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OPTIMIZATION OF BRAKE DISC FOR IMPROVED THERMAL AND STRUCTURAL BEHAVIOR USING THERMO-MECHANICAL ANALYSIS

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Abstract:
During braking, friction between the brake disc and pads converts the vehicle's kinetic energy into heat, resulting in a temperature rise at the contact interface. This temperature rise induces thermal gradients and corresponding stresses within the disc material. Excessive thermal loading and stress concentration can adversely affect braking efficiency and structural integrity. The present study investigates the thermal and mechanical behavior of a ventilated brake disc subjected to frictional heating. A coupled thermo-mechanical analysis was performed to determine the peak temperature distribution and the resulting thermal stresses. Transient thermal and structural simulations were conducted using the finite element method (FEM). Subsequently, key geometric parameters influencing the thermal and stress responses were identified, and a dimensional optimization was carried out using Response Surface Methodology (RSM). The optimized configuration revealed that an increase inboard and outboard disc thickness (from 8.5 mm to 9.497 mm), a reduction in vane thickness (from 6 mm to 4 mm), and a decrease in ventilation gap (from 7 mm to 5.194 mm), with the number of vanes held constant, led to improved thermal performance. The optimized design exhibited a 3.68% reduction in maximum temperature and an 8.86% reduction in thermal stress, without a significant increase in overall weight.

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1. INTRODUCTION

The brake is a very critical component that ensures the safety of the vehicle as well as the occupants. When the vehicle stops due to the application of brakes, the total kinetic energy possessed by it is converted into heat due to the friction between the rotating disc and the squeezing pads. This generated heat results in an increase in the temperature of the disc as well as the pad. If the heat generated is not dissipated at a faster rate than the temperature of the disc continuously increases due to repetitive braking, which results in different adverse effects such as brake fade, local scoring, thermo-elastic instability, premature wear, brake fluid vaporization, thermal judder and cracks, etc. Automobiles commonly use two types of braking systems: drum brakes and disc brakes. Among these, disc brakes offer better performance and are therefore widely used in modern vehicles. Disc brakes are further classified into solid and ventilated types. Ventilated discs have a greater surface area, allowing them to dissipate heat more quickly than solid discs. The efficiency of the disc

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brakes largely depends on how fast it dissipates the frictional heat generated to the atmosphere. For better heat dissipation to the atmosphere, the ventilated disc uses vanes, either curved or radial, sandwiched between the inboard and outboard discs. They form a cooling passage for the flow of the air. In addition, the design of the ventilation channels and vanes helps to improve airflow, enhancing cooling efficiency. The vanes can be either straight radial, curved radial, single or double airfoil [1]. Use of heat pipes [2] between the inboard and outboard discs also enhances the heat dissipation. Holes are also drilled either straight or cross on friction tracks to increase the surface area [3-5]. In certain cases, heat dissipation is further improved by incorporating holes and slots on the friction surfaces in addition to curved vanes, as noted by Afzal [6]. In addition to the geometry of the disc, the material used for brake discs also plays a very important role because it affects how much heat and stress the disc can handle, as well as its weight. Cast iron is one of the best materials for handling heat, according to Belhocine and Afzal [7]. A study by Ahmad et al. [8] showed that aluminum performs the worst compared to carbon-carbon and titanium alloys. Another study by Al Riyami et al. [9] found that cast carbon steel works better than grey cast iron in terms of strength and heat performance. Singh et al. [10] developed a new hybrid aluminum matrix composite (Al6061/SiC/Gr), which can replace the conventional brake disc material-cast iron. The finite element method was used to test the performance of the new hybrid composite and concluded that it performs better than cast iron.

Various optimization techniques have been used by different researchers to optimize the brake disc performance. The Taguchi method was used to determine optimal values of different geometrical parameters that impacted the cooling time [11], fatigue life [12,13], braking time [14], and volume flow of air and heat exchange area [15]. Response surface methodology was employed to increase fatigue life and to decrease the weight [16], to improve thermal performance [17], to minimize thermal distortion and the temperature [18], and to minimize brake squeal tendency [19].

The finite element method (FEM) has become popular as an alternative to time-consuming and costly experimental testing. It is commonly used to simulate disc brakes and evaluate their thermal and structural performance. A review focused on the thermo-mechanical behavior of brake discs has been presented by Patil and Chavan [20]. The finite element method is used by Senbagan et al. [21] to enhance the disc cooling by using an axial ventilator fitted on the wheel rim. Krishnan et al. [22] used ANSYS to determine the temperature and heat flux in the disc. Hariram et al. [23] performed thermo-mechanical analysis to predict the thermal and structural behavior of the brake disc. In order to improve the cooling efficiency of the system, Orlandi et al. [17] investigated horizontal and vertical lubricating oil inlets and concluded a slight improvement in cooling efficiency with horizontal inlets.

Although several studies have optimized specific parameters (using methods like Taguchi or RSM), there is still scope to optimize critical geometrical parameters using transient thermo-mechanical analysis to better represent real braking conditions. Thus, the study is motivated by the need to design an optimized brake disc geometry that enhances heat dissipation and reduces thermal stresses using transient FEM-based analysis.

