http: //www.zju.edu.cn/jzus; E-mail: jzus@zju.edu.cn
ISSN 1009-3095 *Journal of Zhejiang University SCIENCE* V.4, No.5, P.514-519, Sep.-Oct., 2003

A disc-type magneto-rheologic fluid damper*

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Received Aug.29, 2002; revision accepted Nov.21, 2002

Abstract: A disc-type magneto-rheological fluid damper operating in shear mode is proposed in this paper, which is based on the special characteristics of the magneto-rheological (MR) fluid with rapid, reversible and dramatic change in its rheological properties by the application of an external magnetic field. The magnetic field of the disc-type MR fluid damper is analysed by the finite element method; the controllability of the disc-type MR fluid damper on the dynamic behaviour of a rotor system; and the effectiveness of the disc-type MR fluid damper in controlling the vibration of a rotor system, are studied in a flexible rotor system with an overhung disc. It is shown that the magnetic flux density of the disc-type MR fluid damper in the working areas can significantly change with the applied current in the coil; and that the dynamic behavior of the disc-type MR fluid damper can be varied by the application of an external magnetic field produced by a low voltage electromagnetic coil. The disc-type MR fluid damper can significantly change the dynamic characteristics of a rotor system, provided that the location of the disk-type MR fluid damper is carefully chosen. The disc-type MR fluid damper is a new actuator with good dynamic characteristics for rotating machinery.

Key Words: Magneto-rheological fluid, Damper, Rotordynamics, Vibration, Active vibration control
Document code: A **CLC number:** TH113.1

INTRODUCTION

Electro-rheological (ER)/Magneto-rheological (MR) fluid is a kind of controllable fluid whose rheological properties, especially apparent viscosity, can be dramatically and reversibly varied in a few milliseconds by the application of an external electrical/magnetic field. Such special characteristics of the ER/MR fluid with rapid, reversible and dramatic change in its rheological properties provide a possibility for controlling the dynamic characteristics of traditional fluid dampers. A multi-disc ER fluid damper operating in shear mode and an ER fluid squeeze film damper operating in squeeze film mode for controlling the vibration of a rotor system were studied by Nikolajsen *et al*. (1990), Vance *et al*. (2000), Morishita *et al*. (1992; 1996), respectively. It was shown that the dynamic characteristics of a rotor system supported on ER fluid dampers could be controlled by the application of an external high voltage (typically 2000V) and that ER fluid dampers could effectively control the vibration amplitude of a rotor system in a wide range of rotating speeds.

In comparison with the properties of the ER fluid, the MR fluid inherently has higher yield strength, and so, 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. This has considerable safety implications, compared with the high voltage required for ER fluid systems; also MR fluids are less liable to deterioration with use. Zhu (2001a) developed an MR fluid squeeze film damper and showed that the MR fluid squeeze film damper could 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

* Project supported partially by the EC BRITE/EURAM program under BRPR-CT97-0544 IMPACT project and State Key Laboratory of Vibration, Shock and Noise, Shanghai Jiaotong University.

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 disc-type MR fluid damper based on shear operation mode is proposed. After giving the basic structure of the disc-type MR fluid damper, we use the finite element method to analyse the magnetic field of the disc-type MR fluid damper and the effect of excitation current on the magnetic flux density in the axial gaps filled with the MR fluid in order to determine if the design of the damper structure is proper and reasonable. Finally, the controllability of the disc-type MR fluid damper on the dynamic behaviour of a rotor system, and the effectiveness of the disc-type MR fluid damper for attenuating the rotor system vibration and of a simple open-loop on-off control based on rotational speed feedback in actively controlling the vibration of rotor systems were experimentally studied in a flexible rotor system with an overhung disc.

STRUCTURE OF A DISC-TYPE MR FLUID DAMPER AND ROTOR TEST RIG

The basic structure of a symmetrical disc-type MR fluid damper based on shear operation mode is shown in Fig.1. Only the top half cross section is given due to the structure symmetry. It consists of a moving disc mounted on the outer race of the journal, two magnetic poles, a coil wound circumferentially and a damper housing, which form the magnetic path with two identical axial gaps between the magnetic poles and the moving disc. The MR fluid fills in the axial gaps and other areas. The moving disc, the damper housing, and the magnetic poles are made of mild steel; the journal and the seal rings are made of aluminium. Two annular flexible rubber membranes seal the MR fluid.

