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# A New Design of Vibration Damping Structure for Positive Displacement Motor

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**Abstract:** Positive displacement motor (PDM) is a downhole power drilling tool, and the bottom of the drilling tool is directly connected to the drill bit. During the drilling process, the drilling tool will be directly affected by the impact load caused by the drill bit breaking the rock. Severe vibration can easily cause drill tool bearings to break and fail, threads to trip and other problems. The new PDM vibration damping structure uses wear-resistant and heat-resistant metal rubber instead of disc springs as energy-absorbing materials, and designs a "keyway type" vibration damping structure; a two-degree-of-freedom "spring - mass" model is established based on actual working conditions. , determine the basic size of the metal rubber pad based on the metal rubber stress-strain curve, and bring it into the drilling tool dynamics equation to verify that the stiffness and damping parameters of the metal rubber pad will not resonate; finally, compare the drilling conditions under different excitations in the dynamics simulation software The amount of impact the tool receives. Finally, the dynamics simulation of the new vibration-damping drilling tool under different frequency excitations was carried out in Adams simulation software, and it was concluded that in the common working environment of 6-15 Hz, the new vibration-damping drilling tool can effectively reduce the stress on the drilling tool. Impact force, the new vibration-damping drilling tool can reduce the impact force by about 40% when excited at 12 , 15 Hz . The conclusions drawn from the study have reference value for the development of long-life drilling tools and the design of drilling tool vibration damping structures.

**Keywords:** Positive displacement motor, Metal rubber, Shock absorber, Dynamic simulation.

## 1. Introduction

PDM is a type of downhole power drilling tool, commonly used in complex drilling operations such as large displacement, horizontal wells, and multi branch wells due to its high torque, low speed, hard output characteristics, and low cost. However, the service life of screw drilling tools is generally only 150-200 hours, and vibration is one of the important reasons for the failure of screw drilling tools. Axial vibration is widely recognized as the most harmful form of drill string vibration, and the main reason for axial vibration is the irregular collision between the drill bit and the rock during drilling in complex formations. Easy to cause failure and fracture of drilling tool transmission shaft and bearings.

In the relevant research on vibration damping of drilling tools, Finnie I and Batley J obtained an approximate natural frequency of the drill string through experimental methods, ignoring nonlinear factors such as damping[1]-[2]; Skaogen et al. tested the vibration damping effects of different types of dampers within a certain speed range using MWD equipment [3]; Liu Ju bao, Zhang Xue hong, [4]and others established a mechanical model for deep well drilling string vibration analysis through dynamic simulation based on on-site data, and obtained the optimal placement position of the shock absorber within a depth range of 3000 meters. APS has developed a magnetorheological active vibration damper (AVD) and has demonstrated through experiments that it can improve mechanical drilling speed [5]; Li Zi feng[6] used the method of separating variables to solve the vibration model of drilling tools, and concluded that incorrect parameter selection of vibration dampers may lead to intensified vibration of downhole drilling tools; Zhang Xiaodong et al.[7] regarded drilling tools as a seven degree of freedom model and studied the influence of shock absorber installation position and stiffness on vibration isolation effect.

The limited size and large working load of screw drilling tools limit the design of vibration damping structures. Metal rubber has the characteristics of large damping, high temperature resistance, strong energy absorption ability, long service life, and wear resistance. Compared with the commonly used disc spring vibration damping structure in downhole equipment, it is more suitable for harsh drilling working environments. Therefore, this article refers to actual operating parameters and designs a new type of vibration damping structure for screw drilling tools. The vibration damping effect under different damping and stiffness is calculated through numerical simulation. The material and size of metal rubber are determined based on the calculation results. Finally, the vibration damping effect is verified through dynamic simulation. The simulation results and calculations can provide reference for future vibration damping structure design.[8]

## 2. Design of New Vibration Damping Structure for PDM

### 2.1. The working principle of the new PDM

PDM is a volumetric downhole power drilling tool. It usually consists of five parts, namely the bypass valve assembly, the anti-dropping assembly, the power assembly, the universal assembly, and the transmission assembly. When the drilling tool is working, the mud drilling fluid enters the motor through the bypass valve. There is a hydraulic pressure drop between the inlet and outlet of the motor, which drives the motor rotor to perform planetary motion around the stator. The torque and speed generated when the rotor rotates pass through the universal shaft and transmission. The shaft is output to the drill bit, and the main function of the anti-drop assembly is to prevent the drilling tool from falling into the bottom of the well after the motor breaks. The specific structure of the drilling tool is shown in Figure 1.

