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A disk-type magneto-rheological fluid damper for rotor system vibration control

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Abstract

Based on the particular characteristic of a magneto-rheological (MR) fluid, i.e., a rapid, reversible and dramatic change in its rheological properties produced by the application of an external magnetic field, a simple disk-type MR fluid damper operating in shear flow mode is presented in this paper. The magnetic field of the disk-type MR fluid damper is analyzed by the finite element method in order to show if the design is reasonable. The effect of excitation current in the coil on the magnetic flux density in the axial gaps filled with the MR fluid is studied both theoretically and experimentally. Finally, the effectiveness of the disk-type MR fluid damper for attenuating the vibration of rotor systems and of a simple open-loop on-off control based on the feedback of rotational speed for controlling vibration of rotor systems are experimentally studied in a flexible rotor system. It is shown that the dynamic characteristics of the disk-type MR fluid damper can be easily controlled by a steady magnetic field produced by a simple electrical magnetic coil with a low DC current (less than 1 A) and that the disk-type MR fluid damper is very effective for attenuating and controlling the vibration of the rotor systems. It is possible to supply the optimum supporting damping for every vibration mode in the rotor system by using the disk-type MR damper, if the location of the disk-type MR fluid damper in the rotor is properly chosen.

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1. Introduction

Electro-rheological (ER)/magneto-rheological (MR) fluid is a kind of controllable or smart fluids whose rheological properties can be dramatically and reversibly varied by the application of an external electrical/magnetic field in a very short time period. The ER/MR fluid has the property which in the absence of an external electric/magnetic field flows as a viscous liquid, but in the presence of a high electric/magnetic field it immediately solidifies to a grease state. Such a characteristic of the ER/MR fluid with rapid, reversible and dramatic changes in its rheological properties provides a possibility to control the dynamic characteristics of traditional fluid dampers. Many ER/MR fluid structures and dampers for controlling structural or machine vibration have been developed and studied. It is shown that the ER/MR fluid dampers or structures can effectively control the vibration of structures or machines in a wide range of frequencies.

The ER/MR fluids are one of the most active of the current range of "smart materials". Most researches in the application of the ER/MR fluids are focused on structural vibration control and flow power system. Stanway et al. [1] and Agrawal et al. [2] made a comprehensive survey in the state of the ER/MR fluids and the application of the ER/MR fluids in mechanical engineering. However, there are only dozens of papers published which deal with the application of ER fluid in controllable dampers or bearings for rotational machinery. Nikolajsen and Hoque [3,4] first proposed a multi-disk ER fluid damper operating in shear flow mode and studied the effectiveness of the multi-disk ER fluid damper in controlling the vibration of rotor systems when passing through the critical speeds. Vance and Ying [5] developed Nikolajsen and Hoque's test rig and demonstrated the dynamic behavior of the rotor systems supported on the multi-disk ER fluid damper. Diamarogona and Kollias [6,7] studied stability of a rotor system supported by journal bearings with ER fluid theoretically and compared the capability of three kinds of ER fluid damper for controlling the rotor vibration. Morishita and his co-authors [8,9] presented an ER fluid squeeze film damper based on squeeze film-operating mode and studied the effectiveness of the ER fluid squeeze film damper in attenuating the vibration of rotor systems. Nikolakopoulos and Papadopoulos [10] studied the dynamic characteristics of a controllable journal bearing lubricated with the ER fluid. Kim et al. [11] studied the vibration control of an overhung rigid rotor by a sealed and ER fluid squeeze film damper. Yao et al. [12] studied both experimentally and theoretically the effectiveness of the multi-disk ER fluid damper to suppress the rotor vibration around the critical speeds and the loose imbalance response. Based on the Bingham fluid, Tichy [13], Jung and Choi [14] analyzed the dynamic characteristics of fluid film force and the existing conditions and manner of "core" in the ER fluid squeeze film damper and journal bearings. These researches have shown that the dynamic characteristics of the rotor system supported on the ER fluid dampers or journal bearings can be controlled by the application of an external high voltage (typically 2000 V) and the ER fluid dampers or journal bearings can effectively control the vibration amplitude of a rotor system in a wide range of rotating speeds.

Up to now, most researchers in the rotating machinery have focused on the application of the ER fluid in the controllable dampers and journal bearings. However, in comparison with the properties of the ER fluid, an MR fluid inherently has higher yield strength, therefore it is capable of generating a greater fluid force. Furthermore, the MR fluid is activated by the application of an external magnetic field, which is easily produced by a simple, low-voltage electromagnetic coil and avoids arcing problem of the ER fluid dampers occurred while two poles with high-voltage in the ER fluid dampers touch or the electrical insulation is not prefect. Zhu et al. [15,16] presented an MR fluid squeeze film damper operating in the squeeze film mode and showed that the MR fluid squeeze film damper can effectively control the vibration of a rotor system, but an unbalanced magnetic pull force existing in the journal due to the eccentricity of the journal with respect to the damper housing may pull the journal to the damper housing and lock up the damper like a rigid support when the applied current in the coil is over a certain value.

In this paper, a disk-type MR fluid damper based on shear operation mode is first presented, then a flexible rotor test rig supported on a disk-type MR fluid damper is reported to demonstrate the ability of the disk-type MR fluid damper to attenuate and control the vibration of a rotor system. Next, the magnetic field of the disk-type MR fluid damper is analyzed by the finite element method and the effect of excitation current on the magnetic flux density in the axial gaps filled with the MR fluid is studied both theoretically and experimentally. Finally, the effectiveness of the disk-type MR fluid damper for attenuating the vibration of rotor system and of a simple open-loop on-off control based on the feedback of rotational speed for actively controlling the vibration of rotor systems are experimentally studied.

2. Structure of a disk-type MR fluid damper and rotor apparatus

2.1. Structure of a disk-type MR fluid damper

After analyzing and comparing several different designs of disk-type MR fluid dampers based on shear mode operation, a simple symmetrical disk-type MR fluid damper as shown in Fig. 1 was designed and built. It consisted of a moving disk, a ball bearing, two magnetic poles, a coil wound circumferentially and a damper housing, which form the magnetic path with two equal axial gaps. The moving disk was mounted on an additional journal in the outer race of the ball bearing. The

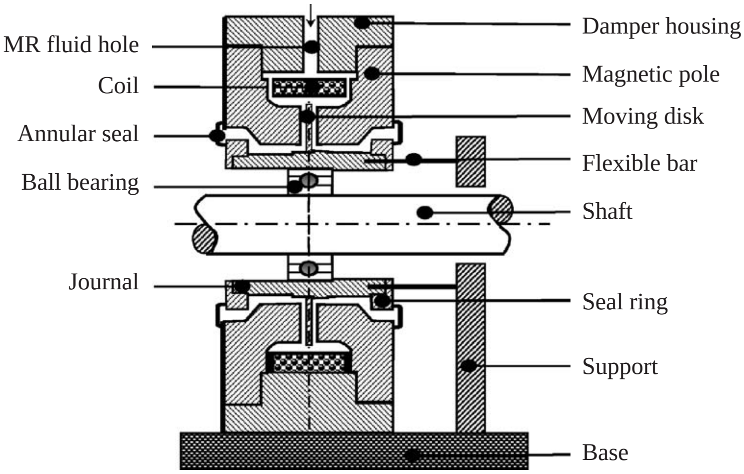

Fig. 1. Cross-section of the disk-type MR fluid damper.

