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RESEARCH OF INTER-IMPACT OF WOOD CIRCULAR SAWS VIBRATION MODES

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ABSTRACT

The paper presents study method which enables to ascertain the influence of separate vibration modes of circular saw on each other, estimating variation of the amplitude and coefficient of damping. Studies were conducted with 9ХФ steel type saw, the diameter of which was 800 mm (thickness s = 4.0 mm). Measurements were carried out within 20-2000 Hz range. It was found that shocks, affecting saw, induce vibrations of several modes. The simultaneous „action" of two modes of the saw in different points causes the increase of sum vibration amplitude on an average by 1.5 times. Also it was determined that, "affecting" saw lower frequency mode with other modes, coefficient of damping of the former in individual cases decreases to 13 %, while "affecting" higher frequency mode coefficient of damping increases in 2-5 %. Increase of saw oscillation amplitude challenge changes in kerf width, while coefficient of damping changes affect saw oscillation dying-out period.

KEY WORDS: circular saw, resonance vibrations, vibration mode, coefficient of damping

INTRODUCTION

In wood processing industry circular sawing machines comprise about one half of all machine-tools. Technical-economic characteristics of the machinery is mainly determined by the efficiency of circular saws.

Increase of the productivity, precision and output of circular sawing machinery is usually achieved through increased rotation frequency of saws and feeding rate, as well as reduced thickness of saws.

The main reasons of technical reject, sawing wood with circular saws, are insufficient rigidity of saws and instability of their form during operation.

Dynamic instability is associated with transverse resonance vibrations of the saw (Stakhiev 1989). Variable sawing resistance force, uneven heating of the saw and other factors cause resonance vibrations of the saw in a wide range of frequencies.

The value of saw resonance frequencies depends on many factors. They are usually estimated applying theoretical calculations of an indiscrete circular plate (Stakhiev 1977).

An important factor determining saw efficiency is vibration damping. Not only the amplitude of vibration, but also its decay and at the same time kerf quality depend on the coefficient of damping.

Practically saw resonance vibrations are reduced by selecting rotation frequency and applying additional vibration extinction measures (Pat. 4979417 1990, Pat. 2048285 1995, Gorin 1997, Pat. EP0819491 1998, Pat. WO9932251 1999, Yu and Huang 2001).

Seeking to increase stability of saws, it is important to estimate saw bend form (mode), size of saw bend and the duration of oscillation decay (coefficient of damping) during operation. Only having evaluated these parameters, it is possible to make real forecast of kerf variation in time and potential noise (Ukvalbergiene and Vobolis 2005).

During sawing a saw experiences shocks which at the same time induce several modes of vibrations. Therefore, theoretical and experimental analysis of separate modes does not allow to evaluate fully the behaviour of saws during operation. When several modes are acting, changes the sum amplitude of vibrations etc. Due to non-linearity of saw mechanical system, the influence of modes on each other may be rather considerable.

The aim of the work is to determine the influence of separate vibration modes of circular saw on each other, estimating variation of the amplitude and coefficient of damping.

MATERIAL AND METHODS

Studies were conducted applying the dynamic method (Vobolis 1996). Special stand was used for the studies (Fig. 1).

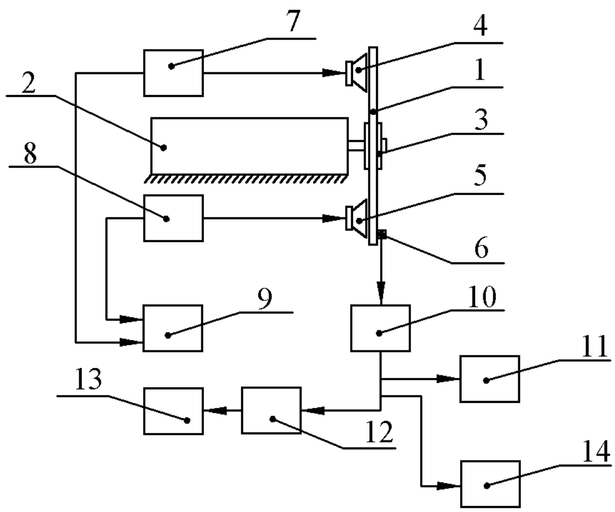


Fig. 1: Study stand of circular saws: 1 – circular saw; 2 – shaft; 3 – flanges; 4, 5 – vibrators; 6 - sensor; 7, 8 – generators of electric signals; 9 – frequency meter; 10 – vibrometer; 11 – oscilloscope; 12 – wide-band filter; 13 – pointer device; 14 – computer

