Research Article

Numerical Investigation on Suppressing High Frequency Self-Excited Noises of Armature Assembly in a Torque Motor Using Ferrofluid

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Attempting to suppress high frequency self-excited noises of armature assembly, ferrofluid is added to the working clearances of torque motor. The mathematical model of resistance force of ferrofluid applied to armature is derived theoretically from parallel plate squeeze flow theory and ferrofluid constitutive model in which shear thickening and shear thinning effects are taken into account. Then the equivalent physical model of ferrofluid is established according to the resistance force, and, through analysis, it can be further simplified as viscous damping model. Finally, the suppressing effect introduced by ferrofluid on high frequency noises of armature assembly is verified by comparing the results of numerical simulation with ferrofluid and that without ferrofluid.

1. Introduction

Hydraulic valve plays an important role in hydraulic control systems [1]. However, high frequency self-excited vibrations appear frequently during its working process due to vibrations of components in torque motor, unpredictable pressure oscillations in flow field around flapper, fluid-structure coupling, and unmatched design parameters [2–5]. The high frequency self-excited noises can deteriorate the control accuracy or even cause the failure of servo valves. So it is crucial to suppress the high frequency noises of armature assembly in torque motor.

Ferrofluid, also known as magnetic fluid, is a kind of colloidal suspension liquid which has the property of magnetism. The flow characteristic of ferrofluid can be controlled by the magnetic field intensity [6]. Ferrofluid consists of base fluid, surfactant, and magnetic nanoparticles whose diameter size is in the order of 10 nm. Magnetic particles remain stable and can be evenly distributed in the base fluid under the cover of the surfactant [6]. Ferrofluids are widely used

in the areas of sealing, lubrication, grinding, speakers, and shock absorbers [7–11]. In recent years, it has begun to be applied in the fields of sensors and microflow controlling [12, 13]. The magnetoviscous properties of ferrofluid after introducing magnetic field were studied [14, 15]. In the case of different compositions and ratios, there is usually shear thinning phenomenon, where the viscosity of the ferrofluid decreases with the increase of shear rate [16, 17]. However, thorough grasping of working principle and some properties is still far from clear.

In recent years, the application of ferrofluid in hydraulic servo valves was studied in [18, 19]. This paper investigates the high frequency self-excited noises suppression of armature assembly by introducing ferrofluid into a torque motor of electro-hydraulic servo valve. Construction of electrichydraulic servo valve is displayed in Figure 1. It consists of the torque motor, the prestage amplifier of nozzle-flapper, and power-stage amplifier as spool valve. Ferrofluid is added to the working clearances between armature and magnetizers in the torque motor.



FIGURE 1: Structure of electric-hydraulic servo valve with ferrofluid.

The equivalent physical model of ferrofluid is studied through theoretical analysis of resistance force due to ferrofluid acting on the armature. The suppressing effect introduced by ferrofluid on high frequency noises of armature assembly is verified by numerical simulation.

2. Constructing Equivalent Physical Model of Ferrofluid

2.1. Constitutive Model of Ferrofluid. Ferrofluid, magnetorheological fluid, and other suspension fluids are non-Newtonian fluids [20]. There are various kinds of constitutive models which are used to describe different ferrofluids and magnetorheological fluids, such as Bingham model, Casson model, Herschel-Bulkley model, and the model which is a combination of Newtonian fluid and non-Newtonian fluid [20–24]. Many researchers have explained existence of yield stress as the transition from a solid-like (high viscosity) state to a liquid-like (low viscosity) state. This phenomenon happens abruptly at very small shear rate [20]. According to these researches, the constitutive model of ferrofluid is formulated as follows:

$$\begin{aligned} \tau_{yx} &= \eta_y \dot{\gamma}, \quad \tau_{yx} < \tau_1 \\ \tau_{yx} &= \tau_0 + k \dot{\gamma}^n, \quad \tau_{yx} \ge \tau_1. \end{aligned} \tag{1}$$

The constitutive model (1) can be simplified to Bingham model, Herschel-Bulkley model, biviscosity model, and a combination model. In (1), τ and $\dot{\gamma}$ represent the shear stress and shear rate; τ_0 is the dynamic yield stress of ferrofluid, determined by the intensity of magnetic field; τ_1 is the yield stress of ferrofluid; η_v is viscosity of ferrofluid in no-yield

zone; *k* is viscosity coefficient of ferrofluid in yield zone; *n* is power-law index. The relationship between τ_{yx} and $\dot{\gamma}$ is illustrated in Figure 2.

