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NEW DESIGN OF CIRCULAR SAW BLADE BODY AND ITS INFLUENCE ON CRITICAL ROTATIONAL SPEED

Oscillation is an unwelcome state as it has adverse effects not only on the work piece but also on the tool. This paper deals with the problem of circular saw blade oscillation and the effect of modification of the circular saw blade body on the natural frequency as well as on the critical rotational speed. The calculation and experimental investigation of the natural frequencies and node shapes on two types of circular saw blades were carried out. Two methods used to determine the natural frequencies are presented: the first, modal analysis (FEM), a theoretical method, and the second, experimental measurement. The results of the investigation were used to compare the FEM and experimental methods and to show which modification achieved a higher critical rotational speed. Both methods were conducted on two circular saw blades for continual cutting with 36 teeth and slots.

Keywords: circular saw blade, critical rotational speed, modal analysis, natural frequencies, slots

Introduction

Inaccurate cutting, low surface quality and high noise level during sawing are the main problems of cutting with circular saws. These adverse effects are related to the oscillation of the circular saw blade.

Strict requirements have been imposed strict for cutting with circular saw blades, such as high surface quality , straight cutting, reduced noise level, etc. To achieve these requirements it is necessary to use a high rotational speed. However, there is then a problem with the critical rotational speed and instability of the circular saw blade. During the cutting process, oscillation occurs in the circular saw blade which may lead to its instability. A reduction in the amplitude of the oscillation is essential in order to reduce the adverse effects.

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Slots are used by some manufactures to reduce oscillation. Similar slots for the reduction of the amplitude were presented for example by Kanefusa (Japan) [2014] a few years ago, and now this manufacturer produces circular saw blades, known as Yield Pro. The Italian company Freud [2014] produces discs with various slots and these are termed “noiseless”.

The phenomenon of oscillation has been the subject of many scientific publications. Determination of the oscillation of circular saw blades using experimental methods has been carried out by Mote [1965], Stakhiev [1970, 1998, 2000], Schajer, Mote [1983], Schajer [1992], Holøyen [1987], Hutton [1991], Yu, Mote [1987], Svoreň [1986], Nishio, Marui [1996], Orlowski et al. [2007] and Veselý et al. [2012].

Application of the finite element method to solve the oscillation of saw blades was the research area of Gogu [1988], Leopold, Münz [1992], Michna, Svoreň [2007], Ekevad et al. [2009], Cristóvão et al. [2012] and Droba et al. [2013].

In order to determine the resonant and critical rotational speed a theory of oscillation is used which says that the oscillation of a circular saw blade is the superposition of two moving waves travelling in opposite directions to each other (a forward- and backward-travelling wave). The frequency of these waves can be expressed as follows [Stakhiev 1970]:

Forward-travelling wave:

$$f_1 = f_{dyn(n)} + \frac{k \cdot n}{60} \text{ [Hz]} \tag{1}$$

Backward-travelling wave:

$$f_2 = f_{dyn(n)} - \frac{k \cdot n}{60} \text{ [Hz]} \tag{2}$$

Where: $f_{dyn\ (n)}$ – frequency of rotating circular saw blade in Hz,
f_1 – frequency of forward-travelling wave in Hz,
f_2 – frequency of backward-travelling wave in Hz,
k – number of nodal diameters,
n – rotational speed in rpm.

In the case that the rotational speed of the circular-saw blade increases, the frequency of the backward-travelling wave at a certain rotational speed (besides the nodal diameters k = 0 and 1) becomes zero. This working speed is called the “critical rotational speed”. At this critical rotational speed, the angular speed of the circular-saw blade is equal to the speed of the wave in the circular-saw blade and the backward-travelling wave appears as if it had stopped in space. This is a resonance point where even a small lateral force will cause a large lateral deflection of the circular-saw blade [Stakhiev 1970], therefore, from equation (2) it is possible to derive the following equation:

$$n_k = \frac{60 \cdot f_{dyn(n)}}{n} \text{ [rpm]} \tag{3}$$

As a result of centrifugal force, the natural frequency of the rotating circular saw blade increases parabolically with the increasing operating speed. The relationship between the natural frequency of the rotating circular saw blade and the rotation speed is expressed in the equation:

$$f_{dyn(n)}^2 = f_{stat}^2 + \lambda \cdot \left(\frac{n}{60}\right)^2 \text{ [Hz]} \tag{4}$$

Where: f_{stat} – natural frequency of non-rotating circular saw blade in Hz,
λ – coefficient of centrifugal force.