The saw 1 is placed on the shaft 2 and fastened by flanges 3. To induce transverse vibrations, vibrators 4 and 5 are used. The saw is induced from two sectors. Saw vibrations are determined by the sensor 6, which with the help of a permanent loadstone is fastened to the surface of the saw. Vibrator 4 is attached to the generator of electric signals 7, while vibrator 5 – to the generator 8. Frequency meter 9 measures the frequency of vibrations, while vibrometer 10 measures the amplitude of vibrations. The form of vibrations is observed in the screen of oscillograph 11. For the sake of sensitivity, the amplitude of vibration acceleration is measured. After free saw oscillation activation by impact oscillation spectrum can be deduced with the help of computer 14. Specific program is used for this.

Firstly, vibrator 4 induces one of the saw vibration modes, and its vibration frequency as well as amplitude are measured. At the same time vibration coefficient of damping is estimated. Later its amplitude is reduced to zero. Then the second vibrator 5 induces another vibration mode (at another resonance frequency) and analogous measurements are performed. Later, with the aid of vibrators 4 and 5, vibrations of both the modes are induced. Vibrometer 10 is used to measure sum amplitude of vibrations, while their form is observed in the screen of the oscillographer 11.

At the same time wide-band filter 12 is used to single out the signal of one of the modes and once again coefficient of damping of the mode is ascertained.

Coefficient of damping is ascertained based on saw amplitude-frequency characteristics (Vobolis 1996). Having ascertained resonance frequency f_{res} and amplitude A_1, additionally two more frequencies f_1 and f_2 are determined, under which amplitude A_1 decreases times (Fig. 2).

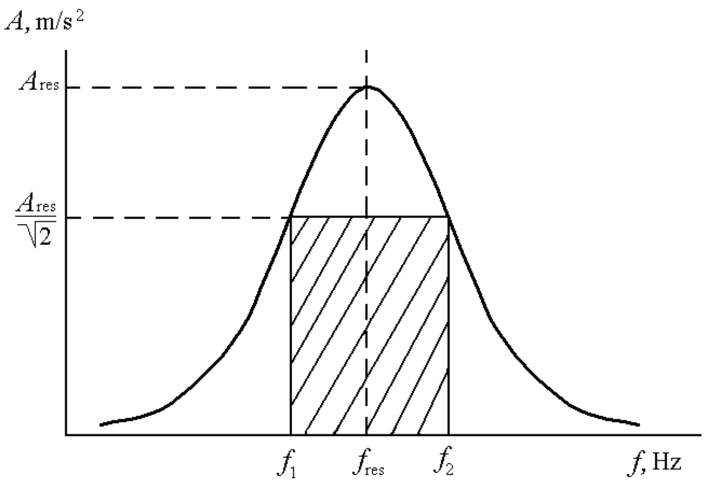


Fig. 2: Saw amplitude-frequency characteristics: A_1 – rezonance amplitude, f_{res} – resonance frequency, f_1, f_2 – frequencies under which resonance amplitude decreases times

Coefficient of damping is calculated as the ratio of band width and resonance frequency :

$$tg\delta \approx \frac{f_2 - f_1}{f_{res}}, \qquad (1)$$

where tgδ tangent of the angle of losses.

For the measurement of vibrations, 8 points of saw surface are chosen (Fig. 3). They are selected in the zones of saw teeth cavities.

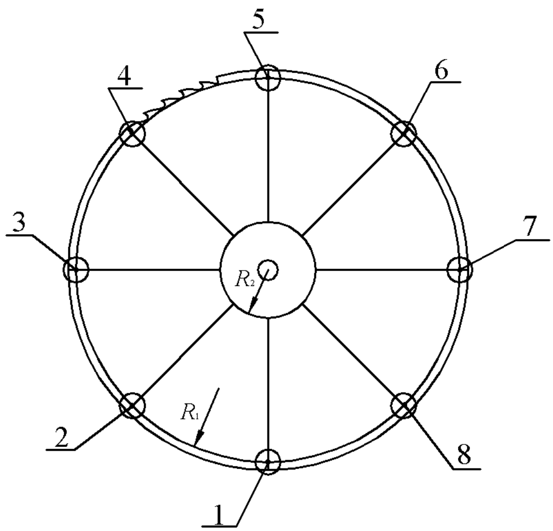


Fig. 3: Measurement scheme of circular saw`s vibrations: 1, 2, 3, 4, 5 and so on – characteristic measurement points of saw vibrations, R_1 – measurement radius of characteristic points (R_1 = 378 mm); R_2 – radius of flanges (R_2 = 70 mm)

The sensor was fastened in these points on the saw and vibration frequency, amplitude as well as coefficient of damping were ascertained.

