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Letter to the Editor

Experimental study on stability of an MR fluid damper-rotor-journal bearing system

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1. Introduction

Magnetorheological (MR) as well as electrorheological (ER) fluids are materials that respond to an applied electric or magnetic field with a dramatic change in rheological behaviour [1]. ER/MR fluid dampers are new type vibration control elements, having the advantage of rapid damping and stiffness changing in the presence of an applied electric or magnetic field. Typical MR fluid has the advantages of higher yield stress (up to 50–100 kPa, which is one order higher than ER fluid), insensitive to general contaminants, using only 12–24 V low voltage, relative broad working temperature range (typically $-40°C$ to $150°C$) [2], so MR fluid damper has the characteristics of large damping force, low-power consumption, easy to control, etc. Many researchers have studied the vibration controllability by ER/MR dampers and some of them were focused on the vibration control of rotor systems by ER fluid dampers or ER squeeze film dampers [3–8]. It is found that the ER damper can reduce high levels of unbalance-excited vibrations and can also shift the critical speed. As MR damper can provide larger damping and stiffness change, the rotor vibration could be better controlled if MR damper is used [9]. Since MR fluid has high yield stress and may solidify at strong magnetic field, the MR dampers in a rotor system may become quasi-rigid, which will increase notably the support stiffness of the rotor system and may cause the system losing stability. As the stability of MR damper is important for the effective and safe use of the damper and almost no literatures till now focus on the stability analysis of this kind of rotor system. This paper will present the experimental results on kinematical stability of a cantilever rotor system supported on an MR fluid damper and a journal bearing.

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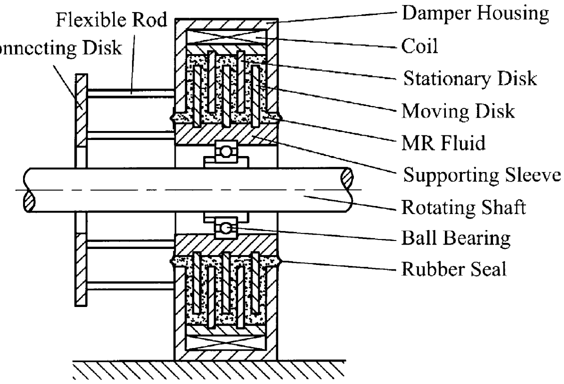


Fig. 1. Cross-section of MR fluid damper.

2. Structure of MR fluid damper and experimental arrangement

The MR fluid damper, shown in Fig. 1, has three moving disks and two stationary disks. The disks are placed uniformly and alternatively with a uniform gap of 1.5 mm forming six relative shear surfaces. Electric current is input to the coil to generate magnetic field, and the damper housing is used as a magnetizer. The flexible rod-type support consisting of the flexible rods, the connecting disk and the supporting sleeve is used to provide a centring flexible support for the bearing. One kind of homemade MR fluid was used in the experimental study. The magnetic particles used are carbonyl iron particles with average diameter of 1 μm. The carrier liquid used is silicone oil with kinematical viscosity of $1 \times 10^{-4}\,m/s^2$. The MR fluid has a particle volume fraction of 35%, and the yield sheer stress is about 20 kPa when the magnetic strength is 240 kA/m.

The experimental rotor system is shown in Fig. 2. The shaft is 9.5 mm in diameter and 500 mm in length. The shaft is driven by a DC motor and the rotating speed can be adjusted continually from 0 to 10 000 r.p.m. by a speed regulator. The measuring equipment consists of a photoelectrical key phasor, 2 eddy current-type displacement transducers and their conditioners, an oscilloscope, a data sampler, and a computer system with rotordynamic analysis software.

3. Experimental process and results

In this research, three types of experiments have been conducted:

(1) *Experiment in accelerating speed and stable current*: The rotor is accelerated from 0 to 7000 r.p.m., the control currents are 0, 0.25, 0.5, 0.75 and 1.0 A in stable, respectively.
(2) *Experiment in stable speed and sudden current*: The rotor is running stably at speeds of 1500, 2000, 3000, 4000, 5000, 6000, and 7000 r.p.m., the control current of 1.0 A is suddenly applied on or taken off.
(3) *Experiment in accelerating speed and sudden current*: To suppress the vibration when the rotor is accelerating across the first critical speed the control current is applied on or taken off at certain speed according to the on/off control method described below.

