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Study of Vibration Dampers Using Magnetorheological Fluids for Rotating Devices

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***Abstract*—This paper presents a method for active vibration control using magnetorheological fluids in rotating devices with protruding shafts. A mechanical model of the rotating device is established, and the vibration amplitude is evaluated as a function of frequency. Based on this model, a magnetorheological active vibration damper is designed and fabricated. The composition of the magnetorheological fluid is determined, and the electrical parameters of the magnetic field coil in the control circuit of the damper are calculated. The performance of the device model is experimentally tested, and the resulting vibration data are measured and analyzed. The experimental results demonstrate the effectiveness of the proposed damper in reducing vibration amplitude, highlighting its potential for practical applications in rotating machinery.**

Keywords*— *Vibration damping methods; Active vibration damper; Rotor; Magnetorheological fluids.

I. Introduction

Vibrations are inevitably generated in industrial machinery during operation, particularly in high-speed rotors. Therefore, the development of effective methods to actively mitigate vibrations in rotating devices is essential. In practice, operating the rotor at high speed is generally advantageous due to the high energy concentration. However, an imbalance caused by eccentricity is inevitable, resulting in unbalanced forces during rotation of the rotor and causing significant vibrations during operation of the device. These vibrations can cause various problems, such as reducing device lifespan, affecting machine accuracy, and generating environmental noise. To reduce vibration during the operation of a rotating device, several vibration reduction methods are currently being researched and applied, such as damping techniques, balancing methods, and isolation systems. In [1], the authors designed and subsequently calculated the parameters of a vibration absorber such that its natural frequency coincided with the excitation frequency, thereby allowing vibrations from the device to be transmitted to the absorber and reducing the overall vibration response. In the study on vibration reduction for rotors with protruding shafts [2], the authors investigated the application of additional active magnetic bearings for radial control, which effectively reduced the lateral vibration of the rotor. In [3], the authors proposed a passive control method to reduce vibrations of the UH-60 helicopter rotor. The results showed that the vibration reduction capability improved by 38%. In [4], the authors used aluminum foam fitted to the rotor shaft to increase damping and absorb vibration energy, thereby reducing rotor shaft vibration.

However, vibration damping methods employing passive dampers are typically effective only within a limited range of operating speeds and entail continuous energy consumption during operation. According to [5], the results showed that to be effective across a wide operating speed range, the damping effect needs to be modified to adapt to operating conditions. The change in damping stiffness is actively controlled by a magnetic field applied to the magnetorheological fluid to reduce vibration. The study was conducted using a damper mounted on the Bently Nevada Rotor Kit [6]. In [7], the authors developed a flexible vibration damping model for the rotor using a magnetorheological fluid combined with an on/off control algorithm to actively reduce vibrations. According to [8], the authors analyzed, designed, and tested a rotor model using a damper with magnetorheological fluid controlled by current flowing through coils. The magnetic field generated by the coils acted on the fluid to change the stiffness of the damper. In [9], the authors proposed a model of squeeze film dampers utilizing magnetorheological fluids for rotating systems. The results demonstrated that the damper effectively reduced vibration amplitudes caused by imbalance and enhanced the dynamic stability of the rotor.

Building upon these studies, this paper establishes a mechanical model and analyzes vibration amplitude as a function of frequency in the resonance region. An active vibration damper is designed using a magnetorheological fluid. The composition of the fluid is determined, and the parameters of the magnetic field coil are calculated. The model is fabricated and experimentally tested, and the vibration damping test results are analyzed and evaluated.

II. MODELING AND CALCULATION OF VIBRATION AMPLITUDE

The mechanical vibration model of the rotor is established as an oscillating system subjected to an unbalanced force, as shown in Fig. 1.

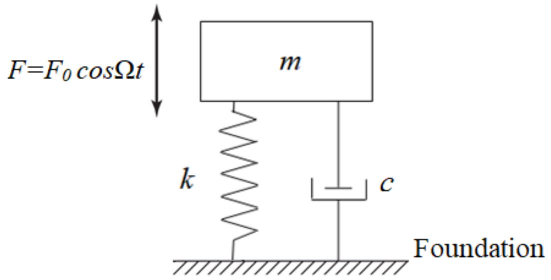


Figure 1. Unbalanced rotor mounted on the isolator

The governing equation of the model, as shown in Fig. 1, can be expressed as follows [8]:

$$m\frac{d^2x}{dt^2}+c\frac{dx}{dt}+kx=F(t), \tag{1}$$

where m is the mass of the system; c is the damping coefficient; k is the spring constant; $F(t)$ represents the excitation force due to rotor rotation, expressed as a harmonic function $F(t)=F_0\cos\Omega t$, where F_0 is the amplitude and Ω is the excitation frequency due to rotor unbalance.

