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Damping Characteristics of Balance-weight Oscillation and Tensile Force Stability under A New Bandsaw Straining Device

By

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Summary : Vibration of a bandsaw blade, which leads to sawing inaccuracies, can stem from several possible causes. One frequently overlooked cause is the change of bandsaw tensile force induced by a change of cutting force and oscillation of the balance-weight. In order to prevent or reduce the oscillation of the balance-weight, the author conceived a new bandsaw straining device incorporating a damper which is simply comprised of two flexible coil springs and an air -cylinder dashpot. In laboratory tests, this straining device had a marked damping effect on balance-weight oscillation while balance-lever movement was unhindered. With the extreme reduction of balance-weight oscillation, bandsaw tensile force was nearly completely stabilized.

1 Introduction

In a bandsawing operation, the bandsaw expands and contracts as bandsaw tensile force changes. This expansion and contraction is manifested in the balance-lever oscillation which in turn causes the balance-weight itself to oscillate up and down. In a previous paper (FUJIWARA, 1996) the author demonstrated that changes in bandsaw tensile forces can be attributed in part to weight changes brought about by the oscillating balance-weight. Excessive changes of bandsaw tensile force can cause vibration or running instability of the bandsaw blade, thereby reducing sawing accuracy and/or inducing kerf bend.

In order to prevent or minimize these undesirable effects, the author has been designing a damper to be incorporated into a conventional balance-weight type bandsaw straining device. The idea is to prevent oscillation or to quickly dampen any induced oscillation of the balance-weight. In the present paper, the author clarifies the oscillation characteristics of balance-weight and changes in bandsaw tensile force under a condition of free-damped balance-lever oscillation.

2 Insulation of Balance-weight Oscillation

2.1 Theory

Fig. 1 shows the oscillation system model of the new bandsaw straining device incorporating a damper composed of spring and dashpot. When the balance-lever oscillates with $a \sin \omega t$ in a one-degree linear oscillation system, the balance-lever oscillation is not transmitted to the balance-weight if the following condition is satisfied (TANIGUCHI, 1966). :

$$\frac{\omega}{\omega_n} \gg 1 \tag{1}$$

where ω is angular frequency (rad/s) and ω_n is angular natural frequency (rad/s). And ω_n in a one-degree linear oscillation is :

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$$\omega_n=\sqrt{\frac{k}{m}}=\sqrt{\frac{kg}{W}} \tag{2}$$

where k is spring constant (kgf/cm), m is mass (kgfs^2/cm), g is acceleration of gravity (cm/s^2), and W is weight of balance-weight (kgf). From Equations (1) and (2),

$$2\pi f \gg \sqrt{\frac{kg}{W}} \tag{3}$$

where f is natural frequency. In order to transform Equation (3) to an equality, we use coefficient M.

$$2\pi f=M\sqrt{\frac{kg}{W}} \tag{4}$$

Setting the conditions to increase M in Equation (4), for example, by using a flexible spring with a smaller spring constant, can improve the performance of the damper , and suppress nearly all transmission of balance-lever oscillation to the balance-weight.

2.2 Damper design and construction

As shown in Fig. 2 and Table 1, a damper composed of two flexible coil springs and an air cylinder as dashpot was assembled and installed as an integral component of a balance-weight type bandsaw straining device. The spring constant k of a coil spring (UDOGUCHI, 1966) is :

$$k=\frac{Gd^4}{8nD^3} \tag{5}$$

where G is the modulus of shearing elasticity (kgf/cm^2), d is the diameter of a spring wire (cm), n

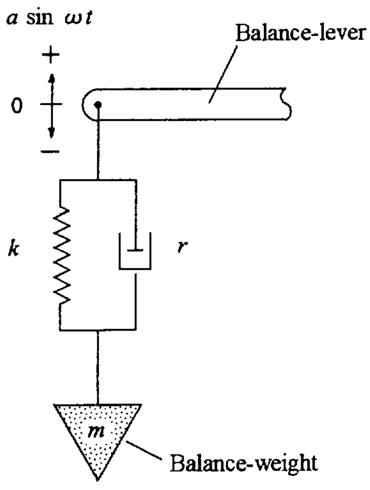

Fig. 1　Oscillation system model of bandsaw straining device composed of balance-lever, spring, dashpot and balance-weight

Legend : a : amplitude, ω : angular frequency, t : time, k : spring constant, r : viscous attenuation constant, m : mass