2. MATERIALS AND METHODS

As braking of a vehicle is a transient phenomenon, the amount of heat generated, temperature and stresses vary continuously with time. For this reason, transient thermal and structural analysis is preferred over steady state analysis. The methodology used is illustrated in Fig. 1. Firstly, the vehicle data is collected and based on the data; calculations for the heat entering into the disc in the form of heat flux, normal force acting due to application of brakes, and convective heat transfer coefficient are done. The existing brake disc is then modeled and coupled transient thermal and transient structural analysis is performed to determine maximum temperature and equivalent stresses. Based on the analysis results of the existing disc, critical areas are identified where maximum temperature is induced and the geometrical parameters in those areas are identified for optimization. Using the design of experiments, 4 geometrical parameters from the critical areas of high temperature are optimised for reducing the maximum temperature, stress and weight of the brake disc. The brake disc is modeled with the optimized values of parameters and coupled transient thermal and transient structural analysis is performed to determine maximum temperature and equivalent stresses. The results are compared with those obtained for the existing brake disc. The research presented in this paper improves the thermal and structural performance, mechanical reliability and safety of the brake disc used in passenger cars. The methodology adapted using finite element analysis and response surface methodology for optimization offers a robust solution to the efficient brake system used in automobiles.

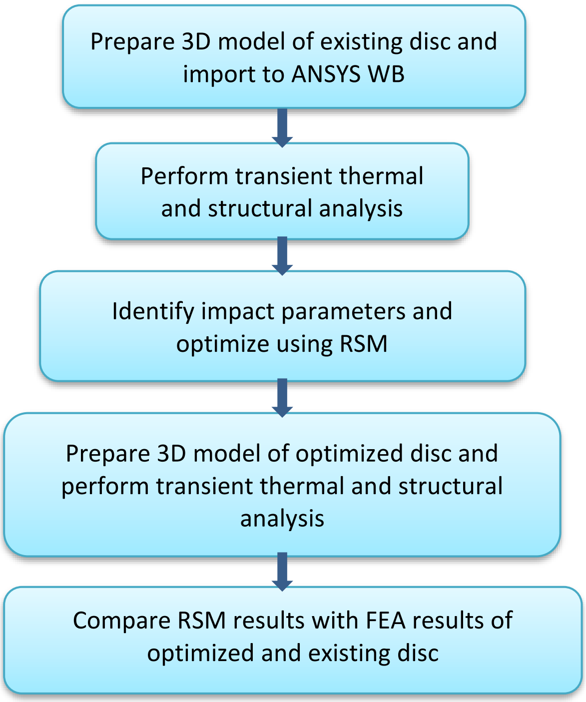


Fig. 1. Methodology Flowchart

2.1 Heat Flux Calculations

When a vehicle slows down or stops, its kinetic energy is converted into heat through friction between the brake disc and pad. This process causes a significant temperature rise and generates high thermal stresses. In thermo-mechanical analysis, the accuracy of results depends on the heat input, represented as heat flux, and the heat loss, which is controlled by the convective heat transfer coefficient. These two factors – heat flux and convective heat transfer – determine the maximum temperature the disc reaches, as noted by Patil and Chavan [20]. During transient thermal analysis, the heat flux varies over time to reflect real braking conditions. Based on vehicle dynamics and the fact that the front brakes bear 60% of the load [5,24], the force acting on each brake rotor (F_{rotor}) is calculated using Eq. (1) [24] and the instantaneous heat flux into the rotor ($Q_{in}(t)$) are calculated by Eq. 2 [24] using the vehicle data provided in Table 1.

Table 1. Vehicle data

Mass of vehicle, M	1710 kg
Initial Velocity, v_o	77 kmph = 21.4 m/s
Time to stop, t_{stop}	6 s
Effective rotor radius, r_{rotor}	0.098 m
Tire Radius, r_{tire}	0.215 m
Brake pad area A_d	0.00509 m^2

$$F_{rotor} = \frac{30\% \cdot \frac{1}{2}(M {V_o}^2)}{\left(2\cdot\frac{r_{rotor}}{r_{tire}} V_o t_{stop}\right) - \frac{1}{2}\left\{\frac{V_o}{t_{stop}}\right\} {t_{stop}}^2} = 2230.076\ \text{N} \tag{1}$$

where are: *M - Mass of the Vehicle (kg), V_0 - Initial velocity (m/s), t_{stop} - time to stop (s), r_{rotor} - effective radius of the rotor (m), r_{tire} – radius of the wheel (m).* Instantaneous heat flux $Q_{in}(t)$ into the rotor is calculated using the relation:

$$Q_{in}(t) = (F_{rotor}) v_{rotor}(t) = F_{rotor}\frac{r_{rotor}}{r_{tire}}\left(v_0 - \left\{\frac{v_0}{t_{stop}}\right\} t\right) = 21758.206 - 3625.68\,\text{t} \tag{2}$$

The heat flux - Q obtained using Eq. (2) is in (W). In order to obtain the heat flux in (W/m^2) it is necessary to integrate over the time interval and divide by the area of the brake pad.

$$Q = \frac{\int_0^t (21758.206 - 3625.68\,(t))\,dt}{(2\cdot 0.00509)} \tag{3}$$

Solving Eq. (3) for different time intervals starting from 0 to 1, 1 to 2, 2 to 3, 3 to 4, 4 to 5 and 5 to 6, the values of heat flux as tabulated in Table 2.

