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# Impact of Circular Saws Surface Hardness and Thickness on Coefficient of Damping

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During sawing, under the influence of different forces and saw speeds close to critical, resonance vibrations occur. Seeking to increase the stability of saw, it is important to estimate saw bend form, the vibration amplitude and vibration damping time (coefficient of damping). Having estimated vibration damping time, it is also possible to forecast the change of kerf size in time. The paper presents study results of four 9ХФ circular steel saws of longitudinal sawing, the diameters of which are 400, 500, 800 and 1000 mm, as well as five 9ХФ circular steel saws of transverse sawing, the diameter of which is 400 mm. Amplitude-frequency characteristics, resonance frequencies and saw bend forms (modes) were ascertained. The variation of coefficient of damping within frequency range and its distribution in the points of saw plane were estimated. The variation of saw thickness was estimated within 0.12 mm – 0.25 mm, while surface hardness within 7 HRC – 10 HRC range respectively. The coefficient of damping as well as the relationship between saw surface hardness and thickness were evaluated. It was found, that the coefficient of damping is inversely proportional to the variation of surface hardness of saw material. Estimating the ratio between saw thickness and surface hardness, it was found, that the value of coefficient of damping is preconditioned by surface hardness.

*Keywords*: circular saw, coefficient of damping, vibration amplitude, surface hardness, saw thickness.

## 1. INTRODUCTION

Circular saws are widely used in the timber processing industry. During sawing, under the influence of different forces and saw rotation speeds close to critical, resonance vibrations occur. These vibrations reduce saw strength, quite frequently are the cause of fissures, lead to unallowable noise and have a great influence on kerf dimensions [1]. Due to resonance vibrations increases the kerf, which leads to reduced timber output.

Seeking to increase saw stability, it is important to estimate the saw bend form, size of vibration amplitude and vibration damping time (coefficient of damping). Having estimated vibration damping time, it is possible to forecast the change of kerf size in time.

One of optimal ways to smother disc vibrations is the application of damping materials [2]. For this purpose elastic polymers are applied, characterized by a high coefficient of damping.

Many studies are carried out with multilayered metal discs [2 – 3], the central layer of which is made of elastic material. This layer increases damping, but at the same time decreases frequencies of the system [2].

To increase the damping of saw vibrations, special damping rings are used [4 – 6]. However, to be more efficient, they have to comprise at least a half of disc diameter and have no contact with the sawn timber. Therefore, such rings are used only producing saws of small diameter. In the other cases, to increase saw form stability, slots of different sizes and forms are made in the saw disc [7 – 10]. Besides, special metal shields are attached to the saw, which cause swaging stresses, when the saw heats up during operation [11].

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Recently many studies are devoted to elaborate a very thin circular saw operating under intensive regimes [12 – 13]. Such a solution increases the output of sawn timber, reduces the losses of consumed sawing energy and the amount of metal used. However, when the saw becomes thinner, its resonance vibrations are especially pronounced. Thus, it is important to estimate the influence of saw thickness and its unevenness in the saw plane on saw stability during operation. Saw stability depends also on saw material and its properties such as modulus of elasticity and hardness.

The aim of this work is to estimate the saw coefficient of damping depending on the saw thickness and surface hardness.

## 2. STUDY METHODS AND EQUIPMENT

The studies of vibrations were carried out using a dynamic study method [14]. A special stand [15] was used for the studies. With the help of a vibrator, resonance vibrations of circular saw are excited. Knowing resonance frequencies and bend forms (modes) of a saw, it is possible to estimate saw modulus of elasticity and coefficient of damping.

For a more precise estimation of saw forms, the whole saw is divided into separate parts, deriving a certain number of diameters and ring circles (Fig. 1). In their intersection places (points $1_1$, $1_2$, $1_3$, $1_4$, $2_1$, $2_2$, $2_3$, $2_4$ and so on) a sensor is fastened and resonance frequency as well as amplitude are ascertained. To obtain a more accurate saw mode, measurements are conducted in 96 points. This is especially actual obtaining higher saw modes.

In measurement points saw thickness and hardness were measured, and coefficient of damping was estimated.

Internal friction (damping) is evaluated having ascertained amplitude-frequency characteristics of the saw [14]. Its form predetermines internal 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 lower than resonance amplitude (Fig. 2).