structure was symmetric about the middle of the moving disk. The material of the magnetic poles, the moving disk and the damper housing were made of mild steel. The additional journal and the seal rings were made of aluminum. Two annular flexible rubber membranes sealed the MR fluid. The rubber membrane was carefully chosen in order to reduce its effect on the static damping and stiffness of the damper as possible. The axial gaps between the magnetic poles and the moving disk and the radial clearance between the outer diameter of the additional journal and the inner diameter of the magnetic poles were filled with the MR fluid. Since the radial clearance chosen was 6–7 mm in order to avoid a significant squeezing film force being generated by the squeezing effect in the radial clearance. Therefore, when the journal is whirling, the moving disk shears the MR fluid in the axial gaps and produces a resisting fluid force to dissipate the vibration energy from the rotor system. The coil of the disk-type MR damper had 1000 turns, the diameter of wire was 0.65 mm, and the resistance of the coil was 43.6 Ω. The axial gap between the magnetic pole surface and the moving disk surface was 2.5 mm in design; the inner and outer radiuses of magnetic poles were 40 and 62 mm, respectively.

The moving part of the disk-type MR damper, together with the additional journal were mounted on a centralizing spring made by four parallel flexible bars for the centralizing journal in the steady state, preventing rotating of the moving disk and providing an initial radial stiffness for the damper. The stiffness of the centralizing spring could be easily altered by varying the length of the flexible bars. In the design of the disk-type MR fluid damper, the structure of the damper was carefully designed in order to easily assemble a multi-disk MR damper if necessary.

2.2. Rotor test apparatus

The experimental apparatus of a flexible rotor system supported on a disk-type MR fluid damper is shown in Fig. 2. It consisted of a uniform flexible shaft with length 900 mm and diameter 20 mm, a uniform disk with weight 3.5 kg located centrally between the bearings, a

Fig. 2. Rotor test rig with a disk-type MR fluid damper.

disk-type MR fluid damper, a flexible coupling, and a variable speed motor with a speed feedback control. The rotor system could run with a given acceleration/deceleration rate controlled by a personal computer or could be maintained any given rotational speed in the region of rotational speed from 500 to 6000 rev/min. For simplicity, the rotor was supported by a self-aligned ball bearing at one side and by a MR fluid damper at the other side. A flexible coupling was used between the shaft and the motor to isolate the vibration transmitted from the motor. In order to measure the vibration of the journal, an additional hollow cylinder was installed in the inner hole of the additional journal. The span of the bearings was 607 mm. The diameter and thickness of the disk were 130 and 25 mm. The weight of the disk-type MR fluid damper was 1.4 kg, which includes the ball bearing, the additional cylinder mounted on the outer race of the ball bearing, two seal rings, and the additional hollow cylinder. The first rigid pin–pin critical speed of the rotor, in which two sides of the rotor were rigidly supported on the ball bearings, was 2843 rev/min, the second rigid pin–pin critical speed of the rotor was over 10,000 rev/min. The first two flexible critical speeds of the rotor system, in which one side of the rotor was supported on the disk-type MR fluid damper in the absence of the MR fluid and the other was rigidly supported on the ball bearing, were 2243 and 3757 rev/min, respectively. It should be pointed out that when the disk-type MR fluid damper was filled with the MR fluid, the first flexible critical speed of the rotor system was reduced to 2080 rev/min, and the second flexible critical speed of the rotor system was about 3640 rev/min, due to the mass effect of the MR fluid in the axial gaps.

The rotor was balanced and then an additional imbalance mass was added in the disk to obtain the required rotor imbalance. The rotor imbalance used in the test was about 130 $g \cdot mm$.

2.3. MR fluid

The MR fluids are typically non-colloidal suspensions of micron-sized, magnetizable particles in base oil. In order to stabilize the suspension, some additives are added in the MR fluid. At first, some commercially available MR fluids, such as MRF-132LD of Lord Corp., USA and MR fluid of Advanced Fluid Development Limit, UK, were used in the test, but no satisfactory results were obtained. These MR fluids were too viscous without the applied magnetic flux to almost lock-up the damper as if a rigid support, which is very similar to that of the disk-type ER fluid damper observed by Vance and Ying [5]. The behavior can be explained by the fact that the initial viscosity of the base oil without the external magnetic field was higher and that number of magnetizable particles in the MR fluids is too large. In order to lower the initial viscosity of the base oil, a MR fluid, composed of base oil with lower initial viscosity, carbonyl iron with a mean particle size of 5 μm and additives, was used. The weight ratio of the base oil to the carbonyl iron was 1:1, i.e. the weight percentage of the magnetizable particles in the MR fluid was 50%. The kinematic viscosity and density of the base oil at 20 °C were 4.5 cSt and 827 kg/m^3, respectively.

2.4. Measurement and control systems

The measurement system was based on the HP 7500 Series C VXI data acquisition system with an E1432A A/D digitizer. The E1432A module has a DSP chip inside the card with 16 16-bit channels. The maximum sample rate is 51,200 samples per second per channel. This module integrates transducer signal conditions, anti-alias protection, digitization and high-speed measurement computation in a single slot card. The control system was based on the d-SPACE 1102 controller board in which the TMS329C40 was used as a processor. The DS 1120 controller board is with maximum 4 12-bit parallel analogue output channels. Only two of them were used in tests, one for controlling the acceleration rate of the motor operation, the other for controlling the DC voltage power supply. The DC voltage power supply for the magnetic coil used was a programmable DC one (HP 4553 type) with maximum output voltage 60 V and maximum output current 2.5 A. The VXI data acquisition system and the d-SPACE control system were connected to a personal computer. All the measurements and control software were written in MATLAB platform with variable sample rate and control parameters.

The vibrations of the rotor system in both horizontal and vertical directions at the journal and disk positions were measured with eddy-current proximity probes. The motion orbits of the rotor system at a series of steady-state rotational speeds and the rotor imbalance responses in the slow acceleration operations were measured. All measurement and control could be automatically done when the acceleration/deceleration rate of rotor run-up/run-down operation was given. The constant acceleration rate used in the test was about 0.15 Hz/s, i.e., about 9 rev/min/s, which was so slow that there was no visual difference between the imbalance response curves measured in the acceleration/deceleration operations and that the rotor imbalance response curve measured in the acceleration operation was basically the same as that in a series of steady-state rotational speeds. While the rotor was run-up or run-down with the given acceleration rate, the time histories of vibrations signals of the rotor were continually and synchronously sampled by the VXI data acquisition system, then on-line displayed, and recorded on the computer hard disk. The sampling frequency chosen at the steady-state rotational speeds and the acceleration operation was so high that there were enough points in the rotor motion orbits or the rotor imbalance response. The signals of the rotor vibration were also displayed on an oscilloscope for observation as a safety precaution. The rotor vibration amplitude given here was obtained by averaging the measured vibration amplitudes in at last two revolutions.