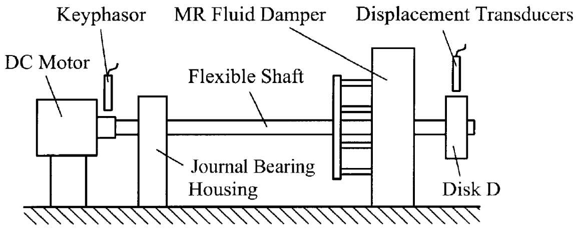


Fig. 2. Sketch of the experimental rotor system.

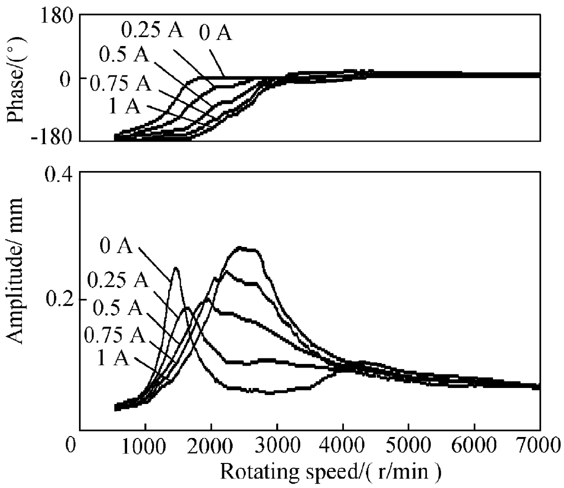


Fig. 3. Response curves in horizontal direction.

Fig. 3 is the response curves of vibration amplitude–speed and phase–speed (synchronous portion) of the system in horizontal direction when the rotor is accelerating and the current is stable. It is shown that when the current is increased gradually, the system appeared firstly damping effect (see the curves for $I = 0.25$ and 0.5 A), and then showed strong shifting of the first critical speed (see the curves for $I = 0.75$ and 1.0 A). The critical speed $n_{cr\,0}$ of the system is 1600 r.p.m. at $I = 0$ A, whereas the critical speed $n_{cr\,1}$ is increased to 2500 r.p.m. at $I = 1.0$ A. The increasing of critical speed with the control current means that the MR fluid in the damper is transformed from liquid state to semi-solid state gradually, and the flexible support of the damper approaches gradually to a rigid support.

Fig. 4 is the waterfall diagrams of vibration amplitude–frequency in horizontal direction at different currents while the rotor is accelerating and the current is stable. The main component in the diagrams is the synchronous one ($1x$), which reflects the imbalance properties of the rotor. In addition, there are also some double-synchronous frequency ($2x$) at lower speed caused by the initial bending of the shaft, and sub- or non-synchronous frequency at higher speed. It is found from experiments that

(1) When $I = 0$ and 0.25 A, the orbits of the rotor look like ellipses and unstable response occurred. There are only two frequency components of $1x$ and $2x$ in the frequency spectrum, as shown in Fig. 4a.

