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Design & Analysis of Brake rotor for an All-Terrain Vehicle and Selection of brake pads material for better performance

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Abstract **The objective of present work is to design and analyze the brake rotor and material selection of brake pads for a BAJA All-Terrain Vehicle (ATV). In the current study, the structural and thermal analysis, of a slotted disc brake rotor of ATV having three different designs by varying the number of holes and position for better ventilation are conducted using finite element analysis (FEA) software package. Zirconia is found as best material for making brake pads using weighted point method by comparing three different materials. The effective radius of brake rotor was obtained as 32.31 mm and 31.69 mm with clamping force of 5.38 kN and 3.6 kN for front and rear respectively.**

Keywords — All-Terrain Vehicle, Structural and Thermal, Brake pad, disc, alloy, effective radius

I. INTRODUCTION

The brake rotor and pad are one of the important parts in the braking system. Brake pads are actuated by either hydraulic or direct means using wire-lever mechanism to apply the force on the brake pads so that it can come in contact with rotating disc. Thus due to friction between the brake pad material and rotating disc the energy of moving vehicle is dissipated as heat, consequently stopping the vehicle by deaccelerating it. Hence, the design of wheel Braking system is of utmost importance in which the Braking disc play a vital role for the reliability of stopping the vehicle at desired location [1]. Various research has been conducted to understand the performance of the rotor disk and pad of the braking system. Ahmed et al. [2] analysed Bajaj Pulsar 150 standard brake disc and compared it with five differently designed discs and found a weight reduction of 81grams with minute change in maximum stress. Jaenudin el al. [3] has designed and analysed thermal characteristics of electric car brake discs and found higher cooling in ventilated braking discs. Yildiz and Duzgun [4] performed stress analysis of brake disc by changing three ventilation geometries and found maximum stress reduction of 19.1% using ventilated discs.

The brake pad material is one of the critical part of braking system. Various studies were carried out to classify and selection of best braking pad material. Kumar and Kumaran [5] reviewed various materials and formulation used for making brake pads by considering their hazardous effect on the environment, the tribological properties were also discussed while the effect of reinforcements, binders, fillers and friction modifiers on brake pads were explained in the review as well. Kumar and Suman [6] conducted a detailed review to classify brake pad materials as metallic, semi-metallic, non-asbestos organic, non-metallic and ceramics. Blau [7] studied frictional and wear characteristics of Zirconium based amorphous alloy in both dry and lubricated condition and found friction coefficient value as 0.74 for unlubricated amorphous alloy. Further, Blau [8] reviewed and explained the use of coefficient of friction, measurement and use shear localization for explaining the friction resistance for different materials and surfaces.

Bashar et al. [9] used cast iron fillings, silica coconut shell powder and epoxy for manufacturing the cold worked composite brake pad and carried out mechanical and corrosion test. Fellah et al. [10] experimentally studied friction and wear characteristics of Ti-6Al-7Nb alloy to be used in biomedical application and simulated the friction behaviour for total hip prosthesis application. The results showed that Ti-6Al-7Nb ha slower wear resistance than Ti6Al4V. Similarly, Nagesh et al. [11] experimentally studied the friction characteristics of an automobile brake pad by varying the composition of constituents in brake pad material and found the maximum shear strength greater than 100 MPa for a sample having steel and resin as 14.6% and 23.8%. Harmukh et al. [12] experimentally studied friction characteristics of Ni-P based multi walled carbon nanotube and Al_2O_3 nano composite coating and found higher friction in case of carbon nanotube coating as compared to the other. García-León et al. [13] numerically compared aerodynamic behaviour of three brake discs of an automobile and found best results for third disc having slotted chambers. Gigan et al. [14] numerically studied the thermal and mechanical characteristics of the different

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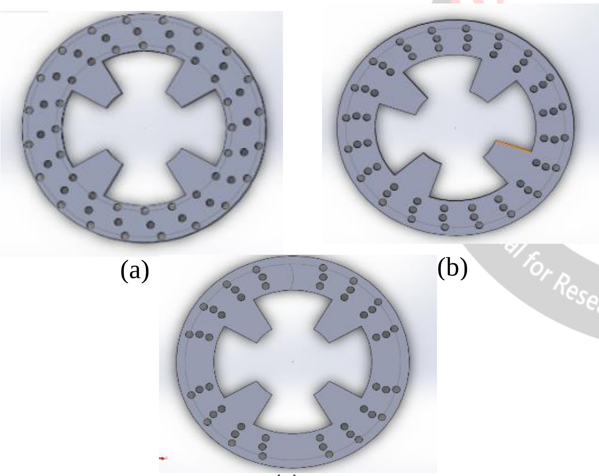
geometries of brake disc for heavy vehicles using FEA tools and Fortran and found a decrease of 13% in mass while considering different pillar arrangement replacing straight vanes with 50% increase in fatigue life of the disc.

