ISSN 1392–1320 MATERIALS SCIENCE (MEDŽIAGOTYRA). Vol. 11, No. 1. 2005

Bend Forms of Circular Saws and Evaluation of their Mechanical Properties

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Received 31 August 2004; accepted 19 October 2004

A circular saw is able to operate properly only when its equilibrium form is stable. Usually it is a flat equilibrium form. In most cases the loss of stability of a circular saw is related to the loss of its dynamic stability. One of the reasons are resonance vibrations of saws. With appearing resonance phenomena, tiny forces cause great deformations. The size of these deformations depends on saw rigidity and damping value, therefore it is very important to know mechanical properties of saws (strength, rigidity, coefficient of damping, etc.). The elaborated study methods and equipment allow to evaluate precisely mechanical properties of saws, separate bend forms, resonance frequencies and damping. Using resonance frequencies, under which saw bend forms are close to theoretical, Young`s modulus was estimated for all four fastened saws (E = 230 ÷ 368 GPa). Using amplitude-frequency characteristics of saw vibrations, coefficient of damping was evaluated (tgδ = 0.00025 ÷ 0.01255). Bend forms of saws were also studied changing their fastening moment, respectively – 50, 100 and 150 Nm. The paper presents study methods, bend forms of saws and obtained relationships, reflecting the change of mechanical properties of saws.

Keywords: circular saw, resonance vibrations of saws, coefficient of damping, Young`s modulus.

INTRODUCTION

Circular saws are widely applied in wood processing industry. They can operate reliably only when their equilibrium form is stable. Analysis of technical defects using circular saws for sawing shows, that major part of defects is caused by insufficient stability of saws [1]. Studies show, that in most cases the loss of circular saw stability is related not to the loss of static stability of flat equilibrium form, but to the loss of dynamic stability. The reason for this are resonance vibrations of saws. These vibrations lead to worse wood processing quality, increased noise, fissures in the disc, etc. Under appearing resonance phenomena, very small forces cause great deformations [2]. The size of these deformations depends on saw rigidity and damping value, therefore it is very important to know mechanical properties of saws (strength, rigidity, coefficient of damping, etc.). The less is the damping force of a saw, the slower is stifling of vibrations, the worse is sawing quality. Producing circular saws, it is sought, that damping of the saw is as high as possible.

The use of damping materials is an optimal way to stifle vibrations [3 – 5]. For this purpose vibration damping rings are used [4, 6]. Owing to them, noise is reduced by 10 dB, while in the case of resonance vibrations – by 20 dB. The amplitude of vibrations decreases 10 and more times, which leads to better sawing quality.

Many studies are carried out seeking to reduce the noise caused by circular saws. For this purpose, in the periphery of saws, cuts of different shapes are made (circular, zigzag, asymmetrical, various holes etc.) [6 – 8]. These cuts may also provide vibration damping effect. The segments of these cuts may be filled with vibration damping material.

Studies are conducted also changing the construction of disc fastening flanges [9]. A layer of vibration absorbing material may be attached to one or both of them. These studies allow to solve some problems related to saw form stability, however, in most cases saw bend form is ascertained only approximately, coefficient of damping and Young`s modulus are not evaluated. Thus, the work is aimed at precise measurement of saw bend forms and evaluation of its mechanical properties by a versatile form stability analysis of circular saws.

THEORETICAL PART

Theoretically, circular saws are studied on the basis of round plates fastened in the centre. In this case the plate is analysed as a linear elastic system, where elastic and viscid properties are uniformly distributed in its whole volume. Analysing vibrations of such an elastic body, it is assumed, that the material is monolithic and isotropic, while its thickness is sufficiently small as compared to other dimensions.

Classical equation of periodical vibrations of the plate is the following [10]:

$$\alpha^4\nabla^2\nabla^2\xi+\frac{\partial^2\xi}{\partial t^2}=0 \tag{1}$$

or

$$\left(\nabla^2+\frac{\omega}{\alpha^2}\right)\left(\nabla^2-\frac{\omega}{\alpha^2}\right)\xi=0\,, \tag{2}$$

where

$$\alpha^4=\frac{Eh^2}{12\left(1-\mu^2\right)\rho}=\frac{D}{\rho}\,, \tag{3}$$

where μ is the Poisson’s ratio (μ = 0.3); E is the Young`s modulus; h is the thickness of the plate; ζ is the function of equation variables; α is the coefficient; ω is the angle frequency of free vibrations; ρ is the density of plate material.

