Research on Vibration Characteristics of Magnetic Suspension Rotor Based on Magnetorheological Damper

Nan Lv [a], Zhao Zhang [b], Zhaojing Su [c]

School of Mechanical and Electronic Engineering, Shandong University of Science and Technology, Qingdao 266590, China

[a]lvnan_xcln@163.com, [b]zhangz5708@126.com, [c]zhaojing_idqd@163.com

Abstract

As for the damping controllability of MRF, structure design and research based on the MRF are made for the magnetic suspension rotor damper in this paper. According to the working principle of MRF damper, damper mechanical design is completed and the mathematical model of magnetic suspension rotor is established. Modal analysis of the rotor is simulated by the means of ANSYS and numerical simulations of amplitude of the rotor is made by the means of MATLAB. The results show that the MRF damper has a good damping effect, and can avoid the rotor resonance appropriately.

Keywords

MRF; magnetic suspension; Finite element method, rotor-bearing system.

1. Introduction

Magnetorheological (MR) fluid is a kind of new intelligent material and its properties of magnetics, mechanics, thermal and optics can be changed under the action of an external magnetic field, and it can achieve the reversible transformation between solid and liquid state(MR effect). Therefore, there will be great application prospect in engineering applications. In this paper, based on the characteristics of the MR fluid, the damping characteristics of the MR damper is studied. We took the magnetic suspension rotor as our research subject, and to achieve the most effective effect of vibration reduction by the means of theoretical analysis and numerical simulation.

2. The working principle of MR damper

Aiming at poor control ability of magnetic suspension rotor on vibration suppression, a kind of MR damper was designed to support the magnetic bearing system. The process of transformation between solid and liquid state inside the MR damper is shown in fig. 1, when there is speed difference between active and fixed parts, the rotating of active parts will be restrained under the action of shear yield stress in the MR effect, and the magnetic field disappears, MR fluid remain Newtonian fluid, at this point, only tiny liquid viscous force remains between the active and fixed parts which can be ignored, so as to ensure the shaft to rotate freely.

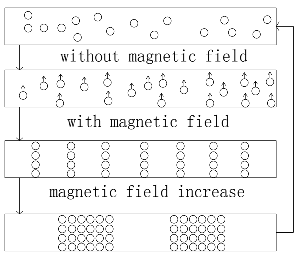


Fig. 1 Diagram of MR effect

2.1 Magnetic circuit of MR damper

Both of MR damper and magnetic suspension rotor system work by magnetic force, but the two magnetic field are independent of each other. So in magnetic circuit design, much attention should be paid to the two magnetic field coupling problems, and should make the magnetic field through the MR fluid to the greatest extent in order to guarantee the performance of MR fluid [1]. Current change in the MR damper coil can change the stiffness and damping of the whole support system, so the vibration of high-speed rotor can be suppressed by the MR damper.

Magnetic field of magnetic bearing and MR damper are both generated by their own control coil respectively, in order to guarantee the independence of the magnetic field, we place two sets of coil vertical and keep the coil on both sides symmetrical. In this way the magnetic field of the magnetic bearing would not be affected by the magnetic field generated by the control coil of MR damper. Considering the coupling can't entirely be avoided, we make current value in MR damper area zero during the analysis of the magnetic field in the magnetic suspension parts.

2.2 Structure design of MR damper

When the magnetic field distance between the two parts of MR damper and magnetic bearing is small, coupling phenomenon may appear among two magnetic fields. Therefore the magnetic field distance should be designed to be large enough directly. And in order to guarantee the damping force provided by MR effect is enough, the magnetic resistance should be small by reducing the thickness or increasing the cross-sectional area of the working MR fluid in MR damper. But if the thickness is too thin , not only will reduce its damping range, but also increase the cost of production. Generally take the thickness for 0.2 ~1 mm [2].

In dampers, the MR fluid,roller ,shaft and protection cover form a closed magnetic circuit, as shown like the dotted line in fig. 2. When there is a current conduction, the closed loop will produce a magnetic field, and the magnetic flux in any closed surface identically equal to zero.