Table 2. Heat Flux

Time interval (s)	Heat flux (W/mm^2)
0-1	1.959
1-2	1.603
2-3	1.246
3-4	0.89
4-5	0.534
5-6	0.178
6-7	0

This coefficient varies with the airflow near the rotor and the vehicle's speed, so it changes constantly while braking. However, measuring the exact value of the heat transfer coefficient is very challenging. Convective heat transfer coefficient – *h* of 230 $W/m^{2o}C$ [3,25,26] is applied on the surfaces from where the heat will be dissipated. Although *h* physically depends on air properties, flow velocity, and rotor geometry and may vary with time, an average value is assumed based on published literature [3,26]. This was justified since heat flux has a dominant effect on the disc temperature evolution, and in this study, the focus is on geometric optimization rather than performing a full conjugate fluid–thermal analysis. The geometry inherently influences the cooling performance, which is reflected in the resulting temperature distribution and stress fields.

2.2 Thermo-Mechanical Analysis of Existing Brake Disc

The 3D model of the brake disc (Fig. 2), prepared using CATIA software, is imported into ANSYS Workbench 2024 R1 software for the analysis. Material properties are assigned, which are tabulated in Table 3, and then the model is meshed. The meshed model is shown in Fig. 3. Element quality and aspect ratio are checked. Tetrahedral elements are used for meshing and the total number of nodes is 1193329, while elements are 820502. A grid independence test was performed to determine the size of elements. By refining the mesh from an element size of 6.5 mm till it was found that there was no difference in the maximum temperature attained. Fig. 4 shows the results of the test. It was decided that the element size of 3 mm should be considered.

Table 3. Material properties for grey cast iron

Density	7200 Kg/m^3
Young's modulus	1.1 E5 N/mm^2
Thermal conductivity	52 W/m°C
Specific heat	447 J/kg°C

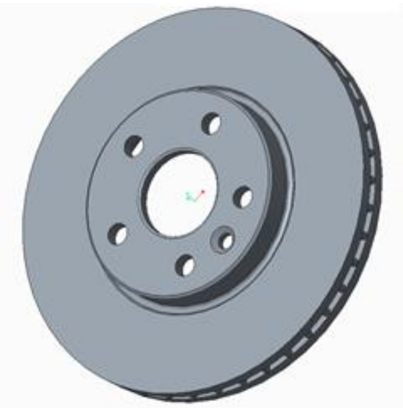


Fig. 2. 3D model of brake disc

Fig. 3. Meshing of brake disc

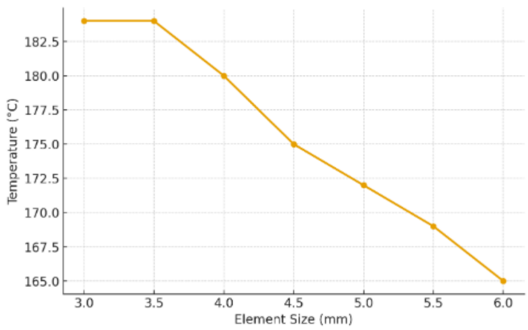


Fig. 4. Selection of mesh size

After meshing, transient thermal boundary conditions are applied as shown in Fig. 5. Heat flux is applied in steps as calculated for 6 seconds on the rubbing surfaces. Convective heat transfer coefficient of 230 W/m^{2}°C is applied on the remaining surfaces.

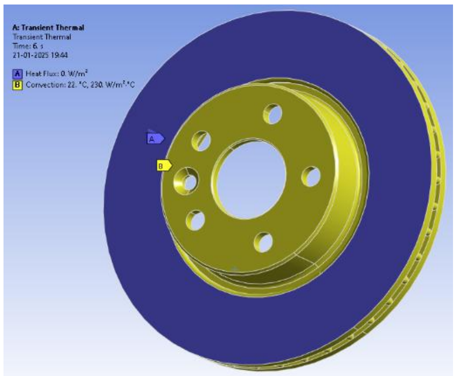


Fig. 5. Transient thermal boundary conditions

After completing the transient thermal analysis, a transient structural analysis is conducted. In this step, the brake disc is constrained at the mounting holes, a normal force is applied on both friction surfaces, and a time-dependent angular velocity is imposed. The boundary conditions are illustrated in Fig. 6.