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Solving the equation of transverse vibrations of a plate firmly fastened in the centre, the obtained frequency of its free vibrations (bend forms) is calculated as follows [11]:

$$f_{res} = \frac{\alpha h}{2\pi R^2}\sqrt{\frac{E}{12\left(1-\mu^2\right)\rho}}, \text{ Hz} \tag{4}$$

where α is the parameter, dependant on the number of knot circles; h is the thickness of circular saw; R is the radius of circular saw; ρ is the density of saw disc material.

Knowing the frequency of free vibrations f_{res}, it is possible, using equation (4), to calculate Young`s modulus E of the vibrating system.

Interior friction (damping) is evaluated having ascertained amplitude-frequency characteristics of the saw [12]. Its form predetermines inner friction of the saw. Having ascertained resonance frequency f_{res} and amplitude A_1, additionally two other frequencies f_1 and f_2 are determined, under which the amplitude is $\sqrt{2}$ times less than resonance amplitude (Fig. 1).

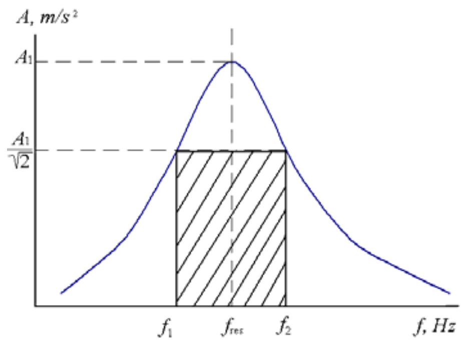


Fig. 1. Amplitude-frequency characteristics of the saw: A_1 – resonance amplitude, f_{res} – resonance frequency, f_1 and f_2 – frequencies, under which amplitude is $\sqrt{2}$ times less than resonance amplitude

The ratio of frequency belt width $\Delta f = f_2 - f_1$ and resonance frequency f_{res} is the measure of inner friction of the saw:

$$\mathrm{tg}\delta \approx \frac{f_2 - f_1}{f_{res}}, \tag{5}$$

where tgδ is the coefficient of damping.

STUDY METHODS AND EQUIPMENT

For the studies, a special vibration stand was used (Fig. 2), which consists of a circular saw 1, fastened on a shaft 2 with flanges 3, a vibrator 4, a piezoelectric sensor 5, a generator of electric signals 6, an amplifier 7, a frequency measurer 8, a phasometer 9, a vibrometer 10 and an oscilloscope 11.

By fastening the sensor in characteristic points, amplitude-frequency characteristics of the saw is ascertained, the values of resonance frequencies and vibration amplitudes as well as coefficient of damping are evaluated.

For an accurate saw form evaluation, the whole saw is divided into separate parts (96 points), obtaining a certain number of diameters and circles (Fig. 3). In the places of their intersection the sensor is fastened and resonance

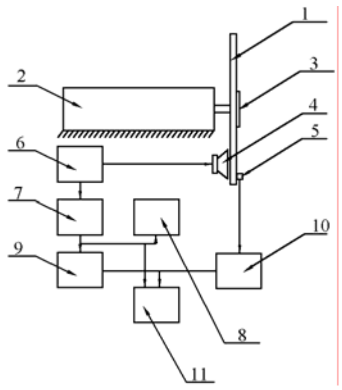


Fig. 2. Study stand of circular saws: 1 – circular saw; 2 – shaft; 3 – flanges; 4 – vibrator; 5 – sensor; 6 – generator of electric signals; 7 – amplifier; 8 – frequency measurer; 9 – phasometer; 10 – vibrometer; 11 – oscilloscope

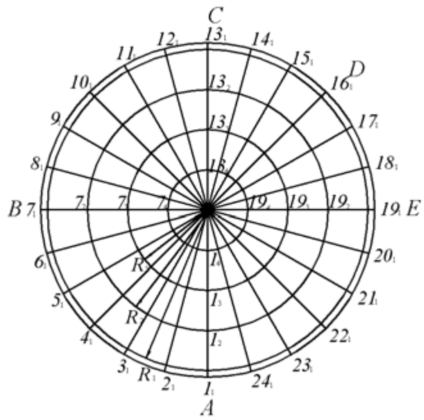


Fig. 3. Circular saw**:** 1_1, 1_2, 1_3 and so on – points, where measurements are carried out, R_1, R_2, R_3 – radius of circles; A, B, C, D, E – characteristic measurement zones of saw vibrations

frequency, amplitude as well as the difference in the phases of signals between the sensor 5 and the generator 6 are ascertained.

