Optimal parameters selection and engineering implementation of dynamic vibration absorber attached to boring bar

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ABSTRACT

Dynamic analysis of damping boring bar with dynamic vibration absorber(DVA) is conducted in this paper. To minimize the vibration amplitude of the boring bar, the calculation method of DVA’s optimal parameters (target stiffness(TS) and target damping(TD) ) is obtained. In order to achieve the TS, the rigid flexible coupling model of the rubber ring is built. By using Abaqus/standard, the rubber ring’s installation process both with the relationship between the radial stiffness and the axial compression are obtained, which shows that when the axial compression increases, the radial stiffness increases non-linearly. In order to achieve the TD, the nonlinear fluid model of the silicone oil is built. By using Abaqus/CFD, the relationship between the damping value and the silicone oil pressure is obtained, which shows that with the increase of the silicone oil pressure the damping value increases non-linearly.

Keywords: Damping boring bar, Dynamic vibration absorber, Optimal parameters

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1. INTRODUCTION

In the manufacture of deep holes, the boring bar is required to have a large length to diameter ratio. As the length of the boring bar increases, the rigidity of the boring bar will decrease sharply, which will lead to the increase of the vibration. At the beginning of the last century, Taylor⁴ has already discovered the phenomenon of vibration in machining. In the 40s of the last century, Arnold⁵ applied vibration theory to tool machining. After 100 years’ development, reducing tool vibration is still a hot issue for many scholars⁶-⁷.

There are two ways to reduce the vibration of boring bar. One is to increase the stiffness of the boring bar, which depends on the development of material science and is not the focus of this paper. The other one is to attach a vibration reduction system, mainly including active vibration reduction system(AVRS) and passive vibration reduction system(PVRS), to the boring bar. AVRS will attach a set of active vibration control system to the boring bar, which will provide additional force or displacement to compensate for the vibration produced by the boring bar. With the development of technology in recent years, even the method of using magneto rheological fluid has appeared, which can change the stiffness of the boring bar in real time to obtain a better vibration control effect⁶-⁷. Generally, AVRS’s structure is complex and also needs frequent adjustments at work⁸. In comparison, PVRS is considered to be a reliable, effective and easy way to reduce vibration of boring bar. Figure 1 (a) is a typical damping boring bar with PVRS.

(a)3D model of damping boring bar

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As shown in Figure 1 (a), the DVA, mainly including the inner core, rubber ring and silicone oil, is placed in the front of the damping boring bar, where the inner core is connected to the inner rod by using the two rubber rings placed on both ends of the DVA, and the silicone oil is filled between the inner core and the inner rod. Figure 1 (b) shows the simplified model of the damping boring bar where the boring bar is modeled as a cantilever beam and the DVA is modeled as a mass-spring-damper system. Figure 1 (c) shows the dynamic model of the damping boring bar modeled as a two-degree-of-freedom system.

2. DYNAMIC ANALYSIS OF THE DAMPING BORING BAR

The kinetic equation of the dynamic model showed in Figure 1 (c) is listed[13]:

$$\begin{align*}
&M\dddot{x}_1 + c\dddot{x}_1 + (k + K)x_1 - c\ddot{x}_2 - kx_2 = p(t) \\
&m\dddot{x}_2 + c\ddot{x}_2 + kx_2 - c\dot{x}_1 - kx_1 = 0
\end{align*}$$

(1)

Where $p(t) = p_0 \sin \omega_0 t$. Referring to the reference [14], by using Lagrange transform, the dynamic magnification factor of the main mass displacement is obtained:

$$A(g) = \sqrt{\frac{(2\zeta g)^2 + (g^2 - f^2)}{(2\zeta g)^2 + (g^2 - 1 + \mu g^2) + [\mu f^2 g^2 - (g^2 - 1)(g^2 - f^2)]^2}}$$

(2)

Where $\zeta$ is the damping ratio given by $\zeta = c / (2\sqrt{mk})$, $\mu$ is the mass ratio given by $\mu = m / M$, $g$ is the frequency ratio given by $g = \omega / \omega_n$, here $\omega_n$ is the natural frequency of the main mass $M$ given by $\omega_n = \sqrt{k/M}$, $f$ is the natural frequency ratio given by $f = \omega_d / \omega_n$, here $\omega_d$ is the natural frequency of the DVA given by $\omega_d = \sqrt{1 - \zeta^2} \sqrt{k/m}$, then $f = \sqrt{1 - \zeta^2} \sqrt{km}$. As shown in Figure 2, given a certain value of $\mu$ and $f$, the curve of formula (2) can be obtained under different value of $\zeta$, and it is easy to find the curve has two common points $A$ and $B$.

$$\begin{align*}
g_A^2 + g_B^2 &= \frac{2[1 + (1 + \mu)f^2]}{2 + \mu}
\end{align*}$$

(3)
For formula (2), when $\zeta \to \infty$:

$$A(g) = \frac{1}{1 - (1 + \mu)g^2}$$  \hspace{1cm} (4)

By changing the natural frequency ratio $f$, the optimal parameters can be obtained. At this point:

$$A(g_A) = A(g_B)$$  \hspace{1cm} (5)

