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# DETERMINATION OF NOISE CAUSED BY VENTILATED BRAKE DISC WITH RESPECT TO THE RIB SHAPE AND MATERIAL PROPERTIES USING TAGUCHI METHOD

**Summary**

Ventilated brake discs may have various configurations of ribs and can be manufactured from different materials. In order to improve the performance in extreme exploitation conditions, it is necessary that they heat up and wear as little as possible, and that they have good heat dissipation capacity and generate low noise. To achieve this, optimization of the influential parameters is required. In this study, the optimization and the analysis of the frequency value were made on the basis of the influential parameters, such as brake disc vane shape, density, Young's modulus, and Poisson's coefficient. A numerical investigation was conducted using the ANSYS software package in the MODAL module. In order to better understand which parameter has the greatest influence on the noise formation, the Taguchi method was applied. By applying the Analysis of Variance – ANOVA, the influence of each parameter on frequency, expressed as a percentage, was determined. The obtained results show that the most influential parameter is the shape of the ribs (90.82%), followed by Young's modulus (8.26%) and density (0.89%).

*Key words:* *ventilated brake disc; Taguchi method; optimization; Young's modulus*

## 1. Introduction

During the braking process, brakes will make noise. Brake manufacturers have been trying to eliminate noise as undesirable phenomenon for several decades. This tendency to noise formation, however, should be predicted before the noise starts [1, 2]. The complaints of users about the noise cause great costs every year, and besides that, the dissatisfaction of users is bad advertising. Naturally, because of the noise during braking, users can decide not to use brakes of some manufacturers. The worst side of the braking noise is its effect on human health. Traffic noise is the second greatest threat to the environment, while air pollution is the first [3, 4]. Noise affects sleep, and the lack of sleep can manifest itself in tiredness, bad memory and low creativity as well as weakened psychomotor skills. Research shows that people who live around airports or roads with high-frequency traffic have headaches more often. Braking noise appears in the high-frequency range, typically from 100 Hz to 20 kHz (which is the upper limit of frequency of human hearing), and classic brakes squeal in the range from 1 to 6 kHz, where the human ear is most sensitive. Braking noise and vibrations can be classified in many subcategories [5, 6], Fig. 1. Braking noise can appear at different frequencies for all types of brakes, and each harmonic can deviate, which creates the characteristic sound. The sound appears at explicit frequencies, and does not appear only during braking, but also while driving. This noise appears because of the airflow through the air gap between the disc ribs.

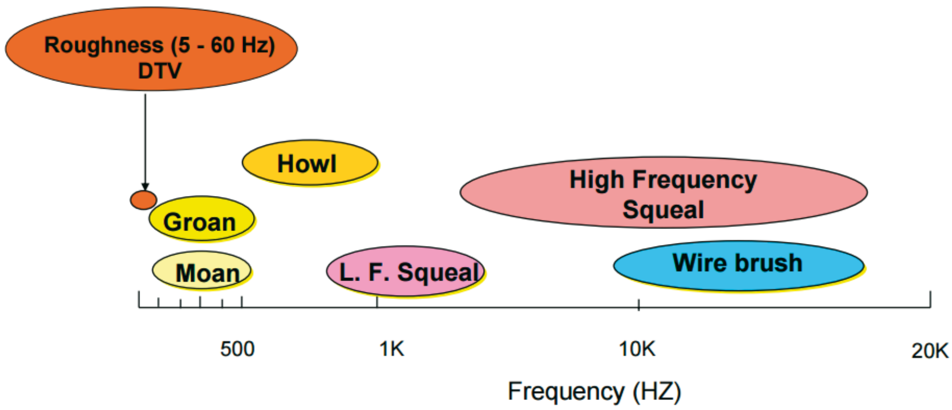



**Fig. 1** Braking noise frequency range and noise types [7]

In the modern computer age, if software including numerical analyses, which would replace experiments, exists [8], it should be used because it can be very useful. It was shown that the error between the numerical analyses and the experiment is within acceptable limits, i.e., it is not over ±2% [9]. Besides that, it is very important to save money during the creation of a new part. It is also important to save time in order to be first on the market. Many factors have influence on the disc brake squeal, e.g., Young's modulus of the back plate, back plate thickness, chamfer, the distance between two slots, slot width and angle of the slot, and they can be investigated by using design of experiments (DOE) techniques.