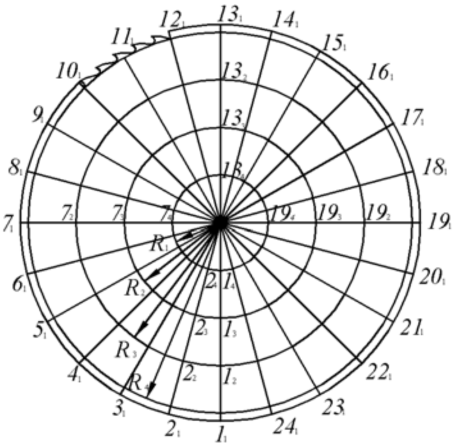



**Fig. 1.** Circular saw partitioning scheme: $1_1$, $1_2$, $1_3$, $1_4$, $2_1$, $2_2$, $2_3$, $2_4$ etc. – points, where vibrations, are measured; $R_1$, $R_2$, $R_3$, $R_4$ – radii of circles

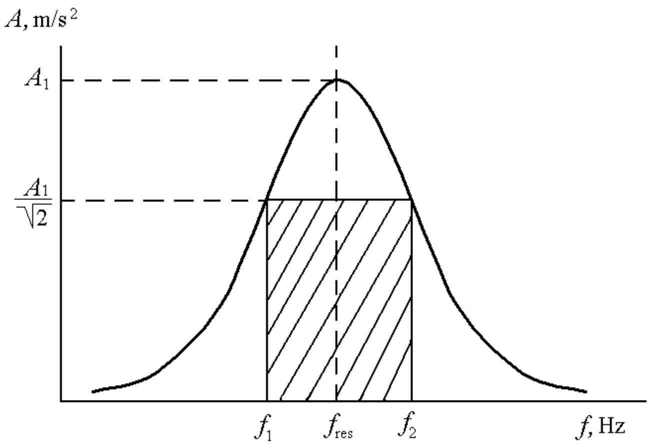



**Fig. 2.** Amplitude-frequency characteristics of the saw: $A_1$ – resonance amplitude, $f_{res}$ – resonance frequency, $f_1$ and $f_2$ – frequencies, under which amplitude is $\sqrt{2}$ times lower than resonance amplitude

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

$$\mathrm{tg}\delta \approx \frac{f_2 - f_1}{f_{res}}, \tag{1}$$

where $\mathrm{tg}\delta$ is tangent of loss angle.

To study the hardness, the hardness measurement device “Hardness Tester” was applied. Measurements using this device were conducted applying Rockwell’s method [16]. Measurement precision – 1 HRC. The thickness measurement device produced by “Vogel” firm was used to study thickness. Measurement precision – 0.01 mm.

## 3. EXPERIMENTAL RESULTS

Four circular 9XΦ steel type saws of longitudinal sawing (I1 – I4) and five 9XΦ steel type saws of transverse sawing (S1 – S5) were studied. Table 1 gives technical parameters of the studied saws. The used saw fastening moment was 150 Nm. Measurements were conducted within 20 Hz – 2000 Hz range.

**Table 1.** Technical parameters of studied saws

| Saw | Diameter of saw $D_s$, mm | Diameter of flanges $d_f$, mm | Number of teeth $z$, unit |
|---|---|---|---|
| I1 | 1000 | 140 | 48 |
| I2 | 800 | 140 | 48 |
| I3 | 500 | 140 | 48 |
| I4 | 400 | 140 | 48 |
| S1 – S5 | 400 | 140 | 72 |

Measuring vibrations, saw resonance frequencies and amplitude-frequency characteristics were determined. Fig. 3 presents amplitude-frequency characteristics of transverse saws. Analogous amplitude-frequency characteristics are obtained for the saws of longitudinal sawing too.

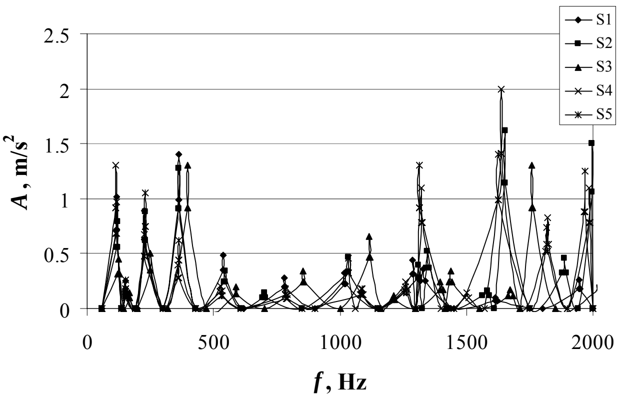



**Fig. 3.** Amplitude-frequency characteristics of transverse saws (diameter of saws S1 – S5 $D_s$ = 400 mm)

From Fig. 3 it can be seen, that within 20 – 500 Hz and 1300 Hz – 2000 Hz frequency ranges vibration amplitude of saws S1 – S5 varies from 1.2 m/s$^2$ – 2 m/s$^2$. Within 500 Hz – 1300 Hz frequency range vibration amplitude varies from 0.2 m/s$^2$ – 0.7 m/s$^2$, i.e within this frequency range it is 2 – 6 times less.

