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The anti-shock performance of fuel cell air compressor supported by water lubricated bearing

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ABSTRACT – This paper presents a theoretical study on the transient characteristics of fuel cell air compressor supported by water lubricated journal bearings. The nonlinear trajectory of the compressor rotor bearing system under a half sinusoidal shock load is numerically calculated by simultaneously solving the rotor motion equations and the Reynolds equation. In order to take the possible contact between the shaft and sleeve into consideration, the pressure-compliance relationship of roughness contact is also introduced. The results show that the anti-shock performance of the compressor satisfies the requirement of vehicle application.

1. INTRODUCTION

As the power source of the fuel cell vehicles, the fuel cell system needs an air compressor to supply pressured oxygen for the stack. Besides the basic requirement on the air pressure and flow rate, the compressor should also qualify for the standards of vehicle application. The anti-shock performance is one of most important ones.

Due to the complex of road surface and driving manners, transient shocks from moving car are inevitable. Thus the rotor bearing system of the compressor should sustain certain load, otherwise direct collision between the spinning shaft and sleeve will occur and results in the damage of the compressor. In the authors' previous studies [1-3], a fuel cell air compressor using water lubricated bearing is developed and the anti-shock performance of the bearing were theoretically investigated. The bearing model and theories are adopted here to analyse the transient characteristics of the rotor bearing system.

2. MODELS AND EQUATIONS

To improve the rotor dynamic performance, the rotor bearing system is specially designed to make sure that the first critical speed is higher than the operating speed and substantial safety margin exits. So the rotor can be regarded as a rigid one in the analysis. The structure of the rotor is shown in Figure 1. The rotor is supported by two water lubricated journal bearings on each side of the permanent magnet synchronous motor. The impeller locates at one end and the water lubricated thrust bearings at the other. The configuration of the water lubricated bearing is shown in Figure 2. The bearing has two pockets, which is defined by the Archimedes helix with the starting point locating at the pocket downstream. The lubricating waters is supplied through the orifices at the pocket upstream. The parameters of the rotor bearing system are listed in Table 1

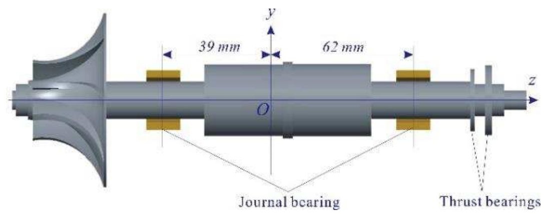


Figure 1 Rotor structure.

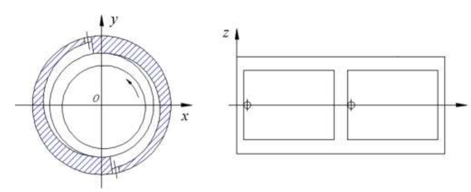


Figure 2 Bearing configuration.

Table 1 Parameters of the rotor bearing system

Properties	Data
Mass of rotor m /kg	0.58
Eccentricity of rotor mass e_u /m	0.5×10^{-6}
Inertia moment I_z / kg·m^2	6.25×10^{-5}
Inertia moment I_x, I_y / kg·m^2	1.40×10^{-3}

The anti-shock performance of the rotor bearing system is studied by numerically calculation of the non-linear trajectory. In the Cartesian coordinate system O-XYZ with its origin point at the rotor centroid, the motion equations of the rigid rotor can be expressed as

$$\begin{cases} m\ddot{x} = \sum F_x + W\sin\theta_S + me_u\omega^2\cos\theta \\ m\ddot{y} = \sum F_y - W\cos\theta_S + me_u\omega^2\sin\theta - mG \\ I_x\ddot{\Theta}_x + \dot{\Theta}_y I_z\omega = \sum M_x \\ I_y\ddot{\Theta}_y + \dot{\Theta}_x I_z\omega = \sum M_y \end{cases} \quad (1)$$

Where ω is the angular speed, F_x and F_y are the reaction forces of the bearings, M_x and M_y are the friction torques of the bearings. The shock load W is assumed as a half sinusoidal wave impulsive force with millisecond order period acting on the rotor centroid:

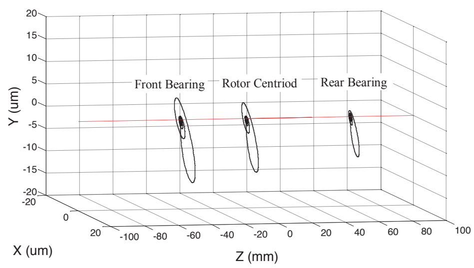
$$W = \begin{cases} mG_S\sin(\pi t/T_S) & 0 \le t \le T_s \\ 0 & t > T_s \end{cases} \quad (2)$$

Where G_S, T_S and θ_S are the amplitude, time and angle of the shock load, respectively.

