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Optimization of Circular Saw Blade Stress Fields Using Numerical Methods of Mechanics

FRANKOVSKÝ Peter[1,a], KICKO Michal[1,b], BOCKO Jozef[1,c], DELYOVÁ Ingrid[1,d] and KOSTAK Ján[1,e]

[1]Technical University of Košice, Faculty of Mechanical Engineering, Letná 9, 042 00 Košice, Slovak Republic

[a]peter.frankovsky@tuke.sk, [b]michal.kicko@tuke.sk, [c]josef.bocko@tuke.sk,[d]ingrid.delyova@tuke.cz, [e]jan.kostka@tuke.sk

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Abstract. The submitted article deals with the design of the cutting saw blade in terms of the location of the anti-vibration grooves on the surface of the disk. The position of the anti-vibration grooves affects vibrations, noise and stress distribution in the disk. Our concern is to investigate the best distribution of anti-vibration grooves to reduce the maximum stress in the critical parts of the cutting blade.
The article presents the numerical stress analysis of the saw blade without and with anti-vibration grooves for their suitable deployment in terms of reducing the stress around the teeth during their cutting process. A suitable layout of the anti-vibration grooves has a significant effect on the reduction of stress around the teeth. The individual steps of numerical analysis are detailly described in the article.

Introduction

Cutting blades whose geometry is provided with different grooves to reduce noise, vibration and stress distribution is at present an increasingly frequent phenomenon, especially in the area of profi-tools. This was one of the main impulses for making this contribution. As it is generally believed that such grooves have in particular, acoustic optimization effects, so they are primarily used to reduce noise. This contribution is also evidenced by their further optimization property, namely the reduction of stress fields caused by centrifugal forces but mainly the reduction of stresses in the area of cutting teeth.

Anti – vibration grooves

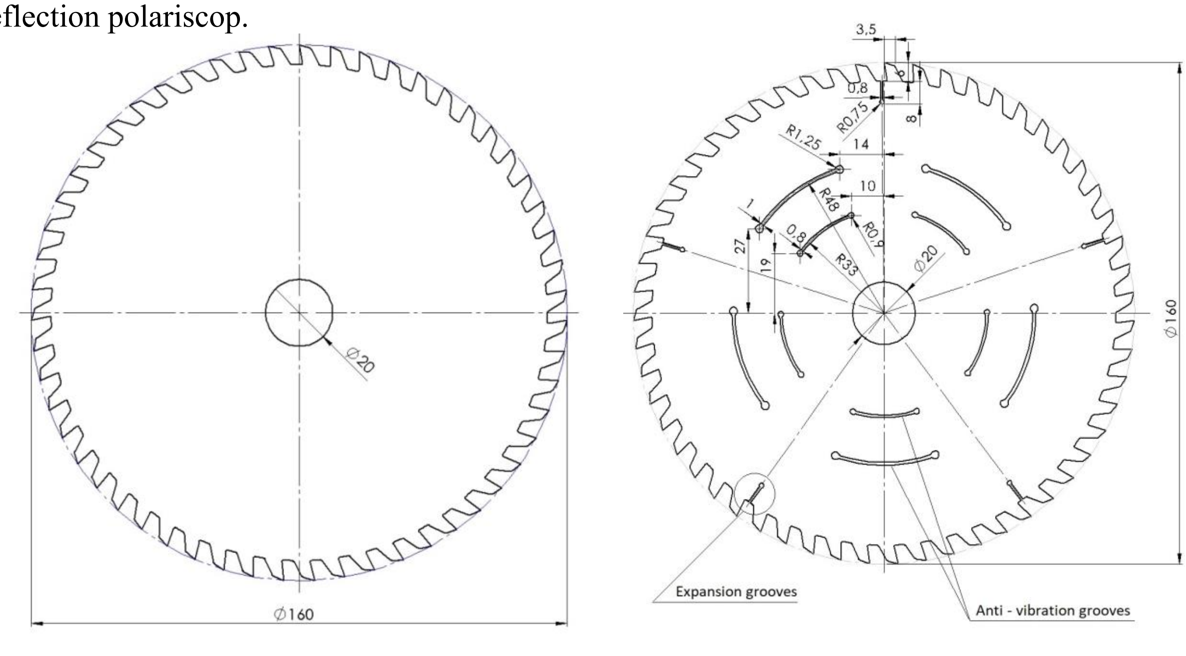
To reduce vibrations caused by centrifugal forces and unbalanced weight distribution, the cutting blades are provided with narrow grooves located in the region between the bearing and the teeth of the disc. Many manufacturers of circular saw blades are compensated with this problem by making holes, usually located under expansion grooves. These openings usually begin and end with rounding, with the laser cut in the shape of the letter "S" (Fig. 1). Laser cutting is essential for dimensional accuracy, as even small geometry inaccuracies can cause great vibration. Thus these holes help disperse vibrations. In some cases, vibration holes are filled with a soft material such as brass or copper, which helps absorb vibrations. Anti-vibration saw holes are designed to prevent unwanted effects and to perform smooth cuts without the use of stabilizers. Anti-vibration grooves also act as noise reduction elements.