That is, the optimal damping ratio $\zeta_{opt}$ needs to satisfy the formula:

$$\frac{1}{1 - (1 + \mu)g^2_A} = -1$$  \hspace{1cm} (6)

Then:

$$g^2_A + g^2_B = \frac{2}{1 + \mu}$$  \hspace{1cm} (7)

Optimal natural frequency ratio can be obtained by comparing formula (7) with formula (3):

$$f_{opt} = \frac{1}{1 + \mu}$$  \hspace{1cm} (8)

Assuming that the mass ratio $\mu$ has been given, then $A(g, \zeta)$ can be obtained by bring formula (8) into formula (2). Then the optimal damping ratio can be obtain by taking the derivative with respect to $g$ and make the value to be zero at point $g_A$ and point $g_B$:

$$\zeta_{i,2}^2 = \frac{\mu[3 + \sqrt{2 + \mu}]}{8(1 + \mu)^3}$$  \hspace{1cm} (9)

As the mass ratio $\mu$ is very small, there is little difference between $\zeta_1$ and $\zeta_2$, so the average value of the two is taken as the optimal damping ratio:

$$\zeta_{opt} = \sqrt{\frac{3\mu}{8(1 + \mu)^3}}$$  \hspace{1cm} (10)

$g^2_A$ and $g^2_B$, obtained by bring formula (8) into the formula (3), is brought into formula (2), then the dynamic magnification factor of the common points can be obtained:

$$A(g_A) = A(g_B) = \sqrt{1 + \frac{2}{\mu}}$$  \hspace{1cm} (11)

As shown in formula (11), with the increase of $\mu$, the vibration amplitude goes down. So when designing the optimal parameters, the mass of the inner core should be as heavy as possible, then the optimal natural frequency ratio $f_{opt}$ and optimal damping ratio $\zeta_{opt}$ can be obtained by using formula (8) and formula (10). Then by using the expression of $f$ and $\zeta$ in formula (2), the TS and the TD can be also obtained:

$$k = \frac{8K\mu(1 + \mu)}{8(1 + \mu)^3 - 3\mu}$$  \hspace{1cm} (12)

$$c = 2\sqrt{\frac{3km\mu^2}{8(1 + \mu)^3 - 3\mu(1 + \mu)^2}}$$  \hspace{1cm} (13)
3. ENGINEERING IMPLEMENTATION OF TARGET PARAMETERS

So far, we have already obtained the optimal parameter, including the TS and the TD. Then how to make the rubber ring and the silicone oil to achieve the target value is what we care about mostly in engineering, and this rely on the clear understanding of the installation and operation information of the the rubber ring and the silicone oil.

3.1 Rubber Ring Installation Process Simulation

As shown in Figure 1 (a), the installation sequence of the damping boring bar is: (1)Assemble the DVA: Bind the head and the inner rod together, then assemble the rubber ring 1, the inner core, the rubber ring 2 and the end cap in turn. (2)Put the DVA into the body of the damping boring bar. It is easy to see that the radial stiffness of the rubber ring can be changed by screwing the end cap.

According to the magnification in Figure 1(a), the rigid flexible coupling simulation model is established as shown in Figure 3.

![Simulation model of the rubber ring](image)

Figure 3 – Simulation model of the rubber ring

Installation process simulation of the rubber ring is conducted in Abaqus:

(1) Components Building: The inner core and the head are treated as rigid bodies. In order to add constraints and loads, reference point 1 and reference point 2 are established as shown in Figure 3.

(2) Material defining: As rigid bodies, the inner core and the head do not need to define material properties. Rubber ring material properties: density $1.3 \times 10^3 \text{ kg/m}^3$, elastic modulus 4.2MPa, Poisson's ratio 0.49.

(3) Assembly defining: Position the three parts as shown in Figure 3.

(4) Meshing: Set global element size and the element type is CPS4R.

(5) Step setting: Initial step: Fix RP2. Step1: RP1 moves to the left 0.001mm to make each contact be established smoothly. Since the problem is nonlinear, turn on the nonlinear button. Step2: RP1 continues to move to the left 1mm, which is used to simulate the extrusion process between the inner core and the rubber ring. The inner core moves to the left 0.01mm per incremental step, so we need 100 time increment steps.

(6) Contact defining: Establish the contact surf 1, surf 2 and surf 3, and then define contact pairs with Coulomb friction (friction coefficient is 0.36) between the surf 1 and the surf 2, also between the surf 2 and the surf 3.

(7) Job submitting: Submit the job and enter the post processing module to get the stress contours of the rubber ring as shown in Figure 5.

As shown in Figure 5, at the beginning of the installation process, the high stress area, which has a symmetrical distribution roughly, occurs first in the left and the right ends of the rubber ring. With the movement of the inner core, the area extends into the interior and gradually leads to the radial stress change (see Figure 4 (b)). Due to the effect of Coulomb friction, stress distribution is no longer symmetrical. The high stress area in the left end goes up while the high stress area in the right end goes down (see Figure 4 (c)). When the inner core's installation is complete, the stress contour is shown in Figure 4 (d).
In the whole installation process, the axial compression leads to the radial movement of the rubber ring, and the radial pressure caused by the radial movement leads to the change of the radial stiffness. At the same time, due to the existence of the Coulomb friction, the friction force on the left and right surface is asymmetric, which will cause some shear stiffness. Generally, the change of the rubber ring’s radial stiffness is the interaction results of the radial movement and the shear stiffness.

