap length can be expressed according to the geometric relationship as:

$$\delta(\alpha,t)\approx R_s-R_r-e\cos(\alpha-\theta)=\delta_0-e\cos(\alpha-\theta) \tag{10}$$

In order to obtain the air gap magnetic field when the air gap is non-uniform, a correction factor needs to be introduced to correct the air gap magnetic field. Since the relative permeability of the air gap and the air gap length are inversely proportional, the correction factor can be expressed as:

$$\varepsilon=\frac{\delta_0}{\delta(\alpha,t)}=\frac{1}{1-\varepsilon_0\cos(\alpha-\theta)} \tag{11}$$

where ε0=e/δ0 indicates the relative eccentricity.

By introducing a correction factor, the air gap magnetic field can be expressed as:

$$B_r(\alpha,t)=\lambda(\alpha)\cdot\left[\sum_{n=1,3,5\ldots}^{\infty}B_{mr}\cos(np\alpha-n\omega_e t)+\sum_{v=1,2,3\ldots}^{\infty}B_{ar}\cos(pv\alpha\mp\omega_e t)\right]\cdot\varepsilon \tag{12}$$

2.3. Non-linear Oil Film Force on Bearings （Fmj, Fsj）

The main and lower bearings of the compressor are sliding bearings, and the oil film pressure can be obtained by solving the Reynolds equation:

$$\frac{\partial}{R_j^2\partial\theta_j}\left(\frac{h^3}{\mu}\frac{\partial P_j}{\partial\theta_j}\right)+\frac{1}{\eta^2}\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial P_j}{\partial z}\right)=6\omega\frac{\partial h}{\partial\theta_j}+12\frac{\partial h}{\partial t} \tag{13}$$

$$h=C(1+\varepsilon_j\cos\theta_j) \tag{14}$$

Many studies have been conducted on the oil film forces of sliding bearings. In this artcile, a finite difference numerical method is used for solving the oil film forces, and a database of oil film forces is established to facilitate the solution of the subsequent kinetic equations.

3. Solving and Analyzing of Crankshaft dynamics model

The main study in this paper is the transverse vibration of the crankshaft, which means only two degrees of freedom in the transverse direction are considered, and the dynamics equation of the crankshaft can be expressed as follows:

$$\begin{aligned}M\ddot{x}&=F_{umpx}+F_{cx}+F_{ex}+F_{b1x}-F_{mjx}-F_{sjx}-F_{b2x}\\ M\ddot{y}&=F_{umpy}+F_{cy}+F_{ey}+F_{b1y}-F_{mjy}-F_{sjy}-F_{b2y}\end{aligned} \tag{15}$$

where M denotes the crankshaft mass and the centrifugal force can be expressed as:

$$F_e=Me_0\omega^2,\quad F_{b1}=m_{b1}e_{b1}\omega^2,\quad F_{b2}=m_{b2}e_{b2}\omega^2 \tag{16}$$

e0, eb1 and eb2 indicate the eccentricity of the crankshaft, upper and lower balancers, respectively.

Considering the nonlinearity of electromagnetic force and bearing oil film force, this paper uses MATLAB to solve the kinetic equations by the fourth-order Runge-Kutta numerical method, and the relevant parameters of the compressor are shown in Table 1. The axial trajectory of the crankshaft is shown in Figure 3, which shows that the axial trajectory is not a line, but also contains many small ripples, which is mainly due to the harmonic effect of the electromagnetic field affecting the electromagnetic force and even the axial displacement of the crankshaft. When the shaft rotates from the suction port θs to the exhaust valve opening angle θp, the radial displacement of the shaft increases obviously, this process belongs to the gas compression process, the gas force increases quickly thus causing the rapid offset of the shaft displacement. When the shaft rotates from θp to θd, the compression chamber starts to exhaust, the effect of the gas force starts to weaken and the radial displacement of the shaft starts to decrease. As the crankshaft continues to rotate, the gas in the compression chamber is slowly discharged and gradually connected to the suction chamber to start preparing for the next cycle. The gas pressure in this process decreases abruptly, and the radial displacement of the crankshaft is also reduced a lot. The change of the force on the crankshaft is shown in Figure 4. It can be seen that the force changes periodically, and the electromagnetic force and the oil film force of the main bearing are obviously larger than other forces, which indicates that the role of electromagnetic force is not negligible when analyzing the vibration of the compressor, and it can be seen that there are many small peaks of electromagnetic force and oil film force, which are generated by the harmonic effect of electromagnetic force. In order to better analyze the vibration characteristics of the rotating shaft, a spectral analysis of the radial displacement of the shaft center is done, as shown in Figure5, which also indicates the coupling effect between the electromagnetic force and the oil film force (main bearing and lower bearing).