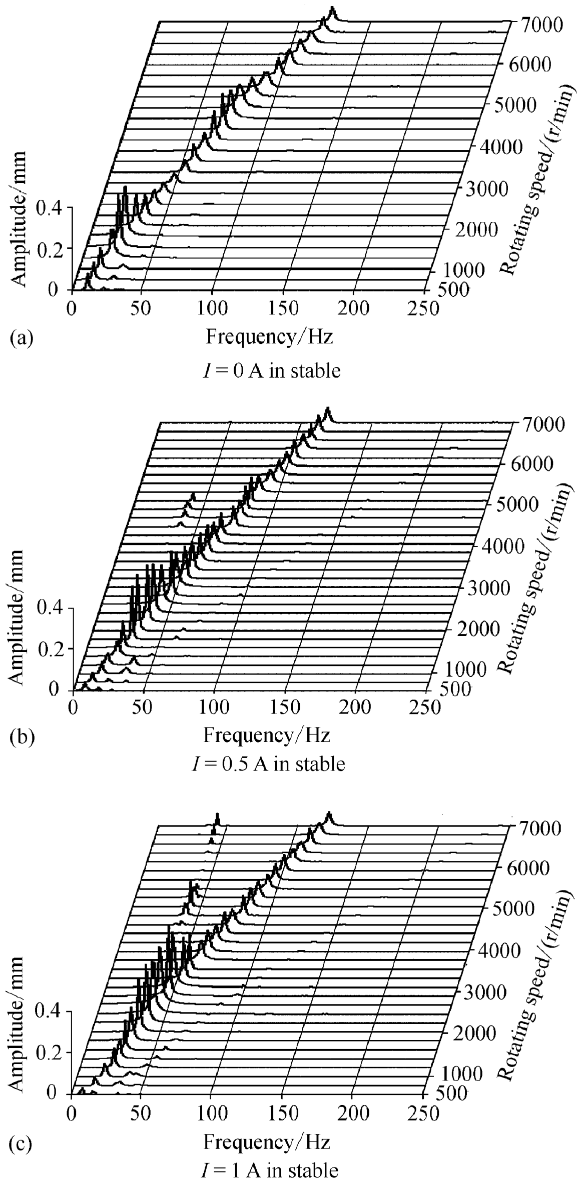
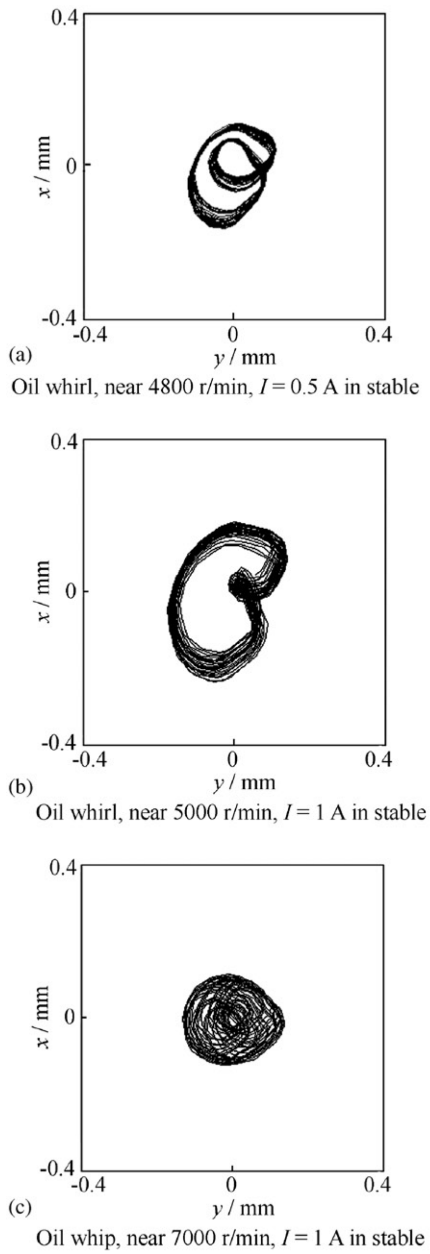


Fig. 4. Waterfall diagrams of displacement in horizontal direction.

(2) When $I = 0.5$ and $0.75\,\mathrm{A}$, the orbits of the rotor may have inner loop in certain speed range, as shown in Fig. 5a. Fig. 4b is a waterfall diagram with unstable situation ($I = 0.5\,\mathrm{A}$). It is clear that a certain sub-synchronous frequency, which is approximately equal to but less than the first critical speed, appeared in the diagram in speed range of 4500–5100 r.p.m. This is the situation like "oil whip" in journal bearings. It is known from Fig. 2 that the critical speed is 2100 r.p.m. for $I = 0.5\,\mathrm{A}$. Obviously, the unstable "oil whip" occurred when rotating speed is near or over two times the first critical speed.



(a) Oil whirl, near 4800 r/min, $I = 0.5$ A in stable

(b) Oil whirl, near 5000 r/min, $I = 1$ A in stable

(c) Oil whip, near 7000 r/min, $I = 1$ A in stable

Fig. 5. Orbits of the rotor in unstable state.

(3) When $I = 1.0\,\text{A}$ and the speed is over 4600 r.p.m., the "oil whip" unstable vibration occurs always, and the vibration amplitude is larger than that at $I = 0.5\,\text{A}$, as shown in Fig. 5b. The "oil whip" vibration disappears near 6000 r.p.m.; but when the speed is over 6500 r.p.m. the "oil whip" comes again and the vibration orbit becomes disordered and looks like chaos

response, as shown in Figs. 4c and 5c. From the logarithmic diagram of Fig. 4c near 7000 r.p.m., it can be seen that continuous frequency components appear. Since the critical speed is 2500 r.p.m. at this condition, the unstable situation has also occurred when the rotating speed is near two times the critical speed and it remains till about 5000 r.p.m.. When the speed is over 5000 r.p.m., the rotor vibration becomes a non-synchronous vibration, the frequency of which does not change with the speed, holding at 42 Hz, which is just equal to the first critical frequency at $I = 1.0$ A ($2500/60 \approx 41.7$ Hz), so "oil whip" happens. The disappearing of non-synchronous vibration within 6000–6500 r.p.m. shows that the rotor system has a narrow stable speed range.

