Design And Analysis of Brake Disc

Sankara Narayanan.D[1], Sowganth.R[2], Thanis Ahamed[3]
[1, 2, 3] Dept of Mechanical Engineering
[1, 2, 3] Dr. Mahalingam College of Engineering and Technology

Abstract- *This paper presents the design and analysis of brake discs using different profiles. The design of the brake disc is carried out using Creo 5.0 software and the analysis is completed by means of ANSYS Workbench. The aim of this revision is to determine the effect of different profiles on the performance of brake discs. Three different profiles are considered, namely straight, curved and slotted profiles. The design of the brake disc is done by creating a 3D model using Creo 5.0, followed by meshing and boundary conditions setup. The analysis is then performed using ANSYS Workbench, considering various loading conditions such as thermal and mechanical loads. The results show that the curved profile exhibits the best performance in terms of heat dissipation and stress distribution. The study concludes that the use of curved profile brake discs can lead to better braking performance and reduced wear and tear.*

Keywords- Brake Disc, Ansys Workbench, Creo 5.0, Thermal Analysis.

I. INTRODUCTION

The design and analysis of brake discs is a critical area of research in the automotive industry, with implications for vehicle safety and performance. Brake discs are essential components of a vehicle's braking system, responsible for converting kinetic energy into heat energy during braking. The ability of brake discs to withstand high temperatures and repeated cycles of heating and cooling is critical for ensuring safe and reliable braking performance. In this project, we aim to analyze the performance of different brake disc designs using finite element analysis (FEA). By comparing and evaluating the results of various designs, we can identify the most effective and efficient brake disc design for optimal vehicle safety and performance. This project is essential for advancing the field of brake disc design and contributing to the development of safer and more efficient vehicles.

II. METHEDOLOGY

A. Design of Existing Model

The previous version of the brake disc rotor was designed using Autodesk Inventor 2019, based on standard values and analytical measurements available from relevant sources. However, the current version of the brake disc rotor was modeled using Creo 5.0 software, taking into account the necessary design criteria and specifications. The modeling process involved creating a new design that met all the required requirements using Creo 5.0.

B. Validating Existing Model

In regards to the analysis of the brake disc rotor, ANSYS Workbench 19.0 was utilized to carry out the necessary simulations. The software was employed to analyze the brake disc rotor's structural integrity, thermal behavior, and other relevant performance criteria. ANSYS Workbench 19.0 was used to ensure that the brake disc rotor was designed to withstand the intended operating conditions and meet all necessary requirements.

C. Properties of Gray cast Iron

Gray cast iron is a popular type of iron alloy that is widely used in various industrial applications. It is known for its excellent casting properties, high wear resistance, and good machinability. The chemical composition and mechanical properties of gray cast iron are as follows:

Chemical Composition:

Carbon (C): 2.5 - 4%
Silicon (Si): 1 - 3%
Manganese (Mn): 0.5 - 1%
Phosphorus (P): 0.1 - 0.2%
Sulfur (S): 0.05 - 0.1%

Mechanical Properties:

Tensile Strength: 180 - 200 MPa
Yield Strength: 120 - 150 MPa
Elongation: 1 - 3%
Hardness: 180 - 220 Brinell
Compressive Strength: 450 - 550 MPa
Shear Strength: 200 - 250 MPa

D. Structural and Thermal Analysis

The structural and steady-state thermal analysis of the brake disc rotor was conducted using ANSYS Workbench 19.0 software. The structural analysis aimed to determine if the brake disc rotor design could withstand the required loads and stresses without any deformation or failure. On the other hand, steady- state thermal analysis evaluated the brake disc rotor's thermal behavior under operating conditions, considering the heat generated during braking. By performing these analyses, the brake disc rotor's design was optimized to ensure its safe and efficient performance under the intended operating conditions. ANSYS Workbench 19.0 accurately modeled and simulated these analyses to ensure that the brake disc rotor design met the required structural and thermal requirements.