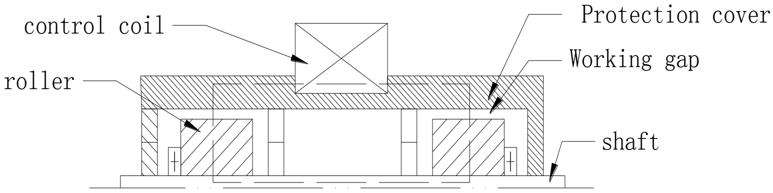


Fig. 2 Circuit structure of the damper

As shown in figure 3, damper structure include: mandrel, outer shaft, roller , protection cover, support frame, end cover ,etc. In order to ensure that the strength of the magnetic field and stability of the structure, we choose pure iron as permeability magnetic material.

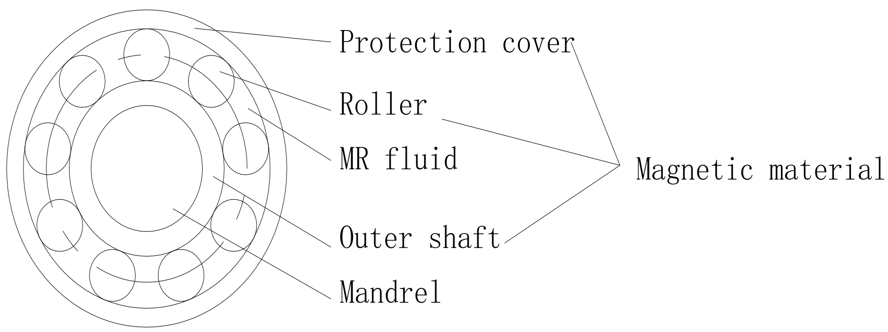


Fig. 3 Structure diagram of working part in MR damper

The structure design of working part in MR damper is a key part of the overall design work. In order to increase torque between the shaft and the protection cover transmitted through the MR fluid transmission torque to control the damping coefficient, the gap between the outer shaft and the protection cover should be full of MR fluid. And to avoid the occurrence of the phenomenon such as uneven ,also to make a great extent of MR effect , the application of the roller supported by support frame between the outer shaft and protection cover is taken in our design.

To make sure the MR fluid have enough work space the distance from the roller to the shaft and to the protection cover are both 0.5mm.[] The outer shaft and the protection cover play a role as two magnetic plate. When the control coil fixed at the protection cover is switched on, the coil can generate magnetic field which will cause MR effect, then by the Shearing action of MR fluid , the damping coefficient increase, so that the rotation of the shaft will be restrained.

3. Vibration characteristic analysis

3.1 Theoretical Model

The rotor system was supported by magnetic bearings and MR damper and an analytical model for a rotor system was developed as shown in fig. 4.

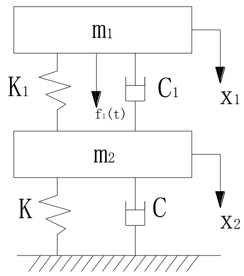


Fig. 4 Simplified model

The equivalent modal properties obtained for the rotor and magnetic bearing mass(m1),MR damper(m2), excited force produced by the rotor when passing through the critical speed(f1 (t)), the rotor displacement (x1), magnetic bearing displacemen t (x2).And the generalized equation of motion for a rotor system has the following form:

$$\begin{pmatrix} m_1 & 0 \\ 0 & m_2 \end{pmatrix}\begin{pmatrix} x_1 \\ x_2 \end{pmatrix} + \begin{pmatrix} c_1 & -c_1 \\ -c_1 & c+c_1 \end{pmatrix}\begin{pmatrix} x_1 \\ x_2 \end{pmatrix} + \begin{pmatrix} k_1 & -k_1 \\ -k_1 & k+k_1 \end{pmatrix}\begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{pmatrix} f_1(t) \\ 0 \end{pmatrix}$$

The rotor vibration amplitude and frequency response function can be expressed as follows:

$$|H'(\omega)| = \frac{\sqrt{(k+k_1-\omega^2 m_2)^2 + (c+c_1)^2\omega^2}}{|\Delta(\omega)|}$$

Where ω is critical frequency of the rotor when passing through the critical speed .

$$\Delta(\omega) = \det\begin{bmatrix} k_1 + j\omega c_1 - \omega^2 m_1 & -k_1 - j\omega c_1 \\ -k_1 - j\omega c_1 & k_1 + k + j\omega(c+c_1) - \omega^2 m_2 \end{bmatrix}$$

The equation of motion for the rotor system in the static coordinates can be expressed as follows:

$$[M]\{\ddot{u}\} + \left([C] + [C_{gry}]\right)\{\dot{u}\} + [K]\{u\} = \{F\}$$

Where C_{gry} is gyroscopic effect matrix .