Table 2 presents coefficient of damping values of saws S1 – S5 under different resonance frequencies.

From Table 2 it can be seen, that coefficient of damping of S1 saw under changing frequency from 116.8 Hz to 539.2 Hz decreases about 47 times, from 539.2 Hz to 1328.4 Hz – increases 22 times, while from 1328.4 Hz to 2000 Hz again decreases about 10 times.

It was found that, under the same vibration mode, the values of resonance frequency in different saw measurement points differ, i.e. having moved transducer from one point, where resonance has been recorded, to another point, resonance undergoes slight changes.

To estimate this frequency variation, it is necessary to change the frequency of the generator (vibrator), i.e. resonance in this case has to be recorded in each measurement point separately.

Fig. 4 presents variation of one of the resonance frequencies, in the points of saw S1 ($D$ = 400 mm) circles,

**Table 2.** Coefficient of damping of saws S1 – S5

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Saw S1 | | | | | | | | | | | |
| $f_{res}$ | 116.8 | 152.2 | 227.7 | 363.0 | 539.2 | 780.7 | 1019.2 | 1288.1 | 1328.4 | 1615.5 | 1946.5 |
| tg$\delta$ | 0.02656 | 0.00920 | 0.00176 | 0.00138 | 0.00056 | 0.00397 | 0.00098 | 0.00396 | 0.01250 | 0.00421 | 0.00118 |
| Saw S2 | | | | | | | | | | | |
| $f_{res}$ | 120.8 | 153.3 | 228.8 | 364.2 | 544.2 | 700.1 | 1034.3 | 1311.5 | 1581.4 | 1651.5 | 1996.4 |
| tg$\delta$ | 0.03477 | 0.01696 | 0.00175 | 0.00055 | 0.00129 | 0.01271 | 0.00058 | 0.00404 | 0.01511 | 0.00242 | 0.00145 |
| Saw S3 | | | | | | | | | | | |
| $f_{res}$ | 124.2 | 166.9 | 249.1 | 396.5 | 588.0 | 855.3 | 1116.7 | 1211.6 | 1397.4 | 1672.2 | 1760.1 |
| tg$\delta$ | 0.04670 | 0.01857 | 0.00241 | 0.00076 | 0.00170 | 0.00304 | 0.00081 | 0.00957 | 0.00751 | 0.01082 | 0.00250 |
| Saw S4 | | | | | | | | | | | |
| $f_{res}$ | 113.3 | 151.8 | 227.4 | 360.5 | 526.5 | 790.6 | 1034.1 | 1260.9 | 1323.1 | 1637.5 | 1983.5 |
| tg$\delta$ | 0.02207 | 0.01713 | 0.00176 | 0.00111 | 0.00513 | 0.00354 | 0.00068 | 0.00928 | 0.00234 | 0.00116 | 0.00187 |
| Saw S5 | | | | | | | | | | | |
| $f_{res}$ | 115.4 | 152.2 | 229.9 | 363.0 | 532.5 | 783.5 | 1086.4 | 1258.7 | 1314.1 | 1625.4 | 1967.9 |
| tg$\delta$ | 0.02513 | 0.01051 | 0.00130 | 0.00083 | 0.00469 | 0.00268 | 0.00828 | 0.01057 | 0.00236 | 0.00135 | 0.00112 |

the radii of which are $R_1 = 30$ mm, $R_2 = 85$ mm, $R_3 = 140$ mm, $R_4 = 195$ mm. It was determined that, the resonance frequency varies within 150.5 – 158.6 Hz range (average resonance frequency $f'_{res} = 155.4$ Hz).