When the MR fluid damper is used in the rotor system, the MR fluid inside the damper is changed from liquid to semi-solid as the control current increases. This will restrict the movements of the moving disks inside the MR damper and increase largely the support stiffness. As rigid support may cause rotor system losing stability, MR damper in large applied current may also cause instability to rotor system.

Fig. 6 are orbits and time histories of the transient response of the rotor system while exerting on (ON) or taking off (OFF) the current of 1.0 A at some test speeds. The very small circles in the time waves are marks of the key phasor. Obviously, the transient responses of the amplitude and the phase at every speed can transform from a stable state to another stable state while the current is applied on or taken off, no unstable phenomena are observed.

When rotating speed $n < n_{cr\,0}$ or $n > n_{cr\,1}$, the response curves for $I = 0$ and 1.0 A in Fig. 3 are all situated in sub-critical range or in super-critical range. When exerting on or taking off the current, the vibration can increase or decrease in single direction, and quickly reach a new stable state because the change in the phase is small. The vibration is shown in Fig. 6a and c. When $n_{cr\,0} < n < n_{cr\,1}$, the response curve is situated in super-critical range for $I = 0$ A, but in sub-critical range for $I = 1.0$ A in Fig. 3. The rotor system changes from a supercritical state to a sub-critical state, or vise visa when exerting on or taking off the current. The vibration time histories appear oscillation, as shown in Fig. 6b.

Even though the rotor system is stable at the instant of exerting on or taking off the 1.0 A current, the system may gradually appear oil whip after the system reaches the stable state when the rotational speed is near 5000 and 7000 r.p.m., respectively. This shows that the increasing of damper support stiffness provides conditions for the occurrence of oil whip, and the occurrence of oil whip needs certain conditions of speed and time in which the vibration energy is accumulated. Obviously, unstable appears when the half-synchronous frequency is near the first critical speed. But when the rotating speed is near 6000 r.p.m., the rotor remains stable after the current of 1.0 A is applied on and the system reaches a stable state, which is consistent with the experiment in accelerating speed (Fig. 4c).

As the applied current can change largely the stiffness of the MR damper and shift the critical speed of the rotor system, a simple ON/OFF control method can be used for the vibration and stability control of the rotor system, especially the large amplitude of the rotor system across the critical speed can be suppressed through simple ON/OFF control scheme. The practical method is as follows: when the rotor is starting up, the current is turned on to 1.0 A, and the response will

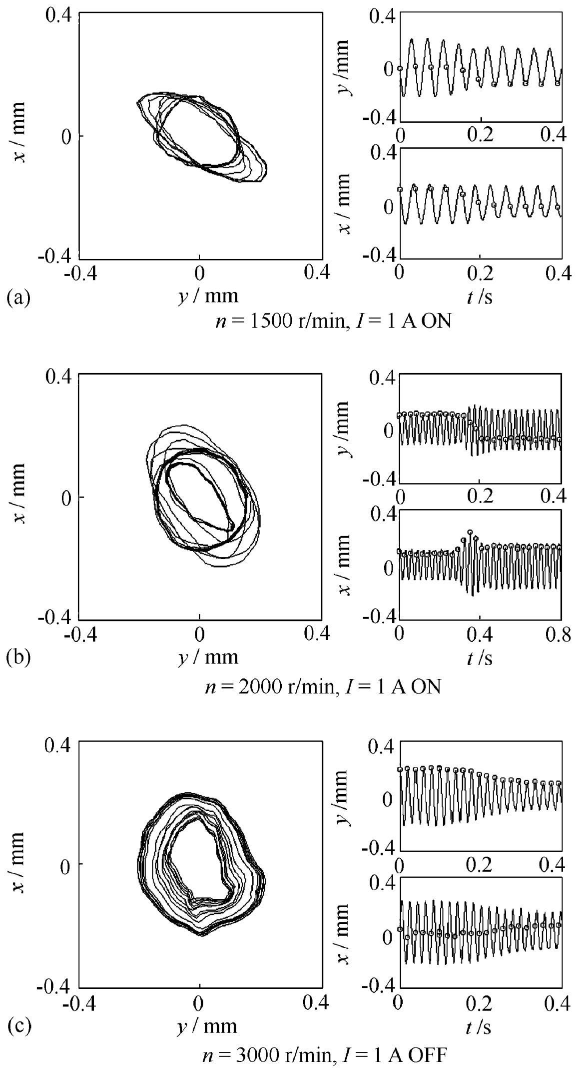


Fig. 6. Transient response of the rotor.