So the equation of motions for the rotor system in a rotating coordinate system can be expressed as follows：

$$[M]\{\dot{u}_1\} + ([C] + [C_{cor}])\{\dot{u}_1\} + ([K] - [K_{spin}])\{u_1\} = \{F\}$$

Where $C_{cor}$is Coriolis effect matrix, K_{spin} is stiffness matrix for spinning effect matrix [5].

3.2 Simulation analysis

The amplitude of the rotor and has a direct relationship with equivalent stiffness and equivalent damping .So that we can effectively restrain the vibration of the rotor by a reasonable parameter.In order to determine the stiffness and damping of the MR damper the rotor when passing through the critical speed, the finite element software ANSYS WORKBENCH is adopted to calculate and analyze the vibration situation of the system.

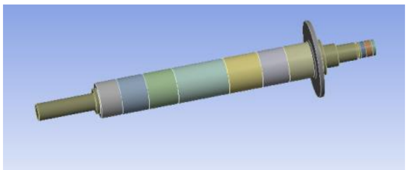


Fig. 5 Simulation model

Firstly the critical speed of the rotor without MR damper is calculated, and the parameters of MR damper are not taken in account. And the rest of the parameters are consistent as shown in tab. 1.The simulation result is shown in tab.2.

Tab. 1 parameters of the rotor system

Shaft material	40Cr
C(magnetic bearing)	7.056 N s/m
K(magnetic bearing)	87700 N/m
m	1.0807kg

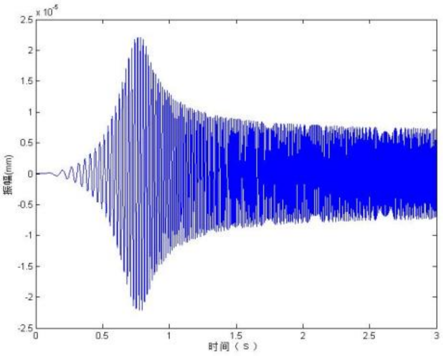
On the analysis of the influence of different stiffness of the MR damper on the critical speed, the parameters of magnetic bearing remain the same. Keep the MR damper c_1 = 100 N s/m and change the stiffness value, the simulation results are shown in table 2. Then keep the MR damper $k_1=10^5$ N/m and change the damping value, the simulation results are shown in table 2.

Tab.2 Changes of the first three order critical speed

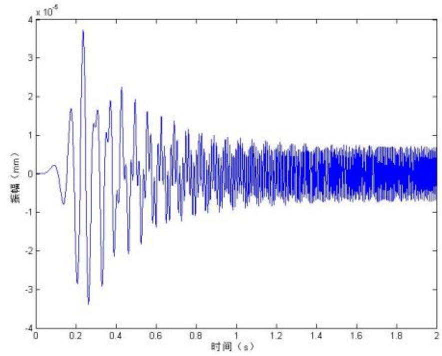
system state parameters		First order critical speed(r/min)	Second order critical speed(r/min)	Third order critical speed(r/min)
Without MR damper		131.35	400.43	697.26
MR damper stiffness (N/m)	10^3	128.65	388.36	691.32
	10^4	124.21	384.84	687.32
	10^5	121.75	379.95	685.89
	10^6	122.2	380.69	686.09
MR damper damping (N s/m)	100	131.70	402.29	696.99
	200	133.22	400.89	698.87
	300	133.09	404.38	699.93
	500	155.98	407.06	672.74

By comparison of the first three order critical speed in different working state, it is found that the along with the increase of the MRF damper stiffness, the critical speed decreases. But when the stiffness value increase to a certain extent, its influence on rotor critical speed is relatively small. And relatively, the change of the damping have little effect on reducing the critical speed which could be ignored.