Figure 2 shows that when shear stress τ is smaller than yield stress τ_1 , the ferrofluid lays in no-yield zone, where ferrofluid follows Newton shearing theorem. In this zone, viscosity η_y is independent of shear rate $\dot{\gamma}$ and it is constant. As shear stress τ is larger than yield stress τ_1 , ferrofluid lays in yield zone, where viscosity of ferrofluid is expressed as $\eta = k\dot{\gamma}^{n-1}$. Generally, it is divided into three kinds of situations: when n > 1, ferrofluid shows shear thickening characteristics; As n = 1, viscosity is constant; $\eta = k$; additionally, when n < 1, ferrofluid shows shear thinning characteristics. Dynamic yield stress τ_0 increases with increasing magnetic field intensity H until H reaches a certain value; then continual increase of H hardly contributes to the change of τ_0 , which reaches a stable value. This is because the magnetization of the ferrofluid reaches saturation.

2.2. Mathematical Formulations of Resistance Force. When torque motor operates, armature assembly rotates around its center and the armature continuously squeezes the ferrofluid introduced into the working clearances. The operation mode of ferrofluid can be seen as squeeze mode, as shown in Figure 3.

Because the armature's rotation angle is in a small range when it is experiencing vibrations, the squeeze flow of ferrofluid in the working clearance is simplified as parallel symmetric squeeze flow between two plates as shown in Figure 4. *L* is half of length of the working clearance. *h* is half of the height of the working clearance. V_0 is squeezing velocity of each plate. *H* is the intensity of applied magnetic field.









In phase II, the constitutive model of ferrofluid can be expressed as

$$\begin{aligned} \tau_{yx} &= \eta_y \frac{\partial u_x}{\partial y}, \quad \tau_{yx} < \tau_1 \\ \tau_{yx} &= \tau_0 + k \left(\frac{\partial u_x}{\partial y}\right)^n, \quad \tau_{yx} \ge \tau_1, \end{aligned} \tag{2}$$

where τ_{yx} and u_x represent the shear stress and the squeeze flow velocity at point (x, y) along x direction; $\partial u_x/\partial y$ is shear velocity gradient along y direction. Because of $\partial u_x/\partial y > 0$ and $\tau_{yx} > 0$ in phase II, it is convenient to carry out theoretical analysis in this phase.

Supposing that ferrofluid is incompressible fluid, the squeeze flow of ferrofluid in the clearances between armature and magnetizers is considered to be steady flow. Because of $L \gg h$ and ignoring gravity and inertial force, the velocity and pressure of squeeze flow are regarded as $u_x = u_x(x, y)$ and $u_y = u_y(y)$ and p = p(x).



FIGURE 4: Diagram of parallel symmetric squeeze flow of ferrofluid.

Flow continuity equation is

$$\frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} = 0.$$
 (3)

Momentum conservation equation is

$$\frac{\partial \tau_{yx}}{\partial y} = \frac{\mathrm{d}p}{\mathrm{d}x}.$$
 (4)

Mass conservation equation is

$$\int_0^h u_x \mathrm{d}y = x V_0. \tag{5}$$

According to (2), (3), (4), and (5), the physical parameters of squeeze flow field in phase II are derived. In (4), by integrating variable y under the boundary condition of $\tau_{yx}(y = 0) = 0$, we obtain the following expression:

$$\tau_{yx} = y \frac{\mathrm{d}p}{\mathrm{d}x}.$$
 (6)

In the region $|x| < |x_0|$, ferrofluid lays in no-yield zone. The squeeze flow velocity can be derived as

$$u_{x}(x, y) = -\frac{3V_{0}}{2h^{3}}x(y^{2} - h^{2})$$

$$u_{y}(y) = \frac{3V_{0}}{2h^{3}}\left(\frac{1}{3}y^{3} - h^{2}y\right).$$
(7)