follow the curve of amplitude versus speed at 1.0 A (referring to Fig. 3); when the rotor speed passes the critical speed and reaches to a certain value (i.e., near the intersection point of curves 0 and 1.0 A), the current is turned off, and the response will follow the curve of amplitude versus speed at 0 A. When the rotor runs down from high speed, the control method is reversed. Fig. 7 is the curves of amplitude and phase versus speed when the rotor speed is increasing and the ON/OFF control scheme is used. It is shown from Fig. 7 that when turning off the current at speed of 1800 r/min, the amplitude and the phase of the response reach to new stable conditions very quickly, and the MR damper effectively suppresses the vibration at the first critical speed.

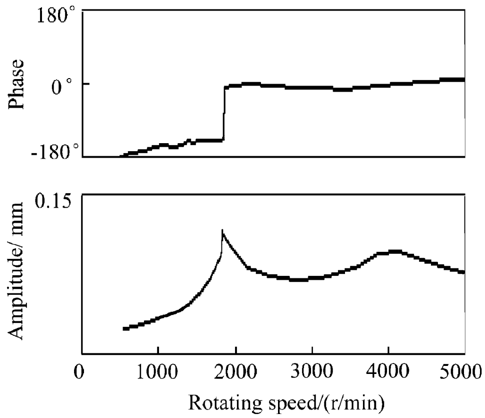


Fig. 7. Response curves while using on–off control.

4. Conclusions

Conclusions can be summarized from the above experiment results:

(1) When the rotor is accelerating and the control current is stable, the stability of the MR fluid damper-rotor-journal bearing system may decrease with the increasing of control current and unstable "oil whip" may occur in some rotating speed range. The main reason is that the increasing of applied current will increase the damper support stiffness, which may provide necessary condition for the occurrence of oil whip. The unstable speed is about two times the first critical speed.
(2) When the rotor is running stably, sudden exerting on or taking off the control current will not cause the system becoming unstable, but oil whip may gradually occur after the system becomes stable due to the large damper support stiffness.
(3) The rotor system is stable when the rotor vibration across the critical speed is suppressed simply by ON/OFF control method, which is a simple and effective vibration and stability control method.

Acknowledgements

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References

[1] J.D. Carlson, Adaptronics and smart structures, in: H. Janocha (Ed.), Magnetorheological Fluid Actuators, Springer, Berlin, 1999, pp. 180–195.
[2] J. Wang, G. Meng, Magnetorheological fluid devices: principles, characteristics, and applications in mechanical engineering, Proceedings of the Institution of Mechanical Engineers Part L. Journal of Materials: Design & Applications 215 (2001) 165–174.

[3] S.Y. Jung, S.B. Choi, Analysis of a short squeeze-film damper operating with electrorheological fluids, STLE Tribology Transactions 38 (4) (1995) 857–862.
[4] S.B. Choi, et al., Control characteristics of a continuously variable ER damper, Mechatronics 8 (2) (1998) 143–161.
[5] J.L. Nikolajsen, An electroviscous damper for rotor applications, Transactions of the American Society of Mechanical Engineers, Journal of Vibration and Acoustics 112 (4) (1990) 440–443.
[6] G.Z. Yao, G. Meng, Vibration control of a rotor system by disk type electrorheological damper, Journal of Sound and Vibration 219 (1999) 175–188.
[7] J.A. Tichy, Behavior of a squeeze film damper with an electrorheological fluid, STLE Tribology Transactions 36 (1993) 127–133.
[8] J.M. Vance, D. Ying, Experimental measurements of actively controlled bearing damping with an electrorheological fluid, Journal of Engineering for Gas Turbine and Power, Transactions of the American Society of Mechanical Engineers, 122 (2) (2000) 337–344.
[9] J. Wang, G. Meng, Vibration control of rotor by magnetorheological fluid damper, Journal of Huazhong University of Science and Technology 29 (7) (2001) 47–49 (in Chinese).