A predictive mathematical model was formulated on the basis of the influential parameters, and its adequacy for simulation validation was tested experimentally. The predicted results indicate that the braking squeal can be reduced by increasing Young's modulus of the back plate, altering the shape of the friction material by adding chamfers on both sides of the friction material and by introducing slot configurations [10]. It can be said that Young's modulus plays an important role in the noise formation [11]. Besides a great number of studies in the field of braking noise, researchers also consider that it is necessary to find an optimal design for disc brakes in order to minimize the noise [12, 13], as well as to find an optimal size of ribs for ventilated brake discs [14]. So, during the first steps of the new part creation, it is necessary to conduct the optimization of parameters that have influence on noise formation [10, 15]. One of the parameters that influences wear and squeal is surface roughness of the brake disc and brake pads. In addition, geometry of the parts is one of the important parameters that affects their wear [16]. The achieved pressure in the contact between the brake disc and brake pads is not stable during the braking process which causes disc vibrations [17]. Even small changes in pressure, speed or friction coefficient can result in the formation of different types of noise [18]. Besides that, during exploitation brake disc are subject to wear, which can also cause squeal due to the reduction in the brake disc natural frequency [19]. The application of layer materials with a purpose to increase the damping effect of the brake system can be a good solution for the reduction of brake noise. Depending on chosen insulators, the noise can be reduced up to 20 dB [20]. However, the selection of insulators is not simple, because many other factors must be included into the selection, such as the influence of temperature and brake pressure, which will also cause additional research costs. Thus, Triches et al. [20] recommend the application of modal analysis and frequency response measurement. The aim of this paper is to show the influence of the vane shape as well as the material properties on the formation of braking noise in passenger cars. Further, the paper considers which of the observed parameters (braking disc vane shape, Young's modulus and Poisson's coefficient) havs the greatest influence on noise appearance.

The aim of the paper is to show which design of the brake disc is more suitable for implementation in the vehicle for the braking noise to be as low as possible. Also, the paper investigates which properties the brake disc material should have in order to prevent the braking noise, i.e. to find optimal parameters which influence the noise reduction.

## 2. 3D model and experimental setup

The design of a brake disc should satisfy several parameters, such as noise prevention or good cooling. This means that brake manufacturers have a very complicated task.

The task is related to the satisfaction of the following conditions:

- Heat should be dissipated into the environment as much as possible;
- The best possible disc cooling, i.e., the best possible airflow is to be achieved;
- The brake disc reliability even in the extreme exploitation conditions is to be achieved;
- Mass reduction with retention of the material properties of the brake disc is to be achieved;
- Noise appearance is to be prevented and noise is to be reduced as much as possible.

In the paper, three different types of brake disc vane shapes are analysed, Fig. 2. The observed brake disc is a brake disc for the passenger vehicle, category M1. The brake disc was modelled as one part, because brake discs are manufactured in such a way. Dimensions of the brake disc can be found in Table 1, and they correspond to a real part. Table 1 also shows a technical drawing of the brake disc with the geometrical parameters and their values.

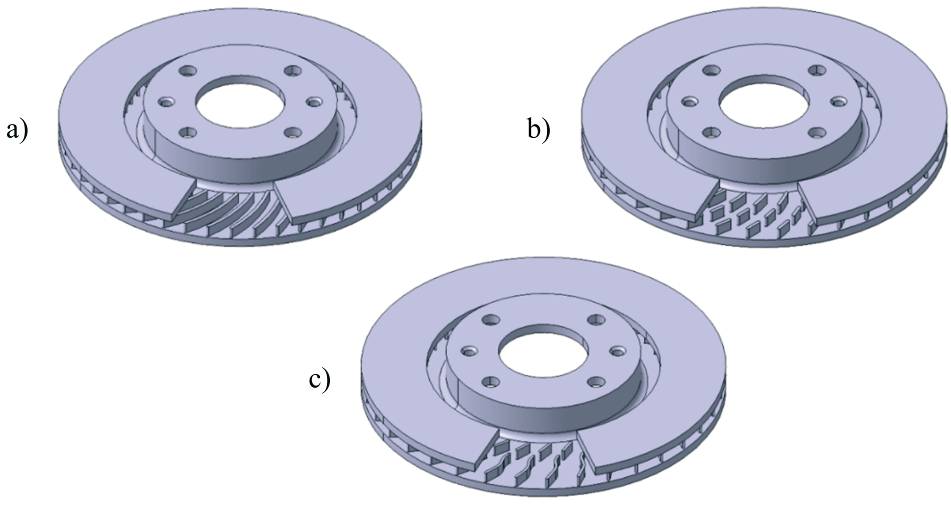
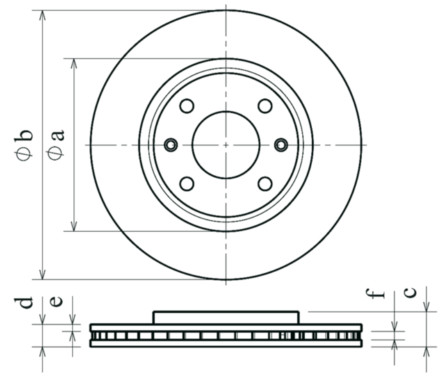



**Fig. 2** Three different ventilated brake discs: a) 48 vane; b) 72 vane, and c) 72 vane flash brake disc

**Table 1** Geometrical parameters of the brake disc




| ***Parameter sign*** | ***Parameters*** | ***Values*** |
|---|---|---|
| *a* | Inner disc diameter, mm | 133.4 |
| *b* | Outer disc diameter, mm | 265.7 |
| *c* | Disc height, mm | 34.3 |
| *d* | Disc thickness, mm | 22.1 |
| *e* | Disc plate thickness, mm | 6.8 |
| *f* | Air gap thickness, mm | 8.3 |

## 3. Defining the finite elements

Three models of the brake disc of the same dimensions were developed as shown in Figure 2 and Table 1. The models differ only in the shape of ribs. The material properties, which are the same in each model, are presented in Table 2. For the modal analysis the ANSYS 14.5 software package, MODAL module, was used. This analysis was based on the determination of the brake disc natural frequency with respect to the shape of the ribs.