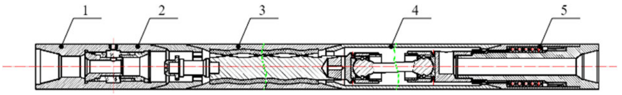



**Figure 1.** Structural diagram of PDM

1 - Bypass valve assembly; 2 - Anti-dropping assembly; 3 - Power assembly; 4 - Universal shaft assembly; 5 - Transmission shaft assembly;

In order to reduce the axial vibration of the drilling tool from the drill bit, a vibration damping assembly is added between PDM transmission shaft and the drill bit. The upper shell and the transmission shaft, the connecting shaft and the drill bit are fixed through threaded connections, and the upper shell The body drives the connecting shaft to rotate and transmit torque through the tapered ring key. At the same time, axial displacement can occur between the housing and the connecting shaft. The metal rubber pad is used to dampen the vibration between the two. The vibration damping assembly structure is shown in figure 2.

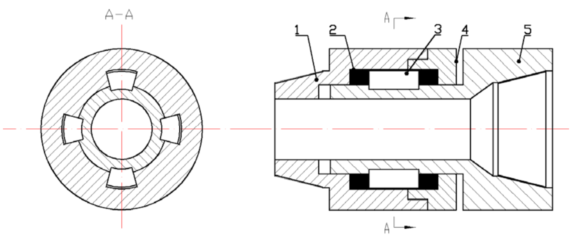



**Figure 2.** Vibration damping assembly structure diagram

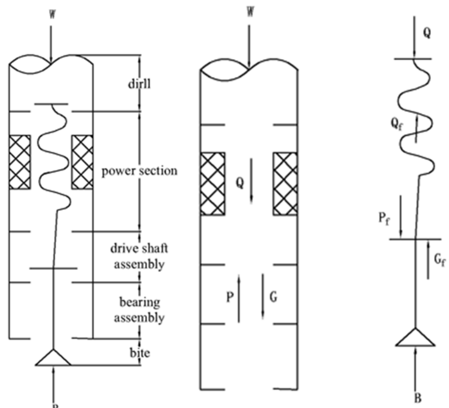
1 - Upper shell; 2 - Ring-shaped metal rubber pad; 3 - Tapered ring key; 4 - Lower shell; 5 - Connecting shaft

The metal rubber pad is a key component of the vibration damping assembly. It is punched and wound by extremely fine spiral stainless steel wire. When being squeezed, the dry friction between the steel wire contact surfaces will produce structural damping, converting the external force into internal energy, giving the metal rubber the ability to buffer and dampen vibrations. At the same time, it also has the high temperature resistance and oil resistance of metal. , fatigue resistance and other properties[9]. It is often used in vibration damping equipment in harsh environments such as aerospace, petroleum, and chemical industries. The new vibration-damping PDM uses metal rubber to replace the commonly used disc springs, which avoids the problems of poor vibration damping effect and short equipment life caused by disc spring failure in underground high-temperature and high-pressure environments . An interference fit is used between the shell and the connecting shaft. , so that part of the drilling fluid takes away part of the heat of the metal rubber and increases the service life of the metal rubber.

**Figure 3.** Metal rubber pads of different shapes

## 2.2. Establishment of simplified model of drilling tools

The drilling tool will be affected by coupled vibration when working downhole. In order to simplify the analysis when studying the axial vibration model of the drilling tool, only the axial force of the drilling tool is studied. Taking the entire PDM as the research object, the drilling tool can be divided into two parts: the drilling tool shell and the shaft system (rotor, cardan shaft, transmission shaft ) . The shell and the shaft system are connected to each other through threads and are hard connections [10]; The shaft system and the drilling tool housing are softly connected through anti-falling nuts, rubber bushings, and string bearings; the transmission shaft and the housing are softly connected through string bearings and disc springs ; PDM shafting system and the drill bit They are connected by threads. The following is a simplified diagram of the force between the drilling tool housing and the shaft system in figure 4 .




