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Experimental research on energy dissipation based on damping of magnetic fluid

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Abstract

Energy dissipation of tall building structures suffering frequent violent shaking under strong excitation is a key research topic for the safety of such buildings. In this paper, a new-type tuned magnetic fluid damper with copper balls immersed is proposed to reduce the vibration under different excitation frequencies. First, the natural frequency of the damper was deduced by the Kinetic equations of magnetic fluid. Meanwhile, the viscosity changes of the magnetic fluid were determined by constructing the equivalent damping ratio model. The kinetic energy of magnetic fluid and copper balls were obtained by establishing the finite element simulation model of the magnetic field and carrying out the dynamic finite element simulation experiment about the magnetic fluid. The viscosity of the magnetic fluid decreases as the strength of the magnetic field increases within a certain magnetic field, which is called the magnetic-viscous characteristic [18, 19]. The tuned magnetic fluid damper has a characteristic that the natural frequency varies with the viscosity of the magnetic fluid. Thus, the damper resonates with the tall building structures by controlling the magnetic field [20, 21]. The most ideal damping effect can thereby be achieved [22]. There are many previous researches about damping of magnetic fluid. Beijing Jiaotong University studied the application of magnetic fluid in the field of spacecraft stabilization

1. Introduction

Wind turbines, television towers, long-span transmission towers and other tall building structures are vulnerable to earthquakes and strong winds. The contributing factors of the situation are the high flexibility and low damping of the components located in their higher horizontal positions of the structures [1–3]. As a result, the anti-vibration control of the tall building structures has come to be increasingly important [4]. At present, some anti-vibration systems are already available to reduce vibration [5–7]. The damping of tall building structures has received extensive attention [8].

Tuned magnetic fluid damper is often used in the vibration reduction of tall building structures because of its high stability and quick response to vibrations [9]. Magnetic fluid is a colloidal mixture [10], which is uniformly dispersed in the base carrier fluid by magnetic nanoparticles through the surfactant [11–15], and it has both magnetism and fluidity [16, 17]. The viscosity of the magnetic fluid increases as the strength of the magnetic field increases within a certain magnetic field, which is called the magnetic-viscous characteristic [18, 19]. The tuned magnetic fluid damper has a characteristic that the natural frequency varies with the viscosity of the magnetic fluid. Thus, the damper resonates with the tall building structures by controlling the magnetic field [20, 21]. The most ideal damping effect can thereby be achieved [22]. There are many previous researches about damping of magnetic fluid. Beijing Jiaotong University studied the application of magnetic fluid in the field of spacecraft stabilization...
vibration reduction [23]. Hebei University of Technology studied passive damping based on magnetic fluid and the application of magnetic fluid in frequency modulation and vibration reduction [24–26].

Metallic magnetic materials are considered as superior energy absorbers [27], which can be applied in the field of vibration reduction based on energy dissipation principle. Sapinski of Poland proposed a magnetorheological vibration energy control system based on electromagnetic induction, which was able to achieve energy recovery [28, 29]. A. Slocum of the United States proposed wave energy collectors attached to offshore wind turbines, which proved to be able to reduce dynamic load on the towers as well as recover energy [30]. Although the above methods can recover the vibration energy, further improvements of the vibration reducing function can still be made in tuning active vibration reduction.

In this paper, a new-type tuned magnetic fluid damper with copper balls immersed is proposed based on the magnetic-viscous characteristic of magnetic fluid. Analyzing energy dissipation under the action of the damper through constructing finite element simulation and carrying out vibration reducing experiments is of profound significance to the research on magnetic fluid damper and vibration energy recovery.

2. Structural model and working principle of the damper

The structural model of the tuned magnetic fluid damper proposed in this paper is shown in figure 1. The model consists of an upper top cover, a lower base, three non-magnetic fluid cylinders, an equal amount of magnetic fluid in each fluid cylinder, three copper balls immersed at the bottom of the magnetic fluid, the compartment 1 and compartment 2 that separate three non-magnetic fluid cylinders, and the electromagnetic coils wound around the fluid cylinders. Move the copper balls at the bottom of the fluid cylinders to shake the magnetic fluid in which they are immersed to achieve better effects of vibration reduction. Connect electromagnetic coils with an external power supply to provide the magnetic fluid with changing magnetic fields in the damper. Enlarge the height of the uppermost fluid cylinder and the distance between its lowermost fluid cylinder of the damper and its lower base to
realize the even distribution of the magnetic fields provided by the electromagnetic coils in the magnetic fluid in three non-magnetic fluid cylinders, whereby to improve the accuracy in regulating the viscosity of magnetic fluid.