Fig. 1 Anti-vibration grooves

Design of the shape and dimensions of the circular saw blade

Fig. 2a shows a conventional wood cutting disc with a hole for bearing 20 mm in diameter and an outer diameter of 160 mm. By creating slides in the areas below the cutting teeth that will create adjacent grooves effect, it will be assumed that the stress drops primarily in the area of the cutting parts.

The proposed shape of the disc, which is shown in FIG. 2b will be made using laser cutting technology, with standardized steel 75Cr1 (DIN 1.2003), (STN 19418). In Fig. 2b two types of optimization elements are shown. The first is the five expansion grooves around the circuit of the disk that serve as a function of protecting the teeth (segments) against unwanted shape change or the tearing of the teeth when the workpiece is pierced during cutting. The other elements are anti-vibration grooves that fill the role of adjacent grooves effect to optimally distribute the principal stresses during cutting. If this design is certified numerically, it will be used to manufacture it from photosensitive material of PS-1A. Thus the produced specimen will then be subjected to experimental analysis by reflective photoelasticity using the LF / Z-2 reflection polariscop.



a) b)

Fig. 2 The shape and dimensions of the saw blades: a) conventional; b) with anti-vibration grooves

Numerical stress analysis

The shape and dimensions of the proposed anti-vibration grooves were analyzed and evaluated numerically in the Abaqus program. The first step was to simulate a conventional disk without optimizing grooves, and then calculate the stress fields in the optimized design of the cutting saw blade (Figure 2b). For the efficient calculation, both samples were created as shell elements [1-6].

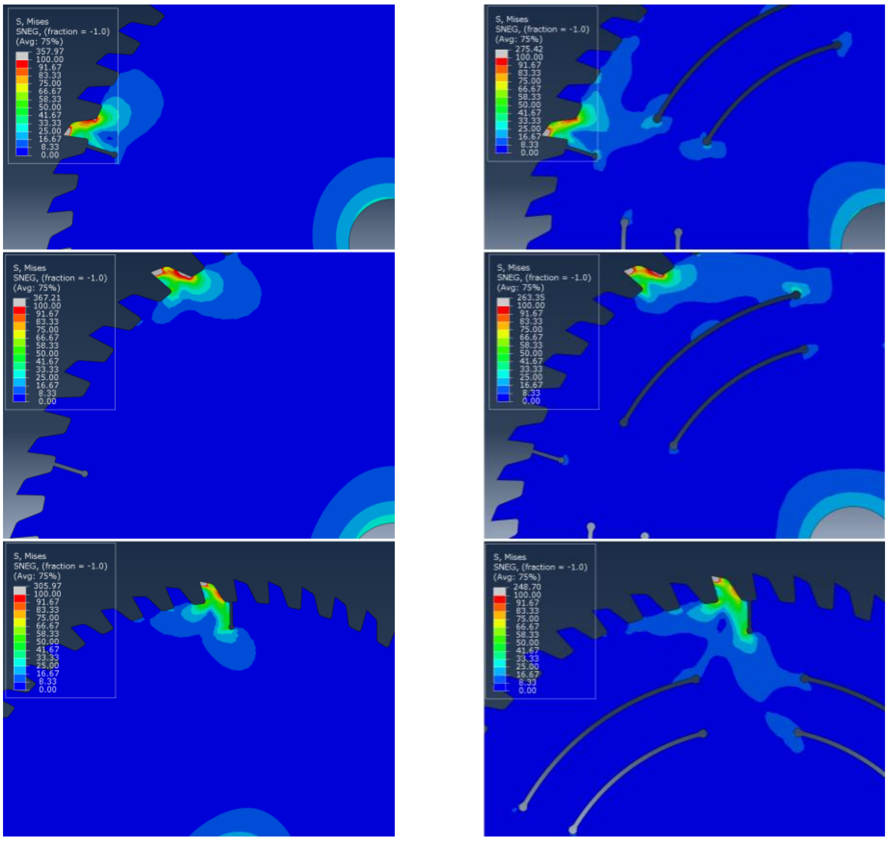
The boundary conditions were chosen as follows: a reference point (RP) was created at the center of the bearing hole, to which the entire internal circumferential edge of the disk was attached (kinematic coupling) to simulate the hinge. In this RP, all degrees of freedom were taken apart excluding the rotation around the Z axis, thus simulating the possibility of movement during cutting. Subsequently, in RP was defined a very low unit of angular rotation for simulating the abovementioned movement of the disc.

