

# Characteristics of Self-Excited Alternate-Tooth Vibration of Circular Saw Blade

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In recent years, circular saw blades with thinner kerfs have been in high demand for improvement of the production yield of wood-based materials and the reduction of sawdust. In the case of ripping of solid wood, the number of teeth of a circular saw blade is normally set to 40 or 50, with a diameter of 305 mm, which allows less cutting resistance and smooth exhausting of sawdust. However, at certain rotational speeds, self-excited alternate-tooth vibration can easily occur in circular saw blades with such thin kerfs and so few teeth. Therefore, the cutting surface quality tends to be worse. In this study, the mechanism of this self-excited alternate-tooth vibration was clarified. The vibration mode and frequency were predicted by the finite element method. In addition, a circular saw blade with a body thickness 1.5 mm and 50 teeth was employed for wood-cutting experiments. The rotational speed range of the self-excited alternate-tooth vibration modes and their frequencies were investigated. When a double of the tooth passage frequency was slightly higher than the frequencies of the alternate-tooth vibration modes, an alternate-tooth vibration of the regenerative chatter type was excited, owing to the forces on the sides of the tooth.

*Keywords:* Circular saw blade; Self-excited vibration; Alternate tooth vibration

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## INTRODUCTION

Circular saw blades for wood machining can easily vibrate in the lateral direction owing to their thin design. This causes deterioration of the cutting surface quality and increases the sawdust and noise. In the 1980s and 1990s, the dynamic stability of circular saw blades was analyzed by many researchers (Mote and Szymani 1977; Szymani and Mote 1977; Leu and Mote 1984; Schajer 1986; Holøyen 1987; Hutton 1991; Nishio and Marui 1996). Some researchers analyzed the influence of roll tensioning on the stability of circular saw blades (Szymani and Mote 1974; Schajer and Mote 1983, 1984; Stakhiev 1999, 2003; Cristóvão *et al.* 2012; Zhang *et al.* 2014; Li *et al.* 2015a, 2015b; Li *et al.* 2016; Li and Zhang 2017). To improve stability, the impact of the temperature distribution and radial slots on the dynamic behavior of circular saw blades has also been analyzed by many researchers (Mote and Nieh 1973; Yu and Mote 1987; Ishihara *et al.* 2010; Yuan 2012). Some of them investigated the influence of temperature differences between the rim and center of a saw blade on its dynamic stability, whereas others analyzed how the saw blades could be stabilized by guides (Mote *et al.* 1981; Schajer 1986; Hutton 1991). Other researchers focused on the vibration behavior during cutting and reported on how to control and reduce this vibration. For example, cutting vibration can be suppressed by uncoupling the two vibration modes of the same nodal diameter number using outer slots on the saw

blade (Yokochi *et al.* 1993; Nishio and Marui 1996). This can make the range of vibration of rotational speed narrow, and if any vibration occurs, the vibration amplitude becomes small. Using a high-damping saw body is also very effective for suppressing the resonance amplitude of circular saw blades. A high-damping saw body can be created using damping alloys for its construction (Hattori *et al.* 1993), putting a damping air film on the saw body (Trochidis 1989), or using inner slots filled with viscoelastic resin (Nishio 2005).

These studies were related to vibration control technology for saw bodies. However, self-excited alternate-tooth vibration cannot be suppressed by these technologies. Self-excited alternate-tooth vibration is defined as two teeth becoming grouped and vibrating alternately in the lateral direction of the saw blade at a particular rotational speed.

Thin-kerf circular saw blades are in high demand for the wood industry for the purposes of yield improvement and the reduction of sawdust. However, they should result in low natural frequencies. Also, higher-order natural frequencies become even lower under the influence of the gullet shape and number of gullets. When using a blade with a thin kerf and lower number of teeth, alternate-tooth vibration is induced in the practical use range of the rotational speed. This is because only the rim part of the saw blade has the same amplitude as the saw blade in these high-order and alternate-tooth vibration modes, even if the saw blade has outer and inner slots with damping material for the suppression of vibration.

In this study, the natural vibration mode and natural vibration frequency for a circular disc and saw blade models were calculated using the finite element method (FEM) modal analysis, and the generated condition of the self-excited alternate-tooth vibration mode was discussed. A tungsten carbide circular saw blade with a high damping efficiency was employed for wood cutting experiments at various rotational speeds. The range of self-excited alternate-tooth vibration modes and their frequencies were investigated, and the characteristic vibration mechanism was theoretically considered.