RESULTS AND DISCUSSION

For the studies, steel 9XФ type saw was chosen with the diameter D = 800 mm, thickness s = 4.0 mm, diameter of flanges d = 140 mm. Fastening moment – 150 Nm.

Initially, after the first shock, the sensor was recording saw vibrations and the spectrum of these vibrations was ascertained (Fig. 4). Measurements were conducted within 20 – 2000 Hz range.

As it can be seen (Fig. 4), shock induces vibrations of several modes (54 Hz, 106 Hz, 185 Hz, 282 Hz, 367 Hz, etc.). The saw in this case vibrated in a rather wide (up to 900 Hz) frequency range.

Later, having induced saw vibrations with one of vibrators (e.g., 4) and recording vibrations by sensor, saw resonance frequencies in characteristic points were measured. The data are provided in Tab. 1.

From Tab. 1 it can be seen that for the saw (D = 800 mm), 16 resonance frequencies were ascertained.

At the same time saw precise amplitude-frequency characteristics was estimated. It was found that it is not symmetric under different resonance frequencies (Fig. 5). Under resonance frequency of f_{res} = 106.3 Hz, the values of other two frequencies, determining coefficient of damping, are f_1 = 106.25 Hz and f_2 = 106.4 Hz. In this case saw may be analysed as a non-linear mechanical system.

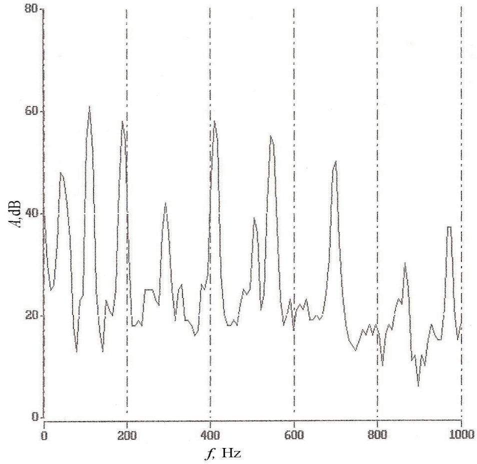


Fig. 4: Vibration spectrum of saw (D = 800 mm, s = 4.0 mm, d_{fl} = 140 mm)

Tab. 1: Resonance frequencies of the saw

	Resonance frequencies of saw, Hz							
Saw (D = 800 mm)	53.9	106.4	185.0	282.8	365.3	491.2	529.8	659.3
	837.4	914.3	1082.9	1194.5	1414.6	1609.4	1828.7	1881.8

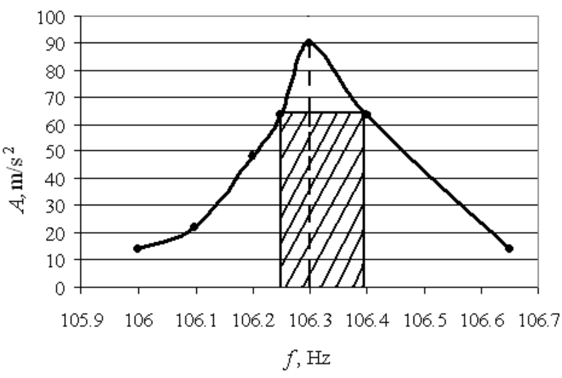
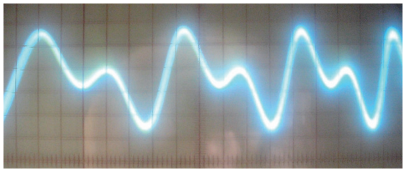


Fig. 5: Saw amplitude-frequency characteristics: f_{res} = 106.3 Hz, f_1 = 106.25 Hz, f_2 = 106.4 Hz

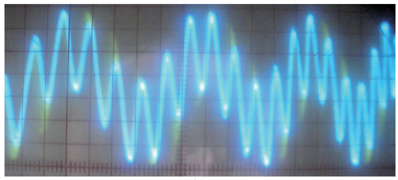
Having evaluated earlier presented initial measurement data, further the influence of saw vibration modes on each other was studied.