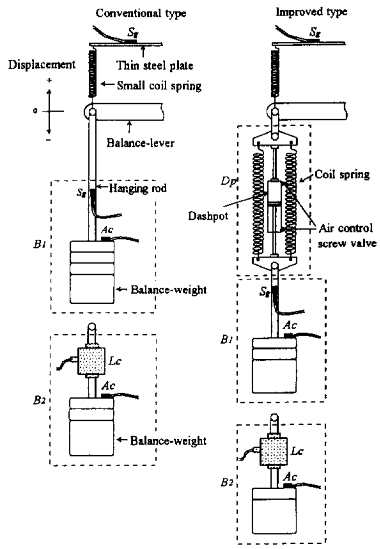

Fig. 2　Bandsaw straining device used

Legend : S_g : strain gauge, A_c : acceleration transducer, L_c : load cell, D_p : damper

Note : Two different balance-weights B_1 and B_2 were used

Table 1. Specifications of damper and experimental conditions

Condition		*1*	*2*	*3*	*4*
Type of balance-weight*		B_1	B_1	B_2	B_2
Weight of balance-weight	(kgf)	21.3	19.8	20.4	17.8
Weight of load cell	(kgf)	0	0	3.4	3.4
Weight of damper	(kgf)	0	1.5	0	2.6
Total weight of balance-weight	(kgf)	21.3	21.3	23.8	23.8
[Damper]					
Number of coil spring		—	2	—	2
Number of coil n		—	79	—	180
Mean diameter of coil D	(cm)	—	2.2	—	2.2
Diameter of spring wire d	(cm)	—	0.3	—	0.3
Spring constant(parallel) k	(kgf/cm)	—	3.00	—	1.33
Modulus of shearing elasticity G	(kgf/cm^2)	—	10.1×10^5	—	10.1×10^5
Number of dashpot		—	1	—	1
Air control screw valve of dashpot		—	full open	—	full open

Note : * : See Fig. 2.

Table 2. Specifications of bandsaw machine and bandsaw blade

Condition		*A*	*B*
Diameter of wheel	(mm)	913	913
Length of balance-lever	(mm)	683	846
Natural frequency of balance-lever	(Hz)	3.00	2.38
Balance-lever ratio		47.8	58.8
Total weight of balance-weight	(kgf)	21.3	23.8
Thickness of bandsaw	(mm)	0.90	0.90
Width of bandsaw	(mm)	93.0	93.0
Straight part length of bandsaw	(cm)	166.0	166.0
Number of revolution	(rpm)	0	0, 500

is the number of coils, and D is the mean diameter of a coil (cm). As previously mentioned, damping can be improved by using a flexible spring having a smaller value of k. In this experiment the author changed the value of k by changing the value of n. The value of k obtained from the experiment and the value of G calculated by Equation (5) using the value of k are shown in Table 1.

Under condition *4* in Table 1 and condition B in Table 2, coefficient M calculated by Equation (4) is 1.91. The value of W used here is 21.2 kgf (weight of balance-weight 17.8 kgf+weight of road cell 3.4 kgf) in a static condition. In order to improve the damper performance under the conditions of $f=2.38$ Hz and $W=21.2$ kgf, we should set $M=3.00$, which means we must use the coil spring with $k=0.54$. Because two coil springs are used in the parallel arrangement shown in Fig. 2, it follows that each of their spring constants is $k/2$, which works out to be 0.27.

3 Experimental Methods

Under two different test conditions, i.e., stationary and running, the balance-lever was oscillated up and down by hand in order to induce oscillation of the balance-weight. Fig. 2 shows the end of a thin steel plate connected to the balance-lever end by a small flexible coil spring. Any displacement of the balance-lever end is detected by a strain gauge affixed to the steel plate. In this fashion, balance-lever movement is not hindered at all by the spring. When the balance-lever is perfectly horizontal, the strain gauge registers a zero reading. Any upward or downward displacement of the balance-lever would register a positive or negative reading, respectively.

In order to detect any weight change of balance-weight due to oscillation, a device comprising a strain gauge put on a hanging rod, load cell between hanging rod and balance-weight and acceleration transducer on balance-weight, as shown in Fig. 2, was employed. Changes in band saw tensile force in the stationary state of the bandsaw were monitored by a strain gauge affixed to the center of the bandsaw blade at the straight portion of the bandsaw.

4 Results and Discussion

In a preliminary run, the most discernible damping effect was obtained with the two air control screw valves of the dashpot fully open. Therefore all subsequent tests were carried out with the screw valves in this position.

The free-damped oscillation waves obtained under the stationary state of the bandsaw for the conventional straining device and the improved unit shown in Figs. 3 and 4, respectively.