3. Magnetic field analysis of the disk-type MR fluid damper

When a DC current is applied in the coil, the magnetic flux density in the axial gaps filled with the MR fluid changes and causes the change of apparent viscosity of the MR fluid. As a result, the dynamic characteristics of the disk-type MR fluid damper can be adjusted in a very simple manner. How to significantly change the magnetic flux density in the axial gaps with a low applied voltage and to keep the magnetic field uniform along the magnetic pole in order to easily model the damper are of importance. In order to show if the design is reasonable and the effect of excitation current and geometric parameters on the magnetic flux density in the axial gaps, the magnetic field of the disk-type MR fluid damper is first analyzed by the finite element method.

Since the disk-type MR fluid damper shown in Fig. 1 is axialsymmetric and is also symmetric about the mid-plane of the moving disk, it is enough to analyze the magnetic field in a quarter of the MR fluid damper. The magnetic field of the disk-type MR fluid damper was analyzed by 2D finite element analysis code MagNet [17]. The size and the mesh of a quarter of the MR fluid damper for the finite element analysis are shown in Fig. 3. In order to get more accurate magnetic flux distribution in some important areas, i.e., in the axial gap regions filled with the MR fluid, a

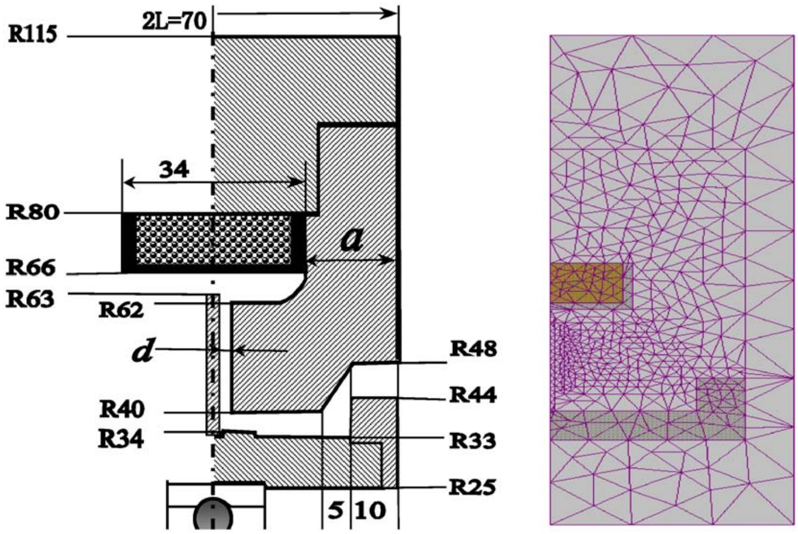

Fig. 3. Size and mesh of a quarter of the disk-type MR fluid damper (all dimension in mm).

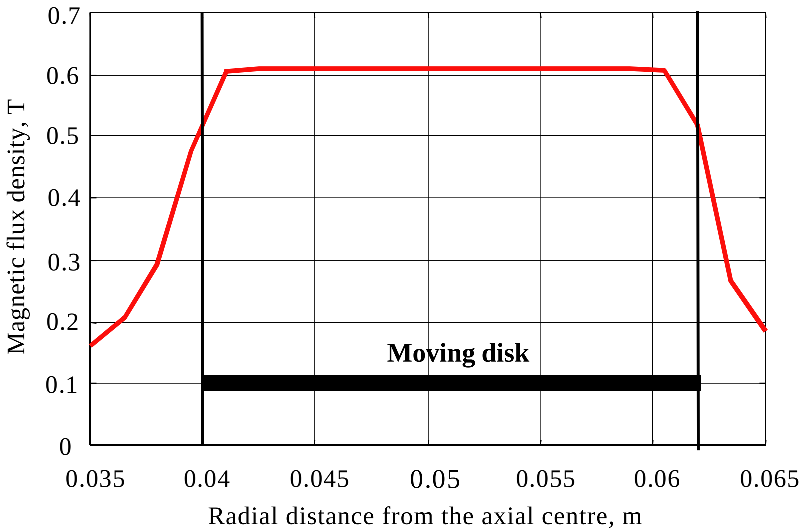

Fig. 4. The variation of the magnetic flux density along the moving disk ($NI = 1000\,\mathrm{A}$, $L = 40\,\mathrm{mm}$, $d = 1.0\,\mathrm{mm}$, and $a = 23\,\mathrm{mm}$).

smaller mesh was used. Since the maximum magnetic flux density required in the axial gaps is less than 1 T and the magnetic field is a steady one, the eddy current, magnetic hysteresis and magnetic saturation were not considered in this analysis.

The variation of the magnetic flux density in the axial gaps along the moving disk direction for excitation $NI = 1000\,\mathrm{A}$, the axial gap $d = 1\,\mathrm{mm}$ and the width of the damper housing $L = 40\,\mathrm{mm}$ is shown in Fig. 4. The corresponding contour plot of the magnetic flux density is shown in Fig. 5.

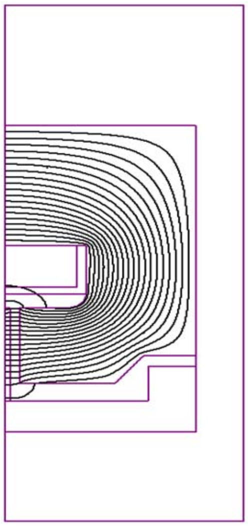

Fig. 5. Contour plot of the magnetic flux density ($NI = 1000\,A$, $L = 40\,mm$, $d = 1.0\,mm$, and $a = 23\,mm$).

It is shown that the magnetic flux density in the magnetic pole areas along the moving disk direction is basically uniform, but with a small end leakage. The average magnetic flux density in the axial gap at the designed excitation current of 1 A is 0.61 T, which is high enough to change the rheological properties of the MR fluid. Since the magnetic flux distribution in magnetic path is uniform and no very high magnetic flux density region existed in the magnetic path, it shows that the shape and geometric sizes of the magnetic path of the damper were properly chosen. The effect of chamfered corners where there were additional seal rings on the magnetic flux density in the axial gaps, is negligible.

Fig. 6 shows the variations of the magnetic flux density along the moving disk and the magnetic flux density in the middle position of the moving disk with the axial gap d. It is shown that the magnetic flux density in the axial gaps decreases linearly with the increase of the axial gaps in the region of small axial gaps, but decreases slowly in the region of the larger axial gaps.