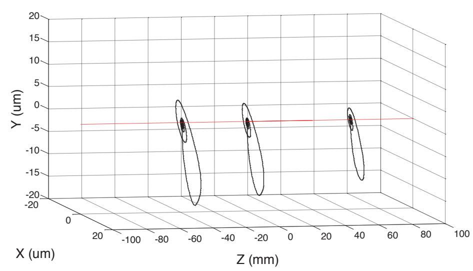
In the calculation of the bearing reaction forces and friction torques, the Reynolds equation with considering turbulence is solved. Meanwhile, in order to consider the possible direct contact between the shaft and sleeve during shock, the pressure-compliance relationship of roughness contact is also introduced. The theories and equations in [3] are adopted here.

3. RESULTS AND DISCUSSION

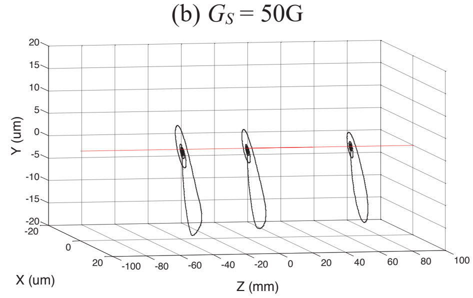
Figure 3 shows the nonlinear trajectories of front bearing, rotor centroid and rear bearing under different shock amplitude. The shock time T_S and angle θ_S are equal to 1ms and 36°, respectively. Due to the different distance from the two bearings to the rotor centroid, the rotor bearing system shows a conical trajectory. Because the front bearing is much closer to the rotor centroid, it suffers more load and its trajectory is larger. When the shock amplitude exceeds the bearing capacity, the roughness contact between the shaft and sleeve occurs at the front bearing first.



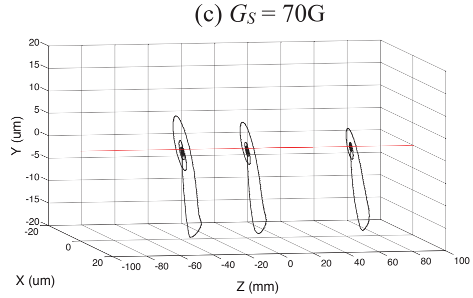
(a) $G_S = 30G$



(b) $G_S = 50G$

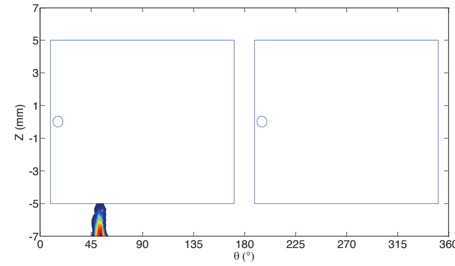


(c) $G_S = 70G$

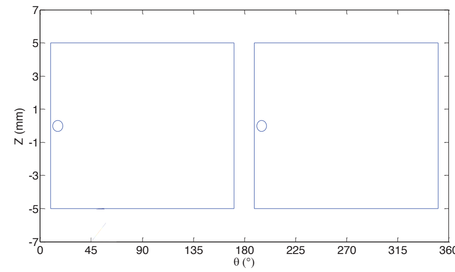


(d) $G_S = 100G$

Figure 3 Nonlinear trajectories of the rotor bearing system.



(a) Front bearing



(b) Rear bearing

Figure 4 Distribution of the roughness contact (G_S=100G).

Figure 4 shows the distribution of the roughness contact of the two bearings when the shock load is 100G. It can been seen that the roughness contact only occurs at the sealing land of the front bearing, which validates the above analysis.

It should be noted that the rotor bearing system can sustain about 100G shock load, the transient performance of the compressor can satisfy the requirement of vehicle application. To further improve the compressor anti-shock performance, the useful conclusions in the previous study [3] can be adopted in the design of front bearing.

4. SUMMARY

The anti-shock performance of the rotor bearing system of the air compressor is numerically studied in this paper. The results show that the rotor bearing system satisfies the requirement of vehicle application. The performance can be further improved by enhancing the front bearing.

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