DEVELOPMENT OF A QUASI-MOMENT DAMPER*

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One of the major obstacles to raising the work efficiency of the machining process using a machine tool in the past was the phenomenon of chattering. Several factors are conceivable as the causes of chattering. It is generated, first of all, by extremely unfavorable dynamic stiffness in the structure of a machine tool. Numerous active dampers have been announced for improving the dynamic stiffness of the structure of machine tool. A Quasi-Moment Damper differs from previous methods in that attention is paid to the bending moment as a control quantity.

This paper includes the experimental results and theoretical analysis of a ram provided with a quasi-moment damper.

1. INTRODUCTION

One of the major obstacles in raising the work efficiency of the machining process using a machine tool in the past was the phenomenon of chattering. Several factors are conceivable as the causes of chattering. It is generated, first of all, by extremely unfavorable dynamic stiffness in the structure of a machine tool. In particular, unfavorable dynamic stiffness of the ram is a serious problem in large-size gate-type machine tools such as planers. Numerous researches have been conducted in the past to overcome this difficulty. However, none of the suggested improvements has been entirely satisfactory. Their applications have been attended with considerable limitations.

We wish to submit here a newly developed quasi-moment damper which, we believe, is free from previous shortcomings and enables efficient machining work.

2. BASIC CONCEPTS OF A QUASI-MOMENT DAMPER

Previous active dampers or similar devices were characterized by the direct damping of the exterior forces acting on the structure of a machine tool by applying

a control force generated in some way.

On the other hand, a quasi-moment damper differs from previous methods in that attention is paid to the moment generated by exterior force in the structure. Undesirable vibration of the structure is damped by directly applying a damping moment.

Let us consider the basic performance of a quasi-moment damper acted on by static exterior force and static damping force.

As shown in attached Fig. 1, we assume an exterior force F_{in} and a damping force F_c which are identical in magnitude and opposite in direction as F_{in} . The forces acting on the beam are now in equilibrium.

Fig. 1 Static Aspect

The bending moments of these forces (M_{in} and M_c) on the cross-sections mn and $m'n'$ also cancel each other. The quasi-moment damper serves to correct exterior force F_{in} by applying M_c directly on the beam, without applying F_c . In the dynamic aspect, a pair of quasi-moment generators are mounted on both sides of the beam's

* Read at the Lecture meeting of the 52nd All-Japan Conference on Nov. 3, 1974. MS was accepted on May 27, 1974.

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Fig. 2 Dynamic Aspect

neutral axis as shown in Fig. 2. Vibration is absorbed through the generation of positive and negative moments.

Here the major question is how to obtain the damping moment corresponding to the moment generated in the structure. As we are going to explain below, we generate a quasi-moment for this purpose. Hence, we wish to designate as "quasi-moment dampers" a series of damping systems described in this treatise.

3. THE APPLICATION OF A QUASI-MOMENT DAMPER ON THE RAM OF A LARGE-SIZE GATE-TYPE MACHINE TOOL

Extremely defective dynamic stiffness of the ram system is the chief cause of the generation of chattering in a large-size gate-type machine tool. In our model for analysis and testing we simulated the ram with a beam whose one end is fixed so as to simplify boundary conditions, since the column and the beam in the gate of a structure have much higher stiffness than the ram.

3-1. Composition of a Quasi-Moment Damper

As mentioned above, the central question in damping a moment is how to generate a counter moment. In the present case, we have conceived the following arrangement. As in Fig. 3, a quasi-moment generator features a pocket with a cross-section A placed against the sides of the beam perpendicular to its length. When a hydraulic pressure P is applied to this pocket, a force $F = AP$ acts in the direction of ram

axis. Since the pocket is located at a distance of l from the neutral axis NN' of the ram, the latter is acted upon by a quasi-moment of the magnitude of $M = Fl$. The generator used in our experiment is shown in Fig. 4, its dimensions in Fig. 5, respectively.

- (1) Cross-section A where pressure is applied
- (2) Hydraulic pressure pocket
- (3) Quasi-moment generator

Fig. 3. Working principle of a quasi-moment damper

Fig. 4. Quasi-moment generators

Fig. 5. Dimensions of a quasi-moment generator

The quasi-moment damper is composed of an electro-hydraulic servo valve, a pair of quasi-moment generators, an amplifier for driving the servo valve and a hydraulic unit. Fig. 6 shows the diagram of a quasi-moment damper.