## **THEORETICAL RESONANCE CONDITION OF SELF-EXCITED ALTERNATE-TOOTH VIBRATION OF CIRCULAR SAW BLADE DURING SAWING**

### **Theoretical Mode Analysis—Finite Element Method of Circular Disc and Circular Saw**

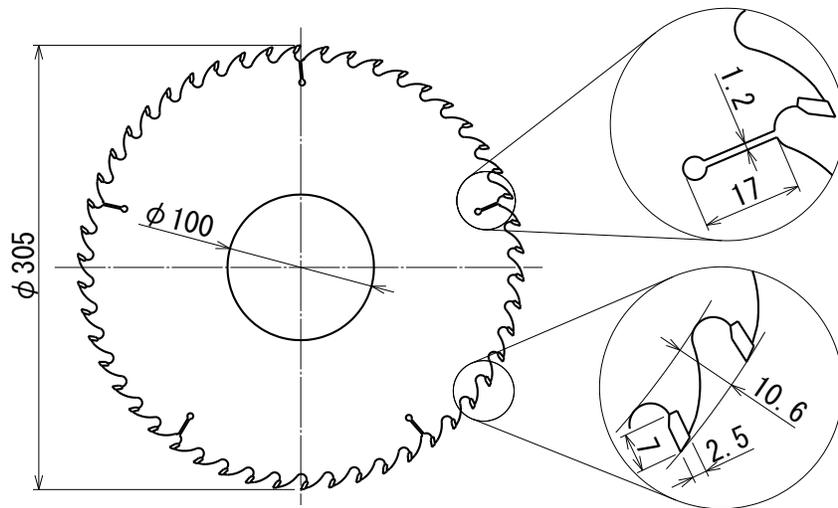
Several studies have reported on the natural vibration modes and natural frequencies of circular discs and circular saw blades (Mote and Szymani 1977; Stakhiev 1998; Orłowski *et al.* 2007), where vibration modes were limited to be under five nodal diameters. This was because the purposes of these studies were for clarification of the critical rotational speed or for investigation of the behavior of a circular saw blade around the critical rotational speed. Therefore, lower natural frequencies, especially the nodal diameters  $n = 2, 3, \text{ or } 4$ , were the targets of these studies. Even if suppression of the vibration during cutting was the purpose of the studies (Nishio and Marui 1996; Tian and Hutton 2001), the number of focused nodal diameters was less than 14 and higher-order modes have not been analyzed.

In this study, modal analyses of circular disc and circular saw models were performed with the aid of FEM to find the high-order natural vibration modes and high-order natural vibration frequencies. Table 1 shows the dimensions and mechanical properties of the circular disc and circular saw models used in the modal analysis. Figure

1 shows a schematic view of the circular saw model.