**Table 2** Material properties of the brake discs

| ***Type of ventilated brake disc*** | *48 vane brake disc* | *72 flash vane brake disc* | *72 vane brake disc* |
|---|---|---|---|
| **Density, kg/m$^3$** | | 7,600 | |
| **Young's modulus, GPa** | | 197 | |
| **Poisson's ratio, -** | | 0.26 | |
| **Mass, kg** | 5.1455 | 4.9294 | 4.9678 |

Second order tetrahedral finite elements were employed for each disc. The reason for the application of this type of mesh are additional mean nodes which influence the precision of results. The results of the deformations which occur on the characteristic frequencies are more precise. The number of elements and nodes can be seen in Table 3. The mesh quality was checked by applying the orthogonal quality test, Table 4.

**Table 3** Number of elements and nodes

| | *48 vane brake disc* | *72 flash vane brake disc* | *72 vane brake disc* |
|---|---|---|---|
| **Number of nodes** | 60,554 | 89,451 | 88,982 |
| **Number of elements** | 33,541 | 50,082 | 50,019 |

**Table 4** Check of mesh quality by applying orthogonal quality test

| | *48 vane brake disc* | *72 flash vane brake disc* | *72 vane brake disc* |
|---|---|---|---|
| **Perfect zone, %** | 69.32 | 78.07 | 77.12 |
| **Good zone, %** | 28.83 | 21.29 | 22.52 |
| **Acceptable zone, %** | 1.84 | 0.63 | 0.35 |
| **Bad zone, %** | 0.01 | 0.01 | 0.01 |
| **Unacceptable zone, %** | 0 | 0 | 0 |
| **Minimum value, -** | 0.0929 | 0.0585 | 0.1737 |

The mesh quality was checked on all three models of the brake disc. Table 4 shows that in the “perfect zone” there are more than 2/3 of elements, in the “good zone” there are a little less than 1/3 of elements, while in the “acceptable zone” there are 1.84% of elements, 0.63% of elements and 0.35% of elements for the 48 vane, 72 flash vane and 72 vane brake disc, respectively. The neglectable number of elements belongs to the “bad zone”. Since there are no elements in the “unacceptable zone”, i.e., the smallest value is greater than 0.001, it can be said that on the basis of the orthogonal quality test, the mesh quality meets the requirements.

## 4. The representation of frequency with respect to the design of the brake disc ribs

Since the diameter mode shapes are dominant in the squeal formation, only they were analysed [21]. The obtained frequency with respect to the number of nodes can be approximated with smooth curves, as shown in Fig. 3, and can be seen in the study by Barton and Fieldhouse [22]. The obtained results can be accepted completely, as they were proven in earlier studies in which deviations were very small ±2% [9].

The obtained numerical results are shown in the shape of nodal diameters, because they are the most dominant in the squeal formation. The brake disc tends to vibrate around one or more nodal diameters at the same time, which results in the disc to take a wave shape [22], Fig. 4. The number of nodal diameters is based on the number of nodes and antinodes which appear on the surface of the brake disc, i.e., on the surface which is in contact with the brake pad. The number of nodal diameters represents the so-called number of vibration modes – the diametrical mode order, so the noise is a consequence of one or more nodal diameters.

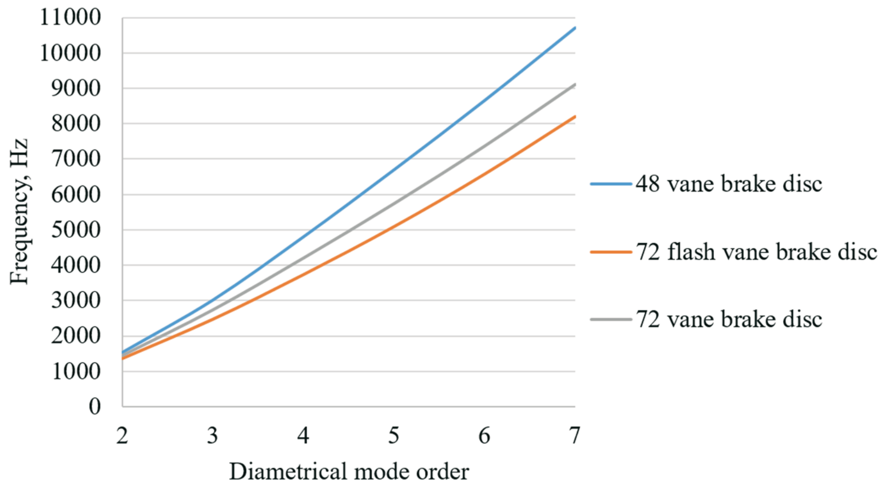



**Fig. 3** Natural frequency of the free-free brake disc with respect to the diametrical mode order

With an increase in the number of nodes, the frequency rises, which directly influences the angle reduction, see Fig. 5. The material properties are the same for all three discs, as shown in Table 2.