(a) Simplified diagram of drilling tool(left)

(b) Shell stress diagram(middle)

(c) Shaft system stress diagram(right)

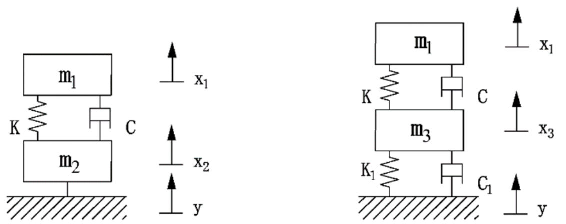
**Figure 4.** Simple diagram of PDM

The forces acting on the shell during the drilling process include: drilling weight *W*, rotor axial force *Q*, transmission shaft piston force *P*, and total shafting weight *G*; according to Newton's third law, the shafting system is also affected by the relative effects of the above forces. The forces $Q_f$, $P_f$, $G_f$ and the drill bit supporting force *B are equal to* the sum of the weight on bit W and the weight of the drilling tool . When the drilling tool is working, high-pressure drilling fluid is injected into the sealed chamber separated by the rotor and stator spirals in the motor assembly. It promotes the planetary motion of the rotor and also exerts the rotor axial force Q on the motor housing. The housing exerts an axial force Q on the rotor. When the two forces are unbalanced, the axial restraint force $Q_f$ will cause the rotor to move axially relative to the stator. At this time, the drilling fluid in the stator sealing cavity is similar to the hydraulic oil of the hydraulic damper, which will produce resistance to the rotor. The magnitude of the resistance It can be obtained from the fluid mechanics formula:

$$F_D = \rho v^2 C_d A \tag{1}$$

Among them: $F_D$ is the fluid resistance; ρ is the drilling fluid density; $v$ is the relative velocity of the fluid; $A$ is the projected area in the direction of movement; $C_d$ is the dimensionless resistance coefficient;

Perform a mechanical model analysis on the drilling tool power assembly. The entire power assembly can be regarded as a hydraulic cylinder with a spring. When the rotor moves under pressure, the rubber bushing converts the pressure into elastic potential energy. When the load decreases, the elastic potential energy is The release, compression and release processes are all affected by hydraulic resistance. Therefore, the screw rotor, rubber bushing, stator shell, and drilling fluid in the stator sealing cavity can be simplified into damping and spring models. In order to study the overall vibration pattern of the drilling tool, the shell and shaft system of the drilling tool are equivalent to masses. block, the entire drilling tool is equivalent to a two-degree-of-freedom "spring - mass block" model. Based on the above analysis, the simplified mechanical model is shown in Figure 5 .



(a) (b)

**Figure 5.** Simplified mechanical model of PDM

Figure 5 (a), $m_1$ is the mass of the drilling tool shell, and $m_2$ is the mass of the drilling tool shaft system. The drilling tool shell and the shaft system are connected through equivalent springs and damping. *k* and *c* represent the shell respectively. The equivalent stiffness and equivalent damping between the body and the shaft system (including between the rubber bushing and the stator, and the transmission shaft ) , the drill bit is equivalent to the fixed connection between the foundation and the shaft system. The impact load transmitted upward by the drill bit is simplified to a simple harmonic excitation y. In order to reduce the impact load on the drilling tool, it is necessary to change the connection relationship between the shaft system and the drill bit, and set up a vibration damping assembly with damping and stiffness between the shaft system and the drill bit. The mechanical model at this time is shown in Figure5 (b ), $m_3$ is the sum of the masses of the shaft system and the vibration-damping structure. The soft connection with the drill bit is achieved through the vibration-damping assembly, and the single-degree-of-freedom vibration model of the drilling tool is converted into a two-degree-of-freedom vibration model. $k_1$ and $c_1$ are Equivalent stiffness and equivalent damping of the vibration damping assembly.

According to Newton's second law, the axial vibration equations of two simplified mechanical models of drilling tools are established[11].