E. Design of Alternative Model

In addition to the previous design of the brake disc rotor, alternative models were also created using Creo 5.0 to explore different design possibilities. These alternative models were designed to meet the necessary design criteria and specifications while also exploring the potential for further improvements in performance. ANSYS Workbench 19.0 was then employed to simulate the structural and thermal behavior of each of these alternative designs to determine their viability. By creating and analyzing multiple alternative designs, the final brake disc rotor design was optimized to meet the necessary requirements and performance criteria while also providing the best possible performance.

III. CALCULATIONS

- For the calculation to be accurate, the dealer's site's initial conditions are used, and the following key assumptions are made:
- It is estimated that the vehicle weighs 300 kg in total.
- The maximum speed of the car is considered to be 110 kph, or v=30.55 m/s.
- The effective radius is assumed to be R eff = 0.12 m, the axial weight distribution is considered to be 0.5, the coefficient of friction is assumed to be 0.5, the kinetic energy to be absorbed is expected to be 0.9, and the standard hydraulic pressure is assumed to be 1 Mpa.
- The friction coefficient, I = O = 0.5, is the same for brake pads and rotors.
- The outside temperature is assumed to be 23 °C.
- The brake pad's total coverage angle was determined to be 43.5°. Because the vehicle's leverage and actuation vary depending on driving conditions, a FOS of 2.5 is taken into account for a single stop surface temperature rise. The vehicle is said to stop using one brake calliper, so the stopping distance is assumed to be 50 m.
- The tangential clamping force (FTRI = FTRO, FRI = FRO) between the brake pad and rotor on the inside is equal to that on the outside.

A. Data for calculation

Total weight = vehicle gross weight + passenger weight
= 150 + 150
m = 300 kg Max. Speed = 110 km/hr
v =30.55 m/s
Axial weight distribution = 0.5 Coefficient of friction = 0.5
Brake pad outer radius Ro = 120mm Brake pad inner radius Ri = 95mm Effective radius Reff = (Ro - Ri)/2
= (120-95)/2
=12.5mm
Reff = 0.12m Ambient Temp = 23℃
Brake pad coverage angle = 43.5º Stopping distance d = 50 m
FOS = 2.5
Hydraulic pressure = 1 Mpa = 1*10^6 n/m

Structural calculations

Brk pad cont area, A = (π (r1) ^2 - π (r2) ^2) * θ/360
= (π (120) ^2 - π (95) ^2) * 43.5/360
= 2040.39 ≈ 2040 mm^2 = 0.00204 m^2
Norm force on inn side, FRI = ((P max/ 2) x A) = ((1 * 10^6)/2 *0.00204)
= ((1000000)/2 * 0.00204)
= (500000 * 0.00204)
= 1000 N
Tang react force on inn side, FTRI = μI x FRI = (0.5 * 1000)
= 500N
Tang react force on outside, FTRO = μO x FRO
= (0.5 * 1000) = 500N
Tang clamping force, FT = FTRI + FTRO
= 500 + 500
= 1000 N
Brake torque, TB = FT * Reff
= 1000* 0.12
= 120 N-m

Thermal calculations

Braking time, d = (u + v) /2 * t 50 = (0+30.55)/2 * t
t = 3.27 s
Kinetic energy, K.E = γ k * (m (u-v) ^2) / 2
= (0.5) (0.9) * ((300) (0-30.55) ^2) /2
= 62997.91 J
Braking power, Pb = K.E / t
= 62997.91 / 3.27
= 14489.44 W