The last step in defining boundary conditions was to prevent the movement in one of the teeth, namely by fixing the tip to simulate an engagement of tooth. In this way, the same trio of teeth for both conventional and optimized discs was used to evaluate and compare the results more objectively.

The results of the strength analysis, the fields of reduced stresses obtained with the HMH strength theory, are shown in Fig. 3. The relation for the calculation of the reduced stress according to the theory of strength HMH has the form

$$\sigma_{red}^{HMH} = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \left(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_1\sigma_3\right)}. \tag{1}$$

For better representation, the upper stress limit was set to 100 MPa in all cases.



a) b)

Fig. 3 Comparison of stress fields for: a) Conventional disc; b) an optimized disc

The strength analysis results for all three of the above cases confirmed a significant reduction of maximum stress due to the application of optimization grooves within the conventional disk. In the first tooth we recorded a decrease of the maximum stress by 81 MPa, in the region of the second considered tooth by 104 MPa and in the last case by 57 MPa.

Modal analysis

By analyzing of eigenshapes, it has been shown that not only optimal distribution of stresses occurs but also the reduction of the noise of the cutting disc, that can be noticed by comparing the results according to the simulations shown in Fig. 4 for a conventional disc and Fig. 5 for optimized one.

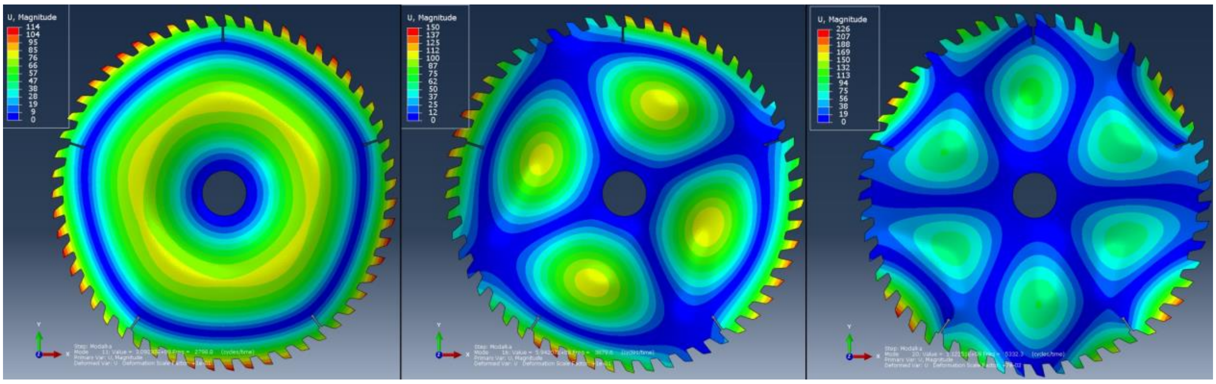


Fig. 4 Mode shapes of conventional disc

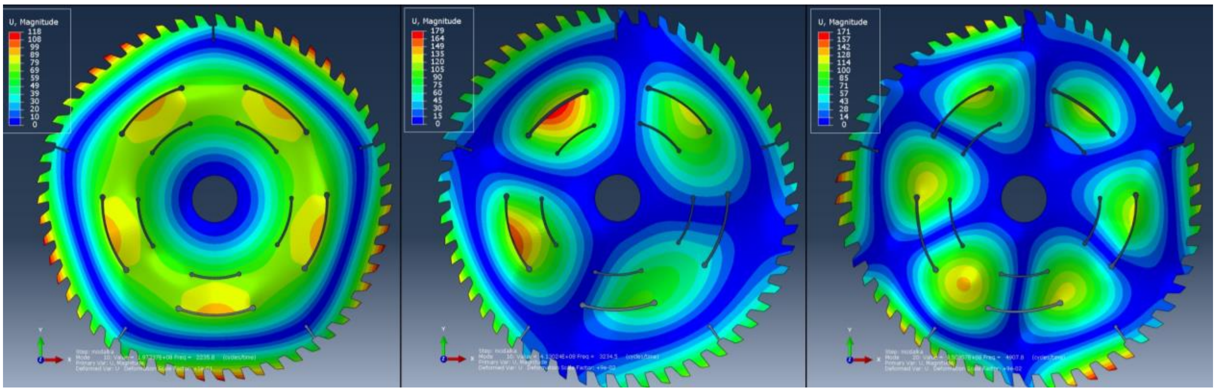


Fig. 5 Mode shapes of disc with anti – vibration grooves

By comparing the results of the modal analysis in Tab. 1, it is obvious that the oscilation frequency magnitudes of the disc with the anti-vibration grooves are at a relatively equal RPM significantly smaller than in the disc with the constant cross-section.