This critical rotational speed can be expressed by substituting equation (4) with (3). Then we receive the following form (5) of equation [Nishio, Marui 1996]:

$$n_k = \frac{60 \cdot f_{stat}}{\sqrt{k^2 - \lambda}} \text{ [rpm]} \tag{5}$$

The aim of this experiment was to find a design solution, which could increase the dynamic stiffness of the saw disc without increasing its thickness. The dynamic stiffness would be evaluated indirectly, i.e. determination of the critical rotational speed, when saw blade disc will lose stability.

Materials and methods

Two methods were used to determine the natural frequencies of the circular saw blades:
- experimental measuring,
- FEM.

The tested subjects were two circular saw blades Ø 350 × 2.4/36 with tooth height h = 13 mm. These circular saw blades have variable slots cut in their body as shown in (fig. 1ab). Both the circular saw blades have a non-uniform pitch, the CB1 repeating after 1/18 of a circle and CB2 repeating after 1/6 of a circle.

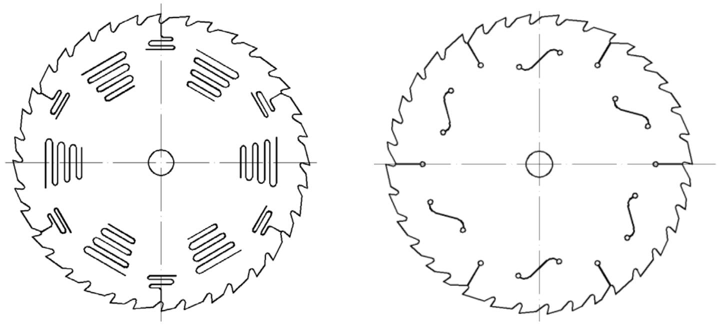


Fig. 1. Construction of circular saw blades used for experimental measuring

Experimental measuring

The natural frequencies f_{stat} of the non-rotating circular-saw blade and frequencies f_2 of the backward-travelling waves were measured experimentally for k = 1[1], 2, 3, 4 on the measuring stand constructed at the laboratory of the Department of Woodworking Machines and Equipment at the Technical University in Zvolen. Clamping collars were used with a diameter d_u = 110 mm (clamping ratio α = 0.314). A diagram showing the connecting instruments is shown in (fig. 2).

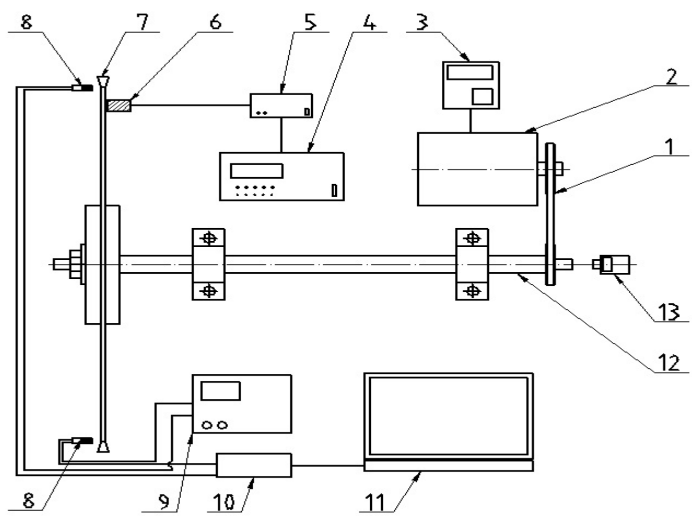


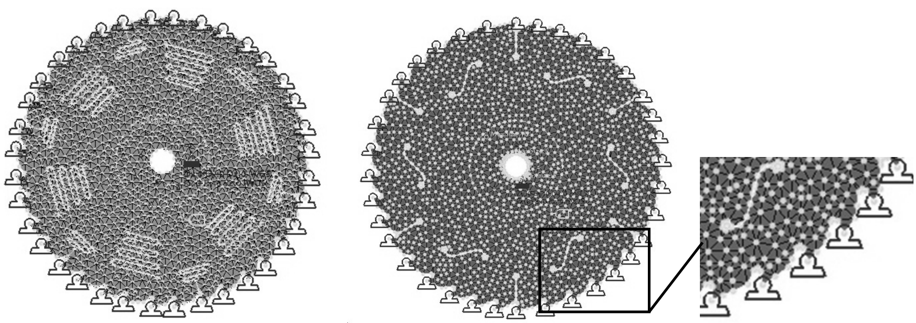
Fig. 2. Diagram of equipment for experimental measuring: 1 – belt drive, 2 – electric motor, 3 – frequency converter, 4 – tone generator, 5 – amplifier, 6 – electromagnetic (solenoid) driver, 7 – circular saw blade mountedwith clamping collars, 8 – oscillation sensor , 9 – electric source 10 – digital oscilloscope, 11 – PC, 12 – spindle in the bearings, 13 – non-contact tachometer