**Table 1.** Dimensions and Mechanical Properties of the Circular Disc and Circular Saw Model Used in the Modal Analysis

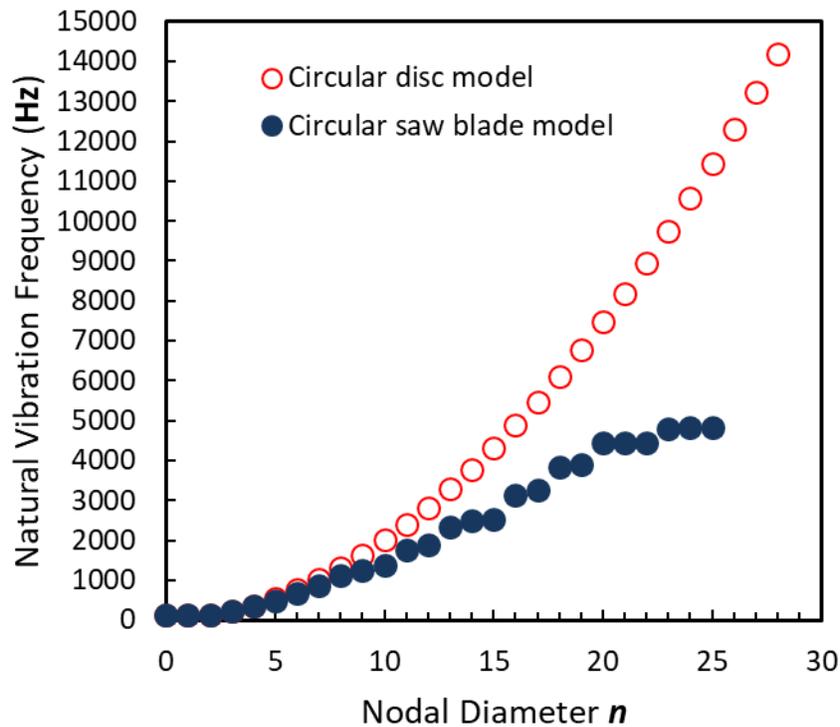
	Circular Disc Model	Circular Saw Model	
Outer diameter	305 mm	305 mm	
Thickness	1.5 mm	1.5 mm	
Number of teeth	---	50	
Slot	---	Length 17 mm, 5 pieces	
Flange diameter	$\phi 100$ mm	$\phi 100$ mm	
		(Saw body)	(Saw tip)
Elements	Shell elements	Shell elements	Shell elements
Mesh size	1 mm	1 mm	0.7 mm
Young's modulus	210 GPa	210 GPa	580 GPa
Poisson's ratio	0.3	0.3	0.23
Mass density	$7.8 \times 10^3$ kg/m <sup>3</sup>	$7.8 \times 10^3$ kg/m <sup>3</sup>	$1.5 \times 10^4$ kg/m <sup>3</sup>



**Fig. 1.** Schematic view of circular saw model for modal analysis

Figure 2 shows the relationship between the number of nodal diameters  $n$  and natural vibration frequency for the circular disc and circular saw models. The natural vibration frequency of the circular disc model increased with increments of the number of nodal diameters. In the case of the circular saw model, the natural vibration frequency also increased with increments of the number of nodal diameters. The difference between the natural vibration frequencies of both models was over 1,000 Hz in the range more than 12 nodal diameters. The maximum number of nodal diameters was 25 (*i.e.*, half the number of teeth), and a higher number of nodal diameters of more than 25 was not calculated in the range under 11,000 Hz.

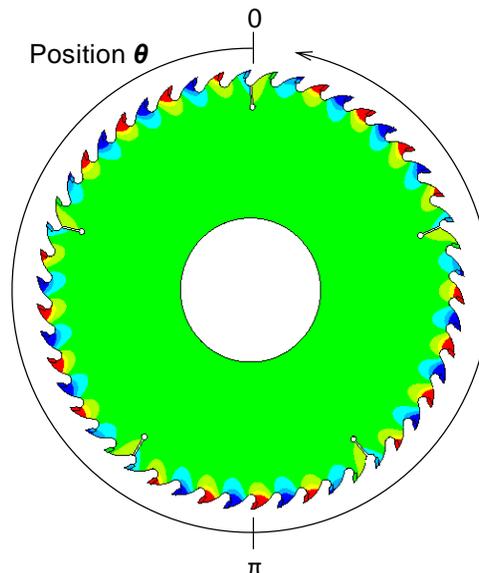
As for the natural vibration modes at 25 nodal diameters, it was observed that two teeth became grouped and vibrated alternately. The higher natural vibration modes, for a nodal diameter of 25, generated characteristic deflections at only the rim part of the saw blade. Its natural vibration frequency was 4,822.5 Hz.



**Fig. 2.** Relationships between nodal diameters and natural vibration frequencies for disc and circular saw models

### Alternate-tooth Vibration Mode

The FEM modal analysis of the circular saw model showed that the alternate-tooth vibration mode (Fig. 3) was  $n = 25$ . Because the alternate-tooth vibration mode was generated only at the rim part of the blade, it underscores the difficulty of suppressing that vibration mode even if the saw blade had inner slots with damping material for a high damping capacity. The amplitudes at the outer slots part were small.