Kadhim et al. [15] deduced a qualitative method which is modification for weight property method of material selection for designing a component. Šimunović et al. [16] explained application for both analytical hierarchy process and weighted property method for the selection of material in steel tube machining using six criteria. Similarly, Athawale et al. [17] carried out a comparative study for selection of material using multi criteria decision making method.

Based on the above review, this research takes on designing a brake rotor for an All-Terrain-Vehicle working under specific conditions. The designing process for it will consist of multiple iterations which will be simulated under multiple parameters on CAD and CAE software. Weighted point method for selection of brake pad material not found in literature in vogue. This is because of a wide variety of material present and selection of a particular material becomes cumbersome as a lot of parameter are required to be studied. After thorough research we found some materials on which brake pads are not compared such as Zirconia (ZrO2), Copper Cobalt Beryllium alloy and Ti6Al4V.



(a) (b) (c)

Fig. 1. Physical model of the (a) Design 1, (b) Design 2, (c) Design 3.

II. Geometrical Design of Brake Disc

The braking disc of BAJA all-terrain-vehicles [18] is considered in this study. The geometrical model of the brake discs is shown in the Fig.1. Three designs of disc having different arrangements and positions of slotted holes are made. The hole diameter in each design is considered as 3 mm so that it can be easily manufactured using standard drilling process [19].

The innermost diameter is considered as 33.38mm with middle diameter as 63.38mm and outermost diameter as 93.38mm [19], [20] .Design 1 is made by making three holes at equidistant position along an arc of circle having radius as 16.91mm, and making total 20 such pattern on the pad thus having 60 holes in which three holes are present at 18° to each other. In Design 1, the minimum distance of upper hole centre from outermost periphery of pad is 2.14mm while in Design 2 this distance is changed to 3.5mm. In Design 3, the number of holes are changed to 48 holes, the holes present in 90^0, $180^{0,}$ 270^0 and 360^0 positions are removed in this design.

The Weight ratio of the vehicle is takes as 40:60 with weight as 2450N. The wheelbase length, L = 1447.8mm. The height of centre of gravity of the vehicle body from the ground is 0.500m while the friction coefficient between tires and roads is assumed as 0.7. The diameters of front and rear tyre are 508mm and 533.4mm respectively. The area of master cylinder bore is 284.87 mm^2 while the area of front and rear piston cylinder bore is 490 mm^2. The Pedal Ratio is taken as 5.2. The front and rear axle static load is 100N and 150N. Frictional Force on vehicle, F_f is given as:

$$F_f = \mu_r N = \mu_r mg \qquad (1)$$

Where, μ_r = Friction coefficient for rear tyre

m = Mass of the vehicle

N = Normal reaction for the tyre

Front axle dynamic load, W_{fd} is given as [20]:

$$W_{fd} = \frac{\{W\left(L_2 + \left(\frac{a}{g}\right)h\right)\}}{L} \qquad (2)$$

Where, W= weight of the vehicle

L_1,L_2 are Longitudinal Distance of C.G. from front and rear axle respectively.

Rear axle dynamic load, W_{rd} is given as [21]:

$$W_{rd} = W - W_{fd} \qquad (3)$$

Frictional torque required to stop the vehicle at front wheels, T_f is given as:

$$T_f = \mu_r \times w_{fd} \times R_f \qquad (4)$$

Similarly, frictional torque required at rear wheels, T_r is given as:

$$T_r = \mu_r \times w_{rd} \times R_r \qquad (5)$$

Where, R_f & R_r are front and rear tyre radius

To achieve the correct amount of braking, brakes are biased to 60% in front wheels and 40% in rear wheels.