The angular node space represents an angle between two adjacent nodes, i.e., between the positive and the negative node, which are next to each other. In the case of two diametrical modes, i.e. in the case of two positive and two negative nodes, the angular node space is determined in the way that the angle of 360° is divided by the sum of the positive and negative nodes, in this specific case by 4, thus it follows that 360°/4=90°. This means, that the third diametrical mode is equivalent to 60°, the fourth diametrical mode is equivalent to 45° and so on, as shown in Fig. 5.

The frequency value was different from disc to disc. The lowest frequencies occurred in the 72 flash vane brake disc, while the highest values were obtained from the 48 vane brake disc, which can be seen by analysing Figs. 3 and 5.

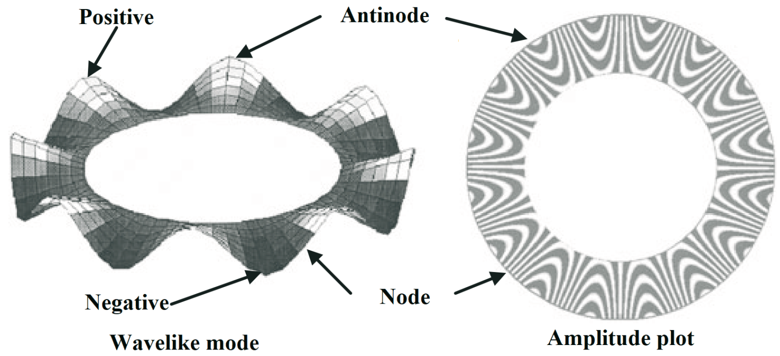



**Fig. 4** Illustration of terms of modal analysis [21]

The frequency value depends on the number of nodes, i.e., with an increase in the number of nodes the frequency rises. Fig. 6 shows wave shapes of the brake disc for each mode. The highest frequencies were obtained for mode 7, and because of this, a further analysis was performed.