**Fig. 3.** Alternate-tooth vibration modes (0, 25)

### Estimation of Tooth Mark Patterns by Alternate-Tooth Vibration

The kind of pattern formed in a sawn surface by alternate-tooth vibration was considered from the results of the FEM modal analysis. The vibration wave on a circular saw blade is mathematically expressed as,

$$D(\theta, t) = d \left| \sin \frac{s}{2} \left( \theta + \frac{\pi}{Z} \right) \right| \cos n\theta \cos 2\pi f_n t \quad (1)$$

where  $D(\theta, t)$  is the displacement of the periphery of the blade at time  $t$ , position angle  $\theta$ , and number of slots  $s$ . Also,  $Z$  is the number of teeth,  $n$  is the nodal diameter number, and  $f_n$  is the natural vibration frequency. On the right-hand side of Eq. 1,  $\left| \sin \frac{s}{2} \left( \theta + \frac{\pi}{Z} \right) \right|$  is a slots mode function and. It expresses that the transverse displacement at the front and back parts of the slots is small. Meanwhile,  $\cos n\theta$  and  $\cos 2\pi f_n t$  express the vibration mode function of a circular saw blade and the time function, respectively.

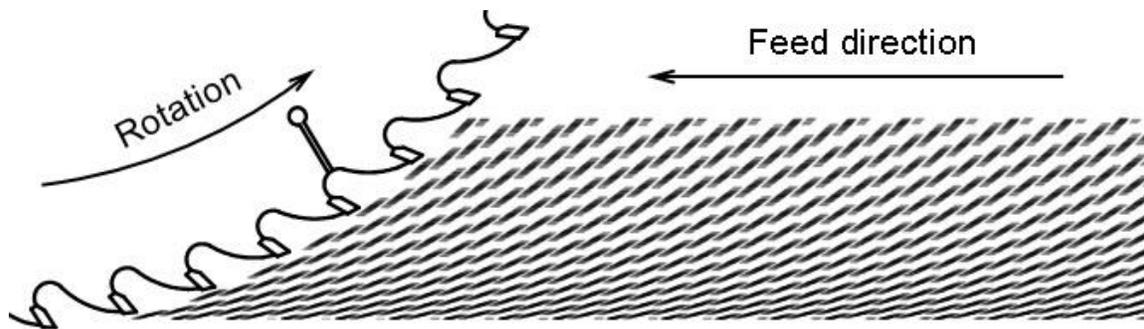
The position angle  $\theta_T$  ( $0 \leq \theta_T \leq 2\pi$ ) of the saw teeth is:

$$\theta_T = \frac{2\pi}{Z} (T - 1), \quad (T = 1, 2, 3, \dots, Z). \quad (2)$$

The outer slots on the circular saw body are located between teeth and equally on the periphery of the circular saw blade. The position angle of the outer slot  $\theta_i$  ( $0 \leq \theta_i \leq 2\pi$ ) is expressed as:

$$\theta_i = \frac{2i}{s} \pi - \frac{\pi}{Z} \quad (i = 1, 2, 3, \dots, s). \quad (3)$$

The estimated tooth mark pattern obtained using Eqs. 1 and 2 is depicted in Fig. 4. In the figure, the following conditions are assumed: blade diameter 305 mm, rotational speed  $N = 70$  rps (4,200 rpm), the number of slots  $s = 5$ , feed per tooth  $S_z = 0.06$  mm, thickness of wood is 45 mm, the saw projection is 2 mm, natural frequency  $f_n = 4,980$  Hz, and the nodal diameter number  $n = 25$ .



**Fig. 4.** Estimated tooth mark pattern on sawn surface using Eqs. (1) and (2).  
 $R$ : 70 rps (4,200 rpm),  $f_n$ : 4,980 Hz,  $n$ : 25,  $Z$ : 50,  $s$ : 5.

### EXPERIMENTAL

A wood-cutting experiment was conducted to investigate the range of the self-excited alternate-tooth vibrations and sawn surfaces. The validity of the above-mentioned theoretical calculation was examined.

## Circular Saw Blade

A tungsten carbide tipped circular saw blade with a high damping efficiency, saw diameter of 305 mm, kerf thickness of 2 mm, body thickness of 1.5 mm, and number of teeth of 50 was employed for the wood-cutting experiments. Five of the inner slots on the saw body were filled with damping material. The thin kerf of 2 mm and the thin body thickness of 1.5 mm were used for improvement of the yield rate and to reduce cutting resistance. And blade with 50 teeth is typically used for rip sawing and produces less cutting resistance. To check the reproducibility, an experiment was conducted using two circular saws of the same dimensions.