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Table I. Calculated values of various parameters

Parameters Details	Values
Frictional Force, F_f	1715 N
Front axle dynamic load, W_{fd}	1565.28 N
Rear axle dynamic load, W_{rd}	844.72 N
Frictional torque (front), T_f	278.33 N.m
Frictional torque (rear), T_r	164.73 N.m
Pedal Force	1000 N
Balance Bar Force	5200 N
Actuation Force (master cylinder, front)	3100 N
Master cylinder Pressure (front)	10.98 MPa
Calliper Force (front)	5383.14 N
Friction force on brake pads (front)	4306.51 N
Friction force on brake pads (front, Both wheels)	8613.02 N
Actuation Force (master cylinder, rear)	2080 N
Master cylinder Pressure (rear)	7.32 MPa
Calliper Force (rear)	3588.76
Friction force on brake pads (rear)	2871 N
Friction force on brake pads (rear, Both wheels)	5742 N
Effective radius for front, R_{df}	32.31 mm
Effective radius for rear, R_{dr}	31.69 mm

Braking torque is the product of Frictional force and Effective Radius of the rotor. Now using the calculated parameters as shown in the Table I the design is made and various boundary conditions are applied on it for further analysis.

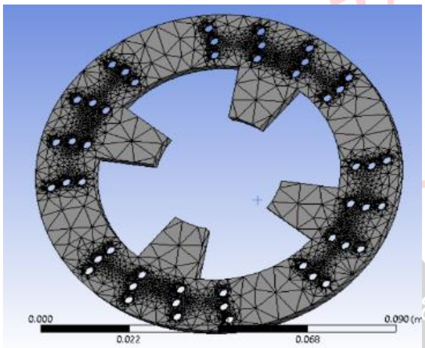

Fig. 2. Meshing for Design 3

III. Numerical Model and Boundary conditions

The model is imported to ICEM where the mesh is generated using medium smoothing; edge sizing is given at the circular holes. The tetrahedral element is taken for the mesh generation. The number of nodes of 24669 with 41607 elements have been generated. The meshing details is shown in fig. 2 which shows that high number of elements are available near holes. The effect due to mounting holes for the brake disc are neglected in the current analysis. The fixed support, clamping force, and rotational velocity have been given at the required positions with centre of gravity of the disc as rotation centre. The clamping force of 2.8 kN with angular velocity of rotation as 214 rad/s is applied to the designed disc. Stainless steel is taken as material which is having 207 MPa as yield strength for the current simulation.

The heat flux is calculated by considering kinetic energy and heat generation rate in the Braking disc [22]. The total kinetic energy of the system is given as:

$$K.E. = \frac{1}{2}\{(MV^2) + \frac{1}{2}(I\omega^2) \quad (6)$$

Where, M = Mass,
V = Velocity,
I = Moment of Inertia
ω = Angular Velocity

$$I = \frac{1}{4} m_d (R^2 - r^2) \quad (7)$$

$$\text{Heat generation (H)} = m_d C_p \Delta t \quad (8)$$

Where, m_d = Mass of disc
C_p = Specific Heat
Δt = Temperature difference

$$\text{Power (P)} = \frac{KE}{t} = \frac{28202.146}{2.2} = 12.82\ kW \quad (9)$$

Where, t = Time taken.
Since only 40% of the mass of the vehicle will be on the rear, the power is reduced and divided = 0.4 x 12.82
= 5.128 kW

$$\text{Heat transfer Coefficient} = \frac{q}{\Delta t} \quad (10)$$

Where, q = Heat flux,
Δt = Temperature difference

Table II. Parameters for thermal analysis

Parameters Details	Values
Mass, M	250 kg
Velocity, V	15 m/s
Moment of inertia, I	28202.15 N.m
Kinetic Energy, K.E.	28125 N.m
Final Temperature, t_f	51.566 °C
Specific heat, C_p	490 J/kg.K
Temperature difference, $\Delta t = (t_f - t_i)$	26.566 °C
Power on each rotor	2.564 kW
Heat Flux	1.28 kW / m²
Heat transfer Coefficient	48.18 W/(m²·K)

Different parameters are calculated as shown in Table II for thermal analysis of the present problem.

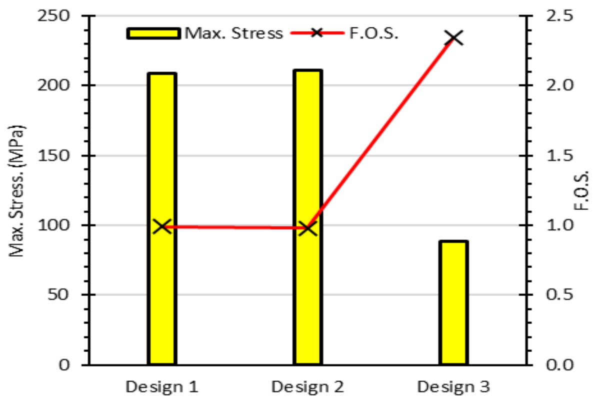

Fig. 3. F.O.S. and Max. Stress values obtained for different designs

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The Factor of safety is defined as:

$$\text{F.O.S.} = \frac{\text{Maximum yield strength of material}}{\text{Maximum stress from testing}} \tag{11}$$

