Optimal Design of Machine Tool Vibration Reduction Based on Magneto-rheological Damper

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Abstract. This paper takes the three-way redundant drive parallel machine tool as research object, designs a magneto-rheological damper with displacement amplification function and simulates the magnetic circuit saturation of the damper by the Ansys workbench finite element analysis software, so that the output of the damper comes to optimization. The research results show that the magnetic field distribution is more reasonable, and the mechanical properties of the damper are improved. The optimization method is feasible and effective.

1. Introduction

In the process of machine tool cutting, there will be varying degrees of vibration, which will not only affect the quality of the processed workpiece, but also aggravate the wear of the tool and generate a certain amount of noise. At present, the methods of machine tool vibration reduction are classified according to the method of suppressing chattering, which can be summarized into the following three categories: vibration control methods using vibration absorbers or additional devices, optimized design or structural improvements, and online monitoring and adjustment of cutting parameters [1]. Vibration control methods can be divided into active, semi-active and passive control methods according to different actuators [2]. At present, the vibration reduction of machine tools mostly uses magneto-rheological (MR) dampers to suppress the vibration of the machine tools. Magneto-rheological fluid is used as the viscous medium in the MR dampers [3]. Magneto-rheological fluid was first proposed by the American scholar Rabinow.J in 1948. It is a controllable fluid. It is made of small soft magnetic particles with high permeability and low hysteresis and a suspension made of non-magnetic liquids. This kind of suspension liquid is characterized by low viscosity and Newtonian fluid characteristics under the condition of no magnetic field; under the action of strong magnetic field, it has the characteristics of Bingham fluid with high viscosity and low fluidity [4]. In recent years, the optimization design of magneto-rheological damper has received extensive attention. Based on the Bingham parallel plate model and considering the optimization of the magnetic circuit, DING Y et al. proposed a simplified design method for a shear valve type magneto-rheological damper [5]; YAIIDIMI et al. used the finite element method to simulate the magnetic field generated by the electromagnetic coil in the magneto-rheological damper, and designed and manufactured the MR damper based on the simulation results [6].

By analyzing the vibration characteristics and force of the parallel machine tool, a magneto-rheological damper with displacement amplification function is designed, the output model of the
Damper is established, and the Ansys Maxwell finite element analysis software is used to analyze the magnetic circuit of the damper. Carry out simulation and adjust the structural parameters of the MR damper to ensure that the magnetic circuit of the piston core is not saturated under the premise that the MRF magnetic circuit at the damping channel is saturated, so that the damping output of the damper is optimized, and the optimal design of the damper is completed.

2. Analysis of machine tool vibration characteristics

2.1. Force analysis of parallel machine tool

The three-dimensional solid model and structure diagram of the parallel machine tool with three translational redundant drive (3-2SPS) are shown in Figure 1. The machine tool mainly includes a fixed platform, a redundant drive linear module, a moving platform and 3 sets of telescopic rods.

![Figure 1. 3-2SPS parallel machine tool three-dimensional model and structure diagram](image)

To study the vibration characteristics of the machine bed, it is necessary to analyze the force of the machine bed. The main technical parameters of the 3-2SPS parallel machine tool are shown in Table 1.

| Technical Parameters                  | Performance index | Parameter value |
|---------------------------------------|-------------------|-----------------|
| C-shaped gantry radius R/mm           |                   | 1080            |
| Processing range /mm                  | X axis            | -1000~1000      |
|                                       | Y axis            | -900~900        |
|                                       | Z axis            | -550~550        |
| Spindle motor power /kW               |                   | 7.5             |
| Spindle speed range /(r/min)          |                   | 0~4000          |

Refer to the metal cutting calculation manual [7], the calculation formula of milling force is

\[
P_m = \frac{F_z \cdot V}{60 \times 102 \times 10}
\]  

(1)

In the formula

\[P_m\] – Milling power, kW;
\[V\] – Milling speed, m/min;
\[F_z\] – Circumferential milling force, N;

Preliminarily given processing parameters, the rated power of the motor is 7.5 kW, and the machine tool is in a certain milling state, and the milling speed \(V\) is 135 m/min. According to formula (1).