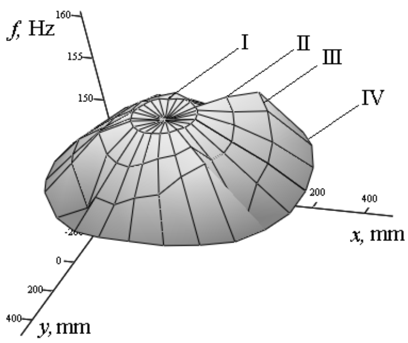
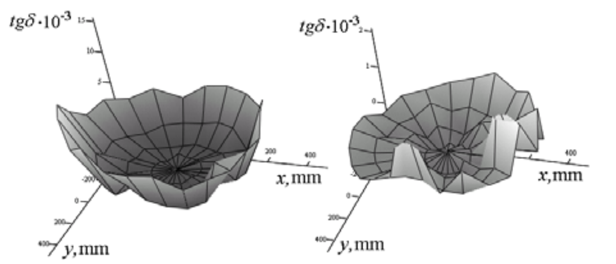



**Fig. 4.** Variation of resonance frequency, measured in different saw S1 ($D = 400$ mm) disc points: I – circle of the radius $R_1 = 30$ mm, II – $R_2 = 85$ mm, III – $R_3 = 140$ mm, IV – $R_4 = 195$ mm

Fig. 4 shows that, the less diameter of the circle correspond higher resonance frequencies. Besides, the resonance frequency in points of the same circle remains practically constant.




a b

**Fig. 5.** Distribution of coefficient of damping in saw S1 plane: a – $f'_{res} = 155.4$ Hz, b – $f'_{res} = 359$ Hz

The variation of resonance frequency causes also the change of coefficient of damping. Fig. 5 shows the distribution of coefficient of damping in saw S1 plane under two resonance frequencies ($f'_{res} = 155.4$ Hz and $f'_{res} = 359$ Hz).

From Fig. 5 it can be seen that, for resonance frequency $f'_{res} = 155.4$ Hz, the coefficient of damping is the highest in the outside circle of the saw. The coefficient of damping is lower approaching to the saw fastening flanges. Analogous results are obtained for resonance frequency $f'_{res} = 359$ Hz.

Fig. 6 shows the variation of the coefficient of damping in the points of saw S1 ($D = 400$ mm) circles, the radius of which is $R_1 = 30$ mm, $R_2 = 85$ mm, $R_3 = 140$ mm, $R_4 = 195$ mm, when resonance frequency $f'_{res} = 155.4$ Hz.

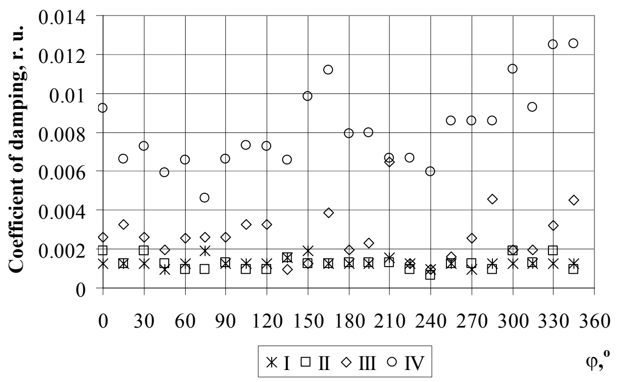



**Fig. 6.** Variation of coefficient of damping, measured in saw S1 ($D = 400$ mm) disc points: I – circle with the radius $R_1 = 30$ mm, II – $R_2 = 85$ mm, III – $R_3 = 140$ mm, IV – $R_4 = 195$ mm

Fig. 6 shows that the coefficient of damping varies within 0.00063 – 0.01254 range. It was found that, the less the radius of saw circle is, the lower coefficient of damping is, i.e. the closer measurement point is to the flanges, the higher resonance frequency and the lower coefficient of damping are. The similar changes of resonance frequencies and coefficient of damping were obtained also carrying out studies with other saws.

Later at each saw vibration measurement point, saw thickness and surface hardness (on both sides of saw disc) were measured. It was found that, the saw thickness changes within 0.12 mm – 0.25 mm, while surface hardness within – 7 HRC – 10 HRC range.

Having analysed the data, it was found that, the variation of resonance frequency, amplitude and coefficient of damping are related to the change of saw thickness and hardness. Fig. 7 presents the variation of vibration amplitude, when $f_{res}$ = 155.5 Hz, and the saw thickness in different measurement points.

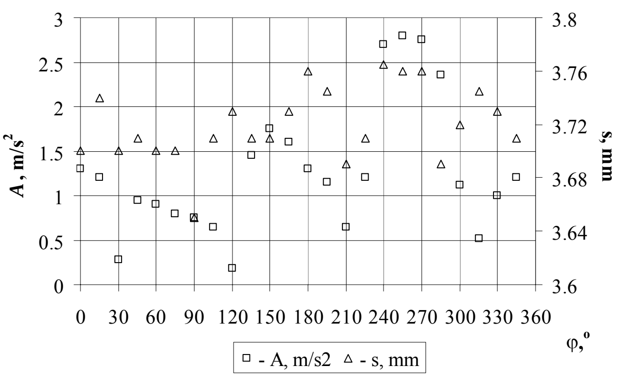



**Fig. 7.** Variation of the vibration amplitude and the saw thickness, measured in the points of saw $D$ = 1000 mm (circle $R_4$ = 470 mm)