Fig. 7 shows the variations of the magnetic flux density along the moving disk and the magnetic flux density in the middle position of the disk with the axial width of the damper housing L. It is shown that the magnetic flux density in the axial gaps increases with the increase of the axial width of the MR fluid damper. When the axial width of the damper housing is over a certain value or the magnetic path area in the damper housing is over the area of the magnetic pole, the magnetic flux density in the axial gaps basically remains unchanged, further increase in the axial width of the damper housing does not greatly change the magnetic flux density in the axial gaps. Therefore, the geometric design with same area in the magnetic path is a basic principle for choosing the structure sizes of such kind of dampers.

Fig. 8 shows the variations of the magnetic flux density along the moving disk and the magnetic flux density in the middle position of the moving disk with the applied currents. It is shown that the magnetic flux density in the axial gaps is directly proportional to the applied excitation current in the coil when the magnetic flux density does not over the magnetic saturation limitation. The

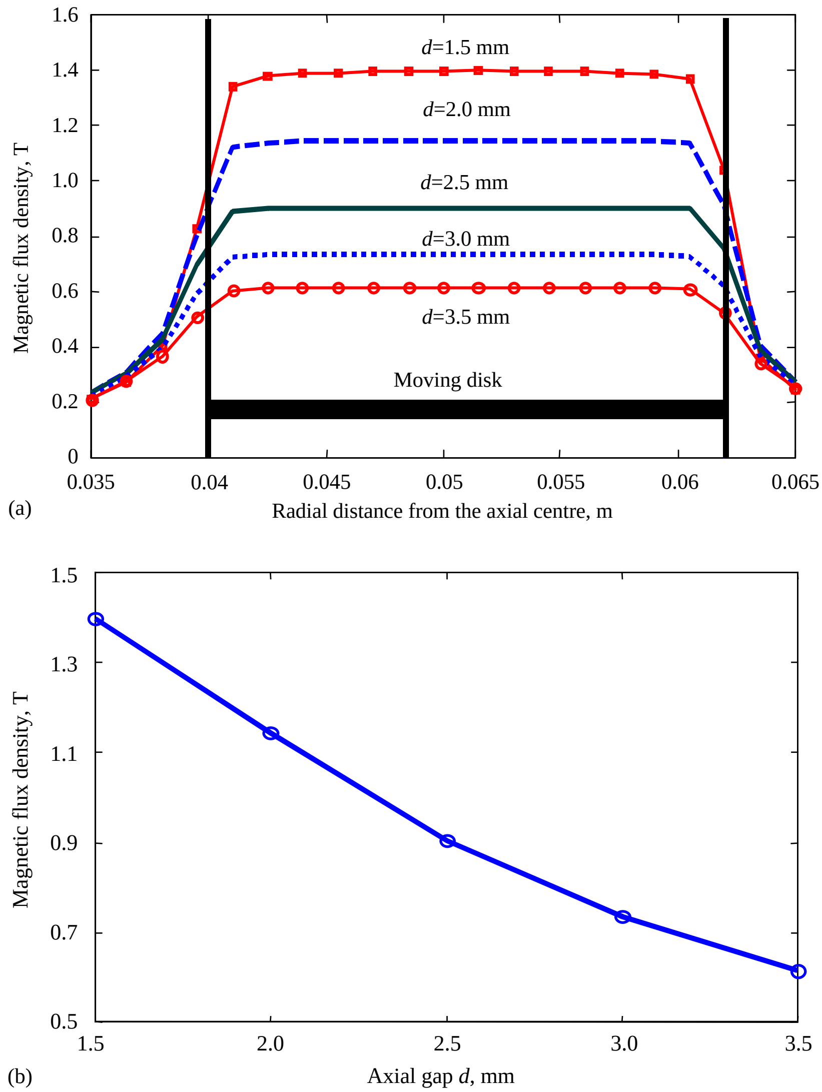

Fig. 6. The effect of the axial gap on the magnetic field in the axial gap ($L = 40$ mm, $a = 23$ mm, $NI = 3500$ A). (a) The variation of the magnetic flux density along the moving disk in different axial gaps. (b) The effect of the axial gap on the magnetic flux density in the axial gap.

linear relation between the magnetic flux density in the axial gaps and the applied currents will be helpful to control the dynamic behavior of the damper by simply varying the currents in the coil.

In order to verify the theoretical results, a magnetic flux sensor of a flux meter was installed at the middle position of the moving disk. When the applied excitation current was changed in steps in steady state, the magnetic flux density at the middle position of the moving disk was measured. Fig. 9 shows the comparison of the numerical results of the magnetic flux density at the middle