The dimensions of three fluid cylinders are \( \varphi 40 \text{ mm} \times 40 \text{ mm} \), \( \varphi 40 \text{ mm} \times 20 \text{ mm} \), \( \varphi 40 \text{ mm} \times 20 \text{ mm} \) respectively from top to bottom and the arc-shaped sandwiches are recessed by 3 mm. The dimensions of compartments are \( \varphi 40 \text{ mm} \times 8 \text{ mm} \). The dimensions of the upper top cover and the lower base are \( \varphi 70 \text{ mm} \times 5 \text{ mm} \). The inner radius and outer radius of the electromagnetic coils are 20 mm and 27 mm respectively, the height of the coils is 121 mm, the number of the coils is 600.

The sizes of copper balls in the damper are determined by the equivalent damping ratio. When the vibration frequency of the tall building structure changes, adjust the input current transmitted to the electromagnetic coil to change the viscosity of magnetic fluid till the damper resonates with tall building structures to achieve better damping effects under various vibration frequencies. Most dissipated energy is converted into the kinetic energy of the magnetic fluid and the copper balls during the process of vibration reduction.

3. The establishment and analysis of theoretical models

3.1. Analysis on natural frequency

The natural frequency of the damper is affected by the natural frequency of magnetic fluid and copper balls. The magnetic fluid near the upper layer shakes with the external vibration without the function of copper balls. The oscillating state of magnetic fluid in the three non-magnetic fluid cylinders are the same. The middle non-magnetic fluid cylinder is idealized as a cylindrical structure and is taken to establish the cylindrical coordinate system, where \( r \) is the radial coordinate, \( z \) is the axial coordinate, \( \beta \) is the rotational coordinate, and the applied excitation is \( A \sin(\omega t) \), as shown in figure 2.

The unsteady Bernoulli equation of magnetic fluid is given by:

\[
\rho \frac{\partial \varphi}{\partial t} + \frac{1}{2} \rho \mathbf{v}^2 + \rho gh + P - \mu_0 \int_0^H \mathbf{M} \cdot d\mathbf{H} = \rho A \omega^2 r \cos \beta \sin(\omega t)
\]

where \( \rho \) is the density of the magnetic fluid, \( \varphi \) is the velocity potential of the magnetic fluid, \( \mathbf{v} \) is the velocity vector of the magnetic fluid, \( g \) is the gravitational acceleration, \( h \) is the height of the magnetic fluid, \( P \) is the pressure, \( \mu_0 \) is the permeability of magnetic fluid, \( \mathbf{M} \) is the magnetization vector of the magnetic fluid, \( \mathbf{H} \) is the magnetic field strength vector, \( A \) is the forced amplitude.

Differentiate the equation (1) to get:

\[
\frac{\partial^2 \varphi}{\partial t^2} \bigg|_{z=h} + (g + g_m)(\frac{\partial \varphi}{\partial z}) \bigg|_{z=h} = A \omega^2 r \cos \beta \cos(\omega t)
\]

where \( g_m = -\frac{\mu_0 H_1}{\rho} \frac{\partial H_1}{\partial z} \), \( H_1 \) is the average magnetic field strength on the wave surface of the magnetic fluid, \( \lambda \) is the magnetic susceptibility of the magnetic fluid.

Introducing velocity potential function as:

\[
\varphi(r, z, t) = T(t)F(r, z)
\]

Substitute the equation \( \frac{\partial \varphi}{\partial r} \bigg|_{z=h} = -g \frac{\partial F}{\partial r} \bigg|_{z=h} \) into equation (2) and yield the following equations:

\[
g \frac{\partial F}{\partial z} \bigg|_{z=h} = -g \frac{\partial \varphi}{\partial z} \bigg|_{z=h}
\]
\[ \frac{d^2T}{dt^2} + \nu^2 F_{12-h} = 0 \] (5)

Suppose functions \( T(t) \) and function \( F(r, z) \) in equation (3) as:
\[ T(t) = C_1 \cos(\omega t + \alpha_t) \] (6)
\[ F(r, z) = R(r) \xi(z) \] (7)

The external force on the wave surface of the magnetic fluid can be expressed as:
\[ \Pi = -\frac{\rho \omega^2}{k_i} h \coth(k_i h) \] (8)
\[ k_i = \sqrt{1 - \frac{d^2R}{R \ dr^2}} = \sqrt{\frac{1}{\xi \ dz^2}} = \frac{\gamma}{R} \] (9)

where \( k_i \) is a constant vector. \( \gamma \) is the wavelength of magnetic fluid wave surface movement.