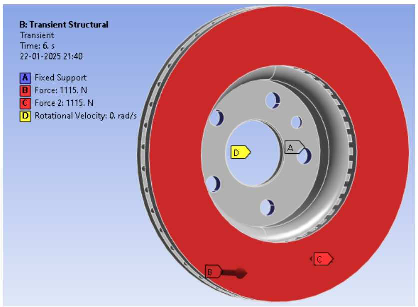
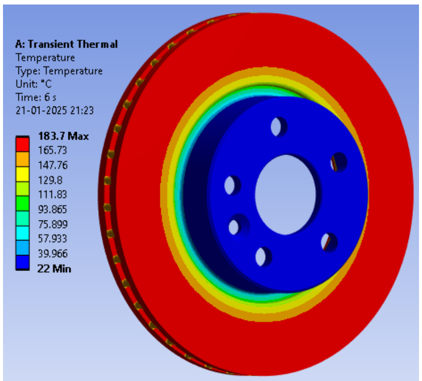
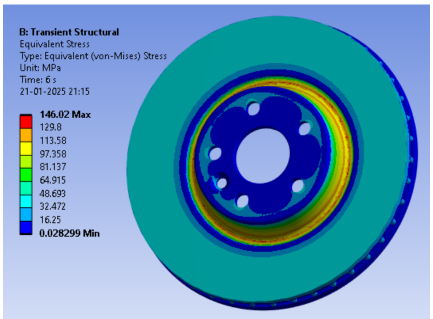


Fig. 6. Transient structural boundary conditions

The results of thermo-mechanical analysis are illustrated in Fig. 7 (a) and Fig. 7 (b)



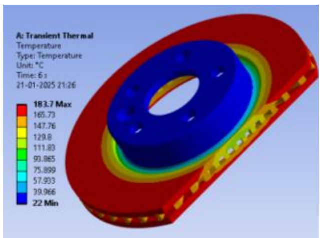
(a)



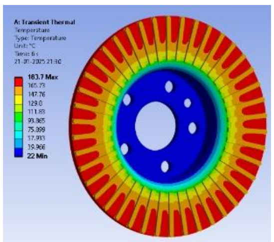
(b)

Fig. 7. Results of Thermo-mechanical analysis on existing brake disc, (a) Maximum temperature at 6 s; (b) Equivalent stress distribution

Fig. 8 (a) and Fig. 8 (b) show the regions of temperature distribution. Based on the thermal analysis, key geometric parameters that influence the maximum temperature are identified, as these factors directly impact the temperature distribution.



(a)



(b)

Fig. 8. Regions of maximum temperature

Inboard and outboard thickness, number of vanes, thickness of vane and ventilation gap are identified as critical parameters as the maximum temperature is found in this region. The original values of these parameters are illustrated in Fig. 9.

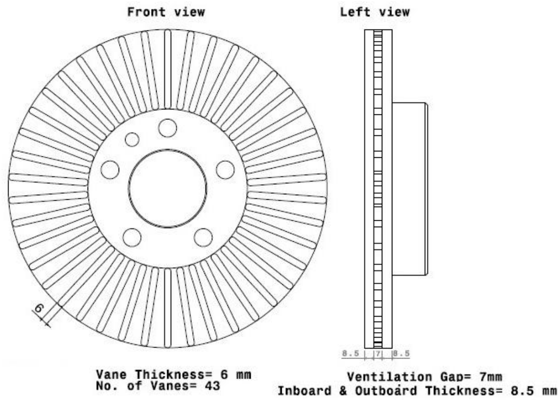


Fig. 9. Original values of geometrical parameters

2.3 Response Surface Methodology

Response surface methodology (RSM) is a statistical, theoretical and mathematical technique for model building in order to optimize the level of independent variables. For this study, RSM was adapted as it efficiently combines experimental design, statistical analysis, and optimization into a single framework. It provides an explicit mathematical model for predicting system behavior, requires fewer simulation runs compared to heuristic algorithms. Along with this, the method provides ANOVA (Analysis of Variance) results to check which parameters significantly affect the response, and to validate model adequacy through R^2, adjusted R^2, lack-of-fit, and residual analysis, which other methods do not providing such solutions.

Table 4. Independent variables and their levels

Independent Variables	Symbol	Coded levels				
		-α	-1	0	+1	+α
Inboard and outboard thickness (mm)	A	4.5	6.5	8.5	10.5	12.5
Number of Vanes (nos.)	B	37	40	43	46	46
Thickness of vane (mm)	C	2	4	6	8	10
Ventilation Gap (mm)	D	3	5	7	9	11

RSM is used to determine the effect of independent variables, which include A: Inboard and outboard thickness, B: Number of Vanes, C:

Thickness of vane, and D: Ventilation Gap on response variables, temperature, stress and weight of the brake disc. RSM design with the levels of independent variables is tabulated (Table 4). Central composite design (Five levels) and quadratic model were used to design the experiment. Thirty experiments, including eight axial points, sixteen factorial points and six central points, were randomly performed according to the Central Composite Design.

3. RESULTS AND DISCUSSIONS

Thirty brake disc 3D models were prepared using modelling software, and thermo-mechanical analysis was performed to determine the effect of independent variables, viz, inboard and outboard thickness, number of vanes, thickness of vane and ventilation gap, on response variables, temperature, stress and weight of the brake disc. The data is given in Table 5. Coefficients of the polynomial equation were computed from experimental data to predict the values of the response variable.