EXPERIMENTAL PART

Four 9ХФ steel type saws were studied. Their dimensions were: saw I – diameter D_s = 1000 mm, thickness s = 3.7 mm, flange diameter d_f = 140 mm; saw II – D_s = 800 mm, s = 4.0 mm, d_f = 140 mm; saw III – D_s = 500 mm, s = 2.6 mm, d_f = 140 mm; saw IV – D_s = 400 mm, s = 2.8 mm, d_f = 140 mm. Measurements were carried out within 0 ÷ 2000 Hz range. Measuring vibrations, resonance frequencies and amplitude-frequency characteristics of the saws were determined. The change of saw bend forms and mechanical properties was studied changing the fastening moment of saws. Fig. 4 presents amplitude-frequency characteristics of saw I. Characteristic is the fact, that the highest amplitudes are attained by saw in the range of low frequencies (50 ÷ 500 Hz). Analogous amplitude-frequency characteristics were measured for other saws too. Besides, these characteristics were obtained under different saw fastening moments (50, 100 and 150 Nm).

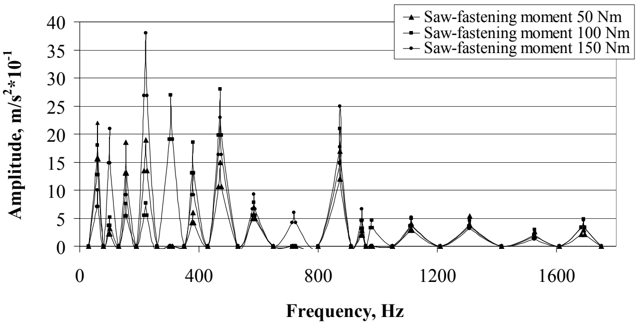


Fig. 4. Amplitude-frequency characteristics of saw I, measuring vibrations in zone *A*

Table 1 presents obtained resonance frequencies of saws. It can be seen, that depending on saw dimensions under similar bend form resonance frequencies slightly differ, besides, additional resonance frequencies occur, which in other saws were not recorded.

Table 1. Resonance frequencies of saws

	Resonance frequencies of saws, Hz				
Saw (I)	30.02	59.50	97.6	155.0	222.6
	305.9	381.3	467.5	583.8	872.2
	948.3	979.4	1111.4	1307.9	1525.7
	1690.2				
Saw (II)	52.9	106.4	185.0	282.8	367.3
	491.2	529.8	659.3	837.4	914.3
	1082.9	1194.5	1414.6	1609.4	1828.7
	1881.8				
Saw (III)	64.1	82.5	144.1	243.2	369.0
	448.4	520.0	674.0	702.0	898.3
	1116.7	1201.3	1349.3	1596.5	1862.6
	1934.3				
Saw (IV)	128.0	166.9	253.7	333.3	407.6
	611.1	888.0	1162.0	1229.3	1499.6
	1710.3	1845.6			

At the same time saw bend form was ascertained under each of resonance frequencies. It was found, that some bend forms are close to theoretical, while others significantly differ from theoretical ones. It was obtained, that analogous bend forms for saws with smaller diameter are observed under higher frequencies.

The disc of a saw is a system with *n* freedom degrees. It has an endless multitude of vibration bend forms. The following bend forms are distinguished: without knot diameters and circles; with knot diameters; with knot circles; combined (with knot diameters and knot circles) [1]. In saw I, saw II and saw III bend forms with knot diameters and combined ones are obtained, while in saw IV only forms with knot diameters are observed.

Some saw bend forms are given in Fig. 5. In Fig. 5a, under diameter $D_s = 400$ mm, frequency $f_{res} = 408$ Hz, bend form is with four knot diameters, in Fig. 5b, under $D_s = 800$ mm, $f_{res} = 185$ Hz, bend form is also with four knot diameters, in Fig. 5c, under $D_s = 1000$ mm, $f_{res} = 154$ Hz bend form is combined (with two knot diameters and one circle).

