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IMPACT OF DOMINANT VIBRATIONS ON NOISE LEVEL OF DIMENSION CIRCULAR SAWBLADES

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ABSTRACT

The paper deals with vibrations of circular sawblades and their effects on increasing the noise level. These effects were analysed at a Pilana 400-72 TFZL circular sawblade intended for large-area trimming to size. Through the direct measurement of noisiness and by the Fourier analysis of the time response of vibrations of the circular sawblade body dominant frequencies of vibrations were determined, which effect increasing the noise level and the disk whistling.

KEYWORDS: Circular sawblade, vibration, resonance, noise level, Fourier transformation.

INTRODUCTION

Vibrations of circular sawblades affect substantially the course of the behaviour of parameters at sawing. Decreasing the amplitude of vibrations is inevitable to improve the quality of cut surface, increasing the yield of material and, last but not least, decreasing the noise level.

Problems of vibrations and noisiness of circular sawblades were already studied in the past. At present, they are rather well processed (Stachiev 1989, Lisičan 1992, Siklienka and Svoreň 1997, Goglia and Lucic 1999, Javorek and Sokolowski 2000, Kopecký et al. 2007) etc. Due to its discoid shape, a circular sawblade can have the infinity of static eigenfrequencies characterized by certain shapes of vibrations.

In practice, asymmetric vibrations occur frequently according to nodal diameters k = 1 to 4 ($f_{0/1}$, $f_{0/2}$, $f_{0/3}$, $f_{0/4}$,) possibly they occur in combination with centrally symmetric vibrations with a nodal circle c = 1 ($f_{1/0}$, $f_{1/1}$, $f_{1/2}$) Thomas et al. (2004), (Fig. 1).

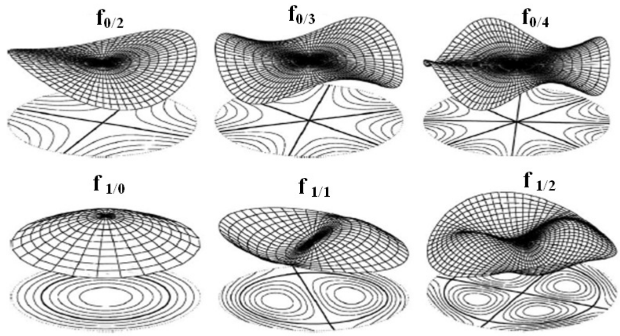


Fig. 1: Shapes of vibrations.

Nishio and Marui (1996) mention a relation (1) for a deformation w in point P of a statically vibrating disc on a radius r, angle of turning φ and nodal diameter *k* (Fig. 2).

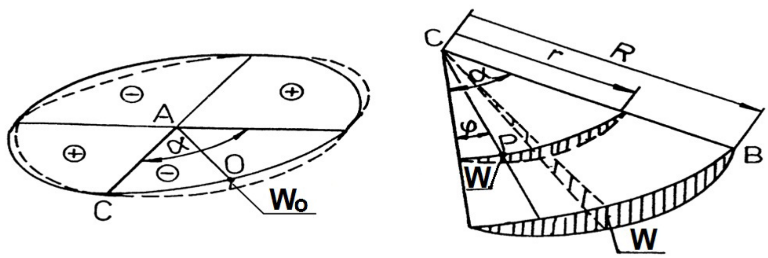


Fig. 2: Sawblade disc deformation.

$$w(r,\varphi,t) = f(r,\varphi)\cdot \sin k\varphi \cdot \cos 2\pi \cdot f_{st} \cdot t \qquad \text{(m)} \qquad (1)$$

where: w (r,φ,t) – the sawblade disc deformation expressed in the polar coordinate system (r,φ) in time t
w_0 – deformation - deviation of point 0
f (r,φ) – the shape function expressing the point P deviation in the polar coordinate system (r,φ)
f_{st} – the frequency of vibrations of a standing disc.