Table 5. Effect of independent variables on response functions

Run	Factors				Response Functions		
	A	B	C	D	Temperature (°C)	Equivalent Stress (MPa)	Weight (N)
1	8.5	43	6	7	183.7	146.02	70.9636
2	6.5	40	4	9	225.08	239.48	56.9127
3	6.5	40	8	5	222.66	248.58	57.5062
4	10.5	40	4	5	166.64	126.68	79.2638
5	8.5	49	6	7	180.87	143.43	71.8053
6	10.5	46	8	5	161.25	118.48	83.3673
7	6.5	46	4	5	231.76	241.92	54.835
8	10.5	40	8	9	158.36	115.44	87.4915
9	6.5	46	4	9	221.03	230.29	57.775
10	4.5	43	6	7	283.45	290.01	45.8971
11	10.5	40	4	9	164.02	121.32	81.8203
12	8.5	43	2	7	192.95	154.22	66.0262
13	8.5	43	6	7	183.7	146.02	70.9636
14	8.5	43	6	7	183.7	146.02	70.9636
15	10.5	46	8	9	156.19	113.23	89.2072
16	8.5	43	10	7	175.05	137.31	75.5115
17	8.5	43	6	7	183.7	146.02	70.9636
18	6.5	40	4	5	233.59	249.5	54.3552
19	8.5	43	6	3	191.96	154.75	66.6943
20	10.5	46	4	9	162.66	120.47	82.6836
21	8.5	43	6	7	183.7	146.02	70.9636
22	6.5	46	8	5	218.77	259.6	58.4588
23	6.5	46	8	9	202.95	214.65	64.2987
24	8.5	37	6	7	185.93	148.06	69.7991
25	10.5	40	8	5	162.75	119.88	82.4148
26	6.5	40	8	9	209.07	219.34	62.5849
27	8.5	43	6	7	183.7	146.02	70.9636
28	8.5	43	6	11	177.82	142.41	74.9111
29	10.5	46	4	5	165.9	125.51	79.7435
30	12.5	43	6	7	149.1	103	95.7083

3.1 Effect of Independent Variables on Temperature

Table 6 summarizes the ANOVA data for the quadratic regression model estimating the temperature of the disc. With an F-value of 245.21, the model explained variance in temperature with a high statistical relevance. With an outstanding fit between experiment performed by simulation and expected values, the coefficient of determination (R^2 = 0.9956) indicates that the model accurately captures data trends. While the sufficient precision value of 63.5848 is above the criterion of 4, it confirms the signal-to-noise appropriateness for navigating the design space.

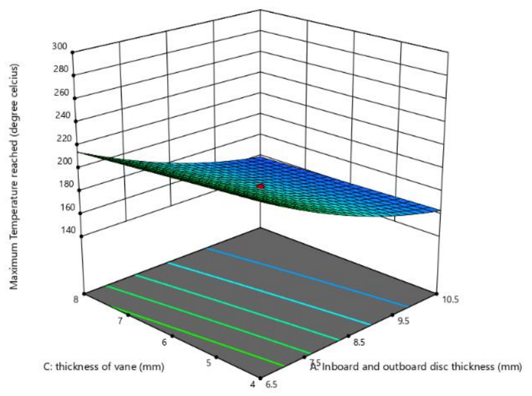
With a F-value of 3032.7, the inboard and outboard disc thickness (A) has the most significant effect on temperature rise among other variables, followed by the thickness of vanes (C) (F value 73.39) and the ventilation gap (D) (F value 47.65).

Eq. (4) is the regression equation developed for the temperature

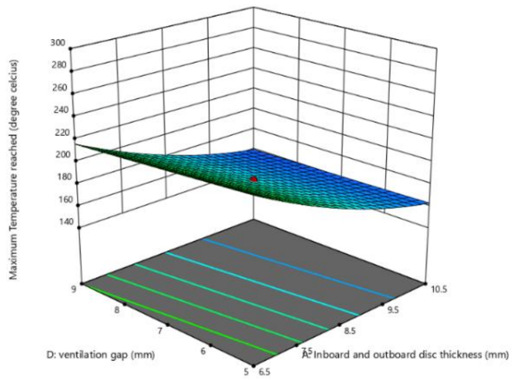
$$\textit{Maximum Temperature} = 524.16 - 61.11A + 1.31B - 3.15C - 3.11D + 0.11AB + 0.58AC + 0.52AD - 0.06BC - 0.06BD - 0.21CD + 2.0A^2 - 0.022B^2 - 0.01C^2 + 0.04D^2 \quad (4)$$

The derived regression equation represents a comprehensive predictive model. The most influential parameter that controls the temperature is the inboard and outboard thickness (A) due to a large negative coefficient (-61.11). An increase in the magnitude of parameter A results in a decrease in temperature. The interaction between the parameters is very small, i.e., each factor can be optimized independently. The analysis of residuals confirmed the model's statistical reliability. The residuals vs. predicted values plot exhibited a systematic distribution pattern.

