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Research on fine boring simulation based on squeezed liquid film damper

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Abstract

The liquid film damping is one of the main technology for the aerospace, navigation, and machine tool. However, the research on the fine boring is very limited. The paper set up the fine boring squeezed liquid film damped system of multi degree of freedom vibration simulation model and solve mathematical model by transfer matrix theory. The damping coefficient for the fine boring system can be obtained. The mechanical device of liquid film damping system is established. 20# machine oil, 40# machine oil, and cutting fluid are used to the system. The tests indicate that the effect of liquid film formed by cutting fluid is better than oil film formed by machine oil in the static test, but oil film formed by machine oil is better than liquid film formed by cutting fluid in the effect of dynamic test. The simulation model has high accuracy to reliability. The simulation model can directly obtain the optimal parameters, so as to provide effective way to guide field processing.

Keywords: Fine boring, Squeezed liquid film, Damp, System simulation

1 Introduction

At present, the liquid film damping technology has widely research and application in the areas such as aerospace and machine tool [1–4]. But the liquid film (including oil film) damping technique is applied to fine boring processing at home and abroad for the first time. And the fine boring liquid film damping system can be carried on through the theoretical analysis and experimental research. A reasonable mathematical model for simulation is established on the basis of the direct analysis on the influence on many parameters of the system [5–8]. So the parameter optimization and the excellent fine boring liquid film damping of shock absorber are designed.

Fine boring liquid film damping system has a simple structure. The inside and outside of the damper is filled with liquid. Under the condition of high-speed rotation, a squeezed liquid film is formed between the workpiece and damping sleeve because of the boring bar's lateral vibration. At the same time, the damping force is produced for preventing the gap between the workpiece and the damping. So it can consume vibration energy of the boring bar, reduce the amplitude, and improve the machining quality of fine boring.

2 Methodology

Over the years, many experts and scholars at home and abroad devoted themselves to the boring bar dynamics analysis. But most of them were based on a single degree of freedom vibration system. The influence of the main modes on more degree of freedom system is ignored because of single degree of freedom system in the vibration of the system. So the analyses of the system are not accurate enough. In this paper, the author will consider the fine boring squeeze liquid film damping system as a multiple degree of freedom vibration system [9–11], and it can significantly improve the accuracy of the vibration analysis. Its dynamic model has lots of types, including: ① lumped parameter model, ② distributed parameter model, and ③ hybrid parameter model. To analyze the vibration characteristics of boring bar concrete parts, the paper use the lumped parameter model. This tool location can be regarded as inelastic quality unit for vibration characteristic analysis [12]. The law of the system vibration is concluded more accurately.

Each part of the boring bar length and the diameter number is as follows: $L_1 = 40$ mm, $D_1 = 56$ mm; $L_2 = 100$ mm, $D_2 = 40$ mm; $L_3 = 100$ mm, $D_3 = 40$ mm; and $L_4 = 40$ mm, $D_4 = 36$ mm. Damper and liquid film clearance of workpiece respectively are 0.05, 0.10, 0.15, and 0.10 mm, damper liquid using emulsion 20# machine oil.

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The equipment starts and begins to process. The eddy current sensor can be used to detect the vibration of the boring bar. The detected vibration signals are input to the computer through the amplifying circuit and the AD converter to carry out the data collection and storage.

The whole system is divided into a series of quality unit and elastic rod unit with no quality. The damping unit and power unit with quality units is combined into a special quality unit. So the whole system is composed of quality unit, elastic shaft unit, damp-quality unit, and power-quality unit. The number of the specific units is decided by the required precision. The more the number of units is, the higher the accuracy of the model analysis is (take nine units in this paper).

The lumped parameter system dynamic model is shown in Fig. 1.

In Fig. 1, "F" for exciting force, these five quality units are calculated according to the length, diameter, and density of the boring bar, respectively as follows: $m_1 = 0.385$ kg, $m_2 = 0.875$ kg, $m_3 = 0.98$ kg, $m_4 = 0.685$ kg, and $m_5 = 0.195$ kg.