During the studies, saw vibrations of the frequencies 54 Hz, 106 Hz, 282 Hz, 365 Hz, 491 Hz and 530 Hz as well as their interaction were estimated.

Fig. 6 shows the oscillograms of two sum vibrations, when at the same time the saw vibrates by 54 Hz and 106 Hz (Fig. 6a) as well as 54 Hz and 491 Hz (Fig. 6b) frequencies.



a)



b)

Fig. 6: Oscillograms of saw (D = 800 mm) sum vibrations: a – 54 Hz, 106 Hz; b – 54 Hz,491 Hz

As can be seen in Fig. 6, sum amplitude of saw vibration changes considerably. It was found that its value differs in different points of the saw. In this case, vibration amplitudes of different modes in each measurement point were maintained equal. The data are provided in Tab. 2. Here are presented vibration amplitudes of different modes, the frequencies of which are 106 Hz and 54 Hz , 365 Hz and 491 Hz, as well as the obtained sum vibration amplitudes. It was found that under simultaneous „action" of two modes, sum vibration amplitude increases on an average by 1.5 times.

Tab. 2: Values of separate modes and sum amplitudes of circular saws

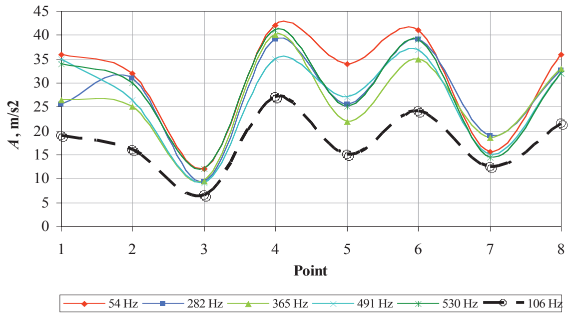
Vibration amplitude, m/s^2*10^{-1}	Measurement point							
	1	2	3	4	5	6	7	8
106 Hz mode	23.5	18.5	8.5	29.5	22	31	11	23
54 Hz mode	23.5	18.5	8.5	29.5	22	31	11	23
Sum vibration amplitude	36	32	12	42	34	46	15.5	36
491 Hz mode	17,5	11	2,9	13	10,2	22	6,8	21
365 Hz mode	17,5	11	2,9	13	10,2	22	6,8	21
Sum vibration amplitude	25,5	16	4,3	19	14,2	35	8,7	29

Fig. 7 presents the change of amplitude in the saw surface when it vibrates by two modes at the same time. Fig. 7a presents amplitude change when the frequency of the „main" mode is 106 Hz, while it is „affected" by modes of 54 Hz, 282 Hz, 365 Hz, 491 Hz and 560 Hz frequencies. It also presents amplitude change of one of the modes (106 Hz) in the saw surface. Fig.7b analogically shows the change of vibration amplitude. When frequency of the „main" mode is 491 Hz, it is „affected" by other modes, the frequencies of which are 54 Hz, 106 Hz, 282 Hz, 365 Hz and 560 Hz. The diagram also shows amplitude change of one of the modes (491 Hz) in the saw surface.

Fig. 7 shows that when the saw vibrates by different modes at the same time, sum vibration amplitude also increases at different rates.

It was found that sum vibration amplitude increases most when the saw vibrates by modes of lower frequencies (e.g. 54 Hz and 106 Hz, Fig. 7a). Besides, sum vibration amplitude is highly dependent on saw surface point. When amplitude of 54 Hz mode was $8.531*10^{-1}$ m/s^2 and 106

Hz mode - 8.531*10^{-1} m/s^2, minimum sum amplitude (12*10^{-1} m/s^2) was registered at point 3 and maximum (46*10^{-1} m/s^2) – at point 6. Analogically, saw oscillation amplitude changes at separate points, when lower frequency modes "affect" higher frequency modes (Fig. 7b).