The 3D surface contour plots illustrating the combined effects of the independent variables on the maximum temperature reached are shown in the Figs. 10 (a) and (b). The plots clearly demonstrate that the inboard and outboard disc thicknesses have a greater influence, with the temperature decreasing from approximately 220°C to below 160°C as the thickness increases from 6.5 mm to 10.5 mm. In contrast, increasing the vane thickness from 4 mm to 8 mm and the ventilation gap from 5 mm to 9 mm results in a smaller temperature reduction, from about 225°C to just under 200°C. This trend occurs because heat dissipation is directly related to the surface area exposed on the disc.



(a)



(b)

Fig. 10. (a) 3D surface contour plots for temperature considering disc thickness and vane thickness, (b) 3D surface contour plots for temperature considering disc thickness and ventilation gap

3.2 Effect of Independent Variables on Equivalent Stress

From Table 6, the model explains the variance in equivalent stress with moderate statistical significance, as indicated by an F-value of 26.66. With an outstanding fit between the experiment performed by simulation and the expected values, the coefficient of determination (R^2 = 0.9614) indicates that the model accurately catches data trends. While the sufficient precision value of 21.0675 is above the criterion of 4, it confirms the signal-to-noise appropriateness for navigating the design space. With a F-value of 332.88, the inboard and outboard disc thickness (A) has the most significant effect on the equivalent stresses induced in the brake disc among other variables.

Eq. (5) is the regression equation obtained for the equivalent stress:

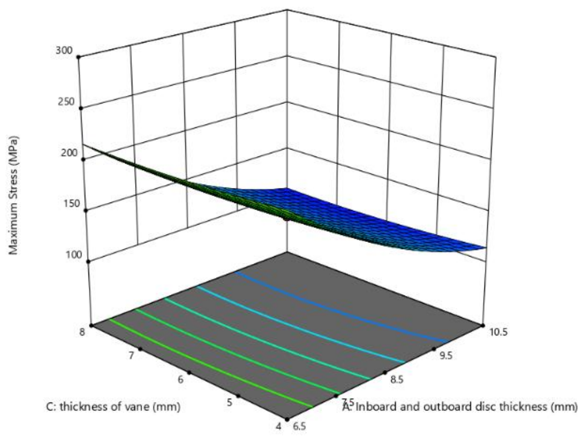
$$\textit{Equivalent Stress} = 1{,}506.95 - 104.84A - 31.82B - 14.32C - 13.98D + 0.05AB - 0.12AC + 1.18AD + 0.22BC - 0.18BD - 0.81CD + 3.98A^2 + 0.36B^2 + 0.81C^2 + 0.98D^2 \quad (5)$$

The regression equation indicates that disc thickness (A) has the most significant effect on the equivalent stress due to its highest negative magnitude of coefficient (104.84), followed by B, while C and D**.** An Increase in parameter A decreases the stress drastically. The positive quadratic terms confirm a nonlinear relationship between the variables and equivalent stress, implying that extreme values of the design parameters can increase stress after an optimal point. The analysis of residuals confirmed the model's statistical reliability. The residuals vs. predicted values plot exhibited a systematic distribution pattern.

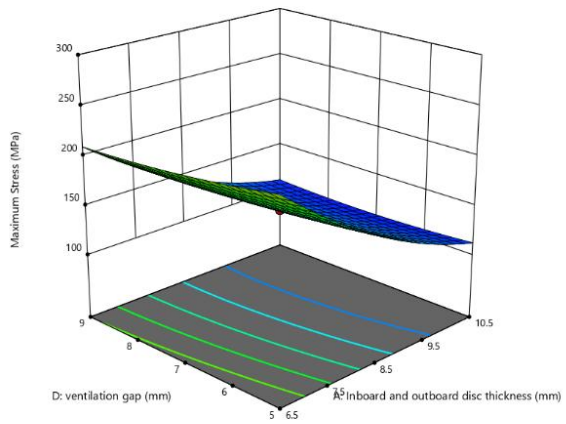
Table 6. Analysis of Variance for quadratic models of temperature, equivalent stress and weight

Source	Temperature °C		Equivalent Stress (MPa)		Weight (N)	
	F-value	p-value	F-value	p-value	F-value	p-value
Model	245.21	< 0.0001	26.66	< 0.0001	1.183E+05	< 0.0001
A	3032.27	< 0.0001	332.88	< 0.0001	1.552E+06	< 0.0001
B	5.66	0.0311	0.1233	0.7304	2514.66	< 0.0001
C	73.39	< 0.0001	1.22	0.2862	56259.11	< 0.0001
D	47.65	< 0.0001	3.80	0.0703	42154.39	< 0.0001
AB	0.8603	0.3683	0.0067	0.9360	0.0002	0.9883
AC	11.71	0.0038	0.0181	0.8947	0.0000	0.9960
AD	9.34	0.0080	1.65	0.2180	0.0002	0.9883
BC	0.2729	0.6090	0.1333	0.7201	182.93	< 0.0001
BD	0.2768	0.6065	0.0914	0.7666	136.71	< 0.0001
CD	1.59	0.2265	0.7741	0.3928	3062.20	< 0.0001
A^2	237.30	< 0.0001	32.11	< 0.0001	18.37	0.0006
B^2	0.1411	0.7125	1.33	0.2668	18.49	0.0006
C^2	0.0077	0.9314	1.33	0.2661	26.95	0.0001
D^2	0.1153	0.7389	1.98	0.1802	18.38	0.0006
R^2	0.9956		0.9614		1	
Adeq. Precision	63.5848		21.0675		1438.4004	