We define the variables as follows: $\omega_0^2=\frac{k}{m}$, $2\beta=\frac{c}{m}$, and $P=\frac{F_0}{m}$. By substituting these into Eq. (1), the equation can be written in the form:

$$\frac{d^2x}{dt^2}+2\beta\frac{dx}{dt}+\omega_0^2x=P\cos\Omega t. \tag{2}$$

By solving Eq. (2), we derive the expression for the amplitude of vibration as follows:

$$X=\frac{P}{\sqrt{(\Omega^2-\omega_0^2)^2+4\beta^2\Omega^2}}. \tag{3}$$

When the rotor system operates at the resonant frequency, with $\Omega_{re}=\sqrt{\omega_0^2-2\beta^2}$, the resulting vibration amplitude can be expressed as:

$$X=\frac{P}{2\beta\sqrt{\omega_0^2-\beta^2}}. \tag{4}$$

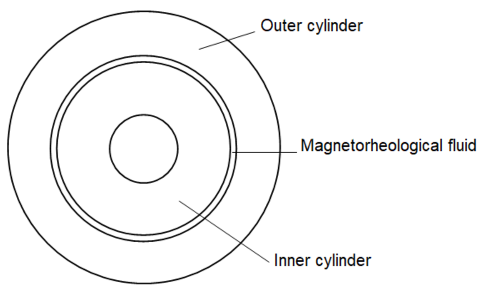
To minimize vibrations caused by the rotor during operation, active vibration control is implemented by adjusting the damping coefficient c and the spring constant k. This adjustment correspondingly changes the distribution of iron particles in the magnetorheological fluid surrounding the bearing under the applied magnetic field. As a result, the vibration amplitude X in Eqs. (3) and (4) can be controlled, thereby minimizing the vibration level of the system.

III. VIBRATION DAMPER DESIGN BASED ON MAGNETORHEOLOGICAL FLUIDS

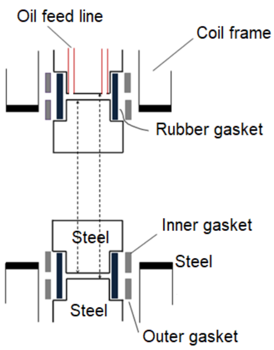
A. *Design of Vibration Damper*

The damper is designed as a controlled bearing, where the application of voltage to two coils generates a magnetic field that interacts with the magnetorheological fluid. This interaction induces a transition of the fluid into a semi-solid state, thereby altering the damping coefficient c and the spring constant k. Mounted at the protruding end of the rotor shaft, the damper is regulated by adjusting the voltage supplied to the coils integrated into the vibration damper. As a result, the vibrational response of the system is reduced, thereby enhancing operational stability.

The main structure of the vibration damper comprises two concentric, nested cylindrical tubes, separated by a 2.5 mm gap and having a length of 20 mm. Rubber gaskets are located at both ends of the tubes, ensuring containment of the magnetorheological fluid within the interstitial gap. The inner surface of the inner cylinder is fixed to the rotor shaft via a ball bearing, while the outer surface of the outer cylinder is secured to the support. The connection between these components forms a support characterized by the variable damping coefficient c and spring constant k. Under controlled excitation, this configuration alters the degree of semi-solid behavior of the magnetorheological fluid. Fig. 2 shows the structural diagram of the vibration damper core based on analytical calculations, and Fig. 3 depicts the design diagram.

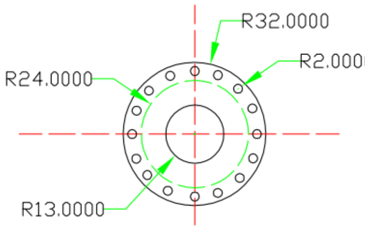


a. Transverse cross-section of the damper core

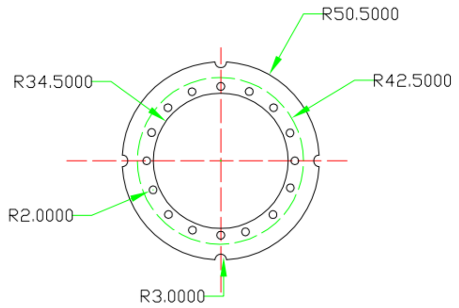


b. Longitudinal cross-section of the damper core

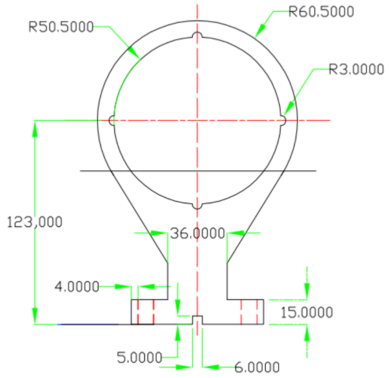
Figure 2. Schematic of the vibration damper core employing a magnetorheological fluid



a. Inner cylindrical tube



b. Outer cylindrical tube



c. Structure of the bearing housing

Figure 3. Schematic design of the vibration damper (dimensions in mm)

