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# Damping force-based cutting force monitoring in ball-screw-driven stage

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**Abstract**
In this study, a sensorless cutting force monitoring technique by damping force was proposed for estimating resonance frequency components in the ball-screw-driven stage. At first, concept of the damping force monitoring was presented based on the equilibrium of forces in the single-degree-of-freedom system. Considering application to the ball-screw-driven stage, estimating equation for dual-inertia model was developed. Damping force was calculated from velocity of the table. The result of the end milling test showed that tooth-pass frequency component around the resonance frequency could be estimated accurately.

Sensorless; Monitoring; Damping force; Ball-screw-driven stage;

## 1. Introduction

Cutting force is one of the most process related information in machining, and process monitoring technology has been developed focusing on cutting force [1]. Sensorless cutting force monitoring on the basis of inner information of the machine tools has an advantage in terms of sustainability and maintainability. A number of estimating methods were proposed using current signal of the servo motor [2], Kalman filter [3], and disturbance observer [4, 5]. Without using additional sensors, however, wideband cutting force monitoring is still difficult in the ball-screw-driven stage due to structural resonances. In this study, a sensorless cutting force monitoring technique by damping force was proposed for estimating the frequency component around the resonance frequency. Damping force was calculated by using velocity of the table. The validity of the proposed method was verified by end milling tests.

## 2. Methodology

### 2.1. Concept of damping force monitoring

Figure 1 shows equilibrium of forces in forced vibration of single-degree-of-freedom system (SDOF) by using bode diagram. Dominant force contributing to equilibrium of load force, $F_l$, varied depending on the frequency of $F_l$. Seen from the figure, $F_l$ equilibrates with damping force, $F_c$, around the resonance frequency, while $F_m$ and $F_k$ are cancelled [6]. In this study, damping force was monitored for estimating cutting force component around the resonance frequency.

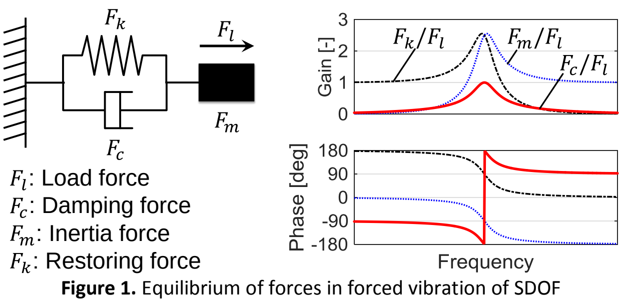


**Figure 1.** Equilibrium of forces in forced vibration of SDOF

### 2.2. Extension to dual-inertia model

Considering dynamic characteristics, ball-screw-driven stage is often modelled as dual-inertia system shown in Figure 2. Here, $M$, $C$ and $x$ represent mass, damping and position, respectively. Rotational and translational components were distinguished by using subscript $m$ and $t$. $K_r$ and $C_k$ denote overall axial stiffness and damping of structure. $F_{th}$ and $F_{cut}$ represent motor thrust force and cutting force. Static coulomb friction terms were ignored for simplification. Thus, $F_{cut}$ corresponded to $F_l$. Proportional damping was considered. In this study, only translational elements were focused in order to regard the system as SDOF. In that case, damping force could be calculated as follows:

$$F_c = (C_k + C_t)v_t \tag{1}$$

where $v_t$ is velocity of the table. On the contrary to the SDOF system, $F_{cut}$ does not necessarily equilibrate with $F_c$ around the resonance frequency in the dual-inertia system. Figure 3 shows gain characteristics between damping forces and cutting force by using mechanical parameters identified by impulse response method. As shown in the figure, gain is not one at the resonance point. Thus, appropriate scaling is required for estimating cutting force. Because resonance frequency can be computed as $\sqrt{\alpha+1}\omega_t$ in dual-inertia system, the peak value of $(C_k + C_t)v_t/F_{cut}$ defined as $A$ can be approximately calculated as follows:

$$A \cong \frac{C_k + C_t}{C_t + \alpha^2 C_r + (\alpha+1)^2 C_k} \tag{2}$$

where $\alpha = M_t/M_m$ and $\omega_t = \sqrt{K_r/M_t}$. Note that Eq. (2) is applicable when resonance frequency is higher than bandwidth of the position and velocity control. If this is not the case, the influence of the controller gain needs to be included in the calculation of $A$. Estimating equation of cutting force is introduced with band-pass filter, $G_{BPF}$, for extracting resonance frequency components around which damping force monitoring was applicable as follows:

$$\hat{F}_{cut} = G_{BPF} \cdot \frac{(C_k + C_t)v_t}{A} \tag{3}$$

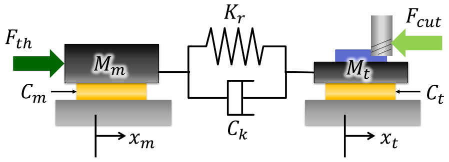


**Figure 2.** Dual-inertia model of ball-screw-driven stage

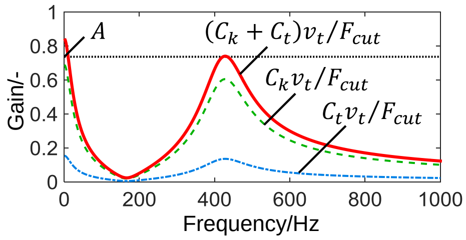


**Figure 3.** Gain characteristics between damping and cutting force

## 3. Experimental verification

### *3.1. Experimental setup*

The single-axis ball-screw-driven stage is shown in Figure 4. An AC servomotor is used as an actuator, and its shaft and screw are directly connected by a coupling. The pitch length of the screw is 5 mm, and the stage is supported by rolling guideways. Feedback signals are provided by the rotary encoder (resolution: 17 bit) built into the motor and the linear encoder attached to the table (LIF471R, resolution: 20 nm from HEIDENHAIN). The sampling frequency of the data acquisition system was set at 20 kHz. A dynamometer (Type 9129A, from Kistler) was mounted on the table to accurately measure the cutting forces for comparison. Frequency response function of the table was shown in Figure 5. Parameters were identified by numerical iteration method, and the value is as follows: $M_t$ = 6.7 kg, $M_m$ = 36.8 kg, $C_t$ = 8.02 x $10^2$ Ns/m, $C_r$ = 4.40 x $10^3$ Ns/m, $C_k$ = 3.59 x $10^3$ Ns/m, $K_r$ = 3.59 x $10^3$ Ns/m, $A$ = 0.736.

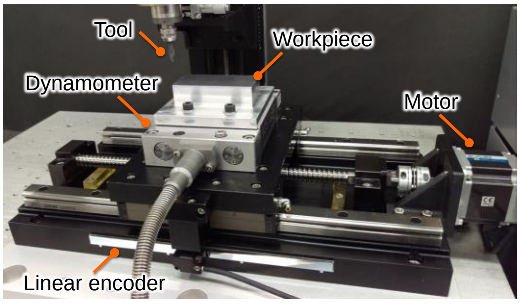


**Figure 4.** Experimental setup for cutting tests

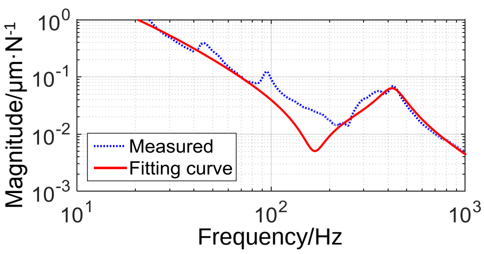


**Figure 5.** Frequency response function of stage

### *3.2. Experimental results*

For evaluating the monitoring performance by damping force, end milling tests were carried out. In the tests, slots were machined under constant feed rate. Velocity of the table was calculated from position signal from linear encoder. Pass band of BPF was set to 350 – 450 Hz so that resonance frequency was included within the pass band. Another cutting condition is listed Table 1. As shown in Figure 6, estimated cutting force and measured cutting force with BPF were in good agreement whole the time. Tooth-pass frequency (i.e. 367 Hz) component of cutting force could be monitored accurately as shown in Figure 7 which is expanded view of Figure 6.

**Table 1** Experimental conditions

| | |
|---|---|
| Spindle speed [min$^{-1}$] | 11000 |
| Axial depth of cut [μm] | 180 |
| Feed per tooth [μm/tooth] | 30 |
| Cutting tool | Square end mill |
| Tool diameter [mm] | ϕ 6.0 |
| Number of flutes | 2 |
| Workpiece material | Al alloy (A5052) |
| Type of BPF | Butterworse (6$^{th}$ order) |

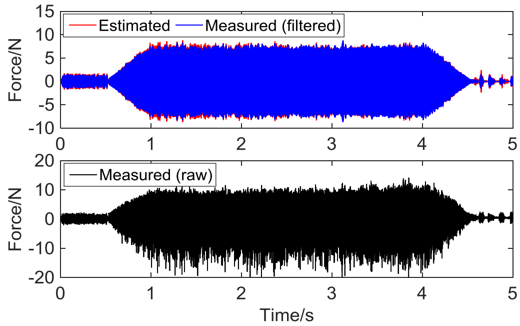


**Figure 6.** Overall view of experimental result (Top: estimated damping force by Eq. (3) and measured force with BPF, Bottom: measured raw force)

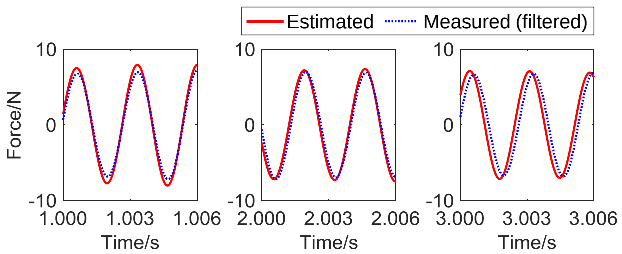


**Figure 7.** Enlarged view of experimental result shown in Figure 6.

## 4. Conclusion

In this study, a sensorless cutting force monitoring technique by damping force was proposed for estimating resonance frequency component. Considering application to the ball-screw-driven stage, estimating equation for dual-inertia system was introduced analytically. The result of the end milling test showed that tooth-pass frequency component around the resonance frequency was estimated accurately. In future work, authors will apply the proposed method to active resonance suppression.

**Acknowledgement**

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