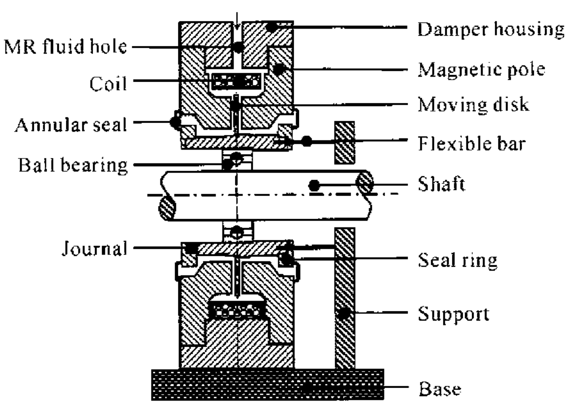

Fig.1 Cross-section of the disc-type MR fluid damper

The moving part of the MR damper, together with the outer cylinder on the outer race of the bearing are mounted on a centralizing spring made by four flexible bars for centralizing the journal in the steady state, thus preventing rotation of the moving disc and providing an initial stiffness for the damper. The effective stiffness of the centralizing spring can be easily altered by varying the length of the flexible bars. The radial clearance between the outer surface of the journal and the inner surface of the magnetic poles is more than 3.5 mm, which is so large that no significant squeeze film force is produced by the squeeze effect in the radial clearance when the journal is whirling.

When the journal is whirling, the moving disc shears the MR fluid in the axial gaps and produces a resistive fluid force to dissipate the vibration energy from the rotor system. A change of magnetizing current alters the fluid characteristics and hence the shear force. Thus, the dynamic characteristics of the damper can be adjusted in a very simple manner to provide optimum system damping at each critical speed and also to improve rotor stability. The coil has 1000 turns, the diameter of wire is 0.50 mm, and the resistance of the coil is 43.6 Ω. Each axial gap is about 2.0 mm in the design; the inner and outer radii of the magnetic poles are 40 mm and 62 mm.

The rotor experimental apparatus, shown in Fig.2, was designed to simulate a low-pressure turbine rotor of a gas turbine engine. It consisted of a uniform flexible shaft with length 900 mm and diameter 20 mm, an over-hung disc with weight of 3.5 kg, an MR fluid damper, a flexible coupling, and a motor. The disc is mounted on one side of the shaft. For simplicity, the rotor was supported by a self-aligned radial ball bearing at one side and by a disc-type MR fluid damper at the other side. A flexible coupling was used between the shaft and the motor in order to isolate the vibration transmitted from the motor. In the absence of the MR fluid in the damper, the first flexible critical speed of the rotor system, in which one side of the rotor was supported on the damper and the other was rigidly supported on the ball bearing, was 1450 r/min. The first rigid critical speed of the rotor, in which two sides of the rotor were rigidly supported on the ball bearings, was 1700 r/min.

Fig.2 Rotor system rig

The MR fluid used was composed of a special base oil and carbonyl iron with mean particle size of 5 μm, the weight ratio of the base oil to the carbonyl iron was 4:1. The vibrations of the rotor system in both the horizontal and vertical directions at the journal and disc positions were measured with eddy current transducers. The power for the magnetic coil used in the test was a programmable DC voltage power supply with maximum current of 2.5 A. All measurements and control could be automatically done when the acceleration rate of rotor run-up or run-down was given. The acceleration rate used in the test was about 9 r/min per second, which was so slow that there was no visual difference between the unbalance response curves measured in the run-up and run-down operations.

MAGNETIC FIELD ANALYSIS OF THE DISC-TYPE MR FLUID DAMPER

The magnetic field of the disc-type MR fluid damper was analyzed by 2D finite element analysis code MagNet(1998). Since the disc-type MR fluid damper shown in Fig. 1 is axial-symmetric and is also symmetric along the mid-plane of the moving disc, it is enough to analyze the magnetic field in a quarter of the damper. To get a more accurate estimate of the magnetic flux density in some important areas, i.e., in the axial gap regions filled with the MR fluid, a smaller mesh was used. In this analysis, eddy current, hysteresis and magnetic saturation were not considered.

Fig. 3 shows the variation of the magnetic flux density of the damper in the axial gaps along the moving disc when each axial gap is 2 mm. It is shown that the magnetic flux density in the axial gaps along the moving disc radial direction is basically uniform, but with a small end leakage. The uniform magnetic field along the moving disc radial direction will be very useful for easily controlling the dynamic characteristics of the damper by varying the current in the coil, and for modelling the MR fluid damper. The average magnetic flux density in the axial gaps at the designed current of 1.0 A was 0.61 Tesla, high enough to solidify the MR fluid. The corresponding magnetic flux density plot showed that the shape and sizes of the magnetic path of the damper were properly chosen as there was no high magnetic flux density region in the magnetic path.