If the disc turns with frequency n (min^{-1}) it is possible to use for the angle of turning an equation

$$\varphi = \frac{2\pi n t}{60} \qquad \text{(rad)} \qquad (2)$$

substituting into equation (1) and through an adaptation by means of the product of trigonometric functions sinα.cosβ, it is possible to transcribe equation (1) into relation (3)

$$w(r,\varphi,t) = \frac{f(r,\varphi)}{2}\cdot \sin 2\pi\left(f_{st} + \frac{k\cdot n}{60}\right)\cdot t - \frac{f(r,\varphi)}{2}\cdot \sin 2\pi\left(f_{st} - \frac{k\cdot n}{60}\right)\cdot t \qquad (3)$$

The first member of the equation (3) represents a wave with the rotation frequency 60 f_{st}/k (min^{-1}) progressing forward, ie in the same direction as the circular sawblade turning. The second member of the equation (3) represents a wave with the rotation frequency 60 f_{st}/k (min^{-1}) progressing backwards, ie counter the direction of the circular sawblade turning. These waves are indicated as a "forward running wave" and "backwards running wave" (Siklienka and Svoreň 1997).

Inner of the rotating disc vibrations fd are affected by the existing frequency of the sawblade rotation n and can be termed as f_d (n). Thus, the frequency of a forward-running wave f_1 and of a backwards-running wave f_2 can be expressed as follows:

$$f_1 = f_d(n) + \frac{k \cdot n}{60} \quad \text{and} \quad f_2 = f_d(n) - \frac{k \cdot n}{60} \qquad \text{(Hz)} \tag{4}$$

Based on a relation (4) for the backwards-running wave f_2 it is evident that once the frequency of the sawblade disc n begins to balance dynamic frequencies of the disc inner f_d(n) at the concrete number of nodal diameters k, the frequency of a backwards-running wave will equal to zero $f_2 = 0$.

Immovable waves will originate in the space; the backwards-running wave appears to be stable in the space, the disk will become unstable and only a very small force is sufficient for its deviation. Thus, sawing is impossible and danger of the disc deformation and its subsequent ripping appears. This limiting (threshold) frequency of the backwards-running wave is termed the turning critical frequency and rpm when this event happens is called critical rpm. Critical rpm can be expressed by the following relation (5):

$$f_2 = 0 \qquad\qquad f_d(n) = k \cdot f_n$$

from here, the disc critical rpm

$$n_k = \frac{60 \cdot f_d(n)}{k} \qquad (\text{min}^{-1}) \tag{5}$$

The inner of the rotating disc vibrations f_d (n) is possible to calculate according to known equation (6). In consequence of a centrifugal force the dynamic inner increases as quadratic with increasing working rpm of the circular sawblade.

$$f_d(n) = \sqrt{f_{st}^{\,2} + \lambda \cdot \left(\frac{n}{60}\right)^2} \qquad \text{(Hz)} \tag{6}$$

where: λ - coefficient of a centrifugal force.

Through the substitution of (6) into equation (5) we can get a relation (7) for critical rpm and nodal diameters k = 2, 3, 4 etc. According to (Strzelecki 1974), critical rpm cannot occur for a nodal diameter k = 1.

$$n_k = \frac{60 \cdot f_{st}}{\sqrt{k^2 - \lambda}} \qquad (\text{min}^{-1}) \tag{7}$$

Static frequencies and frequencies of a backwards running wave are usually determined experimentally on special trial stands (Siklienka and Svoreň 1997) where the creation of Chladny figures on an experimental circular sawblade and Lissajous figures on the oscilloscope screen

(Kopecký 2007) serve as an indicator for reading the frequencies.

Direct measurements of amplitudes of the circular sawblade vibrations are a more exact method, which can be used for the relatively accurate determination of the disc deformation (±0.01 mm) and resonance rpm at higher harmonic frequencies. Advantages of this method consist in the experimental determination of amplitudes of the disc vibrations or of the disc rim directly on the saw (on an experimental stand) including all effects, which influence the disc inner.

MATERIAL AND METHODS

The direct measurement of amplitudes of the sawblade disc vibrations, determination of resonance rpm, which are frequently accompanied by the increased noise level, was carried out experimentally on a trial stand designed for research into cutting by circular sawblades (Fig. 3).