(1) The vibration equation of ordinary PDM is expressed as:

$$\begin{cases} m_1\ddot{x}_1 = W - k_1(x_1 - x_2) - c_1(\dot{x}_1 - \dot{x}_2) \\ x_2 = Y\sin\omega t \end{cases} \tag{2}$$

Convert the absolute displacement of the drilling tool housing and the shaft system into the relative displacement $z = x_1 - x_2$ , and set the basic excitation function as:$\omega$ is an integer multiple of the drilling speed of the drill bit , which is determined by the type of drill bit. Plugged into the formula (2) we get:

$$m_1\ddot{z} + c_1\dot{z} + k_1 z = W + m_1\omega^2 Y\sin\omega t \tag{3}$$

Solve the differential equation to obtain the drilling tool shell displacement equation:

$$z = \frac{W}{k_1} + \frac{m\omega^2 Y\sin(\omega t - \varphi)}{\sqrt{(k_1 - m\omega^2)^2 + (c_1\omega)^2}} = \frac{W}{k_1} + \frac{(\frac{\omega}{\omega_n})^2 Y\sin(\omega t - \varphi)}{\sqrt{(1 - (\frac{\omega}{\omega_n})^2)^2 + (2\eta\frac{\omega}{\omega_n})^2}} \tag{4}$$

$$\varphi = \arctan\frac{c_1\omega}{k_1 - m\omega^2} = \arctan\frac{2\eta(\frac{\omega}{\omega_n})}{1 - (\frac{\omega}{\omega_n})^2} \tag{5}$$

In the formula: the frequency ratio is $s = \frac{\omega}{\omega_n}$, $\omega_n$ which is the natural frequency of the drilling tool shell; the damping ratio is $\eta = \frac{c}{2m\omega_n}$, and the phase difference is $\varphi$ .

from the formula that (4)under the action of the drilling weight *W, the mass* $m_2$ will produce a constant displacement $\frac{W}{k_1}$ . When studying the steady-state response, the constant force can be ignored, and the focus is on the system response under the action of external excitation. The displacement of the drilling tool housing $m_2$ can be obtained by $x = z+y$ , assuming $x = X\sin(\omega t - \varphi)$ . The available amplitude ratio $\frac{X}{Y}$ is:

$$\frac{X}{Y} = \frac{\sqrt{1 + (2\eta\frac{\omega}{\omega_n})^2}}{\sqrt{(1 - (\frac{\omega}{\omega_n})^2)^2 + (2\eta\frac{\omega}{\omega_n})^2}} \tag{6}$$

(2) The vibration equation after adding the vibration damping assembly is expressed as:

$$\begin{cases} m_1\ddot{x}_1 = -k_1(x_1 - x_3) - c_1(\dot{x}_1 - \dot{x}_3) \\ m_3\ddot{x}_3 + k_3(x_3 - y) + c_3(\dot{x}_3 - \dot{y}) = k_1(x_1 - x_3) + c_1(\dot{x}_1 - \dot{x}_3) \end{cases} \tag{7}$$

The above equation is solved using the complex number method. Assuming that the basic excitation can $y(t) = Ye^{i\omega t}$ be represented by the imaginary part of , the special solution of the displacement of the two mass blocks in complex form is $x_1 = \overline{X}_1 e^{i\omega t}; x_3 = \overline{X}_3 e^{i\omega t}$ . Putting the special solution into the formula (7)we get:

$$\begin{cases} -(k_1 + i\omega c_1)\overline{X}_1 e^{i\omega t} + [k_1 + k_3 - m_3\omega^2 + i\omega(c_1 + c_3)]\overline{X}_3 e^{i\omega t} = (i\omega c_3 + k_3)Ye^{i\omega t} \\ (-m_1\omega^2 + k_1 + i\omega c_1)\overline{X}_1 e^{i\omega t} - (k_1 + i\omega c_1)\overline{X}_3 e^{i\omega t} = 0 \end{cases} \tag{8}$$

Write the above formula in matrix form and eliminate it

$e^{i\omega t}$ to get:

$$\begin{bmatrix}[k_1+k_3-m_3\omega^2+i\omega(c_1+c_3)] & -(k_1+i\omega c_1) \\ -(k_1+i\omega c_1) & (-m_1\omega^2+k_1+i\omega c_1)\end{bmatrix}\begin{bmatrix}\overline{X_3} \\ \overline{X_1}\end{bmatrix}=\begin{bmatrix}(i\omega c_3+k_3)Y \\ 0\end{bmatrix} \tag{9}$$