After the system dynamics model is set up, according to transfer matrix theory, the system mathematical model is established and to solve mathematical model, so as to lay foundation for the establishment of the system simulation model. It is known from the transfer matrix theory that the left and right side system units have a state vector to represent the unit state, denoted as Zl (left side) and Zr (right side); the relationship between the state vector can be represented as:

$$\mathrm{Z_l} = \mathrm{CZ_r}$$

C is a unit transfer matrix or a transfer matrix, and it can be introduced from this: The transfer matrix between any two unit state vectors is multiplied to all unit transfer matrices of the two state vectors. The following is expressed as follows:

$$\begin{bmatrix} \mathrm{z_1} \\ 1 \end{bmatrix} = \begin{bmatrix} \mathrm{C'} & \mathrm{F} \\ 0 & 1 \end{bmatrix}_{\mathrm{i}} \begin{bmatrix} \mathrm{C'} & \mathrm{F} \\ 0 & 1 \end{bmatrix}_{\mathrm{i-1}} \cdots \begin{bmatrix} \mathrm{C'} & \mathrm{F} \\ 0 & 1 \end{bmatrix}_{\mathrm{n+1}} \begin{bmatrix} \mathrm{Z_r} \\ 1 \end{bmatrix}_{\mathrm{n}} \tag{1}$$

In the formula:

$i > n \in I$

C' is the first four-order square matrices of each transfer matrix.

According to the force balance relations, every each unit can obtain the following transfer matrix.

1. Inelastic quality unit transfer matrix C_{im}

$$c_{im} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ m_i\omega^2 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

In the formula, ω is circular frequency.

2. Inelastic mass element transfer matrix C_{iL}

$$c_{\mathrm{iL}} = \begin{bmatrix} 1 & L_{\mathrm{i}} & \frac{L_{\mathrm{i}}^2}{2\mathrm{EI}} & \frac{L_{\mathrm{i}}^3}{6\mathrm{EI}} & 0 \\ 0 & 1 & \frac{L_{\mathrm{i}}}{\mathrm{EI}} & \frac{L_{\mathrm{i}}^2}{2\mathrm{EI}} & 0 \\ 0 & 0 & 1 & L_{\mathrm{i}} & 0 \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

In the formula, EI is moment of inertia.

3. Damp-quality unit transfer matrix C_{iF}

$$c_{\mathrm{iC}} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ m_{\mathrm{i}}\omega^2 + j\omega c & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

In the formula, j is imaginary unit.

4. Power quality unit transfer matrix C_{iF}

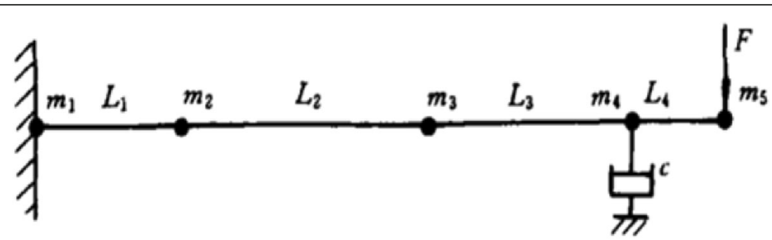


Fig. 1 Boring bar lumped parameter dynamic model

$$c_{\mathrm{iC}} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ m_{\mathrm{i}}\omega^2 & 0 & 0 & 1 & \mathrm{F} \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

Both sides of system state vector of transfer matrix equations can be concluded by plugging formula 4 into formula 1.

$$\begin{Bmatrix} \mathrm{Y} \\ \theta \\ \mathrm{M} \\ \mathrm{F} \\ 1 \end{Bmatrix}_1^1 = C_{1\mathrm{m}}C_{1\mathrm{L}}C_{2\mathrm{m}}C_{2\mathrm{L}}\cdots C_{5\mathrm{m}} \begin{Bmatrix} \mathrm{Y} \\ \theta \\ \mathrm{M} \\ \mathrm{F} \\ 1 \end{Bmatrix}_5^{\prime} \tag{2}$$

All the matrices in the formula are both five-order square. Y, θ, M, and F respectively represent the displacement, angle, torque, and force. The left side vector represents left side state vector of the system, and the right side vector represents right side state vector of the system. So the product of the matrix is still 5 square, the C_g is as follows:

$$C_g = \begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & 0 \\ C_{21} & C_{22} & C_{23} & C_{24} & 0 \\ C_{31} & C_{32} & C_{33} & C_{34} & 0 \\ C_{41} & C_{42} & C_{43} & C_{44} & F \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

Because the boring bar is closely connected with machine tool axle shaft of Morse taper hole, the system can be as a cantilever beam fixed at one end. So the system boundary conditions are as follows:

$$Y_{11} = 0 \quad \theta_{11} = 0 \quad M_{5r} = 0 \quad F_{5r} = F$$

The equation can be gotten:

$$\begin{cases} 0 = C_{11}Y_5 + C_{12}\theta_5 + C_{14}P \\ 0 = C_{21}Y_5 + C_{22}\theta_5 + C_{24}P \end{cases}$$