Fig. 6. Composition of a quasi-moment damper

- a. Vibration meter
- b. Structure
- c. Quasi-moment generator
- d. Electro-hydraulic servo valve
- e. Amplifier for driving servo valve
- f. Hydraulic unit

3-2. Experimental Device and Theoretical Analysis of a Ram Provided with a Quasi-Moment Damper

The experimental device is shown in Fig. 7. Chattering vibration in a defective

Fig. 7. Ram system and vibration detecting unit

ram system is most frequently brought about by its primary frequency resonance. Therefore, the experiment and analysis are carried out at the fixed end of the ram where the moment of the primary mode is at its maximum. In our analysis of the system, the symbols employed are defined as in Table 1.

Table 1. Definitions of symbols used in the treatise

- Y_i : Displacement at the i -extremity of beam
- F_i : Shearing force at the i -extremity of beam
- M_i : Moment at the i -extremity of beam
- ϕ_i : Angle of deflection at the i -extremity of beam
- F_{in} : Exterior force, kg cm/s^2
- Y_{out} : Displacement at the extremity of ram, cm
- m_0 : Added mass of ram, kg
- m_i : Mass of beam, kg

- l_1 : Length of beam, cm
- E : Young's modulus, $2.1 \times 10^9 \text{ kg/cm s}^2$
- I : Moment of inertia of cross-section: cm^4
- E_o : Servo amplifier input voltage, V
- I_o : Servo valve input current, mA
- P : Pressure, kg cm/s^2
- Q : Flow at discharge port of servo valve, cm^3/s
- F : Force, kg cm/s^2
- X : Relative displacement of generator, cm
- A : Area exposed to pressure on generator, cm^2
- r_p : Internal resistance of servo valve, $35 \text{ kg/cm}^4\text{s}$
- V : Volume of enclosed oil, cm^3
- β : Compressibility of oil, $\text{cm s}^2/\text{kg}$
- K : Gain of servo amplifier
- $G_1 = 1 / \{ (m_1 l_1^3 / 8EI) s^2 + 0.01 \sqrt{m_1 l_1^3 / 8EI} s + 1 \}$
- $G_2 = 1$
- $G(s) = 1 / (\frac{1}{440} s + 1) (\frac{1}{(1880)^2} s^2 + \frac{0.6}{1880} s + 1)$
- $I(s)$: Feedback transfer function $= K_1 + K_2 s$
- K_1 : Displacement feedback gain at the extremity of ram, V/cm
- K_2 : Velocity feedback gain at the extremity of ram, V/cm/s
- K_3 : Relative displacement feedback gain of generator, V/m
- μ : Pressure amplification degree, $\text{kg/s}^2\text{mAcm (100)}$

8-terminal matrix, (1)(2) as conceived by Seto, was employed for the analysis of the device shown in Fig. 7. This represents in analog model the dynamic relations among 4 variables of deflection Y , angle of deflection ϕ , moment M and shearing force F .

For the convenience of analysis, shown in Fig. 8, the components are divided as 5 beams and one mass. The relations among components are expressed in a series of boundary conditions listed in Table 2.

Fig. 8. Closed loop vibration model

Fig. 9. Definition of coordinates and variables

Table 2. Boundary Conditions

- (1) At the free extremity of the component
 $M_1 = 0, -F_1 + F_{in} - m_0 S^2 Y_1 = 0$
- (2) At the joints of components (I), (II) & (III)
 $Y_2 = Y_3, \phi_2 = \phi_3 = \phi_7$
 $M_2 + M_7 - M_3 = 0, -F_2 + F_3 + m_4 S^2 Y_2 = 0$
- (3) At the joints of components (II), (III) & (V)
 $Y_4 = Y_5, \phi_4 = \phi_5 = \phi_9$
 $M_4 + M_9 - M_5 = 0, -F_4 + F_5 + m_5 S^2 Y_4 = 0$
- (4) At the fixed end of the component (III)
 $Y_6 = 0, \phi_6 = 0$
- (5) At the upper extremity of the component (IV)
 $M_8 = 0$
- (6) At the upper extremity of the component (V)
 $M_{10} = 0$
- (7) At the joints of components (IV), (V) and generators
 $Y_{10} - Y_8 = X, F_{10} = F, -F_8 = F$

Each signal flowchart is connected according to these boundary conditions. The coordinates of 4 variables at the terminal of each beam are taken as in Fig. 9. Since the moments $M_1 = M_8 = M_{10} = 0$ at the input terminals of beams (I), (IV) and (V), the adopted signal flow charts should be such as to allot said moments to the input terminals of these beams.

If, furthermore, we remember that ϕ_1 , ϕ_8 and ϕ_{10} cannot apply to input terminals as they are indeterminate, the disposition of variables for the signal flow chart of the entire system becomes as Fig. 10. Necessary flow of signals according to boundary conditions is expressed in thick line.