According to Kramer's law, the solution to the above equation can be obtained. The amplitude ratio of each part of the drilling tool is:

$$\begin{aligned}\frac{\overline{X_3}}{Y}&=\frac{(-m_1\omega^2+k_1+i\omega c_1)(k_3+i\omega c_3)}{[k_1+k_3-m_3\omega^2+i\omega(c_1+c_3)](k_1-m_1\omega^2+i\omega c_1)-(k_1+i\omega c_1)^2} \\ \frac{\overline{X_1}}{Y}&=\frac{(k_1+i\omega c_1)(k_3+i\omega c_3)}{[k_1+k_3-m_3\omega^2+i\omega(c_1+c_3)](k_1-m_1\omega^2+i\omega c_1)-(k_1+i\omega c_1)^2}\end{aligned} \tag{10}$$

## 2.3. Structural design of metal rubber

Metal rubber plays the role of energy absorption and vibration damping in the vibration damping assembly. Its material and size selection determine the performance of the vibration damping structure, and it needs to be designed. Select 172 *mm* PDM as the research object. The mass of the drilling tool is*M*. The shell of the drilling tool is generally larger than the shaft system. The mass of the drilling tool shell is 0.7*M* and the mass of the shaft system is 0.3*M*. The Rayleigh model is selected. Express the equivalent stiffness and damping to construct a theoretical model of the bushing. The specific calculation formula is:

$$\mathrm{k}=\frac{E(D^2-d^2)}{4l} \tag{11}$$

$$c=\alpha m_e+\beta k \tag{12}$$

In the above formula , $k$ is the stiffness coefficient; the elastic modulus E of nitrile rubber is 2GPa; the outer diameter of the bushing is 1.50 mm ; the inner diameter *d of the bushing* is 1.20 mm; the length of the bushing $l$ is 2m; $m_e$ is The mass of the bushing is 15 kg; $\alpha$ and $\beta$ are proportional coefficients. Based on relevant literature, the proportional coefficients $\alpha$ and $\beta$ *are selected to be* 0.1 and 0.001 respectively . According to the actual working conditions of PDM, the relevant parameters of the drilling tool are obtained in the following table 1.

**Table 1.** PDM calculation parameters

| Performance Parameters | value |
|---|---|
| **PDM size/(mm)** | 172 |
| **Overall Length/(m)** | 7.98 |
| **Weight/(kg)** | 992 |
| **WOB/(KN)** | 90 |
| **Speed/(RPM)** | 90-172 |
| **k/(N/m)** | $2.025\times10^6$ |
| **c/(N·s/m)** | 2025 |

To design the size of metal rubber, there are three static stiffness models of metal rubber: small curved beam model, porous material model, and micro-element spring model. Use the micro-element spring model with higher accuracy for calculation. The metal rubber load-displacement formula is as follows[12]:

$$\mathrm{F}=0.54G\bullet\sqrt[3]{(\frac{d}{\mathrm{D}})^{10}\bar{\rho}SH}\,x-0.26\bullet\sqrt[3]{(\frac{d}{\mathrm{D}})^{10}\bar{\rho}SH}\,x^2+0.1\bullet\sqrt[3]{(\frac{d}{\mathrm{D}})^{10}\bar{\rho}SH}\,x^3 \tag{13}$$

$d$ is the wire diameter; $D$ is the middle diameter of the spring; $\bar{\rho}$ is the relative density; $S$ is the bearing area of the metal rubber; $H$ is the height of the metal rubber; $G$ is the shear modulus.

According to the structural dimensions of the 172*mm* PDM, the total mass of the designed vibration-absorbing structure is 100 kg, and the size of the metal rubber pad is $\phi$ 155 mm × $\phi$ 135*mm* × 10*mm*. Considering that the center of the vibration-absorbing structure is the connecting shaft, the shape of the metal rubber is selected Ring shape, material selection: Cr18Ni9Ti austenitic stainless steel wire,shear modulus G=71GPa,d = 0.1*mm*,D = 7*mm*, $\bar{\rho}$ =0.15,S=4555.3$mm^2$ , H=10mm . By bringing in the above parameters, The stiffness displacement curve of metal rubber can be obtained as shown in Figure 6.