Table 1. Compressor related parameters

Parameter	Value	Parameter	Value
Shaft mass (M)	1.24 (kg)	Main bearing length	50 (mm)
Bearing clearance (c)	20 (μm)	Lower bearing radius	7 (mm)
Main bearing radius	8 (mm)	Lower bearing length	17.3 (mm)
Suction pressure (Ps)	0.916 (MPa)	Inner radius of cylinder	19.5 (mm)
Discharge pressure (Pd)	2.588 (MPa)	Inner radius of roller	12 (mm)
Shaft speed (ω)	60 (Hz)	Cylinder thickness (Hc)	18 (mm)

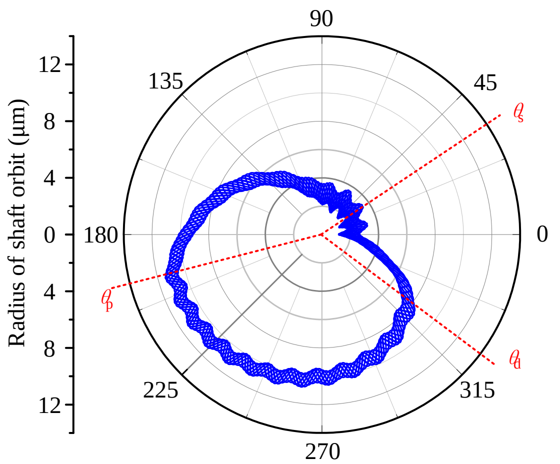

Fig. 3 Axis trajectory of the crankshaft

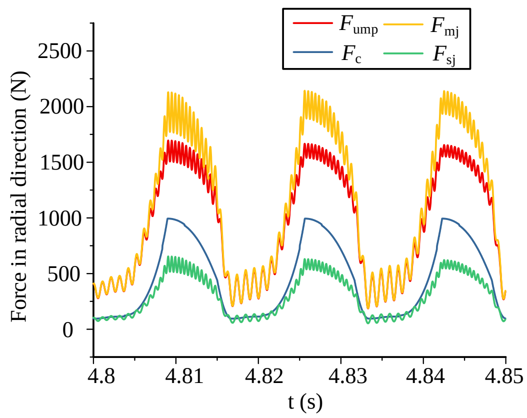

Fig. 4 Radial component of the force on the crankshaft

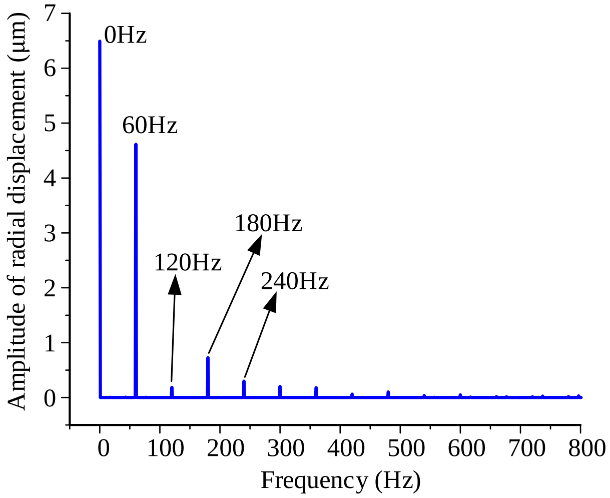

Fig. 5 Spectrogram of axial radial displacement

4. Summary

In this paper, the dynamics of the crankshaft of a rolling rotor compressor is studied by taking into account both electromagnetic and bearing forces. Firstly, the loads on crankshaft are analyzed in detail, and then the established crankshaft dynamics model is solved and analyzed by numerical methods, and the transverse vibration of the crankshaft is discussed in detail on the time and frequency domains, and the main conclusions are as follows:

(1) Among the loads on the crankshaft of the rolling rotor compressor, the electromagnetic force is not negligible and plays a major role in the dynamic characteristics of the crankshaft, while the harmonic effect of the electromagnetic field will be reflected not only in the electromagnetic force and bearing force, but also in the axial displacement of the crankshaft;

(2) The frequency component of the radial displacement of the crankshaft is mainly an integer multiple of the rotating frequency, and 0Hz and single frequency are the most dominant frequency components. The amplitude of triplet frequency will be higher than that of double frequency, reflecting the coupling effect of electromagnetic force and bearing force;

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