The root of the equation:

$$Y_5 = \frac{C_{12}C_{24} - C_{22}C_{14}}{C_{11}C_{22} - C_{12}C_{21}}F \tag{3}$$

As a result of the existence of system with damping device, Y_5 is plural

$$Y_5 = \zeta_r + j\zeta_i = A\cos(\omega t + \phi) \tag{4}$$

In the equation:

$A = |\mathrm{Y}_5| = \sqrt{\zeta_{\mathrm{r}}^2 + \zeta_{\mathrm{i}}^2}\ \phi = \arctan\frac{\zeta_{\mathrm{i}}}{\zeta_{\mathrm{r.}}}$

ζ_{r}——The real part of the complex Y_5

ζ_{i}——The imaginary part of complex Y_5

The first one is Y_5 real part, and the other one is Y_5 imaginary part.

Through the above model, we know that the system response to external simple harmonic excitation is still the harmonic vibration, the amplitude of response is a function of damping coefficient c and excitation frequency ω, where it can be expressed as:

$$A = F(C, \omega) \tag{5}$$

The resonance amplitude is:

$$A_{\max} = \max F(c, \omega) \tag{6}$$

3 Results and discussion

3.1 Reliability analysis of the simulation model

Static test was done under the condition of the machine not working, and vibration adopts the method of external excitation. Its purpose was to test the vibration of the boring bar in the two cases: with damper or not. In the static test, study is done to get the relationship of the damper's parameters and vibration damping effect. The CA6140 lathe was modified to carry boring machining with the boring bar installed in the lathe spindle Morse taper hole [13]. The workpiece was clamped by the fixture and installed on the sliding board. Emulsion and the 20# machine oil were used as damping liquid respectively. The test data signals was obtained through acceleration sensors, which was sent to a computer to process after signal amplification and A/D conversion. Table 1 is the results with different damping sleeve width and damping fluid (oil) film thickness in the cases with fluid (oil) film and no fluid (oil) film. Trial dampers' inside diameter is $D = 60$ mm, the absolute liquid viscosity coefficient $\eta = 5.949$ Pa · s. The results of the data test are shown in the following Table 1:

An empirical formula for damping coefficient can be got by the relevant parameters of the dampers. It represents the law of the liquid film under the finite long axis in the table.

$$c = \frac{\eta L R^3}{\delta^3 a (1-\varepsilon)^b} \tag{7}$$

In the equation, $R = D/2$.

a and b are constants associated with damper parameters.

$$a = 0.6(D/L) - 0.4, b = 1.5 + 0.3\sqrt{D/L}$$

δ is the damping sleeve and the processing hole clearance value, the thickness of liquid film

ε is the damper hole with hammer tool rod eccentricity, and its little impact on c so can be ignored.

Table 1 Field test raw data

Separation δ/μm	Frequency f/Hz amplitude A/μm				
	The length of the damper L/mm 50	The length of the damper L/mm 60	The length of the damper L/mm 80	The length of the damper L/mm 100	The length of the damper L/mm 120
75	1177/2.6	1429/2.9	1355/2.0	1577/1.6	1335/2.2
100	1337/2.6	1440/2.8	1598/3.5	1524/3.3	1724/4.5
150	1459/1.4	1434/1.9	1492/1.1	1609/4.9	1660/2.7
400	1345/1.8	1330/1.7	1337/2.1	1556/2.8	1335/2.8

Under the single damping and complex damping, the static test with liquid film and non-liquid film is carried out. Some important data can be measured as shown in Tables 2, 3, and 4.

From Tables 2, 3, and 4, it can be seen that squeeze liquid (oil) film damper can inhibit the vibration of boring bar no matter in the static excitation or dynamic cutting. In the static test, the effect of liquid film formed by cutting fluid is better than oil film formed by machine oil. In the effect of dynamic test of oil film formed by machine oil is better than liquid film formed by cutting fluid.

From Table 4, it can be seen from all the test data, with the increase of damping sleeve width, the boring bar vibration amplitude was decreased gradually, but when the damping sleeve width continue to increase, the amplitude had a tendency of increase. The change rule can be explained as follows.