The 3D surface contour plots showing the combined effect of independent variables on equivalent stresses induced are shown in Fig. 11 (a) and Fig. 11 (b). The plot clearly indicates that inboard and outboard disc thickness has more impact as the stresses reduce from just above 200 MPa to around 100 MPa with thickness increasing from 6.5 mm to 10.5 mm, while the stresses induced have little effect due to a change in thickness of vane, number of vanes, or ventilation gap. This may be because the forces on the brake disc are applied directly at the friction surfaces, which are supported by the inboard and outboard disc thickness.



(a)



(b)

Fig. 11. (a) 3D surface contour plots for stress considering disc thickness and vane thickness; (b) 3D surface contour plots for stress considering disc thickness and ventilation gap

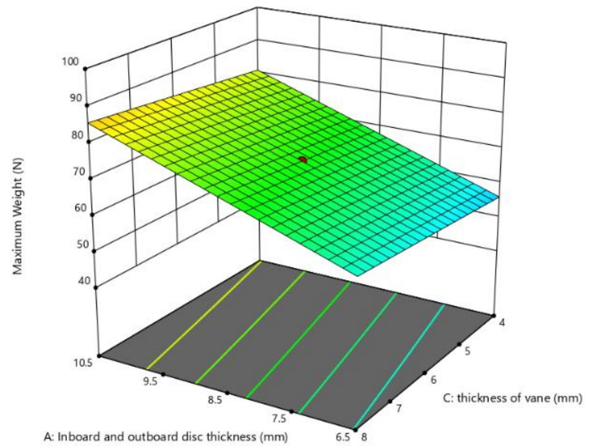
3.3 Effect of Independent Variables on Weight

From Table 6, the model explains the variance in weight with very high statistical significance, as shown by the F-value of 1.183E+05. With an excellent fit between experiment performed by simulation and expected values, the coefficient of determination (R^2 = 1) indicates that the model accurately catches data trends. While the sufficient precision value of 1438.4004 is far above the criterion of 4, it confirms the signal-to-noise appropriateness for navigating the design space. With a F-value of 1.152E+06, the inboard and outboard disc thickness (A) has the most significant effect on the weight of the brake disc among other variables, followed by all three variables with substantial F-values. Also, from Table 6, it is clear that all these variables have a combined effect. Eq. (6) is the regression equation developed for the weight

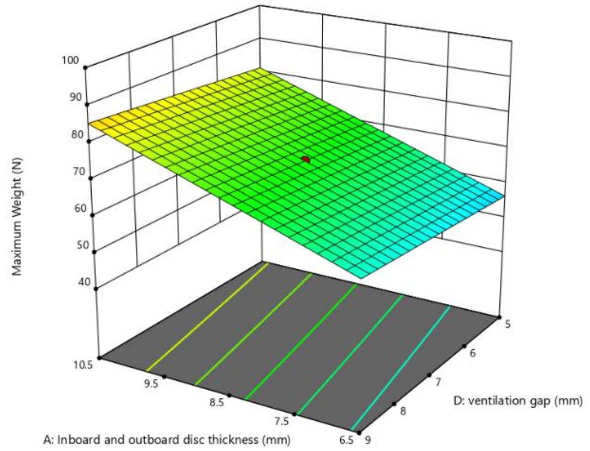
$$Weight = 8.06 + 6.39A + 0.22B - 1.04C + -0.87D + 3.03e\text{-}05AB - 1.57e\text{-}05AC - 4.55e\text{-}05AD + 0.02BC + 0.02BD + 0.17CD - 0.01A^2 - 0.004B^2 - 0.01C^2 - 0.01D^2 \quad (6)$$

According to the regression equation developed for the weight, an increase in disc thickness results in a substantial increase in the overall weight. Vane thickness and ventilation gap having negative coefficients imply that increasing their values reduces the weight, whereas the number of vanes has only a minor positive effect. The analysis of residuals confirmed the model's statistical reliability. The residuals vs. predicted values plot exhibited a systematic distribution pattern.

The 3D surface contour plots in Fig. 12 (a) and Fig. 12 (b). shows the combined effect of independent variables on weight. As per the contour plot, it is clear that the weight increases from around 58 N to 85 N as the thickness of the inboard and outboard discs increases from 6.5 mm to 10.5 mm while the weight slightly increases from around 55 N to 60 N with the increase in vane thickness from 4 mm to 8 mm.