## Experimental Condition

Western hemlock (*Tsuga heterophylla* Sarg., moisture content 12%, thickness: 45 mm × length: 400 mm × width: 120 mm) was ripped using circular saw blades, which was clamped by a 100 mm diameter of flange. The rotational speeds of the spindle during sawing were varied from 40 to 90 rps (2,400 to 5,400 rpm) at each step of 2 rps (120 rpm). The feed rate per tooth was set to a constant 0.06 mm. The lateral displacement during cutting was measured using a gap detector (SENTEC LS-700 HA-50S, Osaka, Japan), which was placed near the cutting point and the resolution of the gap detector was 1 μm. The data were collected with a data logger (KEYENCE NR-500, Osaka, Japan). The sampling frequency of measurement was 100,000 Hz, and the number of samples in FFT (Fast Fourier Transform) was 16,384 for each measurement. In addition, the quality of the sawn surface was observed.

## RESULTS AND DISCUSSION

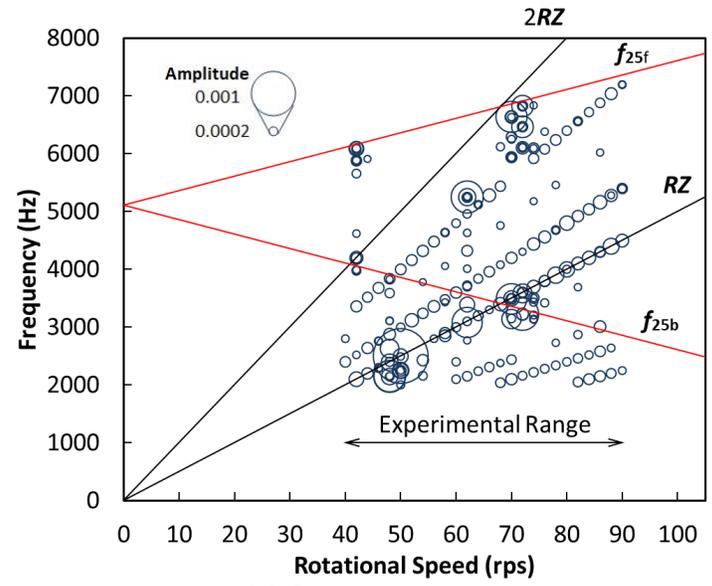
### Vibration Frequency During Sawing

Figures 5(a) and (b) show the relationships between the rotational speed and frequency of the cutting vibration with circular saw No. 1 and No. 2, respectively. The diameter of the circles represents the amplitude level of the saw blade vibration, which was calculated with FFT. Plots below 2,000 Hz and an amplitude level below 0.0002 have been cut off for clarity. The two black lines show the tooth passage frequency  $R \times Z$  and its integer multiples, where  $R$  is the rotational speed (rps) and  $Z$  is the number of teeth (= 50). The red lines show the forward wave  $f_{25f}$  and the backward wave  $f_{25b}$  of the alternate-tooth vibration mode  $f_{25}$ . The forward wave  $f_{25f}$  and backward wave are expressed, respectively, as follows.