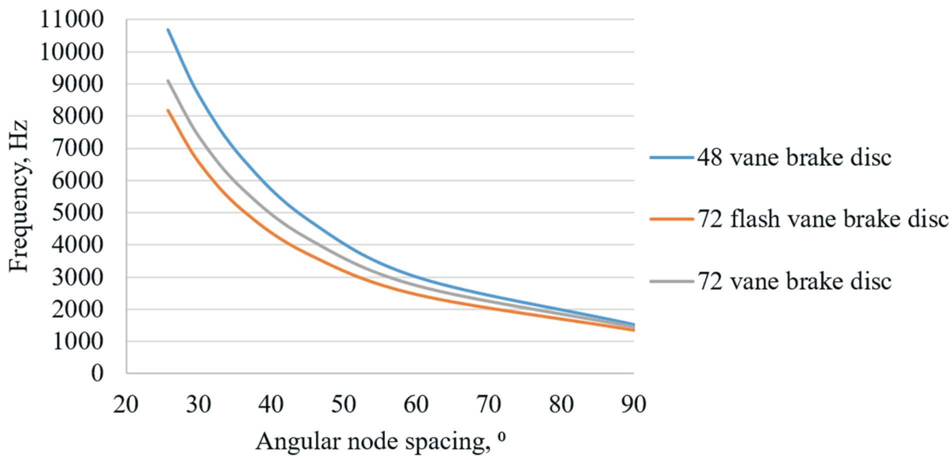



**Fig. 5** Frequency in function of angular node spacing

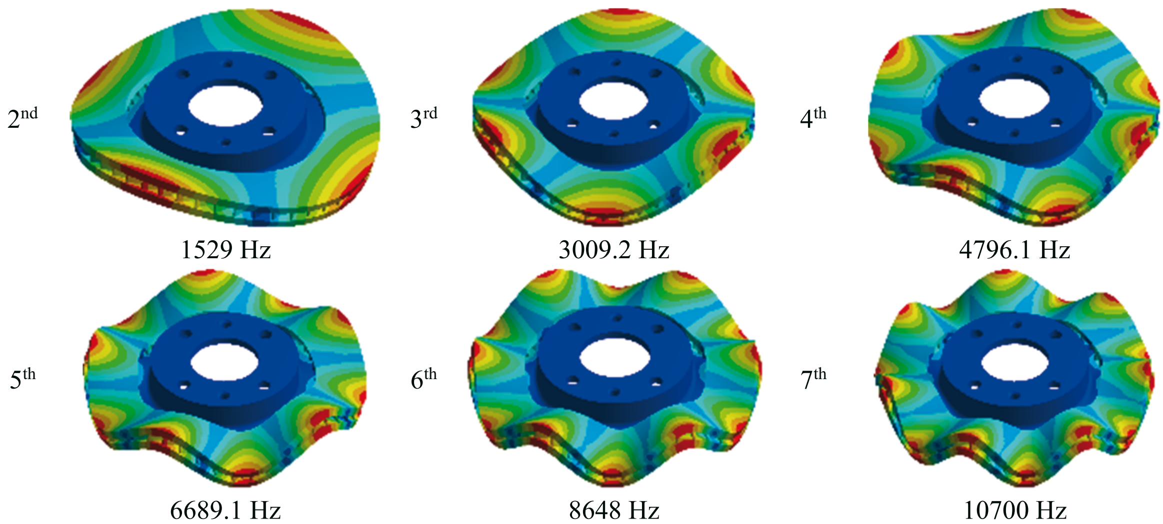



**Fig. 6** Brake disc modes in free-free boundary conditions (48 vane brake disc, according to the material properties from Table 2)

The frequency values for the second diametrical mode of the brake discs are: 48 vane brake disc – 1529 Hz, 72 flash vane brake disc – 1358.9 Hz and 72 vane brake disc – 1450.6 Hz. All these values for each diametrical mode can be read from the diagrams shown in Figs. 3 and 5. Common to all brake disc designs is that the shape of waves is the same for all modes.

## 5. Design of experiments

The Taguchi method was applied in order to investigate which from the observed parameters (Table 5) had the highest influence on noise formation. This is a method used for the evaluation of influence of some parameter on a specific variable. It can also be very easily used to solve various engineering problems [23-25]. The application of the Taguchi method makes the analysis easier and reduces the time necessary for the analysis and finding the best solution. In this case, it helped in finding which type of design of the brake disc would be the best one and which properties the material from which the brake disc is to be made should have, with the aim of the braking noise be as low as possible. The same occurs when some product needs to be optimized, and the results need to be obtained as fast as possible including the best possible quality. In this analysis, four control factors were defined and each of them had three levels, Table 5.

**Table 5** Analysed parameters and their levels

| *Parameters* | *Level 1* | *Level 2* | *Level 3* |
|---|---|---|---|
| **A –Shape of ribs** | 48 vane brake disc | 72 vane brake disc | 72 flash vane brake disc |
| **B – Density (kg/m$^3$)** | 7,400 | 7,600 | 7,800 |
| **C – Young's modulus (GPa)** | 168 | 181 | 197 |
| **D – Poisson's ratio (-)** | 0.26 | 0.29 | 0.32 |

In this study, the smaller-is-better-type [26] for determination of the SN (signal-to-noise) ratio, relation 1, was applied. The signal represents the mean value, while noise is the standard deviation. The SN ratio was used for minimising the brake disc noise, and by analysing the SN ratio variables the statistic important parameters can be determined:

$$SN = -10 \log \frac{1}{n}\left( y_1^2 + y_2^2 + ... + y_n^2 \right), \tag{1}$$

where $y_1, y_2 ... y_n$ are the results of the numerical analysis, and *n* is the number of the numerical analysis.

## 6. Results and discussion

### 6.1 Frequency results and SN ratios

Since the highest frequency was obtained for mode 7 (Fig. 3), this mode was selected for further analysis. The results of the modal analysis for mode 7 are shown in Table 6, together with the calculated SN ratios. If SN ratio is higher, the frequency is lower, which means that the combination with the highest SN ratios gives the best results.

### 6.2 Parameters influencing values of frequency determined by using ANOVA

The results of the modal analysis were used to determine the parameters influencing the value of the frequency. On the basis of the value of delta (Table 7), it can be concluded that the parameter with the highest value of delta has the greatest influence on noise formation, and this is the shape of ribs, while the parameter with the lowest value of delta, that is, the parameter with the smallest influence, is Poisson's coefficient.