IV. Results and Discussion

A. Static Analysis

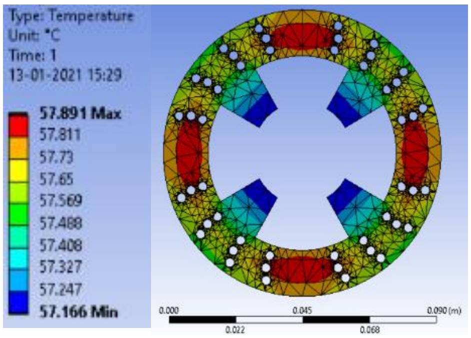

Fig. 4. Stress distribution in design 3

After meshing designed discs are imported one by one in the ANSYS Structural and proper boundary conditions are applied to the corresponding designs. In case of design 1 the boundary, after the analysis, maximum von mises stresses at the periphery of the disc are obtained since only 2.14mm distance between the upper holes and periphery is considered in current design and the stress concentration effect have produce large value of stresses at the location. To decrease the stresses at the periphery of the disc due to stress concentration, the distance is increased to 3.5mm in design 2 and design 3. Different values of maximum equivalent von-mises stress have been obtained for different designs as depicted in the fig.3. The same material testing was simulated for the three designs in which the most robust design comes out to be design 3 as it have highest value of F.O.S. = 2.344. The other designs are likely to fail since the F.O.S. for both the designs is less than unity. The stress distribution for design 3 is depicted in fig. 4 where maximum stresses occur near the holes and four mounting supports. The total deformation for design 3 is shown in fig. 5 in which higher deformation is occurring near the periphery of the disc in the region where holes are absent. The total deformation is shown in fig.5 in which the higher values are found near the region of disc where the holes are absent.

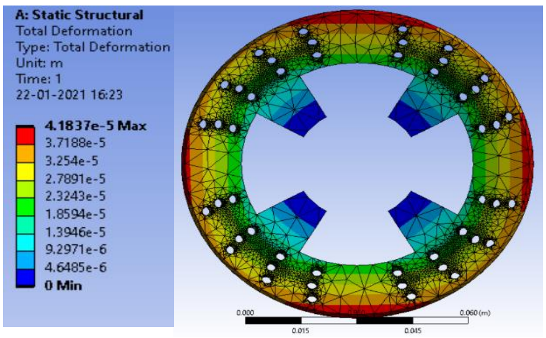

Fig. 5. Total deformation contours for Design 3

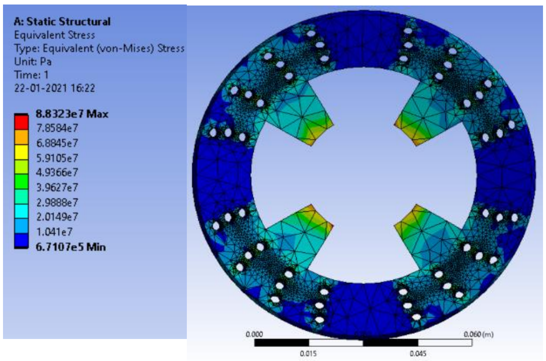

Fig. 6. Temperature Distribution for Design 3

B. Thermal Analysis

A heat flux of 1.28 kW with heat trasfer coefficent of 48.18 W/m^2 is proveded to the disc for analyzing the thermal characteristics of the disc and proper boundary conditions using Table II are applied to the model in the thermal module of ANSYS. Steady state thermla analysis is performed for all the three designed discs.

For design 3, the maximum temperature is obtained as 57.89 ^{0}C as shown in fig.6, the temperature near the holes is lower than the area where the holes are not present, indicating higher convection near the holes and in the inner region of the disc.

V. STUDY OF BRAKE PADS

The brake pads are the heart of Braking system which is used for making contact with the rotating disc for stopping the vehicle when the force is applied on it. Hence, it is necessary to choose the optimum material for manufacturing of braking pads.

The steps used in Weighted-point method for material selection [15], [19]–[22] is summarized as follows:

1. Various properties are studied and a list is made.
2. After than properties of utmost importance are selected.
3. These values are now given weightage according to their importance in the system or product.