Fig. 7 shows that, the vibration amplitude increases as the saw thickness increases. In the zone between 225 and 255 degrees the amplitude increases, between 255 and 285 this is constant, while between 285 and 330 this decreases. An analogous dependence can be observed with saw thickness: between 225 and 255 degrees saw thickness is higher, between 255 and 285 is constant, while between 285 and 330 is lower. It was found that, the amplitude changes within 0.28 m/s$^2$ – 2.9 m/s$^2$, while saw thickness within – 3.65 mm – 3.77 mm range.

Fig. 8 presents similar dependencies of the coefficient of damping and surface hardness.

Fig. 8 shows that, the variation of the coefficient of damping is inversely proportional to the change of average surface hardness of saw material. It has been ascertained that in the zones where the coefficient of damping increases, the surface hardness in analogous zones decreases.

It was found, that coefficient of damping varies within 0.000965 – 0.001928, while surface hardness within – 39 HRC – 49 HRC range.

It was also determined, that the saw coefficient of damping depends on the ratio between surface hardness and thickness (Fig. 9).

It can be seen that, with the increasing of ratio between surface hardness and thickness, the coefficient of damping decreases. On the other side, as it is seen from Fig. 7 and Fig. 8, both the saw thickness and the hardness increase at the same time. This shows that the decrease of the coefficient of damping is preconditioned by a more pronounced increase in hardness.

Therefore, it is possible to estimate "volume" hardness of saw material which depends on the temper depth and the saw thickness.

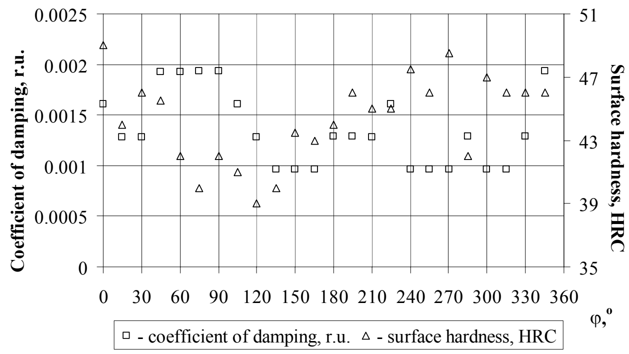



**Fig. 8.** Variation of saw coefficient of damping and surface hardness, measured in the points of saw $D$ = 1000 mm (circle $R_4$ = 470 mm)

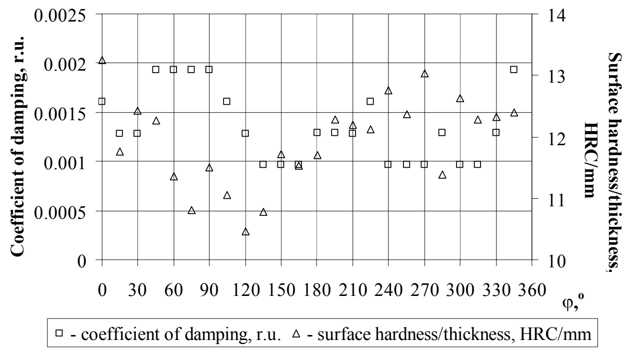



**Fig. 9.** Variation of the ratio between surface hardness and thickness and the coefficient of damping, measured in the points of saw $D$ = 1000 mm (circle $R_4$ = 470 mm)

Analogous dependencies were obtained conducting measurements also in the points of other saw circles, the radii of which are $R_1$ = 140 mm, $R_2$ = 250 mm, $R_3$ = 360 mm, and studying other saws.

The presented study methods allow to estimate some parameters of saw material and to forecast the stability of its form.

## CONCLUSIONS

1. For the investigated saws the saw thickness in saw plane varies within 0.12 mm – 0.25 mm, while the surface hardness varies within 7 HRC – 10 HRC range. We have found that the vibration amplitude varies proportionally to the saw thickness. In the case of longitudinal saws, under thickness increment in separate points by about 1 %, the vibration amplitude increases by about 78 %.
2. The saw coefficient of damping changes inversely proportionally to the surface hardness. It was obtained that in the case of longitudinal saw, under surface hardness increment in separate points by about 20 %, the coefficient of damping decreases 2 times.

3. The variation law of the saw coefficient of damping is mainly preconditioned by the variation of saw surface hardness.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