[1] The critical rotational speed does not exist for k = 1.

FEM

For the theoretical experiment, Pro/Engineer WF4 software was used which enabled a simulation of static and dynamic modal analysis. This software used the FEM (finite element method). Modal analyses were created of the models of the circular saw blades. These models were created according of real circular saw blades (fig. 1a, 1b).

Both models of the circular saw blades were clamped using function constraint displacement with a diameter of 110 mm. This absolutely rigid area was as clamping collars used. This simplification was conducted on both saw blades, where the model was defined as a thin shell disc. The mesh for analysis was created using shell elements with a maximum size of 5 mm (fig. 3). The zoom on the right side shows the mesh in more detail and the interconnection of finite elements; the symbols round the saw disc () represent the mass of the cutting plate (in this case, from tungsten carbide) which is centralized to pinpoint of saw disc tooth. This type of mesh was selected because of the higher accuracy of the calculated results.



CB1 **CB2**

Fig. 3. Mesh used for modal analysis for CB1 and CB2

The calculated values of the natural frequencies and graphical results of the displacement of the circular saw blades for nodal diameter k = 1, 2, 3, 4 were obtained from the modal analysis. In this research, the most important the values are k = 3.

Results and discussion

In this paper it is shown that the relevant values of the natural frequencies were determined by using both the experimental and FEM methods. The values of the natural frequencies of cosine and sine components of the split modes [Yu, Mote 1987] are shown in the tables; in table 2 from the experiment and in table 3 from the modal analysis.

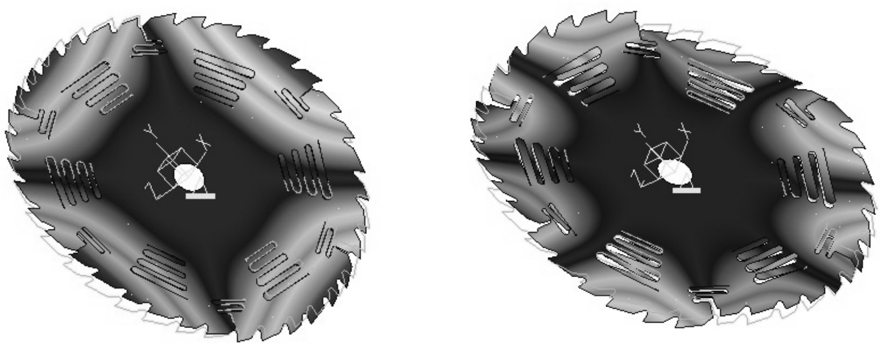
Table 2. Experimentally measured values of natural frequencies of CB1 and CB2

k	1		2		3		4		Unit
	cosine	sine	cosine	sine	cosine	sine	cosine	sine	
CB1	101	101	124	126	182	188	276	280	*f* [Hz]
CB2	106	106	142.7	142.8	235	240.5	381.1	381.15	

Table 3. Calculated values of static natural frequencies of CB1 and CB2 from modal analysis

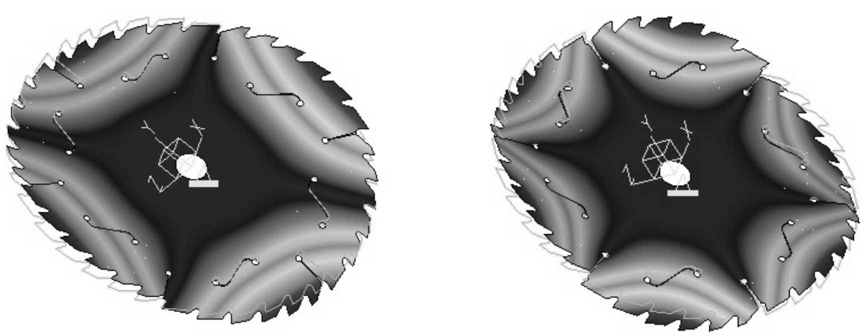
k	1		2		3		4		Unit
	cosine	sine	cosine	sine	cosine	sine	cosine	sine	
CB1	123.7	123.7	143	143	193.2	200.4	280.6	280.7	*f* [Hz]
CB2	135.5	135.5	157.4	157.4	224.8	236.2	347.1	347.1	

The graphical results of the modes of CB1 and CB2 from the modal analysis are shown in fig. 4 and 5.