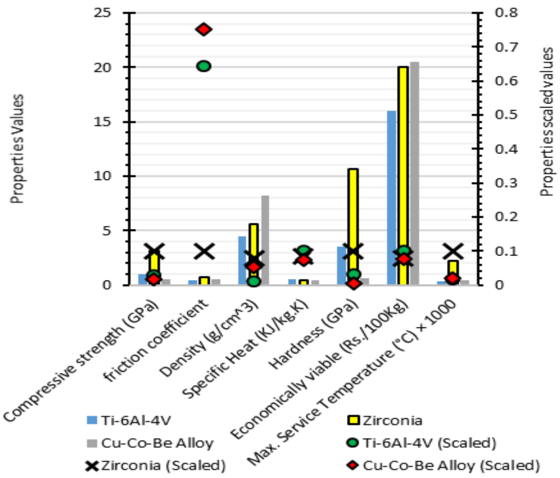

Fig. 7. Different properties for different materials having real as well as scaled values

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The following properties are important while considering a material:

A. Good compressive strength.
B. Higher friction coefficient.
C. Wear resistant.
D. Density
E. Specific heat
F. Optimum hardness.
G. Economic viability
H. Max. Service Temperature

The possible material selection according to maximum service temperature (temperature after which the materials starts fading) are chosen from the following materials:

a. Ti-6Al-4V
b. Zirconia
c. Copper cobalt beryllium alloy

A. Weighted Point Method

The weighted point method is used for determining the appropriate material for the present application of brake pads. This method uses weights for defining the severity or importance of various material properties [15], [19]–[22]. The weights are indication of the effect of the respective properties on the performance, ease of manufacturing and cost, etc. for which corresponding weights are allotted. The properties as well as weights of different materials is shown in fig. 5, the different properties can be easily compared based on the design specifications. Weightage for the different properties is shown in Table III [15].

Table III. Weight values for various properties

Properties	Weighted Factor (α)
Compressive strength (MPa)	5
friction coefficient	6
Density (kg/m^3)	4
Specific Heat (J/kg.K)	7
Hardness (MPa)	2
Economically viable	3
Max. Service Temperature (°C)	1

The scaling up of the values to a factor for making them dimensionless and comparable is done by using following scaling factors:

Scaling factors [15] for properties where lowest value is suitable such as Density, economic viability is given as:

$$\beta = \frac{\text{Lowest value in the list}}{\text{Numerical value of the property}} \times 100 \qquad (12)$$

Scaling factors for properties where highest value is suitable such as Compressive strength, friction coefficient, Specific Heat, Hardness, Max. Service Temperature is given as:

$$\beta = \frac{\text{Numerical value of the property}}{\text{Largest value in the list}} \times 100 \qquad (13)$$

Now, putting the calculated values in the table:

Performance Index, $\gamma = \sum (\beta * \alpha)$

The scaled dimensionless values obtained from Eqs. 12 and 13 are further scaled and shown in the fig. 5 which depicts the property values on primary y-axis and scaled dimensionless values on the secondary y-axis with x-axis showing corresponding material properties.

Table IV. Performance rating of different materials

Materials	Performance rating	Ranking
Ti-6Al-4V	2313.77	2
Zirconia	2725.38	1
Co-Cu-Be alloy	1761.65	3

Hence, it is clear from the Table IV and fig. 5, that the performance rating of Zirconia is high hence it is the best suited material among the three, but if economic viability is given more priority then Ti-6Al-4V will be the best suited among the three.

VI. CONCLUSIONS

In present study, static and thermal analysis of brake disc of an automobile is performed using Finite Element Method. Three brake disc for the automobile are designed by changing the orientation of ventilation holes. The weighted point method is used for selection of appropriate materials by considering the severity of different factors and material properties. The important findings is summarized as:

- Highest factor of safety is obtained for design 3 having the value as 2.344 for the applied boundary conditions, hence it is the optimum design in the analysis.
- The design 3 can handle stresses with intensity of 134.4% more than the allowable stress value for stainless steel.
- Highest deformation of only 41.83 microns is obtained for design 3 near the periphery of the disc where holes are absent.
- The highest temperature of value 57.89 ^{0}C is obtained for design 3 near the region where vent holes are absent.
- The slots and vents made on the disc for heat dissipation should be at least 3mm away from the outer and inner diameter otherwise the total deformation is very high.
- In the current analysis using weighted point method, Zirconia is found as best material for brake pad manufacturing for higher performance which can bear 427% more temperature then Ti-6Al-4V which is one of the best materials for brake pads having 20% lower cost than the former material.

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