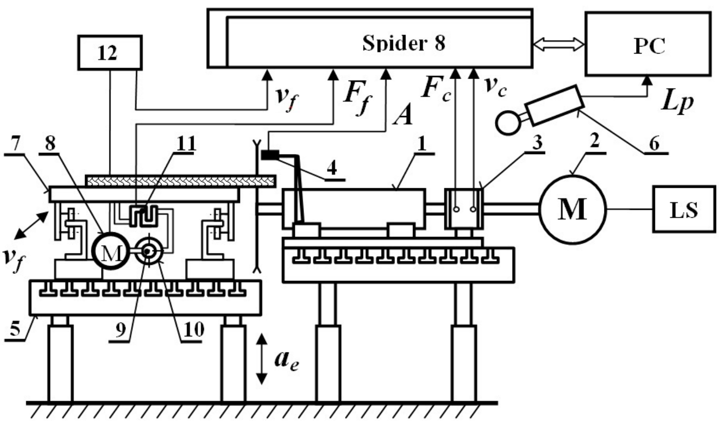
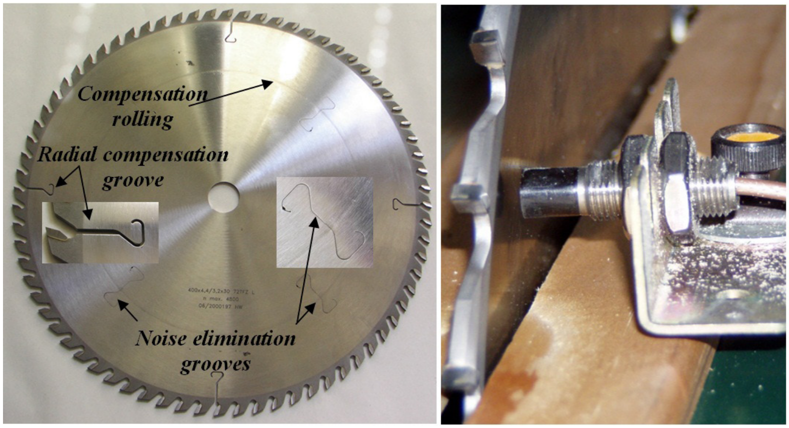


Fig. 3: Diagram of an experimental stand.
1 – spindle, 2 – electric motor with rpm regulation LS, 3 - cutting force $\boldsymbol{F_c}$ and speed $\boldsymbol{v_c}$ sensor, 4 – contactless sensor of vibrations A, 5 - grate table, 6 - noise meter, 7 – feeding carriage, 8 –electric motor for the carriage feed, 9 - ball screw, 10 – nut, 11 – feeding force sensor F_f, 12 – frequency converter for the feeding speed change $\boldsymbol{v_f}$.

The tested Pilana 400-72 TFZL circular sawblade (Fig. 4a) intended for large-area trimming to size of agglomerated materials shows following parameters: diameter of a circular sawblade D = 400 mm, number of teeth z = 72, material of teeth HW, thickness of the disk body s = 3.2 mm, tooth width s_T = 4.4 mm, radial compensation grooves 4, noise elimination grooves 4, compensation rolling – Yes, tooth shape TFZL, radius of the edge fillet ρ_0 = 7 µm, diameter of a flange d_p = 110 mm and fixative ratio $\alpha_p = d_p/D$ = 0.275.

Vibrations were measured using the vibration sensor EPRO PR6423/000-001 (Fig. 4b), which operated on the principle of eddy (vortex) currents. The sensor was placed at the disk rim in a plane under teeth.



a) b)

Fig. 4: Tested circular sawblade and the sensor of vibrations.