***k* = 2** ***k* = 4**

Fig. 4. Results of deformed CB1 from modal analysis for *k* = 2 and 4



***k* = 2** ***k* = 3**

Fig. 5. Results of deformed CB2 from modal analysis for *k* = 2 and 3

As shown (fig. 6), the values of the natural frequencies of CB1 for $k = 1, 2, 3, 4$ obtained from FEM were a little higher than the values measured experimentally. However, for CB2 (fig. 7) the values of the natural frequencies obtained from FEM were higher only for $k = 1$ and 2. For $k = 3$, they are relatively similar, and for $k = 4$, the experimental values were higher than from FEM. This phenomenon was observed on some other circular saw blades with a similar shape of slots to those on CB2 [Ekevad et al. 2009].

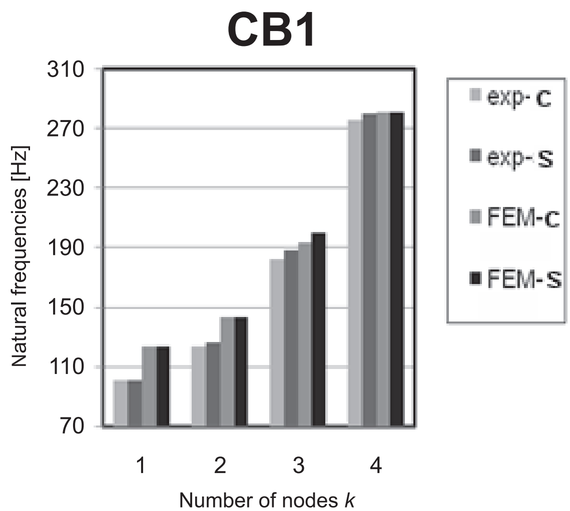


Fig. 6. Comparison of natural frequencies obtained from experimental measurement and modal analysis for CB1

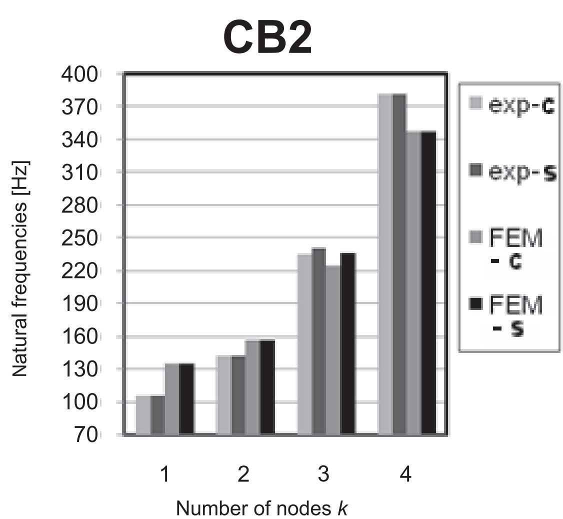


Fig. 7. Comparison of natural frequencies obtained from experimental measurement and modal analysis for CB2

Table 4 shows the average values from the experimentally measured values of the centrifugal coefficient of each circular saw, which were used to calculate the critical rotational speed of both circular saw blades. Table 5 shows the centrifugal coefficient of each model of circular saw blade, which were used to calculate the critical rotational speed of both models of circular saw blade.

The values of coefficient λ (table 4) were computed using formula (4) from the experimentally measured values of the natural frequencies f_{stat} and f_2.

The values of coefficient λ (table 5) were computed using formula (4) from the natural frequencies f_{stat} and $f_{dyn(n)}$ obtained by FEM.