B. *Magnetorheological Fluid*

The magnetorheological fluid employed in this study consists of a blended mixture of three components: a carrier fluid, magnetic iron powder, and an additive. Based on experimental testing, the optimized mass ratios are 62% refined spherical iron powder with diameters ranging from 3.0 to 8.0 µm, 33% silicone oil, and 5% stearic acid. When the magnetic field is applied to the magnetorheological fluid mixture, it acts on the iron powder particles, aligning them along the magnetic field lines. At this stage, the fluid transforms into a semi-solid state. Consequently, the damping coefficient *c* and spring constant *k* of the damper increase, thus reducing rotor vibration during operation. The variation of *c* and *k* is controlled by the applied voltage. Furthermore, in the absence of the magnetic field, the fluid instantly returns to its free-flowing state.

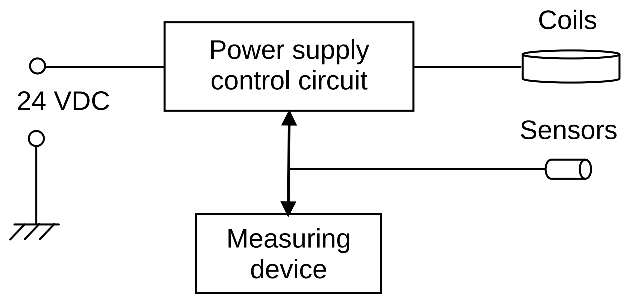
C. *Design and Calculation of a Magnetic Field Coil*

Based on the analysis and literature review [6,7,9], the wire and winding frame dimensions are determined. The winding frame is designed with a ferromagnetic tube core, and the wire diameter is specified in [10]:

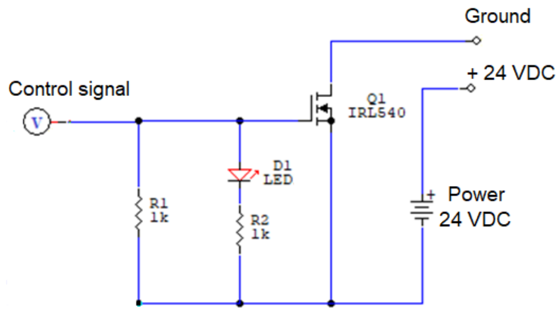
$$d = 2\sqrt{\frac{s}{\pi}} = 2\sqrt{\frac{I}{\pi . J}}, \tag{5}$$

where the current density on the winding is taken as $J = 7\ \text{A/mm}^2$ [6,10], the maximum current is determined to be $I = 2.2\,\text{A}$. From Eq. (5), the wire diameter is obtained as $d = 0.63\,\text{mm}$. In this study, a commercially available wire with a diameter of 0.64 mm was selected, in accordance with the data reported in [6]. Therefore, the magnetic field supply coil operates stably within a current range of 0–1.54 A and is capable of reaching 2 A for short-term limited operation. Based on these parameters, the number of turns was calculated to be 240.

The coils generating the magnetic field are powered by a 24 VDC control circuit, which supplies current when the rotor induces significant vibration and disconnects it when the vibration is low. Sensors measure the vibration of the model and set the on/off threshold of the control circuit, thereby adjusting the damping coefficient *c* and spring constant *k*. Fig. 4 shows the power supply control circuit diagram used to generate the magnetic field for the vibration damper and measurement.



a. Block diagram of the power supply control and measurement circuit



b. Schematic of the power supply control circuit

Figure 4. Power supply control and measurement circuit for the active vibration damper

Fig. 4b illustrates the functions of the circuit components. The IRL540 power MOSFET operates as an electronic switch, which is used to switch the power supply circuit on and off in response to the received control signal from the sensor. Specifically, it connects the negative terminal of the power supply to the ground terminal of the load when the control signal is

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received, and disconnects it when the control signal is absent. Resistor R1 ensures that the gate of the IRL540 remains at 0 V in the absence of the control signal. LED D1 provides a visual indication of circuit operation when the control signal is applied, while resistor R2 limits the current and protects LED D1 during operation.