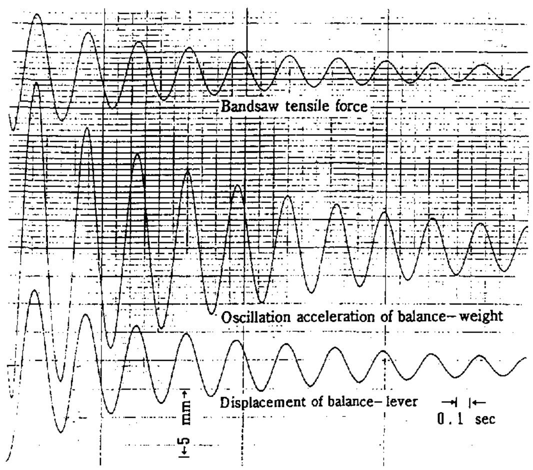

Fig. 3 Free-damped oscillations of balance-lever and balance-weight, and change of bandsaw tensile force in conventional straining device

Note: Under conditions *1* and *A*, and stationary state of bandsaw

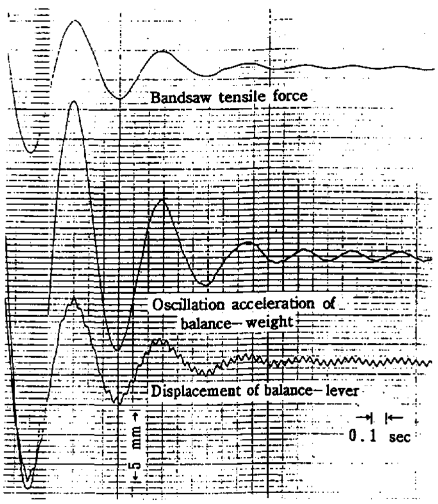

Fig. 4 Free-damped oscillations of balance-lever and balance-weight, and change of bandsaw tensile force in improved straining device

Note: Under conditions *2* and *A*, and stationary state of bandsaw

The damping effect of balance-weight oscillation in the latter was very pronounced indeed, as can be seen in comparison. It should be noted that the corresponding bandsaw tensile force was also rapidly stabilized.

In a previous experiment, FUJIWARA (1996) found that when the balance-lever position was shifted from its horizontal position without balance-weight oscillation, the corresponding change in bandsaw tensile force did not register clearly on the strain gauge affixed to the center of the bandsaw blade. Therefore, the change of bandsaw tensile force due to the shifting of the balance -lever position has been omitted from consideration in this paper.

The mean values of attenuation ratio obtained from free-damped oscillation waves were 1.22 for the conventional straining device and 2.24 for the improved unit. Using these values and setting the damped oscillations to start at $W_s=5.0$ kgf (with initial balance-lever displacement $D_s=9.4$ mm) as the initial weight change, we could get the two exponential damped curves shown in Fig. 5. For the improved straining device, pronounced damping was obtained and the weight change was reduced to 0.64 kgf after only 1.0 sec.

An even more remarkable damping effect was achieved under conditions *4* and *B*, as shown in Fig. 6. At only 0.5 sec after the start of induced oscillation (at point S), both balance-lever and balance-weight oscillations subsided. The obviously improved performance of the damper with spring constant of 1.33 over that with 3.00 is evidenced by a comparison between Fig. 4 and 6.

The free-damped oscillations of balance-lever and balance-weight for the conventional bandsaw straining device under conditions *3* and *B* and at wheel speed of 500 rpm are depicted in Fig. 7. It should be noted that the balance-lever displacement cycles coincided with those of the balance-weight. At point *T*, balance-lever displacement was −16.3 mm while the corresponding weight change was 3.09 kgf.

Fig. 8 shows similar oscillation cycles for the improved straining device under conditions *4* and *B* and 500 rpm. The balance-lever displacement at point *S* is 16.3 mm while the correspond-

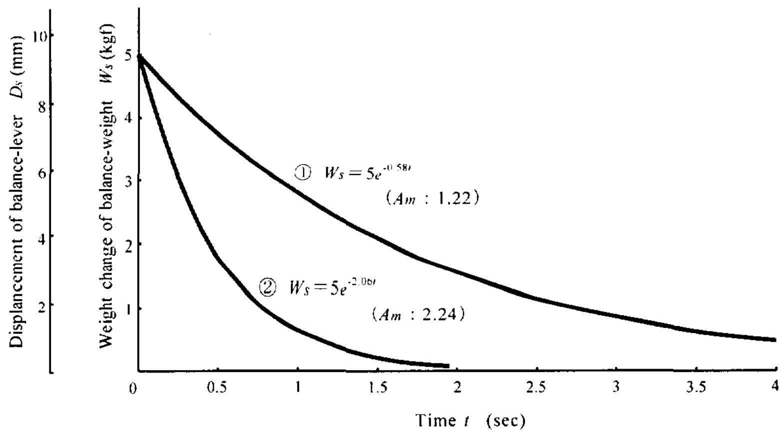

Fig. 5 Weight change curves of balance-weight under free-damped balance-weight oscillation

Note : On the assumption that the free-damped balance-weight oscillation starts at $W_s=5.0$ kgf (with $D_s=9.4$ mm). A_m is the mean value of attenuation ratio.