\[
F_z = \frac{60 \times 102 \times 10 \times P_m}{V} = \frac{60 \times 102 \times 10 \times 7.5}{V} = 3400(N)
\]

According to some research work in the early stage of the mechanism, the resistance force \(F_h\) and the axial force \(F_o\) taken away are:

2
\( F_h = 0.4F_z = 1360\, \text{(N)} \)
\( F_o = 0.5F_z = 1700\, \text{(N)} \)

Then the resultant force \( F \) is

\[
F = \sqrt{F^2_z + F^2_h + F^2_o} \approx 4000\, \text{(N)}
\]

2.2. Analysis of vibration characteristics of parallel machine tool

Based on the above analysis of the force of the machine tool, a method to study the vibration characteristics of the machine tool is formulated, that is, the hammer method is used to simulate the vibration during the working process of the parallel machine tool, and the CCD high-speed photography method is used for data collection to test the vibration parameters of the machine tool during the working process. And use Matlab for data processing, output the measured vibration parameters of the machine tool as the vibration curve of the machine tool, and obtain the vibration period and amplitude of the machine tool. The vibration curve is shown in Figure 2.

![Machine tool vibration curve](image)

By analyzing the vibration curve of the machine tool, it can be known that the vibration of the machine tool is a superposition of vibrations of different frequencies. The maximum vibration period is 26.9 ms and the maximum amplitude is 0.16 mm. It can be seen that the vibration characteristics of the machine tool are high frequency and low amplitude, and from the vibration curve of the machine tool, the vibration attenuation is very slow. The main reason is that the damping of the machine tool is small. Therefore, it is necessary to design a corresponding damper for the machine tool to increase the damping of the machine tool and accelerate the machine tool. Attenuation of vibration.

3. Damper design

3.1. Magneto-rheological damper performance requirements and installation location

1) Damper performance requirements  According to the above analysis of the vibration characteristics of the machine tool, it can be seen that the vibration characteristics of the machine tool are high frequency and low amplitude, and the magneto-rheological damper can produce a large damping force under high frequency vibration. Therefore, the type of damper is selected as the magnetorheological damper; the amplitude of the machine tool is small, so the designed damper needs to have a certain displacement amplification function.

2) Installation position of damper  The dampers are installed at the positions 1, 2, and 3 shown in Figure 3, that is, the connection between the moving branch chain and the moving platform. Installing the damper here can effectively accelerate the attenuation of vibration from the machine tool spindle and branch chain.
3.2. Structural design of the damper

According to the above determination of the performance index of the damper, the damper needs to have the function of displacement amplification, and its specific structure is shown in Figure 4.

![Figure 4. Magneto-rheological damper structure](image)

The damper is composed of 8 parts in Figure 4, in which the material of the cylinder is 45 steel; the piston is equipped with an excitation coil and a fixed rack structure. The piston divides the cylinder into two chambers on the left and right, both of which are installed There is a magneto-rheological fluid; the large and small gears have the same modulus, and the number of teeth of the large gear is twice that of the small gear. The two gears are connected by a keyed shaft. The specific dimensional parameters of the gear are shown in Table 2; The left half is a cylindrical structure, and the right half is a flat structure with a rack. The output rod and the end cover prevent the magneto-rheological fluid from flowing out of the box through a dynamic sealing structure; there is a small gap between the piston and the cylinder. When a high-viscosity liquid passes through the gap, it will produce a force that hinders the flow, which is the damping force.

| Number of gears | Modulus | Gear width | Gear top height | Gear root height |
|-----------------|---------|------------|-----------------|------------------|
| gear            | 36      | 2          | 5               | 2                |
| Big gear        | 18      | 2          | 5               | 2.5              |

The designed damper is a shear valve type magnetorheological damper. Since the large and small gears have the same speed, the theory of gear transmission shows that this damper has a displacement amplification function, which can amplify the amplitude of vibration, so that The piston of the damper produces a larger displacement during the vibration process, thereby generating a larger damping force, and achieving the effect of suppressing the vibration of the machine tool.