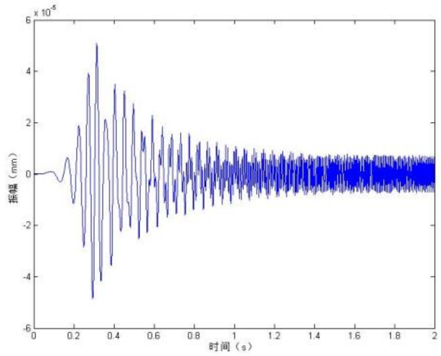
The parameters of magnetic suspension rotor system remain unchanged, and take different stiffness and damping of MR damper to simulate the vibration amplitude of the rotor by MATLAB. And the simulation results are as shown in fig. 6.It can be seen from the graph that with the increase of the damper stiffness, the amplitude of the rotor increases gradually. In addition, with the increase of stiffness, the critical speed of the rotor also increases, and its change is pretty larger. Therefore, when analyzing transient response of rotor system, the stiffness could be changed to adjust the critical speed of the rotor, and to reduce the amplitude of the rotor near the critical speed to passing through the critical speed smoothly.



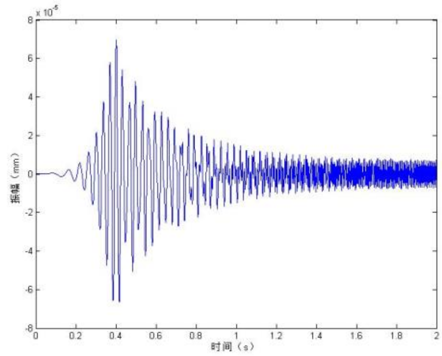
(a)k=10N/m



(b)k=20N/m



(c)k=50N/m



(d)k=100N/m

Fig. 6 Simulation result

4. Conclusion

Based on magnetorheological technology and magnetic suspension technology, the mechanical structure of magnetic suspension rotor based on MR Damper was designed in this paper, and a theoretical model is developed, and with the simulation results made by ANSYS WORKBENCH and MATLAB, the feasibility of MR damper are verified. By making adjustment to the MR damper of its stiffness value and damping value within a certain scope, along with the increase of stiffness value, the maximum amplitude of the rotor is reduced and the critical speed decreased but the change of the damping value effect on critical speed is not big which can be ignored. On rotor vibration control, increasing the damping of MR damper could help to reduce the amplitude of the rotor near the critical speed. But when damping value increases to a certain range, the amplitude decreasing of the rotor gradually slowed down. The analysis above indicates that the structure developed in the paper can get good controllable damping effect, and can help the rotor to passing through the critical speed smoothly.

References

[1] H.K. Roya; A.S. Dasb1;J.K. Dutta.An efficient rotor suspension with active magnetic bearings having viscoelastic control law[J].Mechanism and Machine Theory.2016.

[2] K. T. Millisaps, G.L.Reducing Lateral Vibrations of a Rotor Passing Through Critical Speeds by Acceleration Scheduling[J].Journal of Engineering for Gas Turbines and Power, 1998, 120: 615 -620.

[3]Wang, J.1,2;Meng, G.1.Study of the Vibration Control of a Rotor System Using a Magnetorheological Fluid Damper [J]. Journal of Vibration and Control.2005(NO.2).

[4]Forte, P.1;Paternò, M.1;Rustighi, E.1.A Magnetorheological Fluid Damper for Rotor Applications[J].International Journal of Rotating Machinery.2004(NO.3).

[5] Zheng Xianda. Study on magnetic suspension damping system [D]. Zhejiang university of technology, 2010.

[6] Feihong Xu, Zhao-dong Xu. Multistage magnetorheological damper coil magnetic circuit analysis [J]. Journal of southeast university (natural science edition), 2016,01:100-104.
[7] Jaroslav Zapomĕl, Petr Ferfecki. A computational investigation on the reducing lateral vibration of rotors with rolling-element bearings passing through critical speeds by means of tuning the stiffness of the system supports[J].Mechanism and Machine Theory, 2011,(46):707-724.
[8] Grzegorz Cieplok, Przemysław Pyzik. Reducing lateral vibration of a rotor passing through critical speed by proper driving torque determination[J]. Active Noise and Vibration Control Methods,2011:151-166.
[9] GuiRong Teng,Zhu Xuli, Lv Nan. A magnetorheological elastomer damper based on the tension and compression mode [P]. CHINA: CN104534010A, 2015-04-22.
[10] Hong-chang Ding, Nan Lv. Protecting structure of a magnetic rotor of high-speed motor [P]. CHINA: CN204334214U, 2015-05-13.
[11] Nan Lv, Xubo Gong, Chuanping Wang, Yamin Fan, Zhaojing Su, Luwei Liu. A high-speed magnetic rotor based on magnetorheological damper [P]. CHINA: CN204794531U, 2015-11-18.