① : Under conditions *1* and *A*, and stationary state of bandsaw,

② : Under conditions *2* and *A*, and stationary state of bandsaw

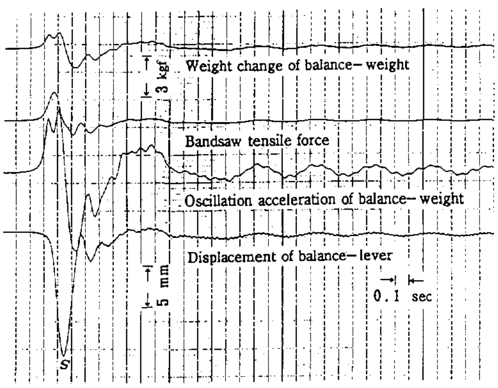

Fig. 6　Free-damped oscillations of balance-lever and balance-weight, weight change of balance-weight, and change of bandsaw tensile force in improved straining device

Note : Under conditions *4* and *B*, and stationary state of bandsaw

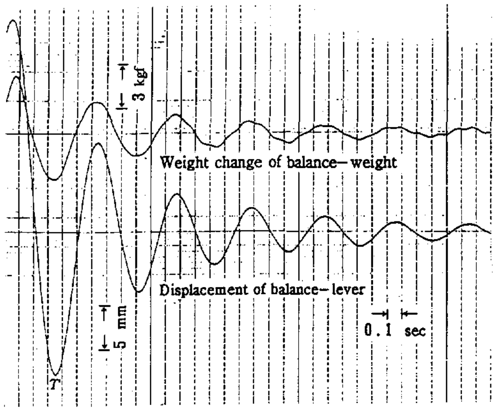

Fig. 7　Free-damped oscillations of balance-lever and balance-weight in conventional straining device

Note : Under conditions *3* and *B*, and wheel speed of 500 rpm

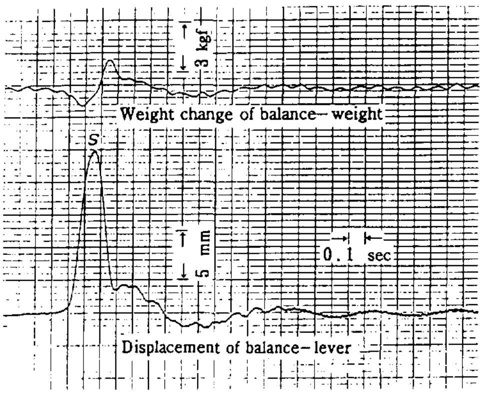

Fig. 8　Free-damped oscillations of balance-lever and balance-weight in improved straining device

Note : Under conditions *4* and *B*, and wheel speed of 500 rpm

ing weight change is 1.74 kgf. It would seem that the balance-lever oscillations were almost instantaneously damped and were not transmitted to the balance-weight. This is a remarkable effect which can also be seen in Fig. 6.

A salient feature of the improved bandsaw straining device is the almost unhindered movement (when induced) of the balance-lever without transmitting oscillation to the balance-weight. Without balance-weight oscillation, changes in bandsaw tensile force arising from the former can be effectively eliminated. Other sawing parameters notwithstanding, the application of this improved bandsaw straining device can significantly improve sawing performance by maintaining the stability of bandsaw tensile force. However, the device should be put into actual service to demonstrate its efficiency under actual operational conditions.

5 Conclusion

The improved bandsaw straining device allows the unhindered movement of the balance-lever but prevents the transmission of oscillation to the balance-weight. This brings about the stability of the balance-weight which in turn stabilizes the bandsaw tensile force.

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新しい帯鋸緊張装置における分銅の振動減衰特性と緊張力安定

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摘　　要

分銅-レバー方式の帯鋸緊張装置においては，挽材中の帯鋸緊張力の変動によって生じる帯鋸の伸縮に追随してレバーが上下にスムースに動く機構になっているため，分銅が同時に上下に振動する。この振動によって分銅の重量変動が生じるため，帯鋸緊張力が変動する。緊張力の変動が大き過ぎる場合には，帯鋸の振動や走行不安定が生じ，挽材精度の低下や挽曲がりが発生する。本研究では，緊張力の変動が少ない帯鋸緊張装置を開発するために，分銅-レバー方式の緊張装置にコイルバネとダッシュポットからなる振動吸振器を組み込んだ新しい装置を試作し，レバーに減衰自由振動を与えた場合の分銅の振動特性と帯鋸緊張力の変動特性を調べた。その結果，レバーの動きを阻害することはないが，レバーの動きを分銅に伝えにくい吸振特性に優れた帯鋸緊張装置を開発することができた。この装置では分銅の振動が大幅に抑制される。従って，分銅の重量変動が極めて小さくなり，帯鋸緊張力の変動が大幅に抑制される。

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