(a)



(b)

Fig. 12. (a) 3D surface contour plots for weight considering disc thickness and vane thickness; (b) 3D surface contour plots for weight considering disc thickness and ventilation gap

4. OPTIMIZATION OF INDEPENDENT VARIABLES

Response surface 3D contour plots were drawn using Design Expert 13 software in order to determine the effect of four independent geometrical parameters on the response function, i.e, temperature, equivalent stresses and weight of the brake disc. The responses were generated by varying the independent variables within a specified range. The plots obtained show a complex interaction among the variables. After that, numerical optimization was executed by the desirability function using the same software. The optimization goal was set to minimize the maximum temperature, reduce the equivalent stresses, and lower the weight. A total of 15 different solutions were generated, each achieving a desirability score of 0.67. Finally, a solution with inboard and outboard disc thickness 9.497 mm, ventilation gap 5.194 mm, no. of vanes 43, and vane thickness 4 mm is selected. The 2D views of the brake disc with optimized dimensions are shown in Fig. 13.

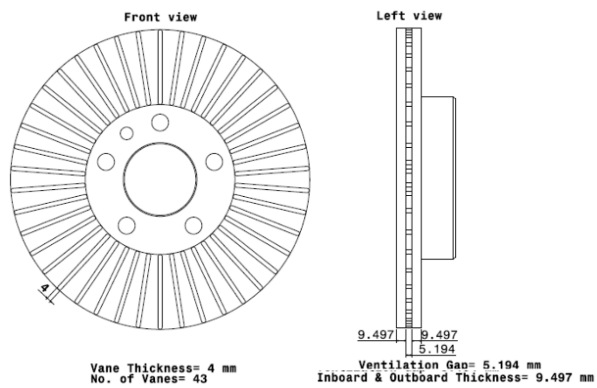
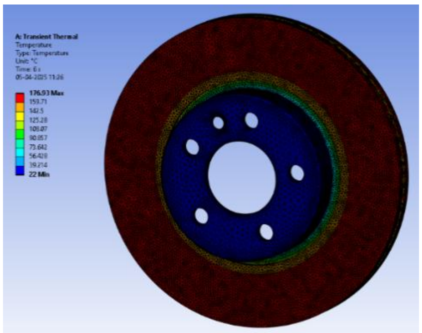


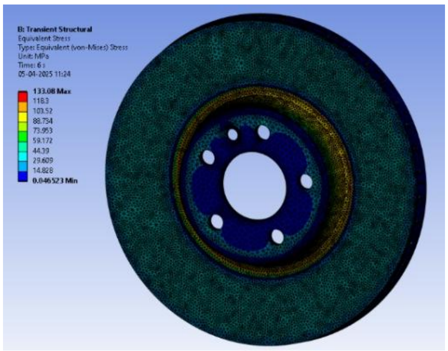
Fig. 13. Optimized values of geometrical parameters

4.1 Verification of RSM Model

A 3D model of the optimised disc was prepared, and a thermo-mechanical analysis with the same boundary conditions was performed. The results obtained from the analysis shown in Fig. 14 (a) and Fig. 14 (b) are closer to those predicted by the design expert software. The comparison of the results for the existing disc, predicted by RSM and verified by FEA, is given in Table 7.



(a)



(b)

Fig. 14. Temperature and stress plots for optimized brake disc

Table 7. Comparisons of results

Response Functions	Existing Disc	Optimized Disc		% Change between existing and optimized disc from analysis
		Predicted by RSM	Obtained through Analysis	
Temperature °C	183.7	176.926	176.93	3.68
Equivalent Stress (MPa)	146.02	131.337	133.08	8.86
Weight (N)	70.963	71.05	71.102	-0.1

5. CONCLUSION

A comprehensive transient analysis along with response surface methodology (RSM) was conducted to optimize the geometry of a ventilated brake disc for improved thermal and structural performance. Transient analysis was preferred over steady state as it realistically captures the time-dependent variations in temperature and stress during braking events. Based on temperature distribution, four critical geometrical parameters – inboard and outboard disc thickness, number of vanes, vane thickness, and ventilation gap – were identified as the key factors influencing disc performance. The coefficients of determination (R^2) exceeding 0.96 confirmed the adequacy of the developed RSM model and the regression equations developed for temperature, equivalent stress and weight clearly explain the effect of individual and combination of the parameters on the response functions. Among the selected parameters, the inboard and outboard disc thickness were significant, followed by vane thickness and ventilation gap. Optimization using the desirability function yielded an optimal configuration with 9.497 mm disc thickness**,** 43 vanes, 4 mm vane thickness, and a 5.194 mm ventilation gap**.** The optimized disc exhibited a 3.68% reduction in maximum temperature, an 8.86% reduction in equivalent stress**,** and a negligible change in weight compared to the existing design. These results show that finite element analysis combined with RSM provides an effective and computationally efficient framework for optimization of the brake disc. The proposed methodology not only enhances heat dissipation and minimizes thermal stresses but also improves overall braking reliability under realistic operating conditions. This research contributes to energy sustainability by improving heat dissipation; supports environmental protection through lower emissions; and provides a cost-effective solution through advanced simulation-based optimization. Researchers can extend this study by validating the results obtained using FEA by experimentation. Material optimization can be incorporated for better thermal conductivity and weight reduction. Use of composite materials and hybrid alloys can yield a better strength-to-weight ratio. Additionally, research can be extended by incorporating noise, vibration, and wear parameters for a better understanding of brake disc performance.

CONFLICTS OF INTEREST

The author declares no conflict of interest.

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