Fig. 11 is a signal flow chart of the servo valve system. The signal flow charts for the beam system and the servo valve system are connected according to

Fig. 10. Signal flow chart of beam

Fig. 11. Signal flow chart of servo valve system

boundary condition, table 2, expressing the signal flow for the entire beam system of this quasi-moment damper.

However, dynamic changes of beams (II), (III), (IV) and (V) can be ignored since they are very short compared with the beam (I). In this case, the signal flow chart may be simplified as Fig. 12.

Fig. 12. Simplified signal flow chart of the entire system

3.2.1. Analysis by Means of Root Locus Method

The characteristic equation, obtained from the signal flow chart of Fig. 12, is reproduced as follows.

$$a_1 s^6 + a_2 s^5 + a_3 s^4 + a_4 s^3 + a_5 s^2 + a_6 s + a_7 + (b_1 s^2 + b_2 s + b_3) I(s) = 0$$

The coefficients are expressed by the following formulas:

$$\begin{aligned} a_1 &= -0.649 \times 10^{-9} A r_p D_1 \\ a_2 &= 0.649 \times 10^{-9} \frac{A^2 r_p l_2 l_4^2}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} - 1.01 \times 10^{-6} A r_p D_1 + 0.649 \times 10^{-9} C_1 \\ a_3 &= 0.649 \times 10^{-9} \frac{A^2 r_p l_2 l_4^2}{EI} + 1.01 \times 10^{-6} \times \frac{A^2 r_p l_2 l_4^2}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} - 2.59 \times 10^{-3} A r_p D_1 \\ &\quad + 0.649 \times 10^{-11} \sqrt{\frac{m_1 l_1^3}{8EI}} + 1.01 \times 10^{-6} C_1 \\ a_4 &= 1.01 \times 10^{-6} \frac{A^2 r_p l_2 l_4^2}{EI} + 2.59 \times 10^{-3} \times \frac{A^2 r_p l_2 l_4^2}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} - A r_p D_1 - 0.649 \times 10^{-9} \\ &\quad + 1.01 \times 10^{-6} \sqrt{\frac{m_1 l_1^3}{8EI}} + 2.59 \times 10^{-3} C_1 \\ a_5 &= 2.59 \times 10^{-3} \frac{A^2 r_p l_2 l_4^2}{EI} + \frac{A^2 r_p l_2 l_4^2}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} + G_K r_p K_3 \mu D_1 \\ &\quad + 1.01 \times 10^{-6} + 2.59 \times 10^{-3} \sqrt{\frac{m_1 l_1^3}{8EI}} + C_1 \\ a_6 &= \frac{A^2 r_p l_2 l_4^2}{EI} - \frac{G_K r_p K_3 \mu A l_2 l_4^2}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} + 2.59 \times 10^{-3} + 0.01 \sqrt{\frac{m_1 l_1^3}{8EI}} \\ a_7 &= 1 - \frac{A l_2 l_4^2 G_K r_p K_3 \mu}{EI} \\ C_1 &= \frac{l_1^3}{EI} \left(\frac{m_1}{8} + \frac{m_0}{3} \right) + \frac{l_1^2 (l_2 + l_3)}{EI} \left(\frac{m_1}{3} + m_0 \right) \\ &\quad + \frac{l_1 (2l_2^2 + 2l_3^2 + 3l_2 l_3)}{EI} \left(\frac{m_1}{2} + m_0 \right) \\ D_1 &= -\frac{A l_2 l_4^2}{EI} C_1 - A l_1 \left(\frac{l_2 l_4}{EI} \right)^2 \\ &\quad \times \left\{ m_1 \left(\frac{l_2}{4} + \frac{l_1}{3} \right) + m_0 \left(\frac{l_2}{2} + l_1 \right) \right\} \\ b_1 &= \frac{A m_1 l_1 l_2 l_4 G_K r_p \mu}{(EI)^2} \left[-\frac{l_1^2}{8} \left(\frac{l_2}{2} + l_1 \right) - \left(\frac{l_2}{2} + l_1 \right) \left(\frac{2l_2^2 + 2l_3^2 + 3l_2 l_3}{8} + \frac{l_1 (l_2 + l_3)}{3} \right) \right. \\ &\quad \left. + \left(\frac{l_2}{4} + \frac{l_1}{3} \right) \left\{ l_1 (l_2 + l_3) + \frac{l_1^2}{3} + \frac{2l_2^2 + 2l_3^2 + 3l_2 l_3}{4} \right\} \right] \\ b_2 &= -\frac{A l_2 l_4 G_K r_p \mu}{EI} \sqrt{\frac{m_1 l_1^3}{8EI}} \left(\frac{l_2}{2} + l_1 \right) \times 10^{-2} \\ b_3 &= -\frac{A l_2 l_4 G_K r_p \mu}{EI} \left(\frac{l_2}{2} + l_1 \right) \end{aligned}$$