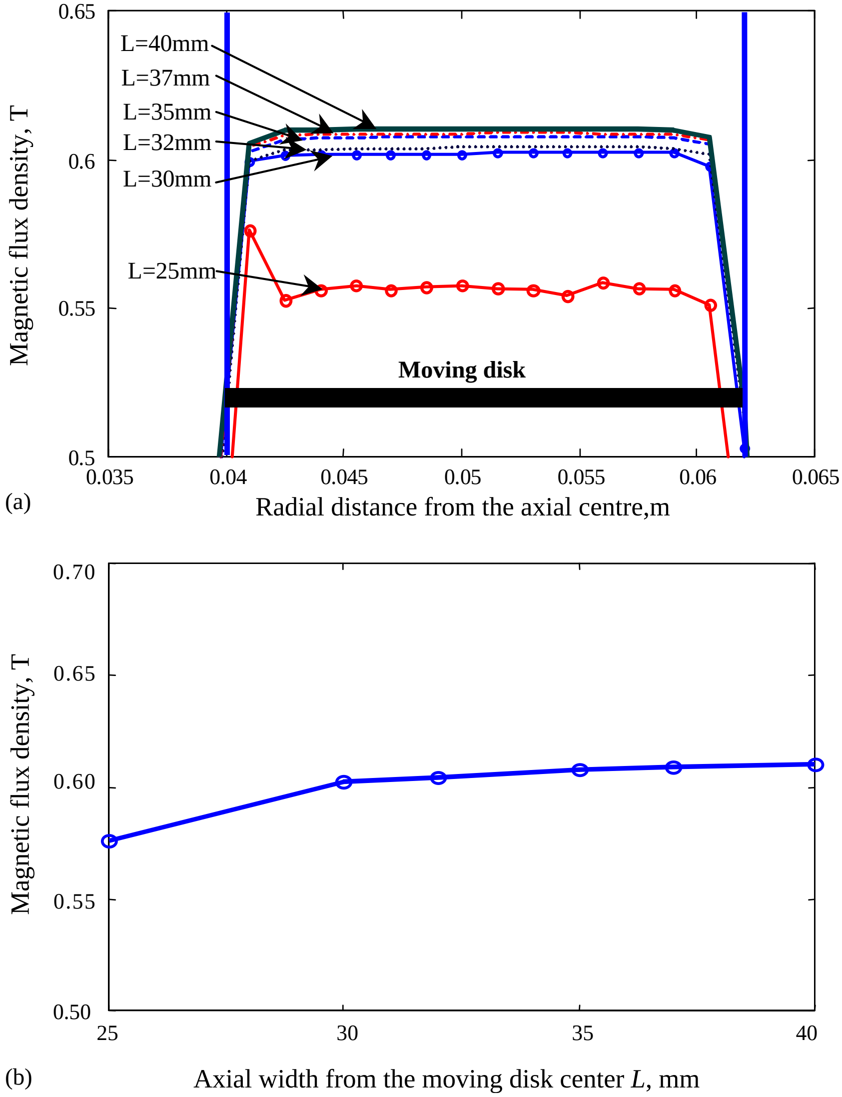

Fig. 7. The effect of the axial width on the magnetic field in the axial gap ($d = 1$ mm, $a = L - 17$ mm, $NI = 1000$ A). (a) The variation of the magnetic flux density along the moving disk in different axial width. (b) The effect of the axial width on the magnetic flux density in the axial gap.

position of the moving disk with the measured results by the flux meter at different applied currents. The numerical results are in reasonable agreement with the measured results. The reason for the difference between the theoretical and experimental results is that two axial gaps were not equal in experiments, which were about 2.0 and 2.75 mm, respectively, and the axial gaps were not uniform in the circumferential position due to disk's skew. The magnetic flux density in the axial gaps varies linearly with the applied current in the coil and the magnetic flux density in the axial

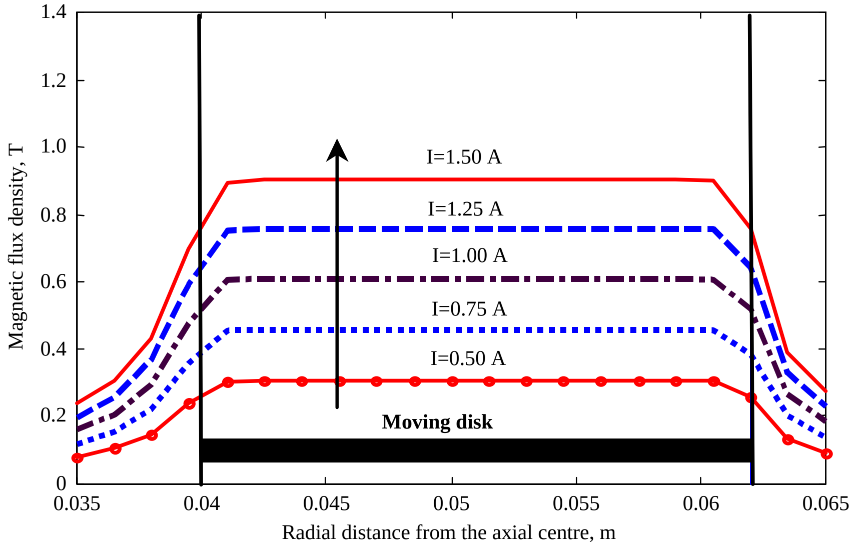

Fig. 8. The variation of the magnetic flux density along the moving disk in the axial gaps in different applied currents ($L = 35\,\mathrm{mm}$, $a = 20.5\,\mathrm{mm}$, $d = 1\,\mathrm{mm}$).

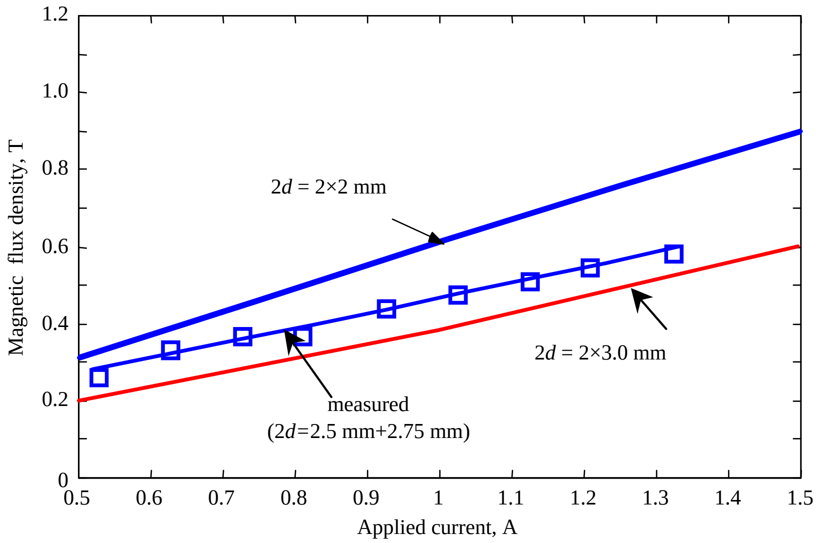

Fig. 9. Theoretical and measured magnetic flux density in the axial gaps at the middle position ($L = 35\,\mathrm{mm}$, $a = 20.5\,\mathrm{mm}$, $NI = 2000\,\mathrm{A}$).

gaps along the moving disk is basically uniform, which will be very useful to more easily control the dynamic characteristics of the damper by varying the current in the coil and to model the disk-type MR fluid damper.

4. Dynamics of a rotor system mounted on the disk-type MR fluid damper

4.1. Effect of applied current on the rotor motion orbits

Firstly, the motion orbits of the rotor system at the disk and journal positions were measured for different applied currents in order to verify the controllability of the disk-type MR fluid damper on the dynamic characteristics of the rotor system.

Fig. 10 shows the variations of the journal and disk whirl orbits with the applied current at speeds of 2200, 2900 and 3400 rev/min, which are close to the first flexible critical speed, first rigid critical speed and the second flexible critical speed, respectively. The arrows stand for the direction of increasing applied current in the order of 0, 0.1, 0.25, 0.5, 0.75 and 1.0 A.

In the speed regions near the flexible critical speeds, as can be seen in Figs. 10(a) and 10(c), both the disk and journal whirl orbits decreased as the applied current increased. Especially the journal whirl orbits were very small and the disk whirl orbits did not significantly change when the applied current was more than 0.5 A. It means the MR fluid has been solidified under the influence of the magnetic field with the current of 0.5 A and a higher applied current is not necessity to change the dynamic characteristics of the disk-type MR fluid damper. In these speed regions, the lower applied current in the coil can obviously change the dynamic characteristics of the rotor system.

However, the disk whirl orbits did not always decrease with the increase of the applied current. In the region of rotational speeds near the first rigid critical speed, as shown in Fig. 10(b), the journal whirl orbits decreased as the applied current increased, but the disk whirl orbits increased. It is shown that in this region, as the applied current increases, the solidified MR fluid damper produces a high fluid force, which leads to the MR fluid damper in the locked-up state like a rigid support.