From Tables 2, 3, 4, and 5, the test curve of amplitude and resonance frequency with the damping value of number can drew. The curve line is shown in Figs. 3 and 4.

In addition, the curve of the amplitude and the resonant frequency from Eqs. (5), (6) paint and the damping value simulation curve can be shown in Figs. 2 and 3. To be sure, in the simulation study, the size of the force F only affects the position of the curve along the longitudinal axis, does not affect the shape of the curve and the changing trends, and so does not affect the essence of the problem. it can be understand of formula (3). In this case, take $F = 1.028$ KN.

In the Figs. 2 and 3: Both amplitude damping curve and resonance frequency damping curve are very close, and it indicates that the simulation results are basically in agreement with the experimental results; it can be seen that establishing the simulation model is reasonable and reliable.

3.2 Parameter optimization of damper

In the design of damper, the main parameters affect the effects of vibration reduction for the thickness δ (mm) and the length L (mm) of the liquid film damper. In

Table 2 Single damping oil film, liquid membrane effects on spindle amplitude

Gap/mm	Width/mm	Amplitude/μm					
		Oil film	Non-oil film	Amplitude reduce	liquid film	Non-liquid film	Amplitude reduce
0.05	40	96	124	28	104	124	20
	60	85	118	33	95	118	23
	80	82	109	27	88	109	21
	100	76	105	29	82	105	23
	120	71	106	35	82	106	24
0.10	40	78	120	42	96	120	24
	60	73	116	43	86	116	20
	80	70	109	39	85	109	24
	100	64	104	40	84	104	20
	120	63	103	40	81	103	22
0.15	40	89	118	29	106	118	12
	60	83	112	29	101	112	11
	80	84	115	31	97	115	18
	100	75	98	23	88	98	10
	120	71	92	21	88	92	4

Table 3 Single damping oil film, liquid membrane effect on main frequency

Gap mm	Width mm	Frequency/Hz					
		oil film	Non-oil film	Amplitude reduce	liquid film	Non-liquid film	Amplitude reduce
0.05	40	1419	1403	16	1392	1403	11
	60	1438	1424	14	1445	1424	21
	80	1466	1424	42	1450	1424	26
	100	1516	1456	60	1482	1456	26
	120	1566	1492	74	1542	1492	50
0.10	40	1425	1389	36	1419	1389	30
	60	1450	1403	47	1447	1403	44
	80	1534	1467	67	1507	1467	40
	100	1521	1426	96	1536	1476	40
	120	1607	1518	89	1605	1518	87
0.15	40	1432	1409	23	1411	1409	2
	60	1414	1392	22	1416	1392	24
	80	1455	1444	11	1444	1432	12
	100	1472	1453	19	1440	1453	7
	120	1574	1524	50	1553	1524	29

order to determine the optimal parameters, more groups of dynamic test have been done under the conditions of the actual processing conditions. Because cost is high, the large amount of test is hard to proceed. The law of various parameters on the vibration reduction is analyzed by the simulation model. The results can directly get the optimal parameters, so as to effectively guide field processing.

The δ value is ensured in a certain range, and the other parameters are constant. Using formula (7) into formula (5), the relation curve of liquid film thickness–amplitude is drew and shown in Fig. 4. In the same way, the liquid film length-amplitude relation curve are drew and shown in Fig. 5.

As seen in Figs. 4 and 5, the optimal thickness of the δ and the length L of the liquid film damper is 0.l and 60 mm respectively. According to the data, the author designed the damping shock absorber, and it has been applied in Dalian Locomotive Plant a TX611C type digital display horizontal boring machine. As the result,

Table 4 Double damping oil film, liquid membrane for the main shaft frequency and amplitude

Gap/mm	Width/mm	Oil film			Liquid film			Non-liquid film		
		Frequency/Hz		Amplitude/μm	Frequency/Hz		Amplitude/μm	Frequency/Hz		Amplitude/μm
0.05	40	1102	1518	72.102	1114	1529	78.107	1023	1503	83.112
	60	1113	1507	68.98	1098	1516	70.100	1098	1464	74.106
	80	1132	1572	60.90	1132	1582	68.92	1135	1454	73.99
	100	1129	1602	63.86	1176	1580	68.88	1097	1570	66.93
	120	1205	1618	59.81	1212	1610	60.84	1153	1592	67.85
0.10	40	1016	1527	71.89	1021	1521	73.91	0986	1508	85.107
	60	1035	1556	69.80	1039	1535	77.83	1024	1527	86.104
	80	1038	1600	68.76	1056	1586	72.81	1038	1580	88.96
	100	1096	1626	60.75	1087	1603	65.78	1069	1596	85.92
	120	1123	1633	52.68	1108	1618	53.71	1123	1624	79.81
0.15	40	1023	1581	77.106	1009	1556	79.108	1010	1545	85.116
	60	1034	1556	74.98	1031	1567	75.99	1056	1531	83.108
	80	1078	1595	70.89	1046	1605	71.91	1035	1584	79.103
	100	1100	1613	66.85	1106	1627	70.88	1078	1615	78.99
	120	1135	1653	60.84	1097	1632	68.85	1110	1665	71.93