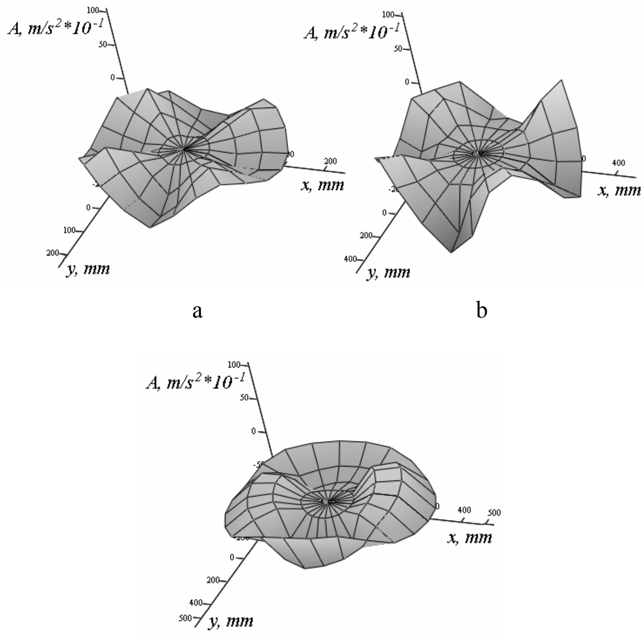


Fig. 5. Saw bend forms: a – $D_s = 400$ mm, $f_{res} = 408$ Hz, b – $D_s = 800$ mm, $f_{res} = 185$ Hz, c – $D_s = 1000$ mm, $f_{res} = 154$ Hz

Using resonance frequencies, under which saw bend forms are close to theoretical, in accordance with expression (4), Young`s modulus was estimated for all four saws ($E = 230 \div 368$ GPa).

Fig. 6 presents the change of saw I Young`s modulus changing saw fastening moment, respectively 50, 100 and 150 Nm. For this saw Young`s modulus was estimated under resonance frequency of 154 Hz. Saw bend form under this frequency is with two knot diameters and one knot circle. It can be seen, that with increasing frequency, Young`s modulus increases. Under fastening moment increment from 50 to 150 Nm, Young`s modulus respectively increases from 230 to 232 GPa.

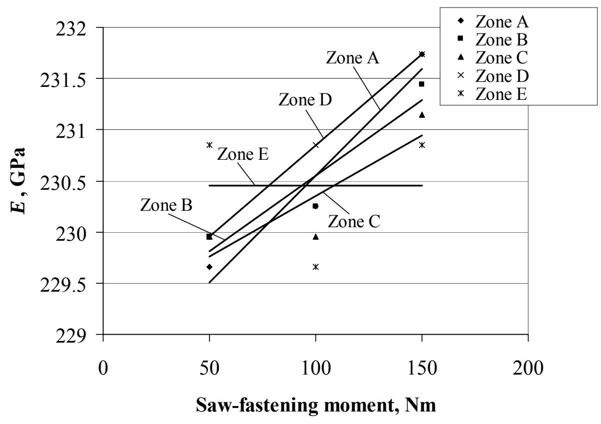


Fig. 6. Change of saw $D_s = 1000$ mm Young`s modulus, changing saw fastening moment

Table 2. Coefficient of damping of saws

	Saw fastening moment 50 Nm												
f_{res}	59.44	97.6	154.9	220.9	380.4	466.4	584.1	870.7	946.7	1108.6	1308.7	1527.3	1690.8
tgδ	0.00034	0.00205	0.00194	0.00091	0.00053	0.00043	0.00086	0.00057	0.0011	0.0036	0.00038	0.00079	0.00024
	Saw fastening moment 100 Nm												
f_{res}	59.3	97.3	154.9	221.2	381.6	467.4	584.0	870.9	947.1	1111.6	1307.6	1526.5	1690.2
tgδ	0.00034	0.0021	0.00194	0.00090	0.00079	0.00043	0.00034	0.00046	0.00137	0.00189	0.00076	0.00072	0.0025
	Saw fastening moment 150 Nm												
f_{res}	59.39	97.0	155.3	221.1	382.3	467.7	584.0	870.5	947.2	1112.1	1306.6	1483.8	1689.4
tgδ	0.00034	0.00103	0.00129	0.00045	0.00092	0.00043	0.00055	0.00034	0.00169	0.00243	0.00069	0.00121	0.00047

Using amplitude-frequency characteristics of saw vibrations, coefficient of damping was also evaluated. Table 2 presents the values of saw I coefficient of damping under changing fastening moment and frequency.

Fig. 7 presents the dependence of coefficient of damping, evaluating saw bend forms and their frequencies. These dependencies were obtained under different saw fastening moments (50, 100 and 150 Nm).