The pressure gradient is expressed as

$$\frac{\mathrm{d}p_1}{\mathrm{d}x} = -\frac{3\eta_y V_0}{h^3} x. \tag{8}$$

Taking (8) to (6) under the boundary condition of $\tau_{yx}(x = x_0, y = h) = \tau_1$, we obtain the following expression:

$$x_0 = -\frac{\tau_1 h^2}{3\eta_v V_0}.$$
 (9)

By integrating dp_1/dx , $p_1(x)$ is formulated as follows:

$$p_1(x) = -\frac{3\eta_y V_0}{2h^3} x^2 + C,$$
(10)

where C is constant of integration.

As pressure distribution in the flow field is continuous, the expression of $p_1(x)$ is derived under the boundary condition of $p_1(x_0) = p_2(x_0)$:

$$p_{1}(x) = -\frac{3\eta_{y}V_{0}}{2h^{3}}x^{2} - \frac{3kV_{0}}{2h^{3}}(x_{0}^{2} - L^{2}) + \frac{3\eta_{y}V_{0}}{2h^{3}}x_{0}^{2} + \frac{3\tau_{0}}{2h}(x_{0} + L) - \tau_{xx},$$
(11)

where τ_{xx} is the normal stress along *x* direction at flow field boundary.

The resistance force applied to the armature is

$$F_1 = 2 \int_{x_0}^0 p_1(x) W \, \mathrm{d}x. \tag{12}$$

In the region $|x| > |x_0|$, as $0 \le y < y_1$, ferrofluid lays in yield zone. As $y_1 < y \le h$, ferrofluid lays in no-yield zone, where $y_1 = \tau_1 (dp_2/dx)^{-1}$.

The squeeze flow velocity can be derived as

$$u_{x}(x,y) = \begin{cases} \frac{n}{n+1}k^{-1/n} \left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}\right)^{-1} \left[\left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}y - \tau_{0}\right)^{(n+1)/n} - \left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}h - \tau_{0}\right)^{(n+1)/n} \right], & y_{1} < y \le h \\ \frac{1}{2\eta_{y}} \frac{\mathrm{d}p_{2}}{\mathrm{d}x} \left(y^{2} - y_{1}^{2}\right) + \frac{n}{n+1}k^{-1/n} \left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}\right)^{-1} \left[\left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}y_{1} - \tau_{0}\right)^{(n+1)/n} - \left(\frac{\mathrm{d}p_{2}}{\mathrm{d}x}h - \tau_{0}\right)^{(n+1)/n} \right], & 0 \le y < y_{1}. \end{cases}$$
(13)

The implicit expression of pressure gradient is

$$\begin{bmatrix} -\frac{\tau_1^3}{3\eta_y} + \frac{n}{n+1} k^{-1/n} \tau_1 (\tau_1 - \tau_0)^{(n+1)/n} \\ -\frac{n^2}{(n+1)(2n+1)} k^{-1/n} (\tau_1 - \tau_0)^{(2n+1)/n} \end{bmatrix} \left(\frac{\mathrm{d}p_2}{\mathrm{d}x}\right)^{-2} \\ -\frac{n}{n+1} k^{-1/n} h \left(\frac{\mathrm{d}p_2}{\mathrm{d}x}\right)^{-1} \left(\frac{\mathrm{d}p_2}{\mathrm{d}x}h - \tau_0\right)^{(n+1)/n}$$
(14)
$$+\frac{n^2}{(n+1)(2n+1)} k^{-1/n} \left(\frac{\mathrm{d}p_2}{\mathrm{d}x}\right)^{-2} \left(\frac{\mathrm{d}p_2}{\mathrm{d}x}h\right)^{-2} \\ -\tau_0 \right)^{(2n+1)/n} = V_0 x.$$

According to the experimental research of viscosity, the power-law index *n* of the applied ferrofluid is approximated to 1. After simplifying the equation above, dp_2/dx is expressed as follows:

$$\frac{\mathrm{d}p_2}{\mathrm{d}x} = -\frac{3kV_0}{h^3}x + \frac{3\tau_0}{2h}.$$
 (15)