The result of the equation \( \frac{d^2C}{d\tau^2} - k_i^2 \xi = 0 \) is obtained according to equation (9):
\[ \xi(z) = C_1 \cosh(k_1 z) + C_2 \sinh(k_1 z) \] (10)

where \( C_2 = -C_1 \tanh(k_1 h). \)

Substitute the equation (10) into equation (9) and yield the following equation:
\[ R(r) = C_1 \cosh(k_1 r) + C_2 \sinh(k_1 r) \] (11)

Substitute the equations (6)–(7) and (9)–(11) into equation (3) in the case of the initial phase \( \alpha_t = 0 \) and yield the following equation:
\[ \varphi = -\frac{\omega h}{k_i} \coth(k_i h) \cos(\omega t) \] (12)

The natural frequency of the magnetic fluid in an external magnetic field is derived from equations (2), (8), (9), (12) as:
\[ f = \frac{1}{2\pi} \sqrt{\frac{\lambda(g + g_m)}{R} \tanh \left( \frac{\lambda h}{R} \right)} \] (13)

where \( R \) is the horizontal section radius of the fluid cylinder, \( g_m \) is a quantity generated by the electromagnetic field.

The magnetization pressure of the magnetic fluid is given by:
\[ P_m = \mu_0 \int_0^H M \cdot dH = \frac{H_0}{2} \lambda H_1^2 e^{-2\alpha h} \] (14)

where \( \alpha \) is the attenuation rate of the magnetic field.

The magnetic field intensity is derived from equation (14) as:
\[ H = \frac{H_1}{2} e^{-2\alpha h} \] (15)

Derive \( z \) from equation (15) and yield the following equation:
\[ \frac{\partial H}{\partial z} \bigg|_{z=h} = -\alpha H_0 e^{-2\alpha h} \] (16)

The magnetic field strength at the wave surface of the magnetic fluid can be expressed as:
\[ H_i = \frac{B}{\mu_0} = \frac{I}{4R} \] (17)

where \( I \) is the input current to the electromagnetic coil.

The equation of \( g_m \) is solved according to \( g_m = -\frac{\mu_0 H_0 \partial H}{\partial z} \) and equations (15)–(17):
\[ g_m = \alpha \mu_0 \lambda^2 e^{-2\alpha h} \] (18)

3.2. Analysis on equivalent damping ratio of copper ball
An energy analysis model of equivalent damping ratio is constructed to study the influence of the copper balls on the natural frequency of the damper, as shown in figure 3.

The copper ball is affected by viscous damping of magnetic fluid during the process of vibration reduction. Its energy dissipated by the viscous damping in one cycle is expressed by the following equation:
where $C$ is the damping coefficient.

The main viscous damping suffered by the copper ball is the flow damping force, which is given by:

$$ F_d = \frac{C_D \rho S}{2} [R \omega \sin(\omega t)]^2 $$

where $S$ is the surface area of the copper ball, $C_D$ is the coefficient of flow damping force. The equation of $C_D$ is given by:

$$ C_D = \frac{12 \mu_0}{\mu_p R} + \frac{6}{1 + \sqrt{2 \mu_p R/\mu_0}} + 0.4 $$

where $\mu_p$ is the average flow rate of the magnetic fluid.

The energy dissipated by the flow resistance in one cycle of the copper ball is calculated using equation (20) as:

$$ E_d = \frac{4}{3} C_D \rho S \omega^2 R^3 $$

The equivalent damping coefficient of the copper ball is obtained in the case of $E_d = E_0$ and is expressed by the following equation:

$$ C_{eq} = \frac{4}{3 \pi} C_D \rho S \omega R $$

The equivalent damping ratio of the copper ball is derived from equation (23) as:

$$ \zeta_{eq} = \frac{C_{eq}}{2 m_q \omega_q} = 0.215 \frac{\omega}{\omega_q} $$

where $\omega$ is the excitation angular frequency, $\omega_q$ is the natural angular frequency of the copper ball.