A quasi-moment damper is mounted on the beam, measuring 100 cm in length and 6.5 by 2.5 cm in cross-section, at a distance of 10 cm from the fixed end. Fig. 13

indicates the root locus of the loop transfer functions obtained from the signal flow chart of Fig. 12. The solid line corresponds to the case where only the displacement at the extremity of the ram is fed

Fig. 13. Root loci (for the ram 100 x 6.5 x 2.5 cm in size, where generators are mounted at a distance of 10 cm from the fixed end)

back. Two-dot chain line corresponds to the case where only the velocity at the extremity of the ram is fed back. One-dot chain line corresponds to the case where displacement and velocity, the latter being weighted with a factor of 1/100, are fed back. The dotted line corresponds to the case where displacement and velocity, the latter being weighted with a factor of 1/50, are fed back.

The root locus of the displacement feedback (solid line in Fig. 13) turns out to be a positive feedback owing to the pole's behaviour in the real axis. A locus starting from the pole due to ram characteristics (+281.2) rapidly approaches the real axis if one increases the feedback gain, meets the real axis and branches off in two curves, one of which turns to the right after passing through the origin. The other branch proceeds to the pole due to servo valve characteristics (-439.9). Hence, an oscillation occurs in the DC component.

The root locus of the velocity feedback (two-dot chain line in Fig. 13) also belongs to the positive feedback. The locus starting from the pole due to ram characteristics goes to the left in parallel to the real axis, while that from the pole due to servo valve characteristics (-439.9) proceeds along the real axis towards the origin.

These root loci evidently belong to positive feedback. The displacement feedback brings the system into a state of overdamping. However, the resonance point

comes at too low frequency. The velocity feedback alone, on the other hand, can keep down the damping factor of the system to about 0.7. These data show that satisfactory results are obtained through velocity feedback. The state of overdamping could possibly be brought about without lowering the resonance frequency of the system by the combined feedback of displacement and velocity as shown by dotted line and one-dot chain line in Fig. 13.

3.2.2. Analysis by Means of an Analog Computer

We used the simulation circuit of Fig. 14. for a more detailed analysis of the effects of this quasi-moment damper. The results of computation using the same beams as shown in root locus are indicated in Figs. 15 and 16. The displacement Y_{out} at the extremity of the ram is plotted against exterior force F_{in} .

Fig. 14. Simulation circuit of analog computer

Fig. 15. Feedback of displacement at the extremity of the ram

Fig. 16. Feedback of velocity at the extremity of the ram

Fig. 17. Feedback of displacement and velocity

In Fig. 15, the feedback is carried out only for displacement, while in Fig. 16 the feedback is carried out only for velocity. And in Fig. 17, the feedback is carried out for displacement and velocity. From these data we can say that position feedback reduces resonance frequency and amplitude but static stiffness deteriorates owing to the increased gain in low frequency range, while velocity feedback promises good results.

Table of feedback gain

Symbol	Fig. 15	Fig. 16	Fig. 17
●	$K_1 = 0$	$K_2 = 0$	$K_1 = 2.0,$ $K_2 = -0.02$
×	$K_1 = 0.5$	$K_2 = -0.1$	$K_1 = 5.0,$ $K_2 = -0.05$
△	$K_1 = 1.0$	$K_2 = -0.2$	
□	$K_1 = 1.5$	$K_2 = -0.3$	
○			$K_1 = 8.0,$ $K_2 = -0.08$

4. EXPERIMENTS AND THEIR RESULTS

An entire device for experiment is

shown in Fig. 18. Model 3F-30L of Tokyo Seimitsu Keiki K.K., with somewhat modified specifications, was used for the servo valve. Fig. 7 shows a vibration detector for feedback. We see, from right to left, a velocimeter making use of the relative

Fig. 18. Experimental device

motion of a magnet and a coil, variable inductance displacement gauge and tiny capacity-type displacement gauge for detecting relative displacements at the upper extremity of the quasi-moment generator.

TEAC's R-200 data recorder served for recording output signals.