Fig. 7 shows, that with increasing frequency, coefficient of damping increases, when saw-fastening moments are 100 and 150 Nm, and decreases when saw-fastening moment is 50 Nm.

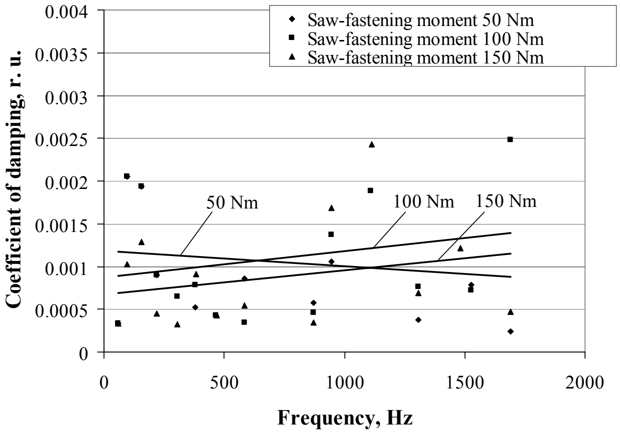


Fig. 7. Dependence of coefficient of damping on frequency, D_s = 1000 mm

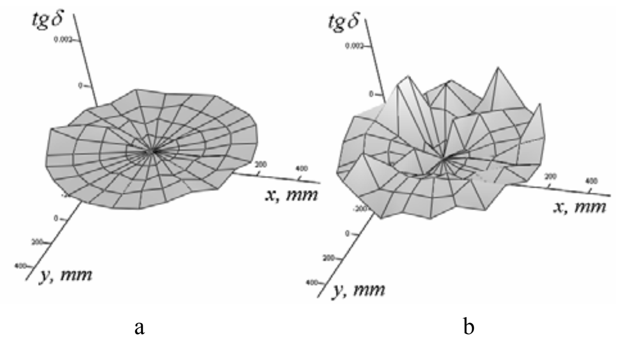


Fig. 8. Change of coefficient of damping on the plane of saw I: a – f_{res} = 305 Hz, b – f_{res} = 1308 Hz

Distribution of coefficient of damping on the saw plane was also studied, i.e. it was measured in each of 96 measurement points. Fig. 8 presents the change of coefficient of damping on saw I plane under resonance frequencies of 305 Hz and 1308 Hz. Under resonance frequency of 305 Hz, coefficient of damping is lower (tgδ = 0.00045 ÷ 0.00089), but it is more evenly distributed on the saw plane than under resonance frequency of 1308 Hz (tgδ = 0.00074 ÷ 0.00195). However, under both resonance frequencies zone CDE (Fig. 3) was distinguished, in which coefficient of damping is higher than in the rest part of the saw.

It was ascertained, that changing saw fastening moment, bend forms remain similar, however, the amplitude of vibrations, frequency and coefficient of damping undergo changes. Fig. 9 presents bend forms of saw I, changing saw fastening, respectively – 50, 100 and 150 Nm, when frequency is 154 Hz, while Fig. 10 shows corresponding changes in the amplitude of vibrations and coefficient of damping.