**Table 6** Results of modal analysis and SN ratios

| No. | A | B | C | D | f (Hz) | SN ratio (dB) | No. | A | B | C | D | f (Hz) | SN ratio (dB) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **1** | 1 | 1 | 1 | 1 | 10,013.0 | -80.0113 | **42** | 2 | 2 | 2 | 3 | 8,742.6 | -78.8328 |
| **2** | 1 | 1 | 1 | 2 | 9,998.7 | -79.9989 | **43** | 2 | 2 | 3 | 1 | 9,107.0 | -79.1875 |
| **3** | 1 | 1 | 1 | 3 | 9,992.1 | -79.9931 | **44** | 2 | 2 | 3 | 2 | 9,110.7 | -79.191 |
| **4** | 1 | 1 | 2 | 1 | 10,394.0 | -80.3357 | **45** | 2 | 2 | 3 | 3 | 9,120.8 | -79.2007 |
| **5** | 1 | 1 | 2 | 2 | 10,378.0 | -80.3223 | **46** | 2 | 3 | 1 | 1 | 8,301.5 | -78.3831 |
| **6** | 1 | 1 | 2 | 3 | 10,371.0 | -80.3164 | **47** | 2 | 3 | 1 | 2 | 8,304.8 | -78.3866 |
| **7** | 1 | 1 | 3 | 1 | 10,843.0 | -80.703 | **48** | 2 | 3 | 1 | 3 | 8,314.1 | -78.3963 |
| **8** | 1 | 1 | 3 | 2 | 10,827.0 | -80.6902 | **49** | 2 | 3 | 2 | 1 | 8,616.7 | -78.7068 |
| **9** | 1 | 1 | 3 | 3 | 10,820.0 | -80.6845 | **50** | 2 | 3 | 2 | 2 | 8,620.2 | -78.7103 |
| **10** | 1 | 2 | 1 | 1 | 9,792.1 | -79.8175 | **51** | 2 | 3 | 2 | 3 | 8,629.8 | -78.72 |
| **11** | 1 | 2 | 1 | 2 | 9,866.3 | -79.8831 | **52** | 2 | 3 | 3 | 1 | 8,989.5 | -79.0747 |
| **12** | 1 | 2 | 1 | 3 | 9,859.7 | -79.8773 | **53** | 2 | 3 | 3 | 2 | 8,993.1 | -79.0782 |
| **13** | 1 | 2 | 2 | 1 | 10,256.0 | -80.2196 | **54** | 2 | 3 | 3 | 3 | 9,003.1 | -79.0878 |
| **14** | 1 | 2 | 2 | 2 | 10,241.0 | -80.2068 | **55** | 3 | 1 | 1 | 1 | 7,665.8 | -77.6911 |
| **15** | 1 | 2 | 2 | 3 | 10,234.0 | -80.2009 | **56** | 3 | 1 | 1 | 2 | 7,680.2 | -77.7075 |
| **16** | 1 | 2 | 3 | 1 | 10,700.0 | -80.5877 | **57** | 3 | 1 | 1 | 3 | 7,701.0 | -77.7309 |
| **17** | 1 | 2 | 3 | 2 | 10,684.0 | -80.5747 | **58** | 3 | 1 | 2 | 1 | 7,956.9 | -78.0149 |
| **18** | 1 | 2 | 3 | 3 | 10,677.0 | -80.569 | **59** | 3 | 1 | 2 | 2 | 7,971.8 | -78.0311 |
| **19** | 1 | 3 | 1 | 1 | 9,753.2 | -79.7829 | **60** | 3 | 1 | 2 | 3 | 7,993.4 | -78.0546 |
| **20** | 1 | 3 | 1 | 2 | 9,739.0 | -79.7703 | **61** | 3 | 1 | 3 | 1 | 8,301.1 | -78.3827 |
| **21** | 1 | 3 | 1 | 3 | 9,732.5 | -79.7645 | **62** | 3 | 1 | 3 | 2 | 8,316.7 | -78.399 |
| **22** | 1 | 3 | 2 | 1 | 10,124.0 | -80.107 | **63** | 3 | 1 | 3 | 3 | 8,339.2 | -78.4225 |
| **23** | 1 | 3 | 2 | 2 | 10,109.0 | -80.0942 | **64** | 3 | 2 | 1 | 1 | 7,564.3 | -77.5754 |
| **24** | 1 | 3 | 2 | 3 | 10,102.0 | -80.0881 | **65** | 3 | 2 | 1 | 2 | 7,578.5 | -77.5917 |
| **25** | 1 | 3 | 3 | 1 | 10,562.0 | -80.4749 | **66** | 3 | 2 | 1 | 3 | 7,599.0 | -77.6151 |
| **26** | 1 | 3 | 3 | 2 | 10,546.0 | -80.4618 | **67** | 3 | 2 | 2 | 1 | 7,851.5 | -77.8991 |
| **27** | 1 | 3 | 3 | 3 | 10,539.0 | -80.456 | **68** | 3 | 2 | 2 | 2 | 7,899.2 | -77.9517 |
| **28** | 2 | 1 | 1 | 1 | 8,522.9 | -78.6117 | **69** | 3 | 2 | 2 | 3 | 7,887.2 | -77.9385 |
| **29** | 2 | 1 | 1 | 2 | 8,526.3 | -78.6152 | **70** | 3 | 2 | 3 | 1 | 8,191.2 | -78.267 |
| **30** | 2 | 1 | 1 | 3 | 8,535.9 | -78.625 | **71** | 3 | 2 | 3 | 2 | 8,206.6 | -78.2833 |
| **31** | 2 | 1 | 2 | 1 | 8,846.5 | -78.9354 | **72** | 3 | 2 | 3 | 3 | 8,228.8 | -78.3067 |
| **32** | 2 | 1 | 2 | 2 | 8,850.1 | -78.939 | **73** | 3 | 3 | 1 | 1 | 7,466.7 | -77.4626 |
| **33** | 2 | 1 | 2 | 3 | 8,860.0 | -78.9487 | **74** | 3 | 3 | 1 | 2 | 7,480.7 | -77.4788 |
| **34** | 2 | 1 | 3 | 1 | 9,229.3 | -79.3034 | **75** | 3 | 3 | 1 | 3 | 7,501.0 | -77.5024 |
| **35** | 2 | 1 | 3 | 2 | 9,233.0 | -79.3069 | **76** | 3 | 3 | 2 | 1 | 7,750.2 | -77.7863 |
| **36** | 2 | 1 | 3 | 3 | 9,243.3 | -79.3165 | **77** | 3 | 3 | 2 | 2 | 7,764.7 | -77.8025 |
| **37** | 2 | 2 | 1 | 1 | 8,410.0 | -78.4959 | **78** | 3 | 3 | 2 | 3 | 7,785.8 | -77.8261 |
| **38** | 2 | 2 | 1 | 2 | 8,413.4 | -78.4994 | **79** | 3 | 3 | 3 | 1 | 8,085.5 | -78.1541 |
| **39** | 2 | 2 | 1 | 3 | 8,422.8 | -78.5091 | **80** | 3 | 3 | 3 | 2 | 8,100.7 | -78.1705 |
| **40** | 2 | 2 | 2 | 1 | 8,729.3 | -78.8196 | **81** | 3 | 3 | 3 | 3 | 8,122.6 | -78.1939 |
| **41** | 2 | 2 | 2 | 2 | 8,732.8 | -78.8231 | | | | | | | |