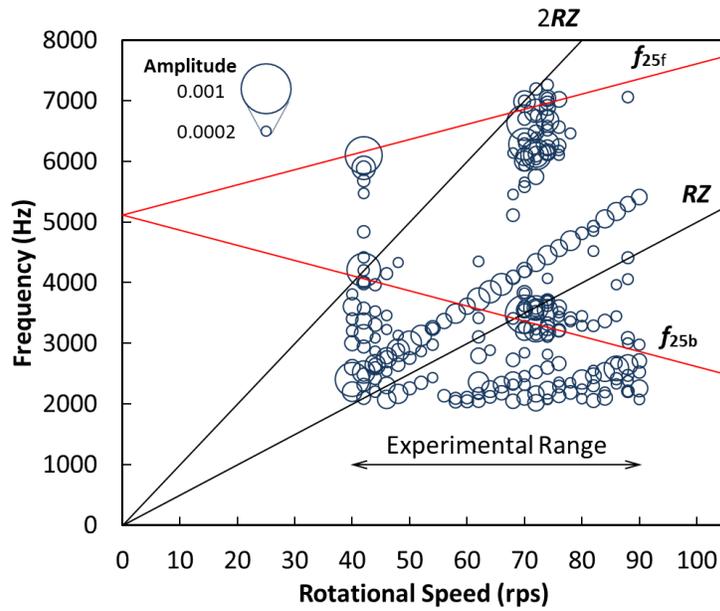
$$f_{25f} = f_{25} + 25R \quad (4)$$

$$f_{25b} = f_{25} - 25R \quad (5)$$

Alternate-tooth vibration was generated, and its amplitude was excited at a rotational speed of 70 and 72 rps (4,200 and 4,320 rpm), where the forward wave  $f_{25f}$  is slightly less than a double of the tooth passage frequency  $2RZ$ . Additionally, alternate-tooth vibration was generated at a rotational speed, 42 rps (2,520 rpm), where the backward wave  $f_{25b}$  is slightly less than a double of the tooth passage frequency  $2RZ$ . The alternate-tooth vibration was confirmed in both Fig. 5(a) and (b), and it was determined that the reproducibility was achieved.



(a) Circular saw No.1



(b) Circular saw No.2

Fig. 5. Relationship between rotational speed and frequency of vibration

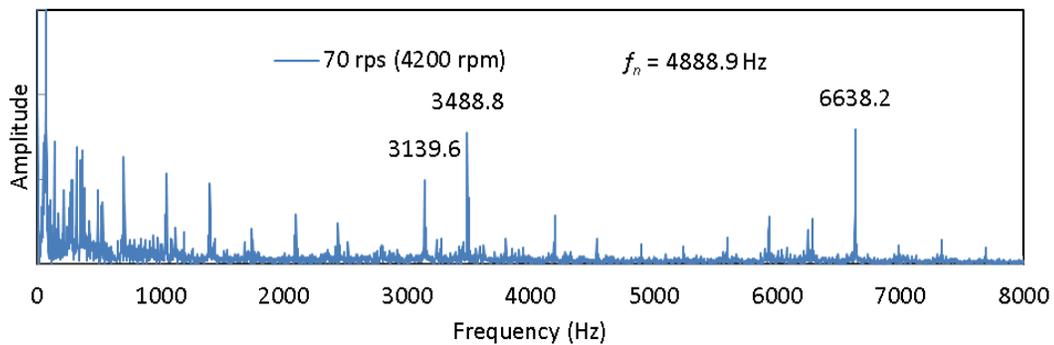
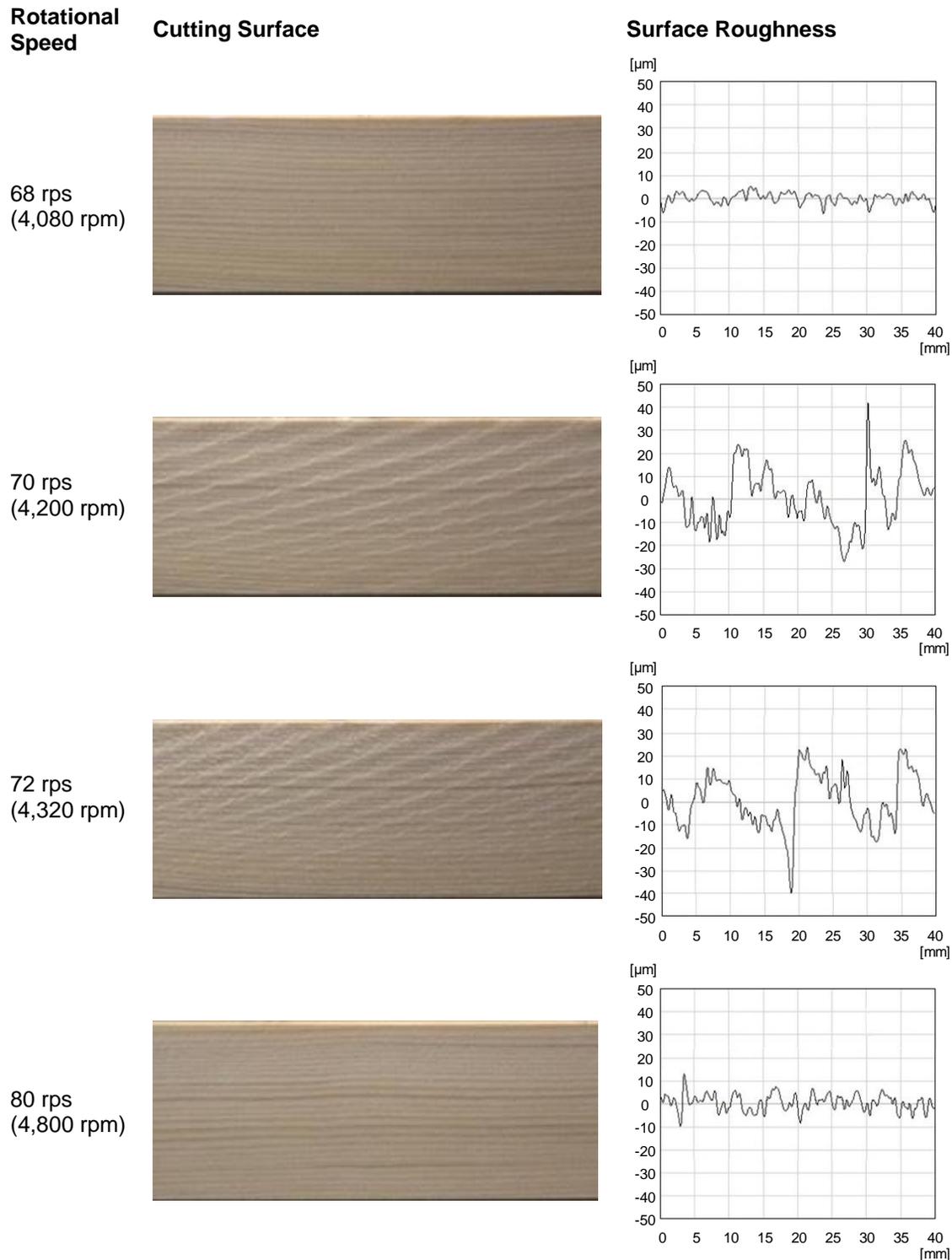