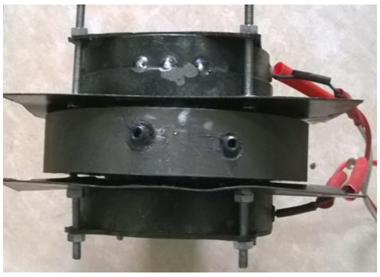
IV. DEVICE FABRICATION AND PERFORMANCE TESTING

A. Device Fabrication

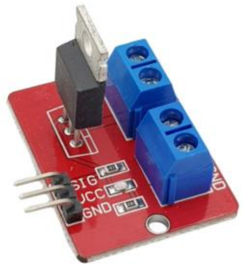
Based on the design calculations, a vibration damper model using magnetorheological fluids was fabricated and tested. Details of the fabricated damper are shown in Fig. 5: Fig. 5a illustrates the core structure; Fig. 5b shows the coil that supplies the magnetic field to the damper; Fig. 5c depicts the complete damper assembly; Fig. 5d presents the control board of the damper system; and Fig. 5e displays the damper installed on the rotor model during testing.

a. Core structure of the damper

b. Magnetic field coil



c. Complete damper assembly



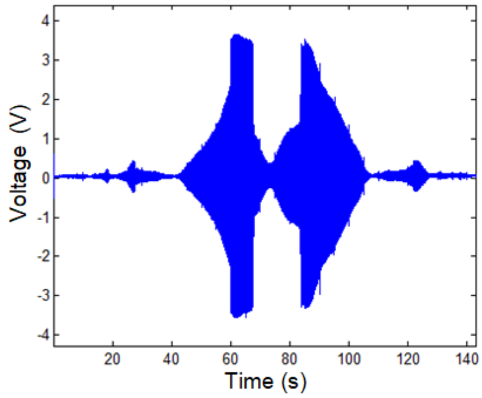
d. Control board

e. Damper installed on the rotor model during testing

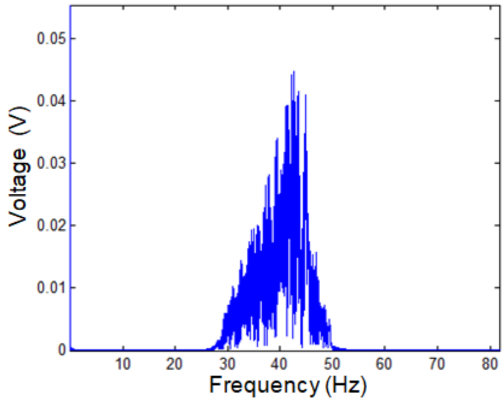
Figure 5. Fabrication and testing setup

B. Testing of Device Operation

The devices used to measure and control vibrations were Omron E2CA-AN4 non-contact proximity sensors and a Picoscope USB oscilloscope 2204A, which was connected to a computer for measurement and data storage. The test results were used to determine the horizontal and vertical vibration levels of the rotor shaft. Vibration levels were analyzed under operating conditions with and without voltage supplied to the damper. Figs. 6–9 present the measured and analyzed vibration voltage graphs of the rotor shaft, comparing conditions with and without power supplied to the damper in the horizontal and vertical measurement directions.