The measured course of the circular sawblade vibrations was analysed by the method of the spectral analysis of a signal in the time domain. The method principle consists in the use of Fourier transformation where the measured course of vibrations U(t) with a period T_0 is distributed to a direct-current component a_0 and further to a number of harmonic (sinusoidal and cosine) components (Fig. 5). The time response of vibrations is then retransformed to a single-sided amplitude spectrum based on a known relation (8).

$$U(t) = a_0 + 2\sum_{k=1}^{\infty}\left[a_k \cos(k2\pi f_0 t) + b_k \sin(k2\pi f_0 t)\right] \qquad (8)$$

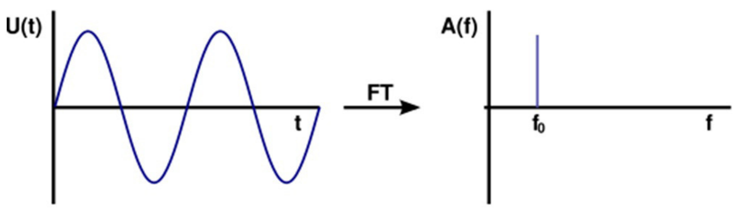


Fig. 5: Spectrum of a sinusoidal wave.

A basic condition of this transformation is periodicity of the measured function of the course of vibrations, its continuity, limitation and the finite number of local extremes. Harmonic components show a frequency corresponding to entire multiples k = 1, 2, 3,… f_0=1/T_0 of the repeating frequency of a basic course (ie, the first harmonic). Basic functions are determined by terms cos ($k2\pi f_0 t$) and sin ($k2\pi f_0 t$).

Amplitudes of harmonic components can be calculated by following relations:

$$a_k = \frac{1}{T_0}\int_{-T_0/2}^{T_0/2} f(t)\cos(k2\pi f_0 t)dt,\ b_k = \frac{1}{T_0}\int_{-T_0/2}^{T_0/2} f(t)\sin(k2\pi f_0 t)dt \qquad (9)$$

and the direct-current component

$$a_0 = \frac{1}{T_0} \int_{-T_0/2}^{T_0/2} f(t)dt \tag{10}$$

The Chauvin Arnoux CA834 noise meter with the digital record of the noise level accurate to ± 1.5 % measured the noise level (Fig. 6). The noise meter was placed 100 cm from the measured disc and 150 cm above the ground in the place of the operator-working zone.

Fig. 6: Chauvin Arnoux noise meter and its placing at the experimental stand.

A basic descriptor for the description of noise in the working environment is the level of acoustic pressure L_p (dB) related to a reference acoustic pressure p_0=20 μPa, which corresponds to the threshold of hearing at a frequency of 1000 Hz. An expression of the level of noise in decibels partly describes the physiology of hearing when the linear increment of an acoustic perception corresponds to the relative change of a stimulus (Fechner-Weber law) and partly makes possible a well-arranged classification of noise data because a dynamic range from the threshold of hearing p = 20 μPa to the threshold of pain p = 200 Pa, i.e. 7 digit places is covered by an extent of 140 dB. The level of acoustic pressure is determined from the relation:

$$L_p = 20 \log \cdot \frac{p}{p_o} \quad \text{(dB)} \tag{11}$$

where: p - acoustic pressure (Pa) - the lowest value of acoustic pressure, the threshold of hearing.
$p_0 = 2.10^{-5}$

From the aspect of the dynamic extent, the audible band is in a zone from the threshold of audibility (corresponding to the level of acoustic pressure 0 dB) to the threshold of hurtfulness, i.e. higher than 130 dB.

The threshold of impermissibility, where irreparable damage to hearing occurs (deafness), is at the level of acoustic pressure 140 dB (p = 200 Pa). At present, allowable levels of noise are dealt with by the CR Government decree No. 148/2006 Gaz., which is concerned with "Acoustics – noise in the working environment". According to these statutory rules a duty results not to exceed an admissible noise level of L_p = 85 dB in woodworking plants.

RESULTS AND DISCUSSION

Based on the primary analysis of the time response of vibrations and the measurement of noisiness an apparent relationship was found between the excessive vibration of the disc body and noisiness. In the area of n = 3200 and 3400 rpm, an apparent peak increase was noted of the disc noise signal by 5 to 7 dB above the standard noise level. An absolute increase reached a level of Lp = 99 dB. These conditions were gone with by unpleasant whistling (see Fig. 7).