The equivalent damping ratio of the copper ball designed for the damper proposed in this paper is given by:

$$ \zeta_{eq} = \sqrt{\frac{3 [m_q/(m_c + m_q)]}{8 [1 + m_q/(m_c + m_q)]}} $$

where $m_c$ is the mass of the magnetic fluid, $m_q$ is the mass of the copper ball.

The ratio of the excitation angular frequency to the natural angular frequency of the copper ball should be in the interval of approximately 1, that is, between 0.9 and 1.1, which not only satisfies the needs of vibration reduction in engineering, but also has certain fault tolerance. Then the value of $\zeta_{eq}$ in equation (25) should be between 0.1935 and 0.2365. It can be determined that the radius of the copper ball should be between 3.6 mm and 4.6 mm under the condition that the horizontal cross-sectional radius of each non-magnetic fluid cylinder is 20 mm and the magnetic fluid height in the fluid cylinder of the damper is 15 mm of the damper. The radius of copper balls in this paper is 4 mm.

4. Simulation and experiment

4.1. Research on magnetic field simulation and magnetic-viscous characteristic

The magnetization characteristic curve of the magnetic fluid is measured with the vibrating sample magnetometer, as shown in figure 4.
Figure 4 shows that the curve passes through the origin of coordinate, which means the magnetic fluid has no residual magnetism when the applied magnetic field is removed. Therefore, there is an exclusive corresponding relationship between the natural frequency of the tuned magnetic fluid damper and the magnetic field.

A magnetic field simulation model is constructed with Comsol simulation package to study the magnetic field distribution in the damper. The material properties are shown in table 1, and the magnetic characteristic parameters of magnetic fluid is acquired based on the magnetization characteristic curve shown in figure 4.

Mesh refinement is conducted for the constructed model to improve the calculation accuracy. Ultra-fine free triangle meshing and sweeping with the maximum element size of 7 mm are carried out on the fluid cylinder and magnetic fluid, and finer free tetrahedral meshing with the maximum element size of 11 mm are carried out on the remaining parts of the damper model. The magnetic field distribution inside the damper is acquired by magnetic field simulation when the input current is 2 A, as shown in figure 5. The conditions that the magnetic induction intensity varies with the radial radius of the fluid cylinder and the height of damper under different currents are obtained, as shown in figure 6.

![Figure 4. Magnetization characteristic curve of magnetic fluid.](image)

![Figure 5. Distribution of magnetic field at I = 2 A.](image)

| Table 1. Material properties. |
|-----------------------------|
| Name                       | Parameter         | Value       |
| Magnetic fluid              | Density [kg m⁻³]  | 1410        |
| Electromagnetic coil        | Conductivity [S m⁻¹] | 5.998e7   |
|                            | Density [kg m⁻³]  | 8700        |
| Fluid cylinder              | Conductivity [S m⁻¹] | 0.1        |
|                            | Density [kg m⁻³]  | 1040        |
|                            | Poisson's ratio   | 0.4         |
|                            | Dielectric strength [kV mm⁻¹] | 13.0 |
|                            | Tensile modulus [MPa] | 2600   |
Figures 5 and 6 show that the magnetic field distribution of magnetic fluid’s positions in the three fluid cylinders are almost the same under three input currents, and the magnetic induction intensity are different. The greater the input current is, the stronger the magnetic field in the fluid cylinder. The changing range of magnetic induction intensity is roughly between 120Gs and 220Gs at the damper’s height of 66 mm when the input current varies from 2 A to 4 A.

The attenuation rates of the magnetic field under different input currents are obtained by the magnetic field simulation. The results of equations (13),(18) are thereby obtained.

The magnetic-viscous experiment of magnetic fluid is carried out by NDJ-5S digital viscometer to study the viscosity change of the magnetic fluid when the magnetic induction intensity varies between 80Gs and 250Gs.

Figures 6 and 7 show that the viscosity of the magnetic fluid increases significantly when the magnetic induction intensity varies from 120Gs to 220Gs, which means the input current of 2A-4A to the electromagnetic coil has a significant regulating effect on the viscosity of the magnetic fluid.

4.2. Experiment on vibration reduction
A vibration experiment is carried out to study the damping effect of the tuned magnetic fluid damper and the damping energy dissipation during the process of vibration reduction. The non-magnetic fluid cylinders and electromagnetic coil frame are made of non-magnetic resin, the horizontal vibration table is made of non-magnetic phosphor copper plate.