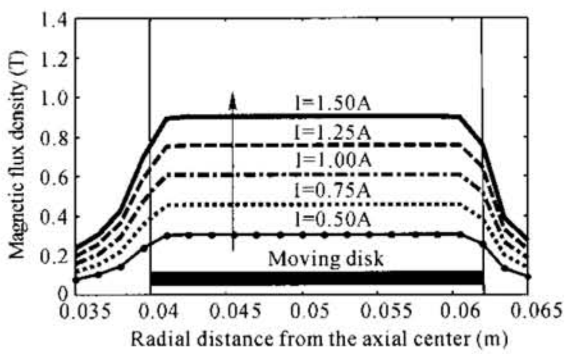

Fig.3 The effect of the applied current on the magnetic flux density in the axial gaps

EXPERIMENTAL RESULTS AND DISCUSSION

The main proposes of dynamic test of the rotor mounted on the disc-type MR fluid damper were to find answers to three questions: are the dynamic characteristics of a rotor system mounted on the disc-type MR fluid damper controllable? If yes, how fast is the response of the rotor system to the external current? And what is the effectiveness of the disc-type MR fluid damper for attenuating and controlling the rotor vibration?

Fig.4 shows the effect of the applied current on the disc orbits near the first flexible and the first rigid critical speeds. The applied currents

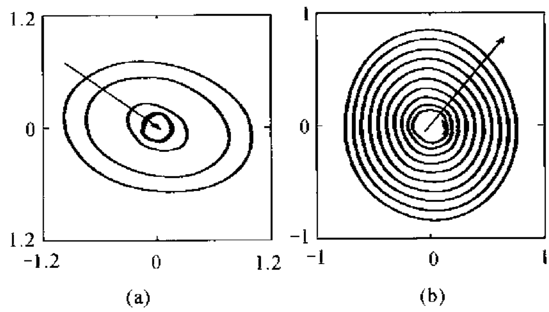

Fig.4 The variation of rotor orbits with applied current(the arrow stands for the direction of increasing current)

(a) at 1275 r/min; (b) at 1600 r/min

are 0 A to 1.0 A with a current increment of 0.1 A. The arrow stands for the direction of increasing applied current. It is shown that the disc vibration amplitude can be greatly changed with the applied current in the coil. Therefore, it is possible to control the dynamic characteristics of a rotor system by the disc-type MR fluid damper.

It is clear that the change of the rotor whirl orbits 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 1.0 A is enough to change dramatically the dynamic characteristics of a rotor system with the disc-type MR fluid damper. The dynamic characteristics of the disc-type MR fluid damper are shown to be controllable.

Fig.5 shows the transient orbits and the disc vibration signals in the vertical direction while the applied current was switched from 0 A to 1.0 A at rotational speed of 1200 r/min. It was found that after the applied current was switched on or off, the rotor would smoothly jump to a new stable state with a different vibration level from a steady state without causing instability. The response of the rotor system to applied current during switching on is faster than that during switching off. The former is about 0.1 second and the latter about 0.5 second.

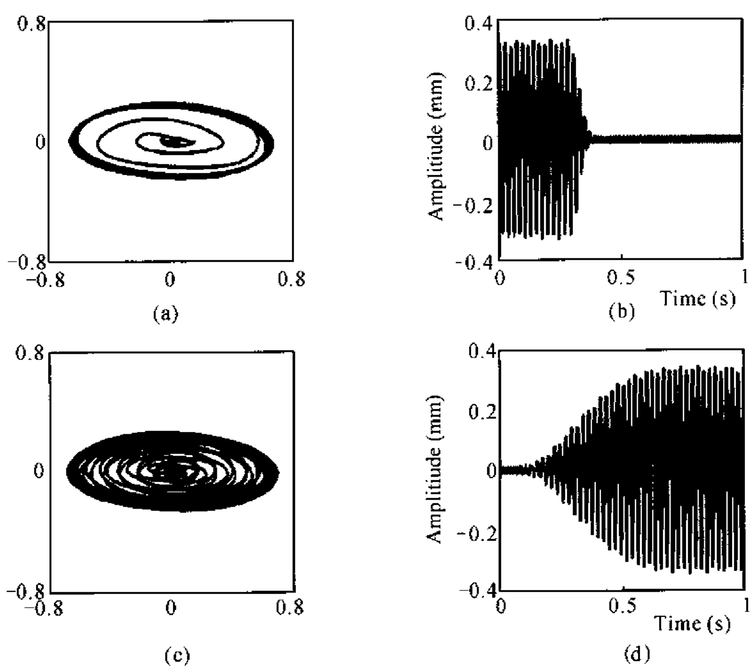

Fig.5 Disc orbits and transient vibration signals while the applied current was switching on (a)&(b), and off (c)&(d), at rotational speed of 1200 r/min