While the applied current is switched on from 0 to 0.25, 0.5, 0.75, and 1.0 A, respectively, the transient vibration signals in the vertical direction both at the disk and journal positions at the speed of 3200 rev/min are shown in Fig. 11. The transient response decayed in a short period less than 0.2 s. The larger the current variation is, the shorter is the time for the transient response to decay. After the current was switched on or switched off, the rotor motion would smoothly jump to a new stable state with a different vibration level from a steady state without causing instability during switching the current.

Therefore, the change of the rotor motion orbits at the different speeds with the applied current is very obvious and very sensitive to the applied current, and it is not necessary to apply a high current in the coil; an applied current of 1 A in the disk-type MR fluid damper can dramatically change the dynamic characteristics of the rotor system. The dynamic characteristics of the disk-type MR fluid damper are shown to be controllable.

4.2. Effect of the applied current on the rotor imbalance response

The imbalance responses of the rotor system with different applied currents were measured to demonstrate the capability of the disk-type MR fluid damper to attenuate the rotor vibration. Fig. 12 shows the imbalance response curves of the rotor system with different applied currents, which provides the entire rotor vibration information in the operating speed region. When no current was applied in the coil, the maximum amplitudes in both the journal and disk occurred at

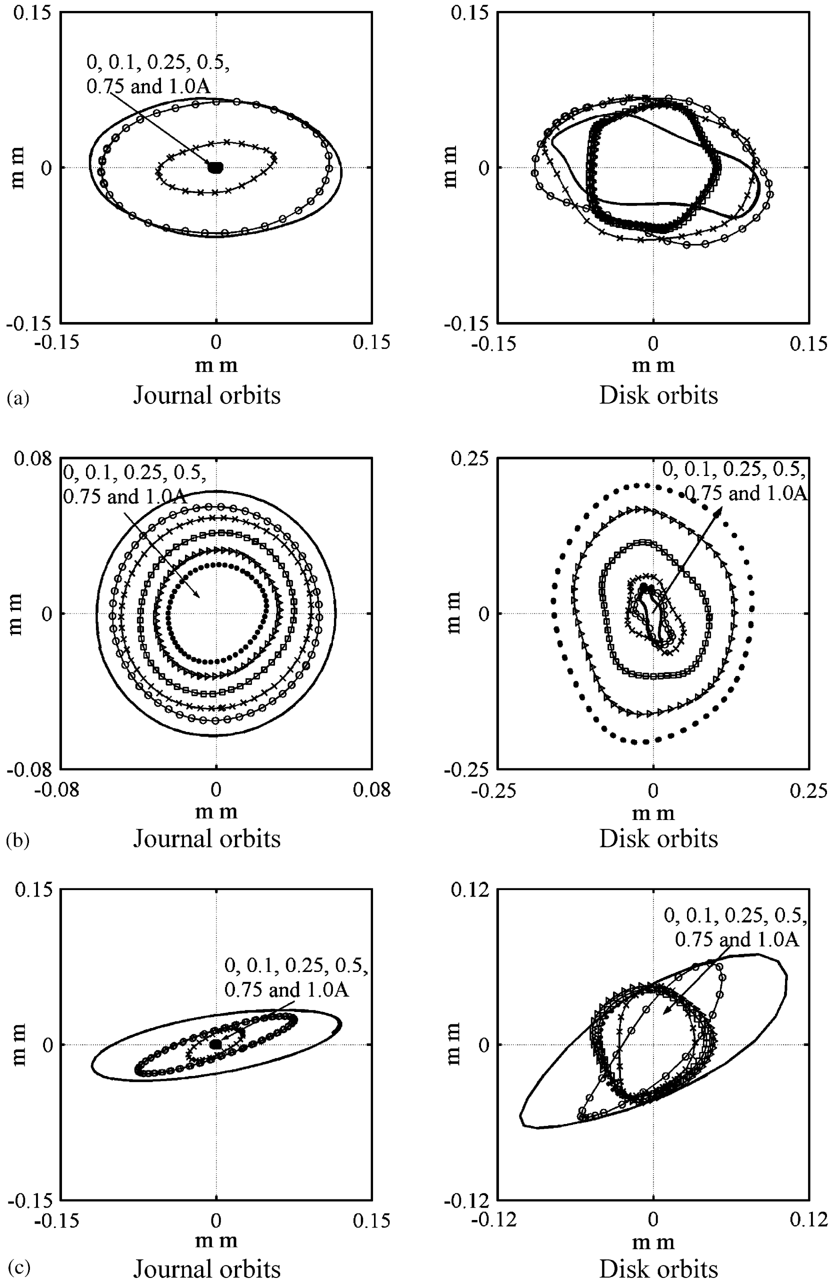

Fig. 10. Journal and disk whirl orbits at different rotational speeds with different applied currents. Key for orbit: — for 0.0 A, ○○○○ for 0.10 A, ×××× for 0.25 A, □□□□ for 0.50 A, △△△△△ for 0.75 A, and ●●●● for 1.0 A. The arrows stand for the direction of increasing applied current in order of 0, 0.1, 0.25, 0.5, 0.75 and 1.0 A. (a) At the speed of 2200 rev/min, (b) at the speed of 2900 rev/min, and (c) at the speed of 3400 rev/min.

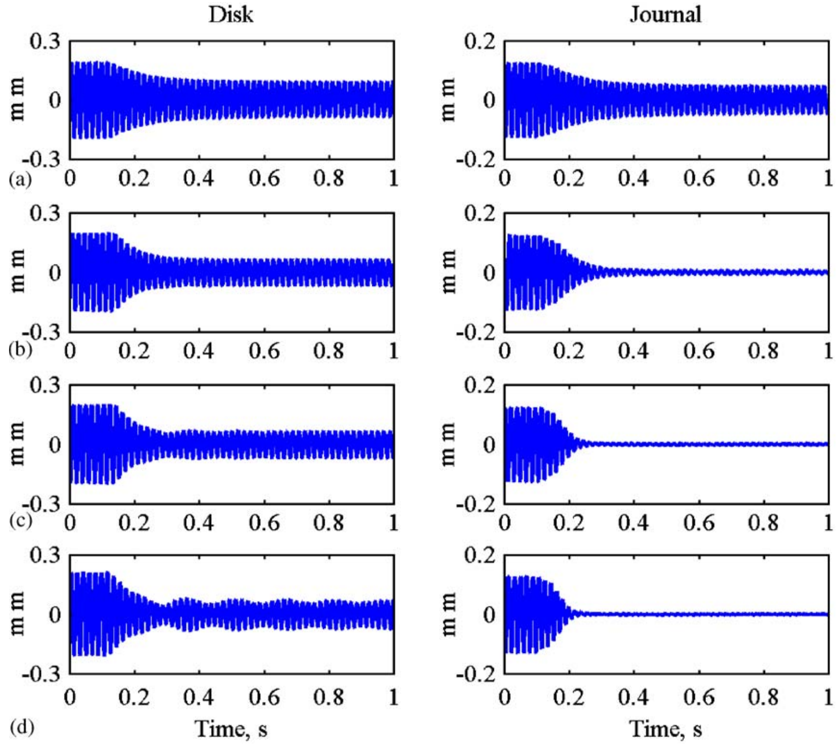

Fig. 11. Transient vibration in vertical direction after current switching at the speed of 3200 rev/min (a) current switching from 0 to 0.25 A, (b) current switching from 0 to 0.5 A, (c) current switching from 0 to 0.75 A, and (d) current switching from 0 to 1.0 A.