By integrating dp_2/dx , $p_2(x)$ is derived as follows:

$$p_2(x) = -\frac{3kV_0}{2h^3} \left(x^2 - L^2\right) + \frac{3\tau_0}{2h} \left(x + L\right) - \tau_{xx}$$
(16)

The resistance force applied to the armature is

$$F_2 = 2 \int_{-L}^{x_0} p_2(x) W \,\mathrm{d}x. \tag{17}$$

The total resistance force applied to the armature is expressed as

$$F = F_{1} + F_{2} = \frac{2\left(k - \eta_{y}\right)V_{0}Wx_{0}^{3}}{h^{3}} + \frac{2kV_{0}WL^{3}}{h^{3}} + \frac{3\tau_{0}W\left(L^{2} - x_{0}^{2}\right)}{2h} + 4LW\eta_{y}\left|\frac{-3kV_{0}\tau_{1}^{2}}{\eta_{y}}\left(\frac{3kV_{0}L}{h^{2}} + \frac{3\tau_{0}}{2}\right)^{-3} + \frac{V_{0}\tau_{1}\left(\tau_{1} - 2\tau_{0}\right)}{6h}\left(\frac{kV_{0}L}{h^{2}} + \frac{\tau_{0}}{2}\right)^{-2} + \frac{3V_{0}}{2h}\right|,$$
(18)

where W is width of the working clearance.

$$h = H_0 - \frac{X}{2},$$
 (19)



FIGURE 5: Resistance forces acting on the armature at different velocities.

where H_0 is half of the height of the working clearance when armature is at zero position. *X* is the squeezing displacement of the armature.

The final expression of total resistance force is derived by taking (9) and (19) to (18):

$$F = \frac{\tau_1^{\ 2}W\left(2H_0 - X\right)^3 \left[4\left(\eta_y - k\right)\tau_1 + 9\eta_y\tau_0\right]}{432\eta_y^{\ 3}V_0^{\ 2}} \\ + \frac{4WL^2\left(4kV_0L + 3\tau_0h^2\right)}{\left(2H_0 - X\right)^3} \\ + 4LW\eta_y \left|\frac{-3kV_0\tau_1^{\ 2}}{\eta_y}\left(\frac{12kV_0L}{\left(2H_0 - X\right)^2} + \frac{3\tau_0}{2}\right)^{-3} \right]$$
(20)
$$+ \frac{V_0\tau_1\left(\tau_1 - 2\tau_0\right)}{3\left(2H_0 - X\right)}\left(\frac{4kV_0L}{\left(2H_0 - X\right)^2} + \frac{\tau_0}{2}\right)^{-2} \\ + \frac{3V_0}{2H_0 - X} \right].$$

2.3. Equivalent Physical Model of the Ferrofluid. In (20), k, η_y , τ_0 , and τ_1 are ferrofluid viscosity parameters obtained by ferrofluid viscosity experiment. L, W, and H_0 are dimensional parameters of torque motor. The resistance forces applied to the armature are derived by using (20) at different squeezing velocities. The results are shown in Figure 5.

Figure 5 shows that the resistance force applied to the armature increases linearly with squeezing displacement at given squeezing velocity. In addition, as the squeezing speed increases, the resistance force is also increasing. From Figure 5, it can be concluded that the squeezing displacement



FIGURE 6: The equivalent physical model of ferrofluid.



FIGURE 7: The spring stiffness and damping coefficient versus armature squeezing velocity.

and the squeezing velocity are crucial for the resistance force introduced by ferrofluid.

Therefore, the resistance force applied to the armature after introducing the ferrofluid can be formulated as

$$F = K \cdot X + B_c \cdot V_0, \tag{21}$$

where *K* is equivalent spring stiffness; B_c is equivalent damping coefficient. *K* and B_c are related to the viscosity parameters of the ferrofluid, dimensional parameters of working clearance of torque motor, and the squeezing velocity.

Equation (21) means that the effect of the ferrofluid between armature and magnetizer can be equivalent to the spring and damper connected in parallel, as shown in Figure 6.

The equivalent spring stiffness K and damping coefficient B_c introduced by ferrofluid are affected by armature squeezing velocity, as shown in Figure 7.