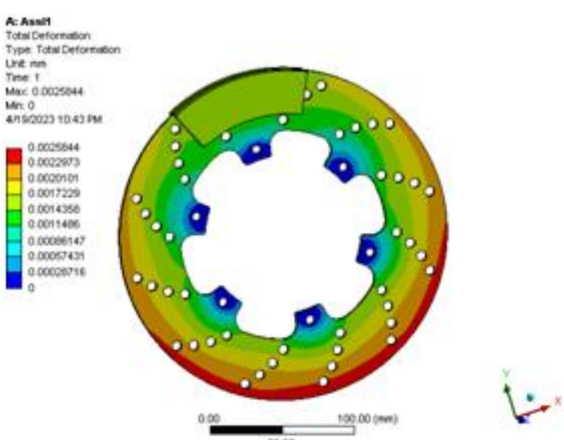
Max. Contact area, $A1 = \pi (r1)^2$
$= \pi * (120)^2$
$= 45238.93 \text{ mm}^2$
Min. contact area, $A2 = \pi (r2)^2$
$= \pi * (95)^2$
$= 28352.87 \text{ mm}^2$
Net disc contact area, $A = A1 - A2$
$= 45238.93 - 28352.87$
$= 16886.06 \text{ mm}^2 = 0.01688 \text{ m}^2$
Heat flux, $q = Pb/A$
$= 14459.44 / 0.01688$
$= 858370.89 \text{W/m}^2$
Max temperature, $T_{max} = 0.527 * q * (\sqrt{t} / \sqrt{(\rho * c * k)}) + T_{amb}$
$= 0.527 * 858370.89 * (\sqrt{3.6} / \sqrt{(6600*460*50)}) + 296$
$= 0.527 * 858370.89 * (1.897 / \sqrt{151800000}) + 296$
$= 818019.77 / 12320.71 + 296$
$= 69.39 + 296$
$= 362.39 \text{ K} = 89.27\ ° \text{c} \approx 89\ ° \text{c}$
Considering, FOS = 2.5
$= 2.5 * 89$
$= 222.5 \approx 250\ °\text{c}$
$T_{max} = 250°\text{c}$

B. Structural Calculations Boundary conditions

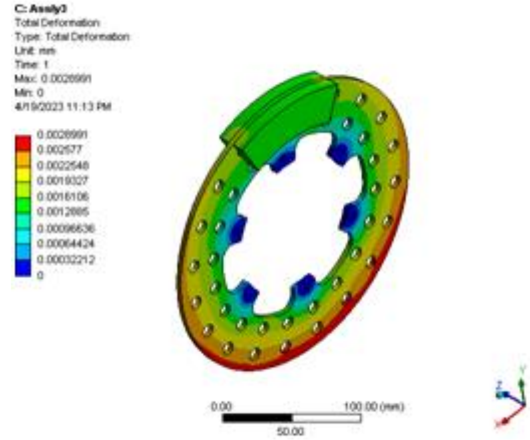
- 6 inner bolt holes - FIXED
- brake torque - 120 N-m
- tangential force - 1000 N (applied usingbrake pads)
- material - grey cast iron
- brake pad displacement - 0,0,free
- mesh – preferred

C. Results – Structural

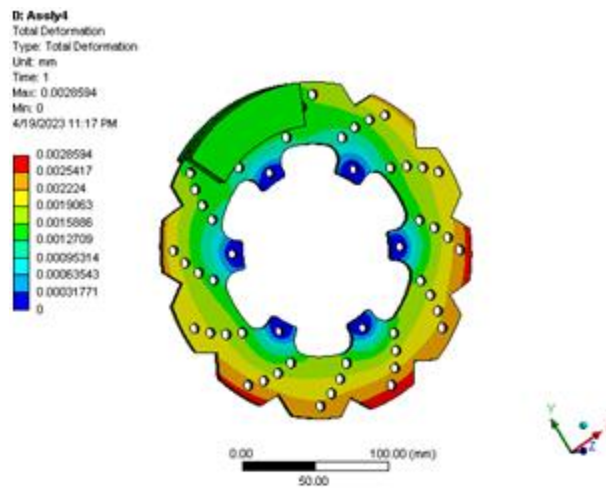
Standard Brake Disc (MODEL-A)