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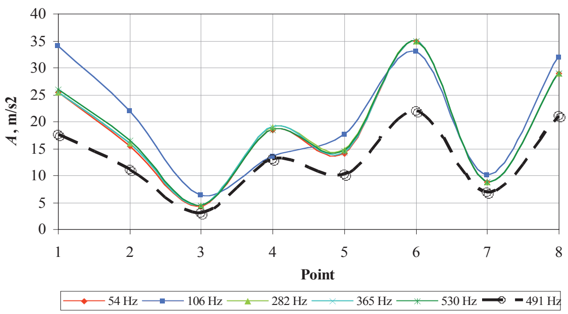


b

Fig. 7: Change of vibration amplitude in saw (D = 800 mm) surface, when the saw vibrates by two modes at the same time: a – "main" mode of 106 Hz frequency; b – "main" mode of 491 Hz frequency

It also was measured coefficient of damping of each of the modes, when two modes were „acting" at the same time.

Table 3 provides the values of 106 Hz frequency mode`s coefficient of damping, simultaneously inducing one of the modes of other frequencies – 54 Hz, 282 Hz, 365 Hz, 491 Hz and 530 Hz.

As it can be seen from Tab. 3, coefficient of damping of the „main" in this case mode (106 Hz) changes, when simultaneously starts „acting" one of the modes of differing frequencies.

Fig. 8 shows the law of coefficient of damping change of the „main" mode (106 Hz), in accordance with the data presented in Tab. 3. A separate curve (dotted black line) shows also the change of coefficient of damping of one 106 Hz mode, when the saw vibrates only by this mode.

Fig. 8 shows that when the saw vibrates simultaneously by two modes of lower frequencies (54 Hz and 106 Hz), coefficient of damping of the „main" mode (106 Hz) decreased by about 20-30%. Vibrating by modes of 106 Hz and 365 Hz frequencies at the same time, coefficient of damping of the „main" mode (106 Hz) in different points increased by about 10-15%.

Analogically in Tab. 4 are presented values of the damping coefficent of 491 Hz frequency mode in saw surface points when the modes of other frequencies are induced simultaneously – 54 Hz, 106 Hz, 282 Hz, 365 Hz ir 530 Hz .

Tab. 3: Values of sum coefficient of damping, when „main" mode is 106 Hz

Coefficient of damping	Measurement point							
	1	2	3	4	5	6	7	8
106 Hz mode	0.0010108	0.0009448	0.0004613	0.0005199	0.0008578	0.0007413	0.0005077	0.0008122
affecting with 54 Hz mode	0.0008477	0.0007995	0.0003767	0.0004237	0.0007071	0.0003771	0.0003766	0.000565
affecting with 282 Hz mode	0.0011093	0.0009578	0.0004889	0.0004333	0.0009	0.0007695	0.0002728	0.0007332
affecting with 365 Hz mode	0.0014127	0.0010844	0.0005184	0.0005188	0.0011324	0.0007554	0.0003774	0.0007549
affecting with 491 Hz mode	0.0009885	0.0009872	0.0003765	0.0005648	0.0004553	0.0006899	0.0009261	0.0008455
affecting with 530 Hz mode	0.0009409	0.0007997	0.0003766	0.0004706	0.0009426	0.0007069	0.0003296	0.0008005

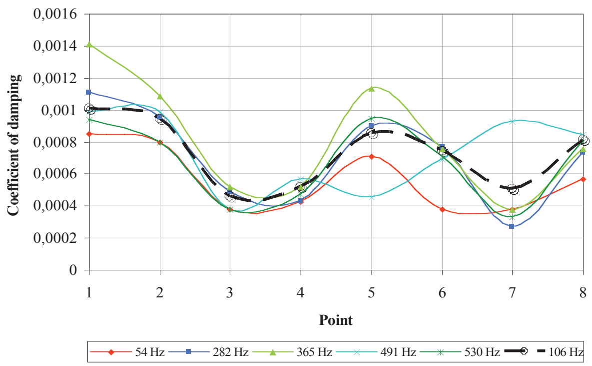


Fig. 8: Change of the coefficient of damping in saw (D = 800 mm) surface, when the saw vibrates by two modes at the same time: frequency of the „main" mode is 106 Hz