Tab. 1 Oscilation frequencies: a) conventional disc; b) optimized disc

Frame					
Index	**Description**				
0	Increment	0: Base State			
1	Mode	1: Value = 6.48817E+06	Freq =	405.40	(cycles/time)
2	Mode	2: Value = 6.49017E+06	Freq =	405.46	(cycles/time)
3	Mode	3: Value = 8.71841E+06	Freq =	469.94	(cycles/time)
4	Mode	4: Value = 1.38924E+07	Freq =	593.21	(cycles/time)
5	Mode	5: Value = 1.38941E+07	Freq =	593.25	(cycles/time)
6	Mode	6: Value = 6.01212E+07	Freq =	1234.1	(cycles/time)
7	Mode	7: Value = 6.01251E+07	Freq =	1234.1	(cycles/time)
8	Mode	8: Value = 1.73387E+08	Freq =	2095.7	(cycles/time)
9	Mode	9: Value = 1.73423E+08	Freq =	2095.9	(cycles/time)
10	Mode	10: Value = 2.49853E+08	Freq =	2515.7	(cycles/time)
11	Mode	11: Value = 3.09237E+08	Freq =	2798.8	(cycles/time)
12	Mode	12: Value = 3.62943E+08	Freq =	3032.1	(cycles/time)
13	Mode	13: Value = 3.63036E+08	Freq =	3032.5	(cycles/time)
14	Mode	14: Value = 3.81131E+08	Freq =	3107.1	(cycles/time)
15	Mode	15: Value = 3.88107E+08	Freq =	3135.4	(cycles/time)
16	Mode	16: Value = 5.94207E+08	Freq =	3879.6	(cycles/time)
17	Mode	17: Value = 5.94328E+08	Freq =	3880.0	(cycles/time)
18	Mode	18: Value = 7.21178E+08	Freq =	4274.1	(cycles/time)
19	Mode	19: Value = 7.21299E+08	Freq =	4274.4	(cycles/time)
20	Mode	20: Value = 1.12251E+09	Freq =	5332.3	(cycles/time)

Frame					
Index	**Description**				
0	Increment	0: Base State			
1	Mode	1: Value = 5.76379E+06	Freq =	382.10	(cycles/time)
2	Mode	2: Value = 5.76630E+06	Freq =	382.18	(cycles/time)
3	Mode	3: Value = 7.65097E+06	Freq =	440.23	(cycles/time)
4	Mode	4: Value = 1.28611E+07	Freq =	570.77	(cycles/time)
5	Mode	5: Value = 1.28623E+07	Freq =	570.79	(cycles/time)
6	Mode	6: Value = 5.64858E+07	Freq =	1196.2	(cycles/time)
7	Mode	7: Value = 5.64927E+07	Freq =	1196.2	(cycles/time)
8	Mode	8: Value = 1.62179E+08	Freq =	2026.8	(cycles/time)
9	Mode	9: Value = 1.62212E+08	Freq =	2027.0	(cycles/time)
10	Mode	10: Value = 1.97337E+08	Freq =	2235.8	(cycles/time)
11	Mode	11: Value = 2.25614E+08	Freq =	2390.6	(cycles/time)
12	Mode	12: Value = 2.43671E+08	Freq =	2484.4	(cycles/time)
13	Mode	13: Value = 2.43798E+08	Freq =	2485.1	(cycles/time)
14	Mode	14: Value = 3.68532E+08	Freq =	3055.3	(cycles/time)
15	Mode	15: Value = 3.72468E+08	Freq =	3071.6	(cycles/time)
16	Mode	16: Value = 4.13024E+08	Freq =	3234.5	(cycles/time)
17	Mode	17: Value = 4.13152E+08	Freq =	3235.0	(cycles/time)
18	Mode	18: Value = 6.97304E+08	Freq =	4202.7	(cycles/time)
19	Mode	19: Value = 6.97429E+08	Freq =	4203.1	(cycles/time)
20	Mode	20: Value = 9.50907E+08	Freq =	4907.8	(cycles/time)

Thus, we can assert that by creating optimization grooves, stiffness of disc was decreased but on the other hand it has allowed dilation in several directions, thereby reducing the oscillation frequency and thus reducing vibrations and noise.

Conclusion

By quantifying and comparing the fields of the reduced stresses in the strength calculations and the oscillation frequencies in the modal analysis, a suitable layout and shape of the optimization grooves has been demonstrated which significantly affected not only the acoustic but also the stress values when comparing the conventional disc with the optimized [7-10].

Acknowledgment

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