Table 5 Cutting, liquid film, and oil film influence on the spindle amplitude

Gap/mm	Width/mm	Frequency/Hz		
		Oil film	Liquid film	No film
0.05	40	52	56	60
	60	49	54	59
	80	42	44	52
	100	45	45	52
	120	51	46	53
0.10	40	50	54	62
	60	47	52	60
	80	44	45	55
	100	41	46	50
	120	40	47	51
0.15	40	53	58	61
	60	50	54	55
	80	43	50	53
	100	43	49	52
	120	44	48	51

the surface roughness has improved to a better vibration damping effect than in the past (seen Fig. 6).

3.3 Experiment results and discussion

The tests indicate that no matter what static and under the cutting condition, liquid membrane (including oil film) for boring bar has good vibration damping effect, the best damping effect at more than 20%. A lot of experimental data show that, when the liquid film thickness and width are 0.1 and 60 mm, respectively, damping effect is best. In simulation, the simulation curve is in agreement with the experimental curve, and it shows the simulation is good and suited for some testing.

The experiment is divided into static and dynamic experiments. In the static experiments, liquid film damper is better than the oil film damper damping effect. In dynamic cutting experiments, the oil film damping effect is better than the liquid film damping effect in general.

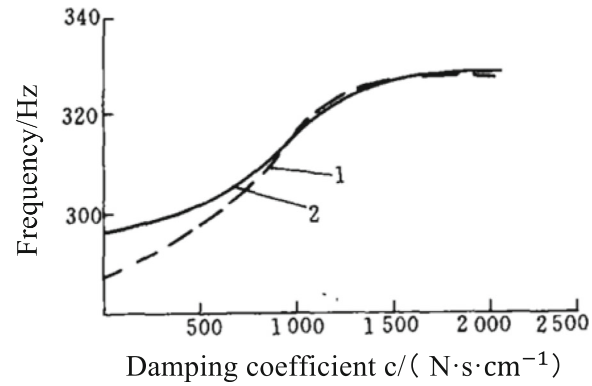


Fig. 3 Resonance frequency and the damping

20Hz

value diagram

500 N·s·cm^{-1}

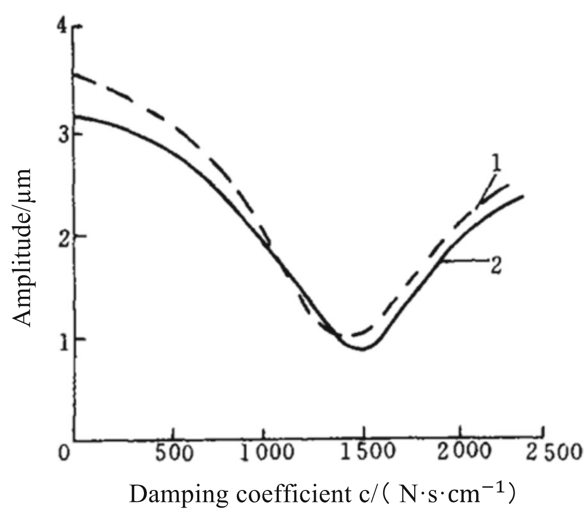


Fig. 2 Response amplitude and damping

1μm

value diagram

500 N·s·cm^{-1}

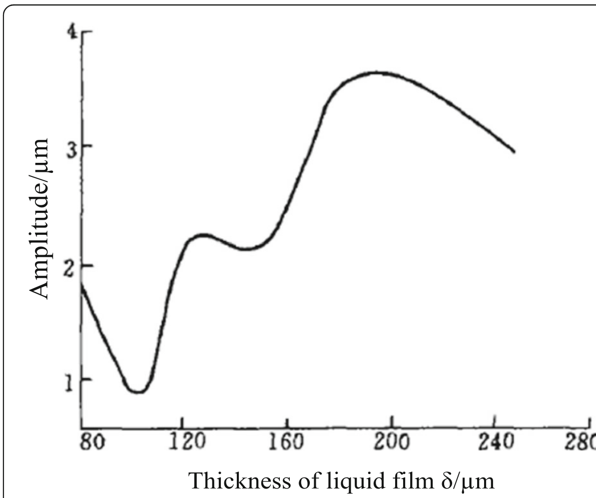


Fig. 4 Liquid film thickness and response amplitude

1μm

relation curve

40μm

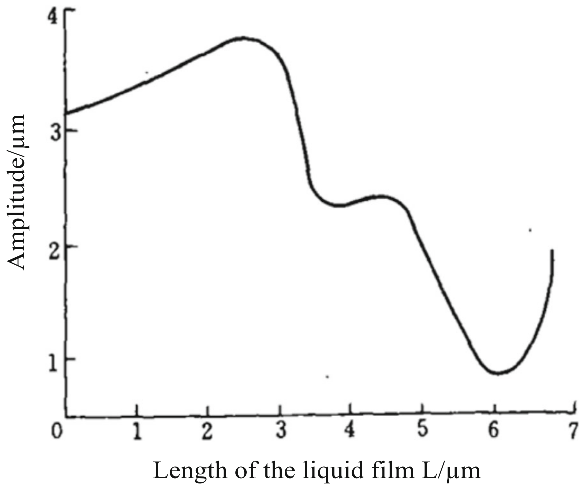
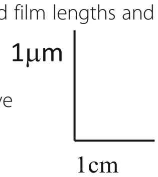


Fig. 5 Liquid film lengths and response amplitude



relation curve