The results for the frequency obtained by using ANOVA are given in Table 8. It can be noticed that the most influential parameter is the shape of ribs (90.82%). So, in the design of a new brake disc, this is the most important parameter that must be taken into consideration. In comparison with this parameter, other parameters do not have a great influence. As regards the material properties, the highest influence has Young's modulus, while Poisson's ratio has almost no influence. In the analysis, the F-ratio was used to indicate the significance of every factor. The F-value expresses which parameter has an important influence on the process response.

**Table 7** Response table for signal-to-noise ratio (smaller is better)

| *Level* | *Shape of ribs* | *Density* | *Young's modulus* | *Poisson's ratio* |
|---|---|---|---|---|
| **1** | -80.22 | -79.11 | -78.66 | -78.99 |
| **2** | -78.84 | -79.00 | -78.99 | -79.00 |
| **3** | -77.93 | -78.89 | -79.35 | -79.01 |
| **Delta** | 2.29 | 0.23 | 0.69 | 0.01 |
| **Rank** | 1 | 3 | 2 | 4 |

**Table 8** Analysis of variance for signal-to-noise ratio for frequency

| *Variance source* | *DoF* | *Swq SS* | *Adj SS* | *Adj MS* | *F ratio* | *Contribution rate (%)* |
|---|---|---|---|---|---|---|
| **Shape of ribs** | 2 | 71.6312 | 71.6312 | 35.8156 | 215,586.7 | 90.82 |
| **Density** | 2 | 0.7053 | 0.7053 | 0.3527 | 2,122.76 | 0.89 |
| **Young's modulus** | 2 | 6.5180 | 6.5180 | 3.2590 | 19,617 | 8.26 |
| **Poison ratio** | 2 | 0.0028 | 0.0028 | 0.0014 | 8.35 | 0.01 |
| **Error** | 72 | 0.0120 | 0.0120 | 0.0002 | | 0.02 |
| **Total** | 80 | 78.8692 | | | | 100 |

## 6.3 Analysis of the signal-to-noise ratio

The influence of the design and materials of the brake disc is shown in Figs. 7 and 8, which also help in easier understanding of the previous tables. The lowest frequencies are obtained when the disc has 72 flash vanes. By reducing Young's modulus and Poisson's ratio, the values of the frequency decrease. For the material density, however, this is not the case. The lowest value of the frequency was obtained for the highest value of the density. If the line of the control parameter is close to the horizontal, then it has the smallest influence. On the other hand, when the control parameter is presented with a line which is inclined with respect to the horizontal line, it has a significant influence. Taking these two rules into consideration, the minimal frequency is obtained in the case of the A3-B2-C1-D1 combination.