Fig.6 shows the unbalance response curves of the rotor system with applied currents of 0 A, 0.1 A, 0.25 A, 0.5 A, 0.75 A and 1.0 A, respectively. When no current was applied in the coil, there was a resonant peak at 1320 r/min, which basically corresponded to the first flexible critical speed of the rotor system. As the applied current increased, the journal vibration amplitudes decreased in the whole operational speed region, the disc vibration amplitudes in the region of the first flexible critical speed were diminished until an optimum value of applied current. However, if the applied current was increased beyond optimum, the disc vibration amplitudes did not change any more in the region of the first flexible critical speed with an increase in the applied current, but would increase in the region of the first rigid critical speed, and the resonant peak shifted to the higher rotational speed. When the applied current of 1.0 A was applied in the coil, the rotational speed with maximum journal or disc vibration amplitude was about 1700 r/min, which was very close to the first rigid critical speed of the rotor system. In this case, the rotor system was characterized by small damping and high stiffness. So, when an external magnetic field was applied to the MR fluid, the solidified MR fluid not only could change the damping of the rotor system, but also its stiffness.

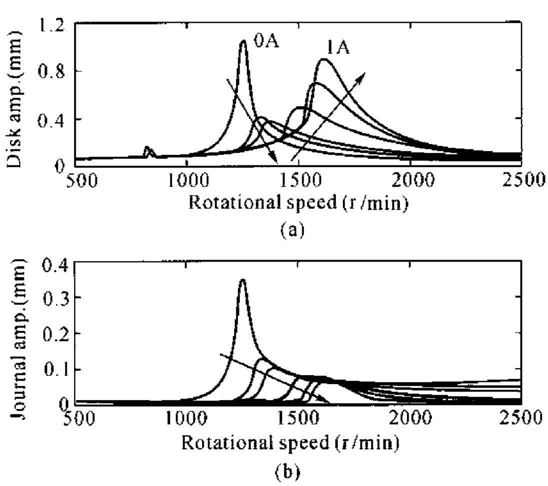

Fig.6 The effect of the applied current on the rotor unbalance response curve (the arrow stands for the direction of increasing current)
(a) disk amplitude; (b) journal amplitude

The results in Fig.6 also showed that within certain speed regions, the disc amplitude either continuously increased or decreased with an increase in the applied current, so, it is not necessary to continuously control the applied current in the coil, only discrete changes in applied current was enough to control the disc vibration; so that within these rotational speed bands, the applied current of the MR fluid damper should be set at either the 0 A state or the 1.0 A state, respectively.

A simple on-off control algorithm based on rotational speed feedback (Zhu *et al.*, 2001b), was used to investigate the effectiveness of the MR fluid damper for active control rotor vibration. In order to control the vibration of the overhung disc, current of 1.0 A should be applied in the coil when the rotational speed was below 1400 r/min, then switched off at 1400 r/min, and then kept at the zero current state until the high speed. The unbalance response of the rotor system after the on-off control is shown in Fig.7 as a solid thick line. As expected, the unbalance response curve after the on-off control was in good agreement with the open loop results in different regions of rotational speeds. When the applied current was switched off at the switching speed, the rotor would run in the expected path after a brief transient response with relatively larger vibration amplitude. The transient response took a short period, generally less than 1 second. It was clear that the rotor vibration amplitude was successfully controlled by the simple on-off control method.

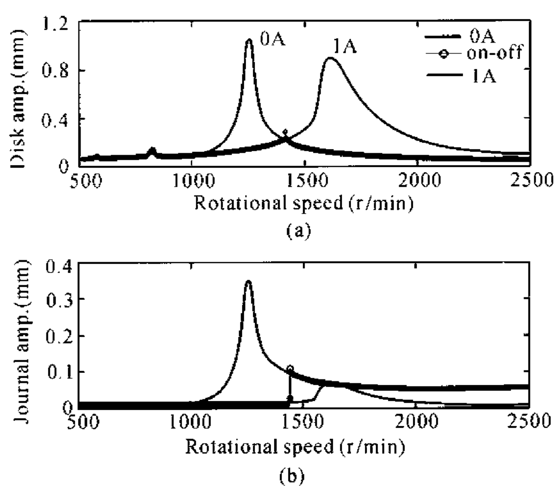

Fig.7 The effectiveness of the on-off control on rotor vibration
(a) disk amplitude; (b) journal amplitude

CONCLUSIONS

It was shown that the dynamic characteristics of the disc-type MR fluid damper proposed can be easily controlled by the application of an external magnetic field to the MR fluid; and that the disc-type MR fluid damper was very effective for controlling the vibration of a rotor system, provided that the rotor dynamics were carefully designed to exploit the changes in both stiffness and damping of the dampers or bearings. The applied voltage in the coil required to change the dynamic characteristics of the rotor system by the disc-type MR fluid damper was much lower than that in the ER fluid damper and could easily be used in real rotating machinery.

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