A heavy block of the same mass as the damper is put on the horizontal vibration table in the vibration experiment without a damper. It ensures the same mass of the vibration system as a whole in the comparative experiments, and thereby reduces the experimental deviation.
The experimental apparatus is shown in figure 8, which consists of the tuned magnetic fluid damper, horizontal vibration table, exciter, DC power supply, signal acquisition instrument, signal generator, power amplifier, and computer.

The natural frequencies of the damper are calculated when the input current is 2 A, 3 A, and 4 A respectively by equations (13), (18), as shown in table 2.

The vibration frequency of the horizontal vibration table in the experiment is 4.70 Hz according to the experimental measurement results of free vibration attenuation. Therefore, the control input current in the experiment ranges between 2 A and 4 A. A continuous horizontal sinusoidal excitation with the initial amplitude of 8 mm is applied to the horizontal vibration table. The comparative experiments are carried out under three input currents respectively, whose damping effects are shown in figure 9.

Figure 9 shows that the amplitude has a significant attenuation when the horizontal vibration table is equipped with the damper. The average sizes of the amplitudes in the positive direction of vibration at three vibration frequencies are taken to calculate the amplitude attenuation percentages, as shown in table 3.

The total mass of the horizontal vibration table and damper is 3 kg measured with the balance. The absolute values of the slopes of each point in the waveform are taken at the three vibration frequencies above, and the vibrating speeds of the horizontal vibration table without damper and with damper are calculated respectively. As a result, the vibration energy converted by the damper, that is, the amount of the damping energy dissipation is obtained according to the kinetic energy formula, as shown in figure 10.
Figure 10 shows that the damping energy dissipation fluctuates as a whole up and down any time under the applied sinusoidal excitation. The damping effects of the damper under different excitation are verified in a quantitative way and the amount of energy dissipated is obtained by carrying out the vibration reduction experiment.

Table 3. Attenuation of amplitude at three vibration frequencies.

| Input current/A | VIBRATION FREQUENCY/Hz | Amplitude attenuation percentage |
|-----------------|-------------------------|---------------------------------|
| 2.0             | 4.60                    | 10.75%                          |
| 3.0             | 4.69                    | 10.65%                          |
| 4.0             | 4.80                    | 10.50%                          |

Figure 9. Experiments of vibration reduction at three vibration frequencies under continuous excitation.
4.3. Research on dynamics simulation

The sizes of the three fluid cylinders were designed under the condition that the magnetic field distribution of the magnetic fluid area in the three fluid cylinders are the same. The intermediate fluid cylinder is taken as an example, and the kinetic energy of the magnetic fluid and the copper ball is simulated with Comsol simulation package. The material properties are shown in Table 4.

The excitation $A \sin(\omega t)$ is applied. The value of $A$ is 7.15 mm, which is the amplitude of the horizontal vibration table after damping. Ten facets perpendicular to the applied excitation are taken to study the oscillating state of the magnetic fluid and the movement of the copper ball at 1 s under the vibration frequency of 4.60 Hz.

### Table 4. Material properties.

| Name            | Parameter                  | Value                  |
|-----------------|----------------------------|------------------------|
| Magnetic fluid  | Viscosity [mPa · s]        | $f = 4.60$ Hz          |
|                 |                            | 6.00                    |
|                 |                            | $f = 4.69$ Hz          |
|                 |                            | 7.00                    |
|                 |                            | $f = 4.80$ Hz          |
|                 |                            | 8.20                    |
| Copper ball     | Density [kg m$^{-3}$]      | 8700                    |
|                 | Young's modulus [GPa]      | 110                     |
|                 | Poisson’s ratio            | 0.35                    |

Figure 10. Damping energy dissipation under the action of the damper.

Figure 11. The oscillating state of magnetic fluid and the movement of copper ball at 1 s under the vibration frequency of 4.60 Hz.
Figure 12. Kinetic energy of magnetic fluid and copper ball varies with time.