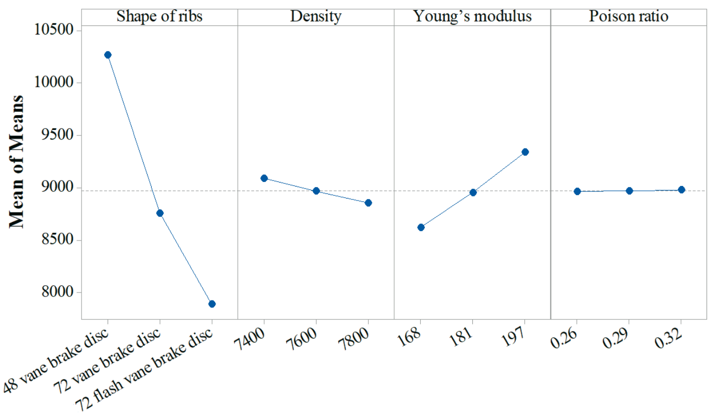



**Fig. 7** Main effects plot for signal-to-noise (smaller is better) ratio for frequency

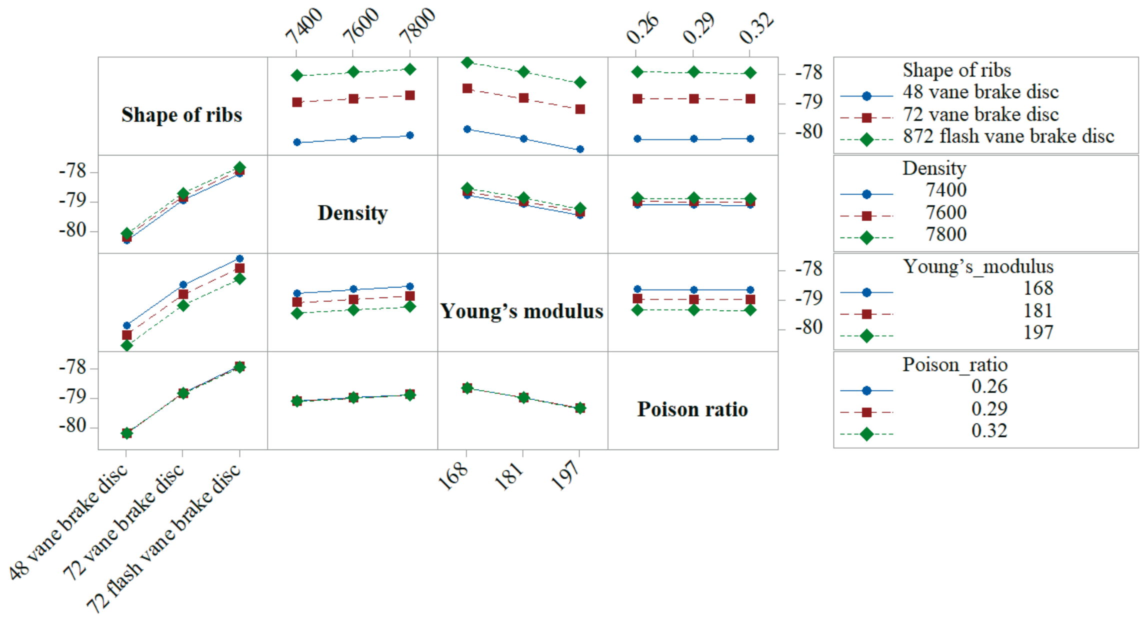



**Fig. 8** Interaction plot for signal-to-noise ratio for frequency

The results of the frequency investigation are shown in Fig. 9. The size of the frequency is presented in the shape of a map, where the dependency of the density and the brake disc shape of ribs (Fig. 9a) as well as Young's modulus and the brake disc shape of ribs (Fig. 9b) are shown. With an increase in the density, the braking noise decreases, while with an increase in Young's modulus, the braking noise rises. It can be noticed that the design of the brake disc, i.e., the shape of ribs has the greatest influence on the noise formation, and the second most important factor is Young's modulus [22].

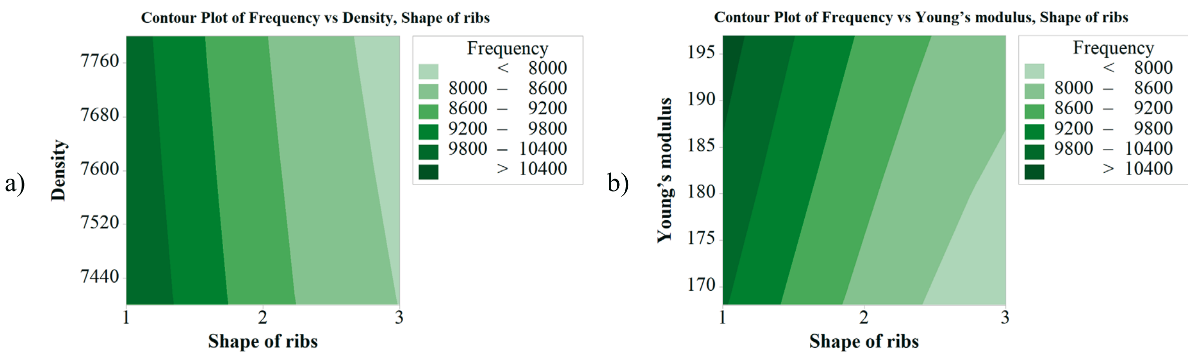



**Fig. 9** Frequency rate with respect to: a) density and shape of vanes; b) Young's modulus and shape of ribs

## 7. Conclusions and remarks

The Taguchi method was used in this study with the aim of determining the parameter which mostly influences frequency values, so that the braking noise that appears during exploitation could be reduced to a minimum. The control factors (shape of the ribs of the brake disc, density, Young's modulus, Poisson's ratio) were also optimized. By applying ANOVA the following conclusions were made: The shape of ribs has the highest influence on the frequency value; The smallest frequencies were obtained for the brake disc with 78 flash vanes; The influence of the share of the shape of ribs on the size of the frequency is not negligible at all, and it affects the size of the frequency with 90.82%. The second most influential factor is Young's modulus with 8.26%. In further studies, one of the influential parameters that should be studied is brake disc temperature that rises during the braking process.

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