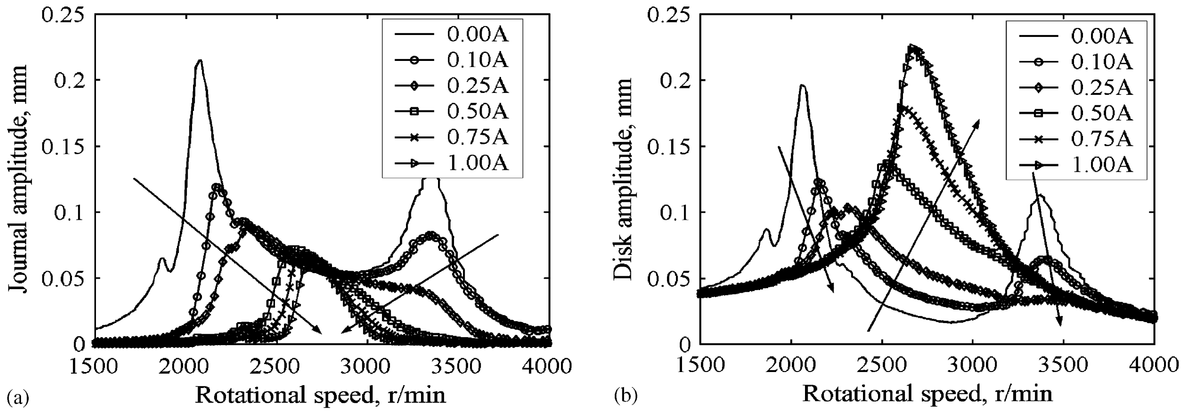

Fig. 12. Imbalance response of rotor system with different applied excitation currents Key for response: — for 0.0 A, ○○○○ for 0.10 A, ◇◇◇◇ for 0.25 A, □□□□ for 0.50 A, ×××× for 0.75 A, ▸▸▸▸ for 1.0 A. The arrows stand for the direction of increasing applied current in order of 0, 0.1, 0.25, 0.5, 0.75 and 1.0 A. (a) Journal and (b) disk.

the 2065 and 3376 rev/min, respectively, which basically corresponded to the first two flexible critical speeds of the rotor system. As the applied current increased, the journal vibration amplitudes decreased in the whole operation speed region, the disk vibration amplitudes decreased in the regions of the first two flexible critical speeds, but increased in the region of the first rigid critical speed. When the applied current of 1 A was applied in the coil, the rotational speed with maximum journal or disk vibration amplitude was about 2683 rev/min, which corresponds to the rigid critical speed of the rotor system. The rotational speed with the maximum vibration amplitude increased with an increase in the applied current. It is shown that since the MR fluid has higher yield strength, the disk-type MR fluid damper with only one moving disk was capable of generating enough fluid shear force to make the damper change to a rigid-like state from a flexible state for our rotor system. Therefore, it is not very necessary to use more disks in the damper. If more moving disks should be used in order to produce larger force as used in the multi-disk ER damper [3–6,12], a very high current should be applied in order to get the same magnetic flux density in the axial gaps. The high current will produce more heat, which adversely affects the property of the MR fluid and is also undesirable.

It is very clear that when the magnetic field was applied in the MR fluid, the disk-type MR fluid damper could change not only the damping of the rotor system, but also its stiffness as well. In order to explain the effects of the applied excitation current on the dynamic characteristics of the rotor system, Fig. 13 shows the variations of disk resonant amplitude, resonant speed, equivalent damping ratio of the rotor system with the applied excitation current. The disk resonant amplitude is the disk amplitude at the resonant speed. The resonant speed is the speed at which the disk vibration amplitude is maximum. The equivalent damping ratio is determined by the half-power point method in the disk amplitude response curve with error less than 10%. As the applied excitation current increases, the disk resonant amplitude decreases or the equivalent damping ratio significantly increases in the region of lower excitation applied currents. For the disk resonant amplitude or the equivalent damping ratio, there is an optimum applied excitation current with which the disk resonant amplitude is minimum or the equivalent damping ratio is maximum. When the applied excitation current is over the optimum applied current, the equivalent damping ratio will decrease, so the disk resonant amplitude will increase in the region of higher excitation currents. Generally, the optimum excitation current for equivalent damping ratio coincides with the optimum applied current for minimizing the disk resonant amplitude. The first resonant speed linearly increases with the increase of the excitation current, but slowly in the higher excitation currents and tends to the first rigid critical speed of the rotor system. It can be found that the effect of the excitation current on the equivalent damping of the damper in the region of higher excitation currents was more obvious than on its equivalent stiffness, but the effect of the excitation current on the equivalent stiffness of the damper in the region of lower excitation currents was more obvious than on its equivalent damping. Since the dynamic characteristics of the disk-type MR dampers could be continuously controlled, it is possible to provide the optimum supporting damping for every vibration mode present in the rotor system.

4.3. Effectiveness of on-off control by the MR fluid damper

It is shown in Fig. 12 that in certain rotational speed regions, the disk amplitude either continuously increases or decreases with an increase on the applied current, therefore it is also not