The displacement at the extremity of the ram is plotted against impulse for a beam 100 cm in length and 6.5 x 2.5 cm in cross-section in Fig. 19. The same response is shown for a beam 100 cm in length and 6.0 x 4.5 cm in cross-section in Fig. 20. (a) is the response without feedback. In (b) a set of generators was mounted on the beam at a distance of 10 cm from the fixed end and the displacement at the extremity of the ram as well as the relative displacement at the upper extremity of generators was fed back. (c) is the same feedback as (b), with a difference in that quasi-moment generators are mounted at the center of the beam. The comparison of (a) and (b) reveals the appreciable damping effect of the quasi-moment damper. The damping ratio has been increased from 0.005 to 0.1. On the other hand, the comparison of (b) and (c) reveals that the damping effect varies with the position of quasi-moment generators. As mentioned at the beginning of the analysis, mounting near the fixed end of the ram is the most effective.

(a) Without feedback.
Horizontal axis: 2s/div.

(b) Generators mounted at
10 cm from fixed end.
Horizontal axis: 0.1 s/div.

(c) Generators mounted
at center. Horizontal
axis: 0.1 s/div.

Fig. 19

(a) Without feedback.
Horizontal axis: 2s/div.

(b) Generators mounted at
10 cm from fixed end.
Horizontal axis: 50 ms/div.

(c) Generators mounted
at center. Horizontal
axis: 0.1 s/div.

Fig. 20

Figs. 21 and 22 show the frequency responses obtained through Fourier transform of impulse responses in Figs. 19 and 20. In Figs. 21 and 22, the responses

Fig. 21. Response for ram, 100 x 6.5 x 2.5 cm

Fig. 22. Response for ram, 100 x 6.0 x 4.5 cm

corresponding to (a), (b) and (c) are marked with $\times$, $\circ$ and $\bullet$, respectively. Higher value of resonance frequency in Fig. 21 than the theoretical value in Fig. 16 is probably accounted for by the greater stiffness of the beam at the position where the quasi-moment damper is mounted. The gain at the resonance point is kept down to ca. 10 dB when generators are mounted at 10 cm from the fixed end. It is further reduced to ca. 5 dB in Fig. 22, showing a sufficient effect of the quasi-moment damper.

For velocity feedback the results of theoretical analysis did not agree with experimental results well. This is probably because the spool of the servo valve did not respond owing to small feedback gain. The agreement of theory and practice was better for displacement feedback.

In order to improve the static rigidity of the beam, we obtained the loop transfer function of $\frac{Y_{out}}{F_{in}}$ consisting of DC component of signal flow in Fig. 12. i.e. assuming $S = j\omega = 0$ and that there is no signal input from S branch. The result is as follows:

$$\frac{Y_{out}}{F_{in}} = \frac{\mathcal{L}_1 AX (1 - BXK_3)}{1 - AYK_1}$$

where

$$AX = \frac{l_1^2}{3EI} + \frac{l_1(l_2 + l_3)}{EI} + \frac{2l_2^2 + 2l_3^2 + 3l_2l_3}{4EI}$$

$$AY = Kr_p \mu A \left(\frac{l_2^2 l_4}{2EI} + \frac{l_1 l_2 l_4}{EI} \right)$$

$$BX = Kr_p \mu A \left(\frac{l_2 l_4^2}{EI} + \frac{l_2 l_4}{EI} \frac{l_2^2 l_4 + 2l_1 l_2 l_4}{2EI} \right)$$

indicating that the static stiffness is improved through suitable selection of K_3 , the feedback constant of relative displacement at the upper extremity of the quasi-moment generator, as well as of the displacement feedback constant at the extremity of the ram. In Figs. 19 and 20, the feedback was effected at approximately $K_3 = 5V/cm$. Actually the damping characteristic was not so well improved as theoretically predicted, probably owing to the saturation of hydraulic power. We are still continuing the analysis to ascertain the cause.

5. CONCLUSIONS

It has been confirmed that the following advantages are obtained by means of the above-mentioned quasi-moment damper.

- (1) The closed loop adopted in this technique is effective for a temporarily changing vibratory system, too.
- (2) The damper features simple structure, compact size and light weight, occupying little installation space on the structural object.
- (3) The pressure control valve, which can be used as servo valve of this device may allow for a wider controllable range of frequencies than the flow control valve.
- (4) The damping of frequency modes of higher orders in the structure is also a simple matter of adjusting positions and number of such dampers.

The damping of vibration through control force appears to contribute to the improvement of static stiffness characteristics. We are continuing the improvement of control method in both theory and practice.

In conclusion we should like to express our heartfelt thanks to Mr. Iwata, Assistant in Engineering Department, Tokyo Metropolitan University, and Mr. Seto, Assistant Professor of National Defense Academy, and we wish to express sincere appreciations to Professor S. Sato (Institute of Industrial Science, University of Tokyo) for his suggestions.

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