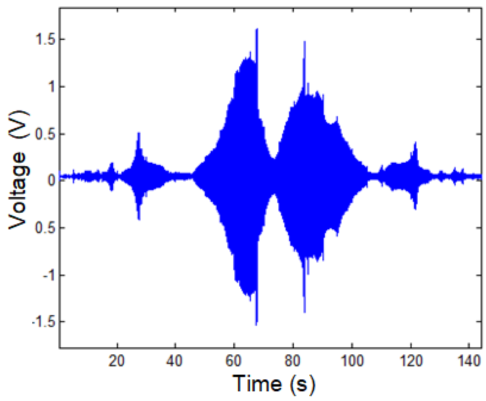


a. Graph of vibration versus time

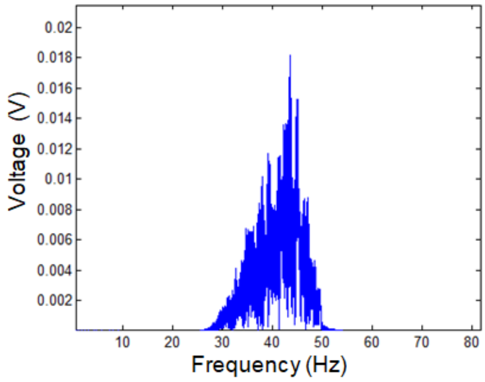


b. Frequency spectrum of vibration

Figure 6. Horizontal vibration voltage measured with the damper unpowered

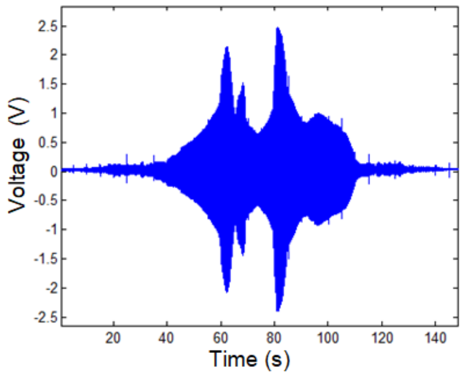


a. Graph of vibration versus time

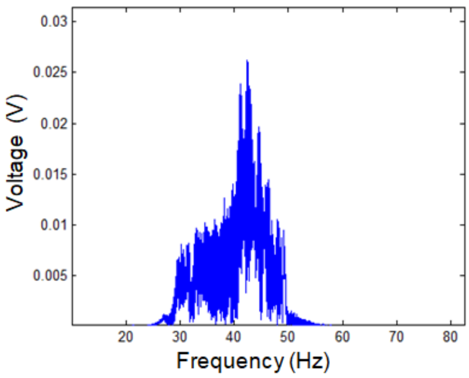


b. Frequency spectrum of vibration

Figure 7. Vertical vibration voltage measured with the damper unpowered

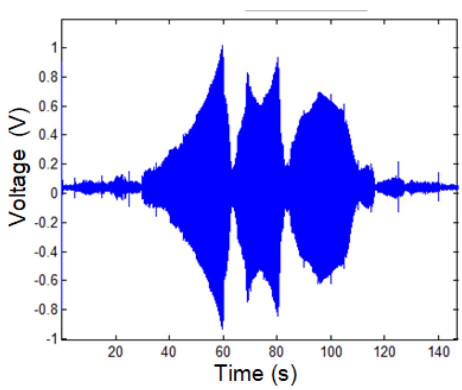


a. Graph of vibration versus time

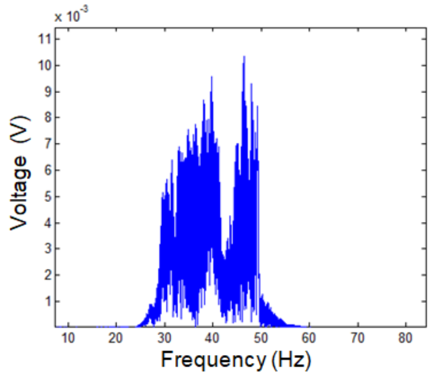


b. Frequency spectrum of vibration

Figure 8. Horizontal vibration voltage measured when power is supplied to the damper



a. Graph of vibration versus time



b. Frequency spectrum of vibration

Figure 9. Vertical vibration voltage measured when power is supplied to the damper

The obtained results indicate that when the 24 VDC voltage is applied, the damping coefficient *c* and spring constant *k* increase. Consequently, the vibration level of the rotor shaft is reduced, with the peak of the horizontal vibration spectrum decreasing from 0.0447 V to 0.0262 V (−41.4%) and the peak of the vertical vibration spectrum decreasing from 0.0181 V to 0.0103 V (−43.1%).

Initial test results indicate that the vibration damping method using magnetorheological fluids demonstrates potential for reducing vibration in rotating devices and civil engineering applications. In particular, for high-speed machinery, during startup and shutdown, the device passes through the resonant frequency region. The vibration damper operates to adjust the damping coefficient *c* and spring constant *k*, thereby reducing vibration in the system, improving vibration stability, and extending the lifespan of the machinery. Active vibration dampers are designed to activate when the system vibration level exceeds the permissible threshold and deactivate when the vibration level is low. Therefore, vibration dampers conserve energy during operation, which is particularly advantageous for devices operating with limited power sources.

V. CONCLUSIONS

In this study, an active vibration damping model was successfully designed and fabricated using magnetorheological fluid composed of 62% spherical iron powder, 33% silicone oil, and 5% stearic acid. Experimental testing on a rotating rotor model demonstrated vibration amplitude reductions of 41.4% in the horizontal direction and 43.1% in the vertical direction when the damper was activated by the magnetic field. The findings confirm the effectiveness of the magnetorheological fluid-based dampers in mitigating vibrations during rotor operation.

The proposed approach validates the feasibility of integrating magnetorheological fluids into vibration control systems and underscores their potential for practical applications in advanced rotating machinery and precision engineering. Future work will focus on optimizing damper design and exploring scalability for industrial applications.

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