Figure 7 shows that the equivalent spring stiffness K introduced by ferrofluid increases linearly with armature squeezing velocity. However, the equivalent damping coefficient B_c introduced by ferrofluid decreases dramatically with armature squeezing velocity. The results show that the squeezing speed has positive effect on spring stiffness K and adverse effect on damping coefficient B_c .

TABLE 1: K and B_c introduced by ferrofluid at each resonant frequency.

Resonant frequency (Hz)	Resonant peak (µm)	Squeezing velocity (m/s)	<i>K</i> (N/m)	B_c (Ns/m)
470	40.473	0.0761	86.72	0.5371
1140	62.694	0.2859	339.51	0.5317
2580	3.314	0.0342	36.47	0.5447
3585	1.42	0.0204	21.95	0.5531

TABLE 2: The results of modal analysis of armature assembly with and without ferrofluid.

<i>K</i> (N/m)	1st order (Hz)	4th order (Hz)	6th order (Hz)	8th order (Hz)	9th order (Hz)	Other orders
0	590.6	1127.9	2595.4	3705.4	3734.6	
21.95	591.2	1128.1	2594.4	3708.3	3735.4	Unchanged
36.47	591.8	1128.2	2594.5	3708.5	3735.4	Unchanged
86.72	593.7	1128.4	2594.6	3709.3	3737.6	Unchanged
339.51	603.1	1129.7	2595.1	3713.2	3736.7	Unchanged

3. Numerical Investigation on Dynamic Behavior of Armature Assembly

3.1. Modal Analysis of Armature Assembly without Ferrofluid. In order to examine the dynamic characteristics of the armature assembly, modal analysis of the armature assembly is carried out in the range of 0~5000 Hz. Total 9 vibration modes and the corresponding natural frequencies are obtained. The 1st-order, 4th-order, 6th-order, 8th-order, and 9th-order vibration modes, as shown in Figure 8, are associated with high frequency vibrations in the working plane.

The 1st vibration mode is the swing movement around rotating center of spring tube in the working plane; the maximum displacement occurs at the feedback rod end part. The 4th-order, 6th-order, and 9th-order vibration modes are bending motion of the armature assembly in the working plane. The maximum displacement also occurs at the feedback rod end part. The 8th-order vibration mode is the warping motion of armature; maximum displacement occurs at the armature end. The high frequency sympathetic vibration of feedback rod end part is mainly affected by the 1st-order, 4th-order, 6th-order, and 9th-order vibration modes and corresponding natural frequencies.

3.2. Modal Analysis of Armature Assembly with Ferrofluid. Through the experiment of dynamic characteristics, four resonant frequencies in the range of 0~5000 Hz and the corresponding resonant peaks of armature assembly with ferrofluid are measured. Average armature squeezing speed can be obtained at each resonant frequency. Then the corresponding *K* and B_c introduced by ferrofluid can be obtained, respectively, at each squeezing speed using parallel plate squeezing theory. The results are shown in Table 1.

The equivalent physical model and finite element model of armature assembly with ferrofluid are shown in Figure 9.

Ferrofluid is simulated using spring-damper elements in Figure 9(b). Four holes on the flange of armature assembly are fixed. The results of modal analysis of armature assembly using different spring stiffness *K* introduced by ferrofluid are compared to the results of being without ferrofluid in Table 2.

Table 2 shows that 1st-order, 4th-order, 6th-order, 8thorder, and 9th-order natural frequencies increase slightly with the increase of K. It is because spring stiffness Kintroduced by ferrofluid raises the structure stiffness of working plane in which these vibration modes would appear. Compared to the natural frequencies of being without ferrofluid, the biggest change of natural frequency occurs at the first order when K is equal to 339.51 N/m. The difference of frequency is 12.5 Hz, and the relative change is 2.12%. Based on the above analysis, it can be known that several natural frequencies increase due to introducing ferrofluid into the working clearances of torque motor. But the increment is very small; the influence of K introduced by ferrofluid can be ignored in modal analysis.