Table 4. Calculated values of coefficient of centrifugal force and critical rotational speed from experimentally measured values of natural frequencies

k	2			3			4		
Type	CB1	CB2		CB1	CB2		CB1	CB2	
Component of the split mode	sine	cosine	sine	sine	cosine	sine	sine	cosine	sine
λ	2.32	1.91	1.91	2.62	2.44	2.68	3.69	3.79	3.79
n_k [rpm]	5740	5928	5928	4323	5506	5733	4720	6544	6544

Table 5. Calculated values of coefficient of centrifugal force and critical rotational speed from natural frequencies obtained by FEM

k	2		3		4	
Type	CB1	CB2	CB1	CB2	CB1	CB2
Component of the split mode	cosine	cosine	cosine	cosine	cosine	cosine
λ	2.07	2.13	2.51	2.87	3.12	3.78
n_k [rpm]	6171.5	6913.8	4550.3	5449.4	4692	5958.7

The influence of the design of the circular saw blade body on the critical rotational speed is shown in fig. 8. There were only small differences (approx. 3%) for *k* = 2 between the critical rotational speed of CB1 and CB2. But for *k* = 3, the increase in the critical rotational speed was over 27%, and for *k* = 4, it was over 38% for the experimental values. CB2 achieved better values for the critical rotating speed in the experimental measuring.

According to the results obtained from the modal analysis for *k* = 2, CB2 achieved a higher critical rotational speed (by 12%), for *k* = 3 it was higher by 19.8%, and for *k* = 4, it was 27% higher than CB1 (fig. 9).

Therefore, it could be said that the types of slots cut in CB2 have a extremely positive effect on increasing the critical rotational speed of the circular saw blades, and it is essential to know the position, shape, and thickness, etc. of the slots cut into the body of the circular saw blade.

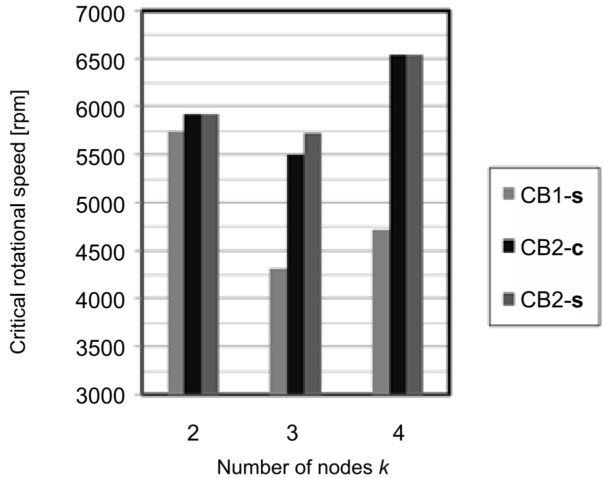


Fig. 8. Comparison of critical rotational speed calculated from experimental measured values of natural frequencies CB1 and CB2

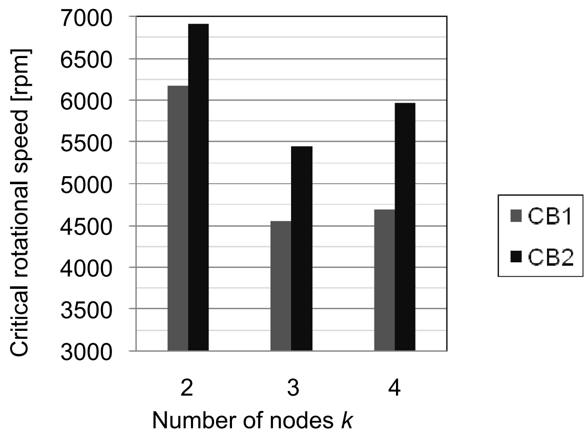


Fig. 9. Comparison of critical rotational speed calculated from natural frequencies of CB1 and CB2 obtained by FEM

Conclusions

On the basis of the experiments conducted, it could be stated that:

1. Both the methods applied confirmed the positive influence of the saw disc body's design on its dynamic stiffness, which was evaluated indirectly (figs. 8, 9). On average, the increase in the critical rotational speed detected in the experiment for $k = 2, 3, 4$ was 1140 rpm; using FEM (for the same "k", i.e. for $k = 2, 3, 4$) it was 969 rpm. The increase in critical rotational speed was in every case, i.e. for all monitored modal shapes.

2. The good conformity of both results can be positive impulse for manufacturers to use of available software that can relatively exactly simulates the behaviour of saw disc. Using software saves the time and money of manufacturers for the production of circular saw blades and their testing.

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