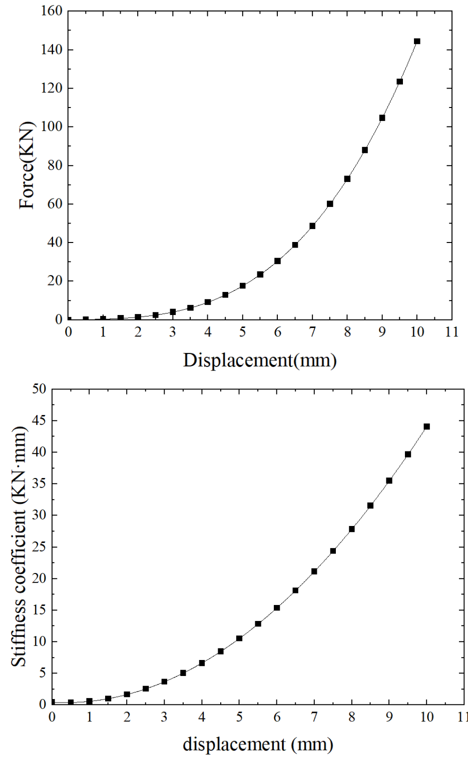



**Figure 6.** Metal rubber stiffness-displacement curve

After deriving the formula(13), the metal rubber stiffness change curve is obtained. It is generally believed that the ultimate compression amount of metal rubber is 30 %~60% , that is, the compression amount is between 3-6 mm. Within this range, the stiffness of metal rubber meets the design requirements, and the floating range is small, which can be approximately regarded as a fixed value. value. There is currently no specific mathematical model for nonlinear damping, which is usually based on the use of the Rayleigh model to represent metal-rubber damping, with a size of 0.01 of the stiffness *K*. Perform data processing on the stiffness within the compression range, and obtain the average value as the design stiffness and damping of the metal rubber. The calculation results are that the stiffness $k_3$ is $5\times10^{\wedge 2}$*KN/ m*, and the damping $c_3$ is $5.2962\times10^{\wedge 3}$*N·s m* , the damping ratio is about 0.37. From the formula (6), (10)the amplitude-frequency response diagrams of different damping ratios can be obtained as shown in Figure 7. [12]

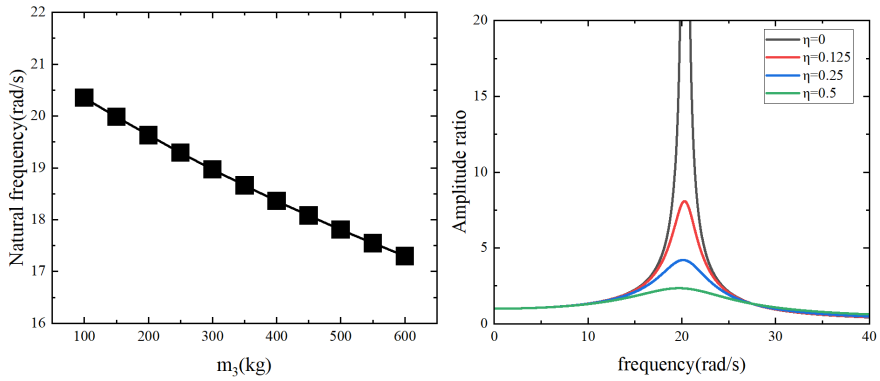


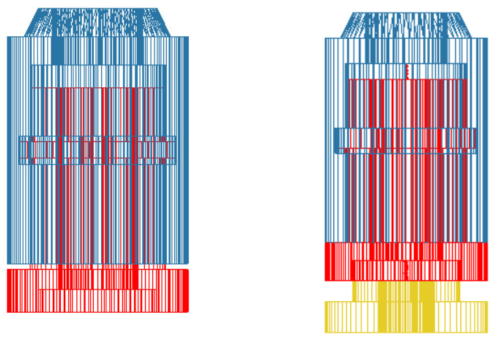

**Figure 7.** Amplitude-frequency response curves of vibration-damping drilling tools and ordinary drilling tools

It can be seen from the figure7 that the natural frequency of the drilling tool decreases as the mass of the vibration damping assembly increases, and the amplitude ratio of the drilling tool decreases as the damping ratio increases. In order to reduce the axial vibration of the drilling tool and avoid resonance, the mass of the drilling tool should be reduced, increase stiffness and damping. According to the data in 0, the natural frequency of the conventional PDM is 54 *rad* /*s*. When the damper mass is 100 kg, the natural frequency of the drilling tool is 20.3 *rad*/*s* , which is less than the maximum drilling speed of the drilling tool. 1 8.01 *rad*/*s* , which can avoid resonance. The designed damping structure has a damping ratio of 0.37 and a relatively small amplitude, which can effectively reduce resonance phenomena. The amplitude-frequency response curve of the shock absorber is basically the same as that of the drilling tool. There will be a difference only when the excitation frequency is higher, so no additional figure is included.