Reasonable design of the size of the damper plays an important role in improving the vibration damping performance of the magnetorheological damper. The initial structure parameters of the designed magnetorheological damper are shown in Table 3.
Table 3. Structural parameters of the damper  

| Structural parameters       | Unit: mm |
|-----------------------------|----------|
| Magnetic core radius        | 15       |
| Damping channel clearance   | 1        |
| Piston diameter             | 80       |
| Effective length of piston  | 40       |
| Thickness of Cylinder       | 5        |

3.3. Characteristic analysis of the damper

Based on the parallel plate theory and Bingham shear model, the annular gap between the piston and the cylinder of the damper can be approximated as a model of two parallel plates [8].

The relationship between the shear stress of the fluid in the magneto-rheological fluid unit and the pressure gradient is

$$\frac{d\tau}{dy} = \frac{dp}{dx} = p'$$

As shown in Figure 5(a).

![Figure 5(a)](image)

(a) Unit fluid forces

![Figure 5(b)](image)

(b) Valve-type velocity distribution

Figure 5. The force and velocity of the fluid between parallel plates

As shown in Figure 5(b), zone 1 and zone 2 are the yield zone, zone C is the un-yield zone, and the damping force of the designed damper can be expressed as

$$F = \frac{12\eta L A_p^2}{\pi D h^3} v + \frac{3L \tau_y A_p}{h} sgn v$$

(2)

In the formula:
- $\eta$ – Zero field viscosity of MRF;
- $A_p$ – The area of the piston;
- $L$ – Effective length of piston;
- $h$ – Clearance of damping passage between piston and cylinder;
- $D$ – The outer diameter of the piston;
- $v$ – The speed of the piston relative to the cylinder;

From the damping force formula shown in equation (2),

$$M = \frac{F}{F_v} = \frac{F_t + F_f + F_f}{F_t + F_f}$$

(3)

In the formula:
- $F$ – Total damping force;
- $F_f$ – Mechanical friction;
- $F_v$ – Damping force without applied current;
The dynamic adjustable coefficient $\lambda$ can be expressed approximately by the ratio of Coulomb force to viscous force:

$$\lambda = \frac{F_c}{F_\eta} = \frac{\tau_y Dh^2}{\eta(D^2 - d^2)v}$$

(4)

From equation (2)-equation (4), the amplitude modulation range of magnetorheological damping force $M$ and $\lambda$ is closely related to the size of each part of the damper. It can be seen from equation (4) that the dynamic adjustable range of the damper is proportional to the gap $h$ of the damping channel.

3.4. Magnetic circuit design

The design of the magnetic circuit of the magneto-rheological damper has a great influence on the maximum damping output and the dynamic adjustable range of the damper. It is a key issue in the design of the damper. The magnetic circuit structure of the designed magneto-rheological damper is shown in Figure 6.

![Magnetic circuit structure](image)

Figure 6. Magneto-rheological damper magnetic circuit structure

Table 4 shows the initial design dimensions of each part of the structure on the piston of the magneto-rheological damper.

| h1  | h2  | h3  | d1  | d2  | L   | R   |
|-----|-----|-----|-----|-----|-----|-----|
| 10  | 20  | 10  | 1   | 5   | 10  | 40  |

According to Ohm’s law of the magnetic circuit, the formula for calculating the magnetic circuit of the damper can be known:

$$NI = \phi\left(\frac{L'}{\mu_1S_1} + \frac{2h}{\mu_0S_0}\right)$$

(5)

In the formula
- $N$ – Number of turns of excitation coil;
- $\phi$ – Total magnetic flux of the loop;
- $I$ – Maximum current;
- $L'$ – Average length of magnetic circuit;
- $h$ – Width of the damping channel;
- $S_1$ – The average cross-sectional area of the magnetic circuit;
- $S_0$ – Average cross-sectional area at the damping channel;
- $\mu_1$ – Permeability of air;
- $\mu_0$ – Permeability of magnetic core;
4. Optimal design of dampers

4.1. Optimization model

In the initial design of the magneto-rheological damper, the material used for the magnetic ring of the excitation coil is electrical pure iron, the cylinder is made of 45 steel, the model of the magneto-rheological fluid is 132DG, the magnetic induction intensity and magnetic field of each part of the material. The relationship of intensity is shown in Table 5-7.