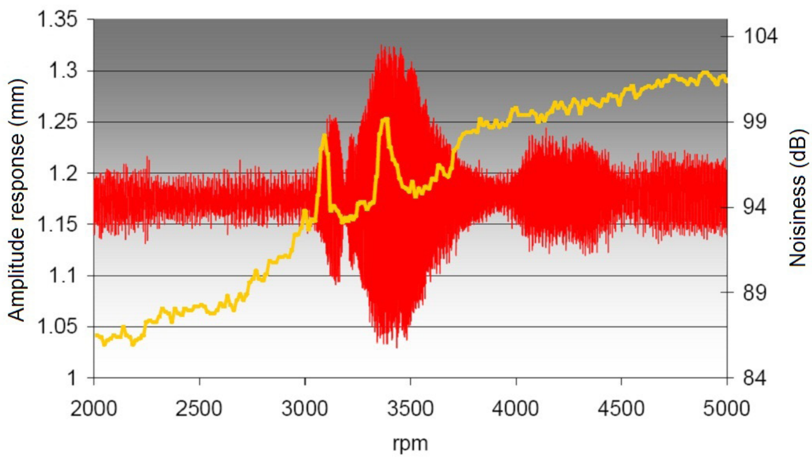


Fig. 7: Course of vibrations and noisiness at the circular sawblade run – idle run.

Increased noise level at the circular sawblade run – idle run were acknowledged by authors from University of Zagreb (Goglia and Lucic 1999). Through the amplitude spectrum analysis, information was obtained on dominant frequencies and amplitudes of the disc vibrations causing increased noisiness. FFT has been used at determination of critical rpm sawblades (Orlowski 2005). In the first stage, peak values caused by the disc dilatation grooves were filtered off from the measured signal by means of a median filter in the MATLAB 7.5 environment (Fig. 8).

In this case (Fig. 9), the disc frequency f_n = 1.33 Hz is a dominant frequency. At this frequency of turning the circular sawblade (n = 80 rpm), it refers only to static eccentric running the circular sawblade disc. Higher harmonic frequencies follow, which are a multiple of the frequency of turning. Then, the sum of all amplitudes at the dominant frequency and higher harmonic frequencies gives the disc deformation 0.0375 mm from the centre position. Thus, the total eccentric running of the sawblade disc is A = 0.075 mm. This value fulfils the manufacturer standards allowing a maximum static eccentric running the disc body up to 0.1 mm.

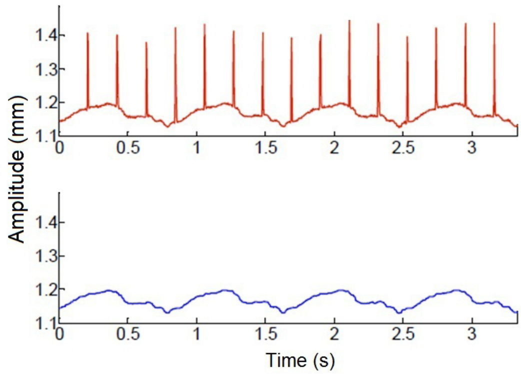


Fig. 8: Filtration of the time response of a measured signal.

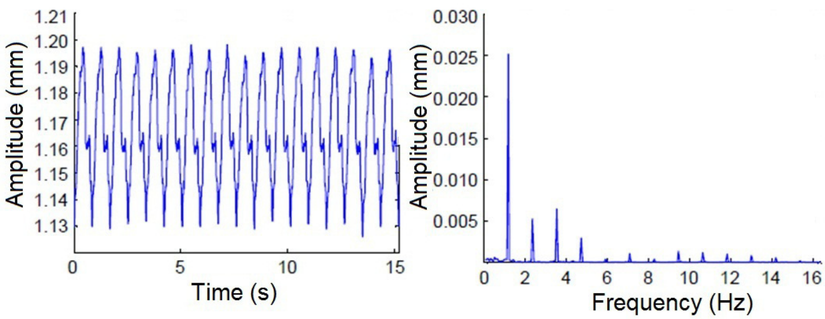


Fig. 9: Time response and the frequency spectrum of vibrations at 80 rpm.