## 3. Finite Element Simulation of New Vibration-damping Structure

After obtaining the design stiffness and damping of the vibration-damping assembly through numerical calculation methods, in order to verify the vibration-damping effect of the vibration-damping drilling tool, the finite element method was used to analyze the differences between the new vibration-damping drilling tool and the ordinary drilling tool under different frequencies of drill bit impact. The difference in the impact force of the drill bit.

on the actual drilling tool structure. According to 0, the ordinary PDM is simplified into two parts: the drilling tool shell and PDM shaft system. The connection between the shell and the shaft system is through the finite element software. spring connection; the new vibration-damping drilling tool is simplified into three parts: the drilling tool housing, the drilling tool shaft system and the vibration-damping assembly. The drilling tool housing includes the part below the drill string and above the vibration damping assembly that does not include the shaft system; PDM shaft system is selected from the rest above the metal rubber pad; below the metal rubber and above the drill bit is the vibration damping assembly. Each part of the drilling tool is connected by a damper. The stiffness and damping size of the damper are consistent with the previous chapter. The simulation model is shown in Figure 8..[13]-[14]



(a) Ordinary PDM (b) New vibration-damping PDM

**Figure 8.** PDM dynamic model

According to the actual working conditions, a 90KN weight-on-bit preload is applied to the drilling tool shell, and displacement excitation is used to simulate the impact of the drill bit on the drilling tool. The displacement excitation size is set to: $y(t) = Y\sin(\omega t)$ . Where Y represents the amplitude and ω represents the speed of excitation. During the drilling process, tri -cone bits are often used for drilling. Each cone of the drill bit will impact the rock formation at the bottom of the hole three times every time the bit rotates. The impact of the bit on the drill string is generally three times the drilling speed. Actual on-site measurements are consistent with Calculations have proven this, and the excitation frequencies were selected as 6, 9 , 12 and 15 Hz to simulate the impact of the drilling tool under different working conditions. The rotation speed can be obtained by $\omega=2\pi f$, and the displacement excitation amplitude is selected as 4mm. The change curve of the axial impact force on the drilling tool is shown in figure 9 below.

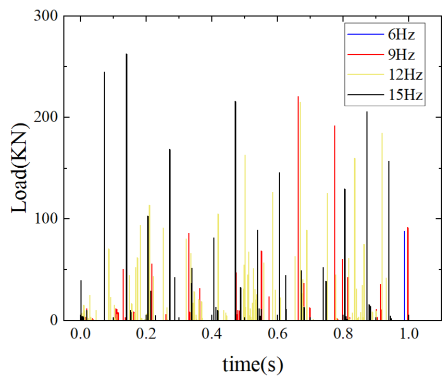



**Figure 9.** Impact force curve of new vibration-damping drilling tools

From the above impact force curves of the drilling tool under different working conditions, it can be seen that at low frequencies of 6 and 9 Hz, the impact force curves are stable within 100KN. When the frequency rises to 12Hz , the impact force distribution is similar to 9Hz , but The impact force of 100-200KN experienced by the drilling tool increases significantly. When the frequency continues to be as high as 15Hz, the impact force curve represented by the black line is mostly distributed in the range of 100-300KN. It can be inferred that the impact force on the drilling tool increases as the drill bit excitation frequency increases, the impact force distribution gradually narrows as the frequency increases, and the impact peak also increases as the excitation frequency increases.