Fig 6. Results of spectral analysis. R: 70 rps (4,200 rpm), circular saw No.1

Figure 6 shows the result of a spectral analysis at a rotational speed of 70 rps (4,200 rpm). The forward wave  $f_{25f}$  was 6,638.2 Hz and the backward wave  $f_{25b}$  was 3,139.6 Hz. In addition to the forward and backward waves, peaks appeared at intervals of 350 Hz.



**Fig. 7.** Sawn surface when self-excited alternate-tooth vibration is generated (70 and 72 rps) or not generated (68 and 80 rps)

Because it appeared as a result of the gap detector measuring the passage of five slits at 70 rps (4,200 rpm) of the revolving blade, it was believed to be noise rather than a vibration component. The peak at 3,488.8 Hz was the pass frequency of the teeth and was believed to be the excitation frequency. At the same time, there was a peak at 3,139.6 Hz, which is a backward wave  $f_{25b}$ . The amplitude of the forward wave  $f_{25f}$  is dominant in this vibration. The frequency of alternate-tooth vibration  $f_{25}$  was estimated to be 4,888.9 Hz.

### Sawn Surface Aspects with Alternate-Tooth Vibration

Figure 7 shows photographs and the roughness of sawn surfaces when self-excited alternate-tooth vibration occurred and when it did not occur. The roughness was measured using a surface roughness meter (SURFTEST SV-3100, Mitsutoyo, Kawasaki, Japan).

When no alternate-tooth vibration was generated, at rotational speeds of 68 and 80 rps (4,080 and 4,800 rpm), the maximum height of the surface roughness was from 10 to 23  $\mu\text{m}$ . On the other hand, when alternate-tooth vibration was generated, at 70 and 72 rps (4,200 and 4,320 rpm), the maximum height of the surface roughness was from 40 to 60  $\mu\text{m}$ . This roughness is not large but represents a loss when compared with a sawn surface under conditions in which alternate-tooth vibration is not generated, as it requires more finishing. The tooth mark pattern was consistent with the one predicted in Fig. 4. Therefore, the generated vibration was confirmed to be the alternate-tooth vibration with the mode shape and mode frequency related to 25 nodal diameters (half the number of teeth), as determined by the FEM modal analysis.

## CONCLUSIONS

1. With a thin-body blade, higher natural frequencies are more likely to produce vibration. Also, the frequencies of the higher modes are reduced by the number and shape of the gullets.
2. The alternate-tooth vibration could not be suppressed, even by a blade with high body damping. For example, having inner slots with damping material did not help because the alternate-tooth vibration mode only involved movement of the teeth and not the inner part of the blade.
3. Alternate-tooth vibration is a self-excited vibration. It is generated at the rotational speed at which the alternate-tooth vibration mode is slightly less than double the tooth passage frequency of the saw blade.

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