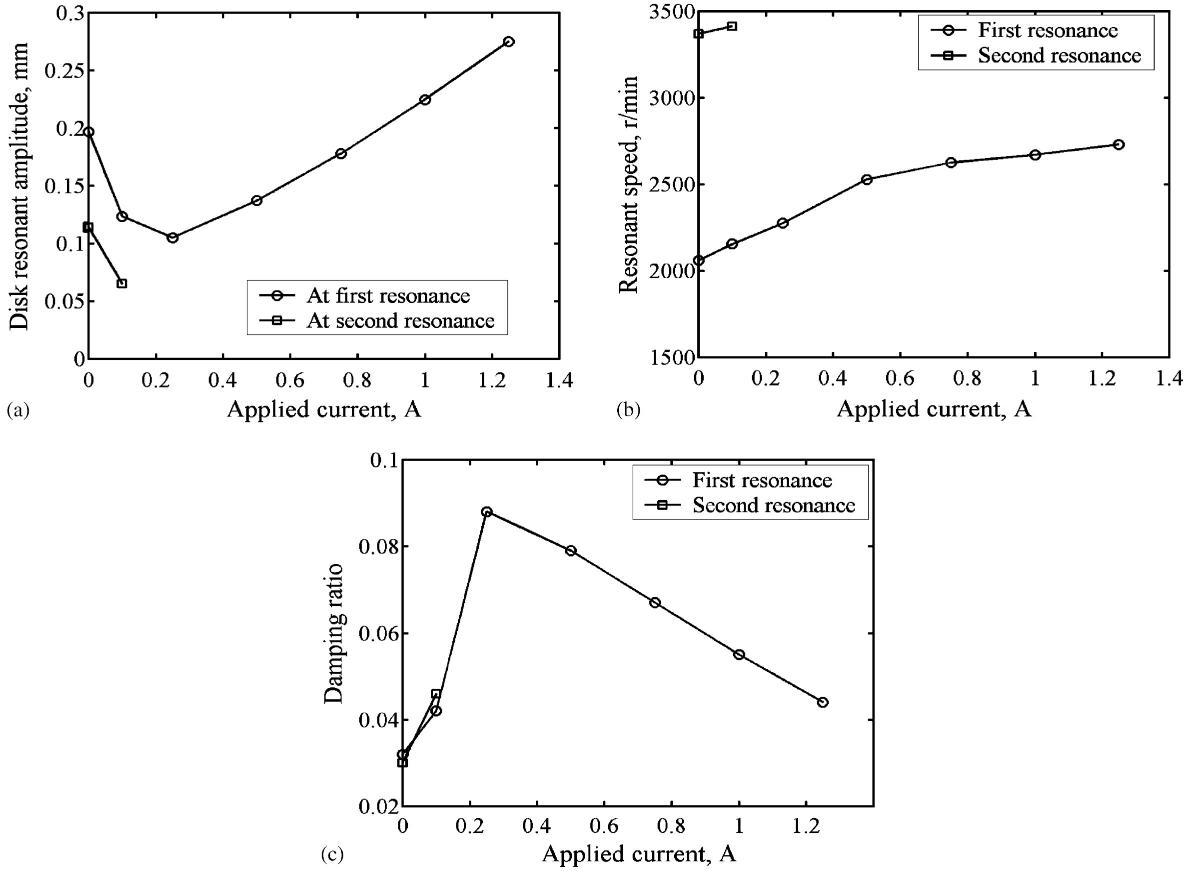

Fig. 13. Variation of disk resonant amplitude, resonant speed and equivalent damping ratio with the applied excitation current. Key: oooo, for the first resonance and □□□□, for the second resonance. (a) The disk resonant amplitude, (b) the resonant speed, and (c) the equivalent damping ratio.

necessary to continuously control the applied current, only discrete changes in the applied current is enough to effectively control the rotor vibration. Within these speed bands, the applied current of the disk-type MR fluid damper should be set at either the no current state or the maximum current state, respectively.

In order to investigate the effectiveness of the disk-type MR fluid damper for active control rotor vibration, a simple on-off control algorithm based on rotational speed feedback was used in the primary stage. A series of the tests were performed, while the rotor system was run-up at a given acceleration rate. From the principle of the on-off control, we know that the applied current of the disk-type MR fluid damper should respectively be set at either the 0 A state (for the minimum stiffness and damping) or the 1.0 A state (for the maximum stiffness) and switched at special speeds, in order to minimize the required objective.

Based on the measured steady-state imbalance response curves of the rotor system at different excitation currents shown in Fig. 12, it is found that if the disk vibration amplitude was selected as the control objective, when the rotational speed was below 2234 rev/min, the excitation current of

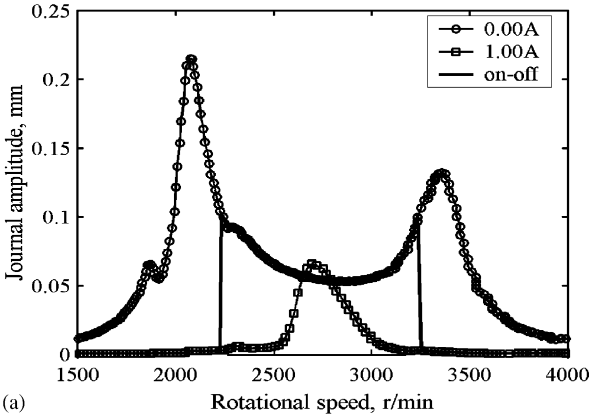

Fig. 14. Effectiveness of on-off control on controlling rotor vibration: Key: ○○○○, for 0.0 A, and □□□□, for for 1.0 A, and ——, for on-off control. (a) Journal and (b) disk.

1.0 A was applied in the coil to make the MR fluid damper in the maximum stiffness mode, the excitation current was switched off at the 2234 rev/min, and then kept at the zero current state until the current switched on again at the 3233 rev/min from 0 to 1.0 A. The excitation current reached its steady state in less than 0.005 s. The imbalance response of the rotor system after the on-off control is shown in Fig. 14 as a solid thick line. As expected, the imbalance response curve after the on-off control is in good agreement with the open-loop results in different regions of rotational speeds. When the applied current was switched at the switching points, the rotor would run in the expected path after a brief transient response with relative larger vibration amplitude. The transient response took a short period, general less than 1 s. The reason for the transient response occurring with high vibration at the switching speeds is that disk amplitudes at the switching points before and after applied current switching are same, but there is great difference in the journal amplitude, it means that energy in the rotor system jumps from a state to the other state at the switching speed [18].

Comparison with the original rotor imbalance response, after applying on-off control, the disk maximum amplitude (i.e., the maximum amplitude in the transient response) was only 0.092 mm in the whole region of rotational speeds, which was about 50% of the maximum amplitude of the disk in the 0 A state. The vibration amplitude of the journal was less than 0.1 mm, which was about 47.5% of the maximum amplitude of the journal in the 0 A state. It is shown that the disk vibration amplitude is effectively controlled and the on-off control is very effective to reach the required objective.

5. Conclusion

A simple disk-type MR fluid damper based on the shear operation mode was presented and the dynamic characteristics of the disk-type MR fluid damper have been experimentally studied in this paper. It is shown that the dynamic characteristics of the disk-type MR fluid damper can be easily controlled by the application of an external DC magnetic field and that the disk-type MR fluid damper is very effective for attenuating and controlling rotor vibration. The applied excitation current required to dramatically change the dynamic characteristics of the rotor system by the disk-type MR fluid damper is less than 1 A (or applied voltage less than 50 V), which makes the disk-type MR fluid damper more attractive than the ER fluid dampers and can be easily used in real rotating machinery. It is possible to provide the optimum supporting damping and stiffness for every vibration mode in the rotor system by using the disk-type MR damper, supposing that the disk-type MR fluid damper is not located at the nodes of rotor system modes.

There are many problems to be studied in detail, such as feedback control loop to provide active control rotor vibration, theoretical model of the damper, development of new MR fluid to avoid particles setting, system stability and nonlinear characteristics etc. All of these investigations are in progress both theoretically and experimentally.

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