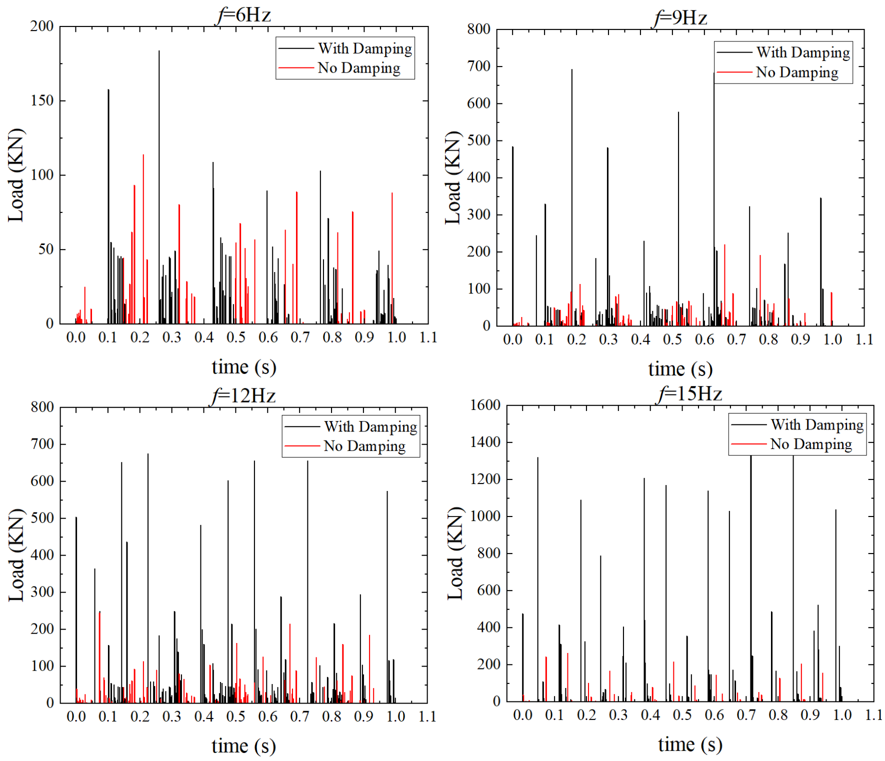



**Figure 10.** Comparison curves of impact force on drilling tools under different frequency excitations

From the comparison curve of the impact force of different drilling tools in figure 10, it can be seen that different drilling tools will have obvious phase differences at the same frequency. This is the phase delay phenomenon of vibration caused by the metal rubber damping structure of the vibration damping assembly absorbing the impact energy. . The new vibration-damping drilling tools can achieve vibration damping effects under different frequency excitations. Especially under high-frequency excitation conditions, the impact force can be reduced to 40 %-60% . However, under high-frequency impact, ordinary PDM It is easy to cause drilling tools to fail and break, which will slow down the work progress and increase the work cost. Therefore, for drilling operations in hard formations and other conditions where high-frequency jumps are prone to occur during drilling, it is necessary to use PDM with vibration-damping structures for drilling work.

## 4. Conclusion

(1) Aiming at problems such as transmission shaft breakage and bearing failure caused by the impact force generated by the drill bit breaking rock during the use of PDM, a vibration-damping assembly structure with vibration-damping and energy-absorbing effects was designed. The vibration-damping assembly is used to softly connect the drive shaft and the drill bit, and corrosion-resistant and high-temperature-resistant metal rubber is used instead of the commonly used disc springs as the energy-absorbing material of the vibration-damping assembly, making the new vibration-damping drilling tool more suitable for harsh drilling environments.

(2) Combined with the actual structure of PDM, the overall stress of PDM was analyzed, and the new vibration-damping drilling tool was simplified into three parts: the drilling tool shell, the shaft system, and the vibration damping assembly. The rubber lining inside PDM was The set is simplified into a spring damping model, and a simplified dynamic model of the drilling tool is established, and the relationship between the amplitude ratio of the drilling tool and the quality, stiffness and damping ratio of the drilling tool is established. Provide a theoretical basis for subsequent metal rubber size design.

(3) Design the metal rubber size required for the vibration damping assembly. Combined with the stress-strain curve of Cr18Ni9Ti metal rubber, calculate the average stiffness and damping within the designed compression range, and bring it into the drilling tool vibration model to check the new vibration damping Whether the drilling tool will produce resonance within the working penetration speed. A model was established in finite element software to compare and analyze the vibration damping effect of the new vibration-damping drilling tool under different working conditions.

From the simulation results, it can be concluded that in the common working environment of 6-15 Hz, the new vibration-damping drilling tool can effectively reduce the impact force on the drilling tool, and the vibration damping effect of low-frequency impact can reach 80%-90%. The vibration effect is average; but at higher frequencies, it can reduce the impact force by about 40% , which can effectively avoid damage to drilling tools by high-frequency impact forces and prevent failure of drilling tools caused by impact.

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