Tab. 4: Values of sum coefficient of damping, when „main" mode is 491 Hz

Coefficient of damping	Measurement point							
	1	2	3	4	5	6	7	8
491 Hz mode	0.0010923	0.0009306	0.0004477	0.0007717	0.0011496	0.0007297	0.0004279	0.0007991
affecting with 54 Hz mode	0.0011617	0.0010974	0.0004479	0.0008147	0.0012843	0.0007099	0.0004599	0.0008556
affecting with 106 Hz mode	0.0007941	0.0006904	0.0004061	0.0007728	0.000876	0.0006931	0.0004063	0.0008862
affecting with 282 Hz mode	0.0012229	0.0009755	0.0004276	0.0008147	0.0012028	0.0006929	0.0004366	0.0008452
affecting with 365 Hz mode	0.0012025	0.0010364	0.0004683	0.0008351	0.0012028	0.0007171	0.0004287	0.0008494
affecting with 530 Hz mode	0.001284	0.0009958	0.0004581	0.0008351	0.0012231	0.0007198	0.0004358	0.0008556

In accordance with the data of Tab. 4, Fig. 9 presents change law of the coefficient of damping of 491 Hz mode in saw surface points. A separate curve shows change law of the coefficient of damping when saw vibrates only by 491 Hz mode.

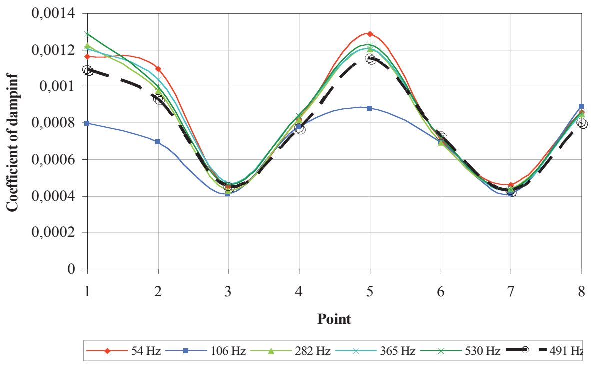


Fig. 9: Law of the coefficient of damping variation on the saw (D = 800 mm) plane, when the saw vibrates by two modes simultaneously: frequency of the „main" mode is 491 Hz

As it can be seen (Fig. 9), in this case the coefficient of damping signally decreases (about 1.4 times) only when the saw vibrates by 106 Hz and 491 Hz modes. In other cases coefficient of damping changed very insignificantly.

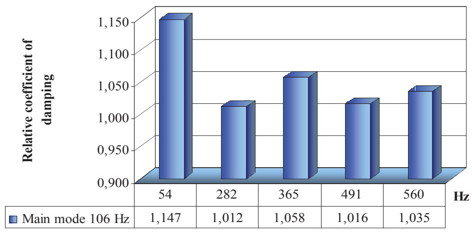
For further analysis of coefficient of damping relative coefficient of damping, i.e., proportion between coefficient of damping of the „main" and "affecting" modes, is introduced. In this case relative coefficient of damping shows how many times coefficient of damping of the „main" mode (for example, 106 Hz) changes, affecting it by one of the other modes (54 Hz, 282 Hz, etc.).

Fig. 10 shows the laws of relative coefficient of damping distribution of two saw modes (106 Hz and 491 Hz).

As it can be seen (Fig.10), coefficient of damping of 106 Hz mode is mostly (up to 13%) influenced by 54 Hz mode, while the least (1%) – by 282 Hz mode. Coefficient of damping of higher frequency (491 Hz) is less influenced by other modes. In this case it changes by 2-3%. Coefficient of damping changes least (0.5%) when 282 Hz mode "affects" 491 Hz frequency mode.

Therefore, when saw vibrates by two modes simultaneously, sum vibration amplitude and coefficient of damping of separate modes change considerably. When saw is affected by shocks during sawing, vibrations of several modes can be induced at the same time. It is obvious that in this case change law of saw vibration parameters would be still more complicated.

In similar research works (Yu 1986, Marui et al. 1994, Nishio and Marui 1996, Aubry et al. 2000, Stakhiev 2003) only saw bend form (mode) and resonance frequency were estimated, though the way in which one mode affects the other, how this changes oscillation amplitude and coefficient of damping is not analysed. There are known works in which are measured resonance frequencies, estimated saw modes and calculated coefficient of damping based on the abovementioned data (Radcliffe and Mote 1980). Berger et al. (2003) offered a method which can be used for excitation of several frequencies at the same time. Saw is struck with a hammer. Several resonance frequencies are excited at the same time, while oscillation is measured using a sensor. In such way several modes excite at the same time and their frequencies are estimated during analysis. However, sum amplitude and coefficient of damping were not determined in such method.