Table 5. Fitting function of energy dissipation.

| $f_v$/Hz | Fitting function                                      |
|----------|-------------------------------------------------------|
| 4.60     | $Q_d = 0.00002807-0.00002544\cos(57.71t) + 0.00000814\sin(57.71t)$ |
| 4.69     | $Q_d = 0.00002783-0.00002249\cos(58.93t) + 0.00001671\sin(58.93t)$ |
| 4.80     | $Q_d = 0.00002664-0.00002199\cos(60.40t) + 0.00001557\sin(60.40t)$ |
| $f_r$/Hz | Fitting function |
|-------|-----------------|
| 4.60  | $Q_k = 0.0002292 - 0.000063 \cos(5.139t) + 0.000035 \sin(5.139t) - 0.0000012 \cos(10.278t) + 0.0000013 \sin(10.278t) - 0.0000042 \cos(15.417t) + 0.0000065 \sin(15.417t) - 0.0000022 \cos(20.556t) + 0.0000096 \sin(20.556t) - 0.0000012 \cos(25.693t) + 0.0000013 \sin(25.693t) - 0.0000035 \cos(30.834t) - 0.0000063 \sin(30.834t)$ |
| 4.69  | $Q_k = 0.0002168 - 0.000066 \cos(5.152t) + 0.000038 \sin(5.152t) - 0.0000002 \cos(10.304t) + 0.0000003 \sin(10.304t) - 0.0000003 \cos(15.456t) + 0.0000006 \sin(15.456t) - 0.0000001 \cos(20.608t) + 0.0000009 \sin(20.608t) - 0.0000033 \cos(25.760t) + 0.0000011 \sin(25.760t) - 0.0000007 \cos(30.912t) - 0.0000004 \sin(30.912t)$ |
| 4.80  | $Q_k = 0.0002076 - 0.000065 \cos(5.233t) + 0.000055 \sin(5.233t) - 0.0000051 \cos(10.466t) + 0.0000072 \sin(10.466t) - 0.0000028 \cos(15.699t) + 0.0000082 \sin(15.699t) - 0.0000036 \cos(20.932t) + 0.0000082 \sin(20.932t) - 0.0000025 \cos(26.165t) + 0.000091 \sin(26.165t) - 0.0000081 \cos(31.398t) - 0.0000035 \sin(31.398t)$ |
state of magnetic fluid and the movement of copper ball at 1 s under the vibration frequency of 4.60 Hz, as shown in figure 11.

The kinetic energy changes of magnetic fluid and copper ball within 1 s under three vibration frequencies are obtained, as shown in figure 12.

Figure 12 shows that the kinetic energy of the magnetic fluid changes more greatly and the amplitude is larger compared to the copper ball at the three vibration frequencies above.

Figure 13. Comparison between total kinetic energy and total energy dissipation.
5. Analysis of damping energy dissipation

The vibration frequency is defined as $f_v$. The fitting functions at three vibration frequencies are obtained according to the curves of the energy dissipation shown in figure 10, as shown in table 5.

The sum of kinetic energy of magnetic fluid and copper balls of three fluid cylinders at three vibration frequencies are calculated respectively according to figure 12. The fitting functions of kinetic energy at three vibration frequencies are obtained, as shown in table 6.

The Cumtrapz function can be applied for continuously accumulating calculations from the first value to the current value. The total energy dissipation within 1 s at three frequencies are calculated respectively based on the Cumtrapz function according to the fitting functions of energy dissipation, as shown by the line with circular marks in the figure 13. The total kinetic energy within 1 s at three frequencies are calculated respectively based on the Cumtrapz function according to the fitting functions of kinetic energy, as shown by the line with triangle marks in the figure 13.

The percentage of the total kinetic energy taken up in the total energy dissipation at three vibration frequencies are obtained when damping lasts for 1 s from figure 13, as shown in table 7.

Table 7 shows that more than 80% of the dissipated energy by means of damping is converted into the kinetic energy of the magnetic fluid and the copper balls while the tuned magnetic fluid damper is used to reduce vibration.

There are boundary layer areas formed near the surface of these three copper balls because the rolling copper balls drive the nearby magnetic fluid to oscillate. The velocity gradients exist between the magnetic fluid in this area and the magnetic fluid that oscillates above, which result in energy losses. Therefore, a part of energy is released in the form of thermal energy.

6. Conclusion

In this paper, a tuned magnetic fluid damper with copper balls is proposed. The damper can be made parameter designs to achieve the purpose of vibration reduction under different vibration frequencies for the tall building structures based on the magnetic-viscous property of magnetic fluid. In the vibration reducing process, most dissipated energy by means of damping is converted into the kinetic energy of the magnetic fluid and the copper balls, and the kinetic energy of the magnetic fluid can be further recycled. Furthermore, the kinetic energy of the magnetic fluid can be converted into electrical energy when the application of magnetic fluid is combined with electrodes and other components. The basis of power generation of magnetic fluid can be provided by the research on damping energy dissipation.

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