Table 5. The relationship between magnetic induction intensity and magnetic field intensity of magneto-rheological fluid

| Parameter | Numerical value |
|-----------|-----------------|
| H/(kA/m)  | 30 60 100 120 140 |
| B/T       | 0.08 0.30 0.50 0.55 0.62 |

Table 6. The relationship between magnetic induction intensity and magnetic field intensity of 45 Steel

| Parameter | Numerical value |
|-----------|-----------------|
| H/(kA/m)  | 164 245 365 545 813 |
| B/T       | 0.2 0.4 0.6 0.8 1.0 |

Table 7. The relationship between magnetic induction intensity and magnetic field intensity of Electrician pure iron

| Parameter | Numerical value |
|-----------|-----------------|
| H/(kA/m)  | 21.2 42 63.2 84.8 104 |
| B/T       | 0.2 0.4 0.6 0.8 1.0 |

4.2. Optimization method

From the above analysis of the damping characteristics of the magnetorheological damper, it can be seen that the piston diameter and the length of the excitation coil have the greatest influence on the magnetic field density of the damping channel. Therefore, the finite element analysis of the damper magnetic circuit is carried out through Maxwell software to continuously optimize the size of the piston diameter. As well as the length of the excitation coil, the output of the damper is finally optimized.

4.3. Finite element analysis of magnetic circuit

According to the above optimization method, the finite element simulation analysis of the magnetic circuit of the magnetorheological damper is carried out using Ansys Maxwell software.

1) Definition of boundary conditions and load conditions Analyze the static magnetic field of the magnetorheological damper. From the principle of continuity of the magnetic field, the magnetic flux leakage is ignored, the magnetic induction line is parallel to the boundary, and the input current is 2A.

2) Solving and result processing Through the setting of initial conditions, it can be solved through Maxwell 2D simulation module. The finite element simulation of the magneto-rheological damper before optimization is carried out. The magnetic field lines and the distribution of the magnetic induction intensity are shown in Figure 7. The figure shows that the maximum magnetic induction intensity at the damping channel is approximately 0.4T. By optimizing dimensions such as the diameter of the piston rod and the length of the excitation coil, the dimensions of each part when the magnetic flux density at the damping channel reaches the best are obtained. The simulation results are shown in Figure 8. After adjusting the size of each part, the maximum magnetic induction intensity at the damping channel is approximately 0.7T, and the magnetic induction intensity is approximately doubled compared to the original one.
5. Analysis of the damping effect of the damper

Magneto-rheological dampers are installed at the positions 1, 2, and 3 shown in Figure 3. The hammering method is used to simulate the vibration of the parallel machine tool, the CCD high-speed photography method is used for data collection, and the MATLAB data analysis software is used for data processing. The analysis shows that after the damper is installed on the machine tool, the vibration is significantly attenuated, the amplitude is reduced, and the period is slightly increased, as shown in Figure 9. It can be seen that the designed damper has a significant damping effect.

![Figure 9. Vibration attenuation curve](image)
6. Conclusion
Taking the three translational redundant drive (3-2SPS) parallel machine tool as the research object, by analyzing the vibration characteristics and force of the machine tool, a magneto-rheological damper with displacement amplification function is designed, and the output model of the damper is established. The magnetic circuit saturation of the damper is simulated through the ansys workbench finite element analysis software, and the structural parameters of the MR damper are adjusted to ensure that the magnetic circuit of the piston core is not saturated under the premise that the MRF at the damping channel is saturated, so that the damper is not saturated. The damping output is optimal. The research results show that the optimization makes the magnetic field distribution more reasonable, improves the mechanical performance of the damper, and the optimization method is feasible and effective. The research results provide a certain theoretical basis for the research of machine tool vibration reduction technology and the optimization design of magnetorheological damper.

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