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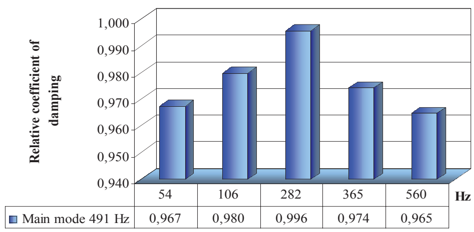

b

Fig. 10: Distribution law of relative coefficient of damping of two saw modes: a – 106 Hz mode, b – 491 Hz mode

Though many researches in this area are published, such "inter-effect" of modes is not known. Therefore, predicting saw behaviour during work and kerf precision, such researches are pending.

It is self evident that increase of saw oscillation amplitude challenge changes in kerf width, while coefficient of damping changes affect saw oscillation dying-out period.

CONCLUSIONS

Analysis of the obtained data leads to the following conclusions:

1. It was found that shocks, affecting saw, induce vibrations of several modes.
2. It was obtained that simultaneous „action" of two modes of the saw (D = 800 mm, s = 4.0 mm) in different points causes the increase of sum vibration amplitude on an average by 1.5 times.
3. It was determined that, "affecting" saw lower frequency (e.g., 106 Hz) mode with other modes, coefficient of damping of the former in individual cases decreases to 13%, while "affecting" higher frequency (e.g., 491 Hz) mode coefficient of damping increases in 2-5 %.

REFERENCES

1. Stakhiev, Y. M., 1989: Workability of flat circular saws. Lesnaya promyšlennostj. Moscow, 384 pp. (in Russian)
2. Stakhiev, Y. M., 1977: Stability and vibrations of flat circular saws. Lesnaya promyšlennostj. Moscow, 296 pp. (in Russian)
3. Pat. 4979417, 1990: Rotating saw blade having improved critical vibrational speed. USA
4. Pat. 2048285, 1995: Device for noise and vibration reduction of circular saws. Russia
5. Gorin, S. V., 1997: Circular saw of lowered vibration and noise. Wood Industry 4: 25-26. (in Russian)
6. Pat. EP0819491, 1998: Noiseless and stable blade disc for circular saws
7. Pat. WO9932251, 1999: Cutting tool form
8. Yu, S. C., Huang, S. C., 2001: Vibration of a three-layered viscoelastic sandwich circular plate. International Journal of Mechanical Sciences 43 (10): 2215-2236
9. Ukvalbergienė, K., Vobolis, J., 2005: Bend forms of circular saws and evaluation of their mechanical properties. Materials Science (Medžiagotyra) 11(1): 79-84
10. Vobolis, J., 1996: Diagnostics of light industry machinery. Technologija. Kaunas, 216 p. (in Lithuanian)
11. Yu, R. C., 1986: Vibration and parametric excitation in rotating asymmetric circular plates. PhD. Dissertation, University of California, Berkeley
12. Marui, E., Ema, S., Miyachi, R., 1994: An experimental investigation of circular saw vibration via a thin plate model. International Journal of Machine Tools and Manufacture 34(7): 893-905
13. Nishio, S., Marui, E., 1996: Effects of slots on the lateral vibration of a circular saw blade. International Journal of Machine Tools and Manufacture 7 (36): 771-787
14. Aubry, E., Fendeleur, D., Schmitt, F. M., Renner, M., 2000: Measurement of the natural frequencies and critical speed of roll-tensioned discs. Experimental Techniques 24 (4): 22-26
15. Stakhiev, Y. M., 2003: Research on circular saw disc problems: several of results. Holz als Roh- und Werkstoff 61: 13-22
16. Radcliffe, C. J., Mote, C. D. Jr., 1980: Active control of circular saw vibration using spectral analysis. Wood Science 13(3): 129-139
17. Berger, S., Aubry, E., Marquet, F., Thomann, G., 2003: Experimental modal shape identification of a rotating asymmetric disk subjected to multiple-frequency excitation: use of finite impulse response (FIR) filters. Experimental Techniques 27(6): 44-50

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