3.3. Further Simplifying Equivalent Physical Model of Ferrofluid. In order to study the suppression of high frequency self-excited noises of the armature assembly, harmonic response analysis of armature assembly is carried out by method of using different ferrofluid parameters (in Table 2) in different frequency domains. The frequency of exciting electromagnetic force applied to the armature ranges from 0 Hz to 5000 Hz. Also the simulation is carried out in the case of K = 0 N/m and $\overline{B}_c = 0.5416$ Ns/m (\overline{B}_c is the average value of $B_{c1} \sim B_{c4}$). The dynamic analyzing results of feedback rod end part are compared in Figure 10.

Figure 10 shows that the biggest gap of resonant peaks between the results of two different methods occurs at the first-order natural frequency of feedback rod, and the difference is smaller than 6μ m. When the frequency of exciting force is larger than 800 Hz, the results based on the two different methods are basically the same. According to the above results of the simulation, the equivalent physical model of ferrofluid introduced into the working clearances can be further simplified as viscous damping model, as shown in Figure 11, and the damping coefficient \overline{B}_c is equal to 0.5461 Ns/m.

3.4. Comparison of Harmonic Response Analysis Results. Then harmonic response analysis of armature assembly is



FIGURE 8: The vibration modes associated with the vibrations in the working plane.

Without ferrofluid		With fe	Amplitude dropped by		
Resonant frequency (Hz)	Resonant amplitude (μ m)	Resonant frequency (Hz)	Resonant amplitude (μ m)	miphtude dropped by	
590	320.02	590	66.35	79.27%	
1125	149.22	1120	92.54	37.98%	
2605	6.83	2610	5.81	14.93%	
3725	7.08	3710	5.26	25.71%	

TABLE 3: Simulation results of dynamic characteristics of armature assembly.



FIGURE 9: Armature assembly with ferrofluid.



FIGURE 10: Harmonic response of feedback rod end part with ferrofluid.

carried out and the results of numerical analysis with and without ferrofluid are displayed in Figure 12.

It can be seen from Figure 12 that sympathetic vibration of the armature assembly may happen during the process of harmonic excitation. The resonant peaks occurred at the 1st-order, 4th-order, 6th-order, and 9th-order natural



FIGURE 11: The simplified physical model of ferrofluid.

frequencies, respectively. The amplitudes of the feedback rod end part at four resonant peaks are displayed in Table 3.

Table 3 shows that, after adding ferrofluid, the amplitudes of resonant peaks are all reduced compared to the situation without ferrofluid. The reduction is from 14.93% up to 79.27%. Amplitudes of high frequency sympathetic vibrations are suppressed obviously because the ferrofluid introduced into the working clearances acts as a damper in the system.

From the above simulation analysis, it can be concluded that introducing ferrofluid can significantly reduce the amplitudes of resonant peaks of the armature assembly.

Considering that different types of ferrofluid are added in the working clearances and assuming that the equivalent viscous damping coefficient \overline{B}_c changes from 0.1 Ns/m to



FIGURE 12: Harmonic responses of end part of feedback rod.



FIGURE 13: Influence of viscous damping coefficient \overline{B}_c on harmonic response.

5 Ns/m, the harmonic responses of feedback rod end part are shown in Figure 13.

In Figure 13, the different damping effects on high frequency noises are investigated through introducing different types of ferrofluid into the working clearances. It also shows that high frequency noises of feedback rod end part are reduced with the increase of \overline{B}_c . Damping ratio ξ of the system becomes larger with the increase of the viscous damping coefficient \overline{B}_c . When \overline{B}_c reaches 3 Ns/m, the high frequency noise at the 1st-order natural frequency will vanish. It is most likely because the damping ratio ξ is larger than 1 as $\overline{B}_c \ge$ 3 Ns/m and the system is an overdamped system.

4. Conclusions

Numerical studies have shown that the amplitudes of high frequency resonant peaks reduce significantly after introducing the ferrofluid into the working clearances of torque motor. It means that ferrofluid can introduce a damping force to torque motor and can suppress high frequency self-excited noises of armature assembly in torque motor. By selecting proper kind of ferrofluid, there is a strong possibility of making the certain high frequency self-excited noises in the servo valve disappear completely. Therefore, ferrofluid can be used to increase the stability and control accuracy of hydraulic servo valve.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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