3.2 Simulation of Rubber Ring’s Radial Stiffness

For a certain type of boring bar, the structure size has been determined, so what we concern mostly is whether the radial stiffness of the rubber ring has reached the target value under different axial compression. Therefore, based on the simulation in 2.1, more increment steps should be added: Applying 5N downward force on RP1 (in order to prevent the rubber ring from rebounding, the force increases from 0N to 2N quickly, then from 2N to 5N slowly). The changing process of the rubber ring is shown in Figure 5.

As shown in Figure 5, with the increase of radial force, the high stress area moves to the upper left and lower right. If the size design and installation are not reasonable, it is even possible to see the "spillover" phenomenon in the upper left corner of the rubber ring (see Figure 5 (d)), which will lead to a serious stress concentration area.
Under different axial compression ranges from 0.2mm-1.4mm, seven repeated experiments have been done and the rubber ring’s displacement-radial force curves are obtained as shown in Figure 6, whose slopes reflect the radial stiffness of the rubber ring. Figure 6 is divided into region A and region B. The radial force changes from 0N to 2N in region A while from 2N to 5N in region B. In region A, when the axial compression is 0.2mm the radial displacement is -0.51mm, but when the axial compression rises to 1.4mm the radial displacement is only -0.06mm, which means the axial compression of the rubber ring has a significant impact on its radial stiffness, and with the increase of the axial compression, it is more difficult for the rubber ring to produce a deformation. In region B, the seven curves are both concave, the absolute value of the slope decreases with the increase of radial force, which means the radial stiffness of the rubber ring is nonlinear. It is worth noting that the curvature radius of the concave curve is very large, so in approximate calculation, the stiffness can be treated as a linear curve. In region B, the radial stiffness can be calculated under different axial compression. As shown in Figure 7, the radial stiffness increases no-linearly with the increase of the axial compression. By selecting different axial compression, the radial stiffness of the rubber ring can reach the target value in engineering.

3.3 Silicone Oil Dynamic Simulation

The purpose of the silicone oil dynamic simulation is to find the relationship between the silicone oil’s force and velocity, then the damping value can be obtained by the formula of $c = F / v$. For a certain type of product, as the structure size has been determined, what we concern about mostly is the relationship between the silicone oil pressure and the damping value. As shown in Figure 1, the silicone oil, filled between the inner rod and the inner core, has a cylindrical structure. As shown in Figure 8, the silicone oil model is established in ABAQUS / CFD.
The process of modeling and analysis is similar to the main process in 2.1, and the differences mainly lies in the material defining and the boundary defining. (1) Material defining: density $97 \times 10^3 \text{ kg/m}^3$, kinematic viscosity $5000 \text{ mm}^2/\text{s}$. (2) Boundary defining: boundary definition is shown in Figure 8. As the simulation mainly focuses on the relationship between the silicone oil’s force and velocity, the displacement and velocity of the inner cylinder are simplified as the sine and cosine signal. The typical velocity distribution of the silicone oil is obtained (see Figure 9).

![Figure 9 – Typical velocity distribution](image)

In status 1, the velocity direction of each point is similar and the silicon oil is in a state of advection. In status 2 and 3, the silicone oil is in a state of contraction and divergence separately. In state 4, the velocity of each point is 0. In the process of simulation, these four states alternate inordinately, which means the movement of the silicone oil is uncertain. Therefore, the maximum force and velocity are chosen to calculate the damping value. Similar to the method in 2.2, repeat the simulation process under different pressure. The damping value can be obtained (see Figure 10).

![Figure 10 – Damping value of the silicone oil](image)
4. CONCLUSIONS

The damping boring bar is modeled as a two-degree-of-freedom mass-spring-damper system. In order to minimize the vibration amplitude of the boring bar, the dynamic equation is listed to obtain the optimal parameters TS and TD of the DVA.

In order to achieve the desired TS in engineering, the nonlinear simulation of the rubber ring’s rigid flexible coupling model is conducted, and the compression process of the rubber ring is observed. The result shows that the change of the rubber ring’s radial stiffness is the interaction results of the radial movement and the shear stiffness. At the same time, the relationship between the rubber ring’s radial stiffness and the axial compression is obtained, and the result shows that the radial stiffness increases no-linearly with the increase of the axial compression. In order to achieve the desired TD in engineering, the nonlinear fluid simulation of the silicone oil is conducted. The result shows that the damping value increases no-linearly with the increase of the silicone oil’s pressure. By adjusting the pressure of silicone oil, the damping value can reach the target value in engineering. This paper provides a theoretical and simulation method for optimal parameters selection and engineering implementation of DVA attached to boring bar, and the experimental research should be conducted in further research.

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