Fine boring liquid film damping system can greatly improve the machine tool dynamic stiffness and cutting vibration resistance, while not reducing other performance indicators. It has a simple structure and reliable performance.

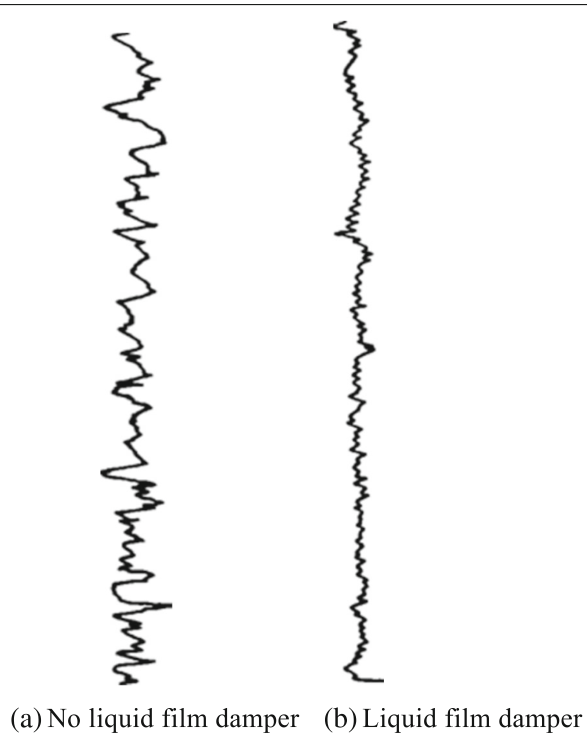


Fig. 6 Surface roughness measuring results

4 Conclusions

This paper presented an experimental study on fine boring squeezed liquid film damped system, and setting up a freedom vibration simulation model. Some mathematical model can be solved by transfer matrix theory. The mechanical device of a liquid film damping system is established. 20# machine oil, 40# machine oil, and cutting fluid are used to the system. The following conclusions can be drawn from this study.

1. The damping coefficient for the fine boring system can be obtained.
2. Using multi degree of freedom system vibration simulation model to analyze the squeeze liquid film damping system has high reliability and accuracy.
3. Using the simulation model can reduce the test cost, direct access to optimize parameters of shock absorber on the basis of parameter design and manufacture of shock absorber, and bring good effect in practice.

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Authors' contributions
QShao is the main writer of this paper. He proposed the main idea, deduced the performance of liquid film damper, completed the simulation, and analyzed the result. LWang introduced the liquid film damper for testing. DWang simulated the liquid film damper by soft. P-SGe gave some important suggestions for the liquid film damper. All authors read and approved the final manuscript.

Competing interests
The authors declare that they have no competing interests.

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