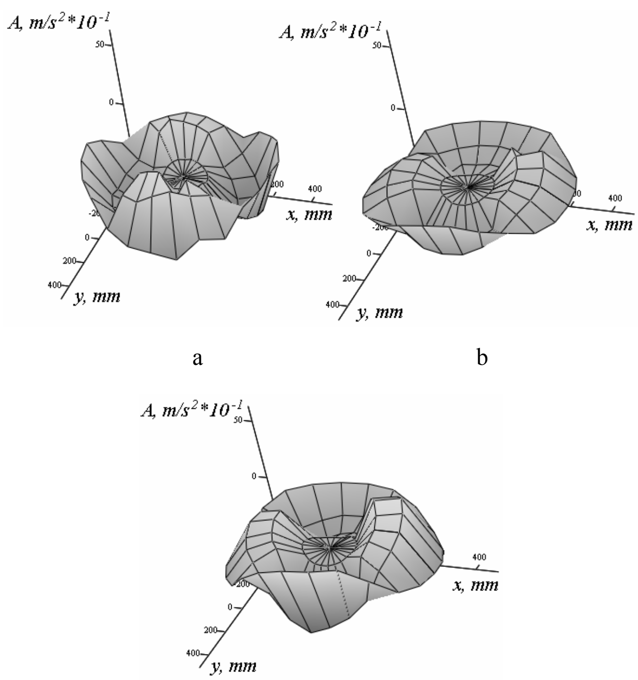
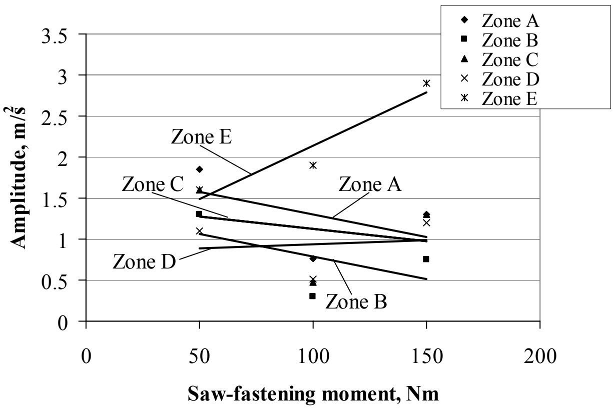
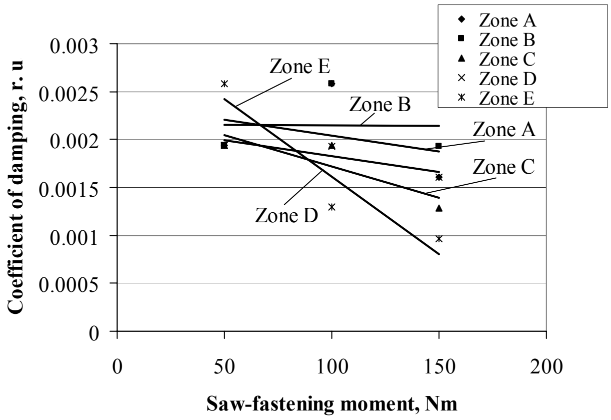


Fig. 9. Change of saw I bend forms, when f_{res} = 154 Hz, while – a – fastening moment – 50 Nm, b – fastening moment – 100 Nm, c – fastening moment – 150 Nm



a



b

Fig. 10. Dependence of the amplitude (a) and coefficient of damping (b) on saw I fastening moment

Fig. 10 shows, that with increasing fastening moment, the amplitude and coefficient of damping decreased respectively from 2.9 to 0.3 m/s^2 and from 0.00258 to 0.00097. Fig. 6 and Fig. 10 show, that with increasing Young`s modulus, coefficient of damping decreases, i.e. with improving elastic properties of the system, its viscid properties become worse.

Thus, having ascertained amplitude-frequency characteristics of the saw, and knowing resonance frequencies, it is possible to evaluate mechanical properties of such a system, i.e Young`s modulus and damping. On the other hand, these properties allow to evaluate the quality of circular saws, forecast their workability and sawing precision.

CONCLUSIONS

1. It was found, that the values of resonance frequencies change depending on circular saw dimensions, when $D_s = 400$ mm, $f_{res} = 166.9$ Hz, *b* – $D_s = 500$ mm, $f_{res} = 82.5$ Hz, *c* – $D_s = 800$ mm, $f_{res} = 52.9$ Hz, $D_s = 1000$ mm, $f_{res} = 30.02$ Hz under the same bend form.

2. It was obtained, that only some saw bend forms coincide with theoretical ones. Analogous bend forms for saws with smaller diameter are observed under higher frequencies. Besides, in saws with 1000 mm, 800 mm and 500 mm diameters bend forms with knot diameters and combined ones are obtained, while in 400 mm diameter saw only forms with knot diameters are observed.

3. Using resonance frequencies, under which saw bend forms are close to theoretical, saw Young`s modulus was calculated ($E = 230 \div 368$ GPa).

4. It was ascertained, that vibrations of higher frequencies fade away slower, i.e. their coefficient of damping is higher than that of lower frequency vibrations. Coefficient of damping $\mathrm{tg}\delta = 0.00025 \div 0.01255$ was calculated.

5. It was determined, that changing saw fastening moment, bend forms remain the same, however, the amplitude of vibrations, frequency and coefficient of damping undergo changes.

6. It was found, that with increasing Young`s modulus of a circular saw, coefficient of damping decreases. When Young`s modulus increases from 223 GPa to 232 GPa, coefficient of damping decreases from 0.00258 to 0.00097.

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DOI: 10.5755/j02.ms.26492