Fig. 10 shows the time response of the sawblade disc vibrations and the frequency spectrum in resonance speed of the disc n_r = 3 400 rpm. Thus, the maximum amplitude of the disc peak value increased more than 5 times to a value of A = 0.39 mm. The disc amplitude 0.17 mm occurs at a dominant frequency of 55 Hz. This point corresponds roughly to the speed frequency f_n = 56.6 Hz at 3400 rpm and, at the same time, to the frequency of a backwards-running wave of the disc vibration f_z = 54 Hz.

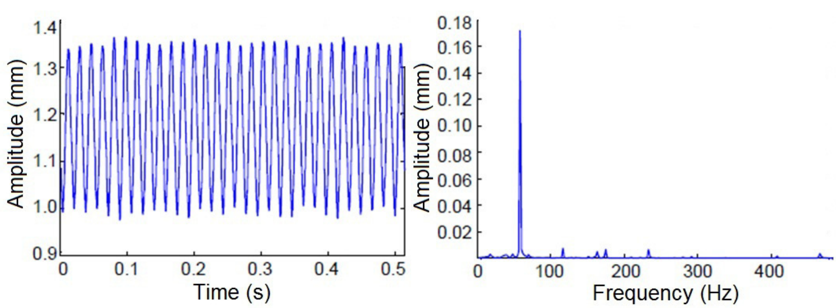


Fig. 10: Time response and the frequency spectrum of vibrations at the disc resonance speed.

The frequency of the backwards-running wave of the disc vibration at 3400 rpm was determined in the course of previous experiments in cooperation with TU Zvolen (Kopecký et al. 2007). It has been also found that resonance rpm and frequencies f_{01}, f_{02}, f_{03} of the 1^{st} – 3^{rd} nodal diameters occur in this zone, namely in the very narrow band of speed 3300 to 3420 rpm.

Similar results were also obtained in research carried out abroad where the noise level reduction was noted at similarly modified circular sawblades with irregular tooth pitch, namely by 3 to 8 dB (Svoreň 2006). These findings correspond well with data mentioned in brochures of some manufactures of circular sawblades (Leitz, Freud).

CONCLUSION

Existing results validate relationships between increased vibrations in resonance speeds n_r 3200, 3400 rpm and the disc whistling (see Fig. 7). The disc whistled intensely and allowable values of noise were exceeded even by Lp = 15 dB. By means of Fourier transformation of the measured signal of the time response of vibrations, it was possible to determine rather exactly dominant frequencies when increased vibrations occurred. These areas usually correspond to reaching the resonance rpm for certain nodal diameters of the shape of vibrations being consistent with the frequency of a backwards-running wave of the disc body vibration at given rpm.

Thus, for the tested Pilana 72 TFZL disc, it is possible to recommend operational speed without increased vibrations in the area of n_{p1} = 3800 rpm (v_c = 79.6 $m.s^{-1}$) at the level of acoustic pressure Lp = 99 dB. In this case, the admissible level of noisiness is exceeded by 14 dB. Through the conversion to acoustic pressure according to equation (11), it refers to the fivefold increasing the acoustic pressure from p = 0.36 Pa (85 dB) to p = 1.79 Pa (99 dB). At sawing, it is possible to expect another increasing the acoustic pressure on average by 3 to 5 dB.

From the point of view of the maximum admissible noisiness in the workplace 85 dB, it is possible to select even lower working rpm in the area of n_{p2} = 2900 rpm (v_c = 60.7 $m.s^{-1}$). In this case, problems concerning the disc noisiness are lower and the noise level reaches a value of Lp = 92 dB. However, also in this case, it refers to more than double exceeding the admissible limit of acoustic pressure. Under these conditions, the circular sawblade will work at the lower limit of cutting conditions for trimming to size of agglomerated materials and its use is, therefore, controversial.

At the conclusion, it is necessary to note that the tested disc surpasses admissible hygienic standards in the workplace. Therefore, it is quite necessary workers to use ear protectors. Potential ways to improve the condition can be found both in the construction of the circular sawblade (e.g., dilatation grooves provided with copper rivets, another shape and placing the noise-elimination grooves, changes in reinforcing the disc by rolling etc.) and in anti-noise measures of the sawmill.

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