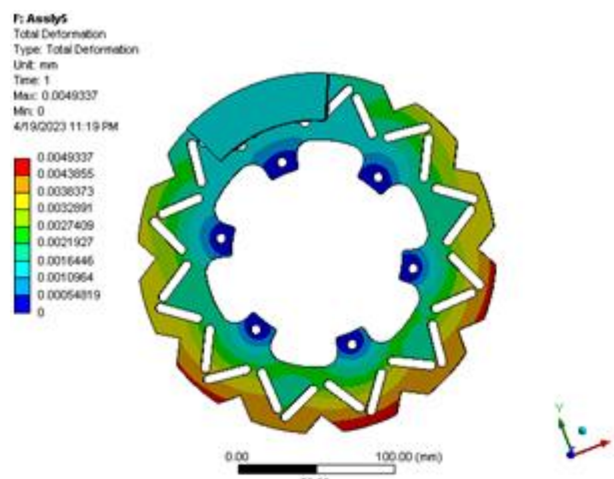
Alternative Brake Disc (MODEL-B)



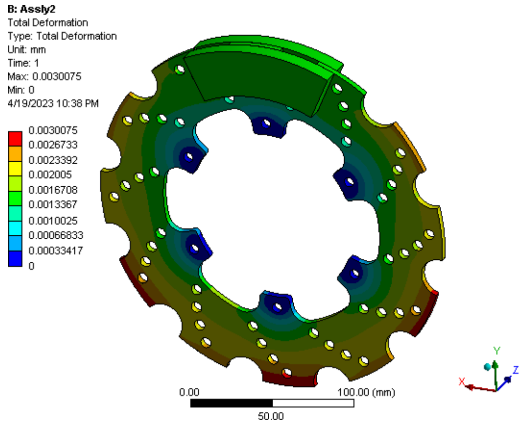
Alternative Brake Disc (MODEL-C)



Alternative Brake Disc (MODEL-D)



Alternative Brake Disc (MODEL-E)

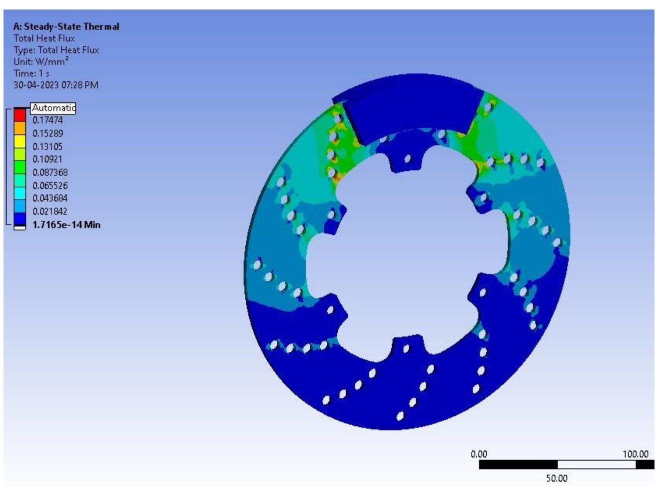


Boundary conditions – Thermal

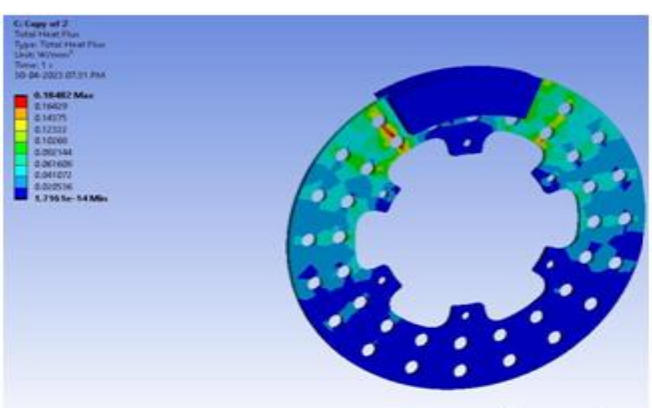
- maximum permissible temperature - 250°c
- ambient temperature - 23°c
- convection - through non-contact area indisc

III. RESULTS – THERMAL

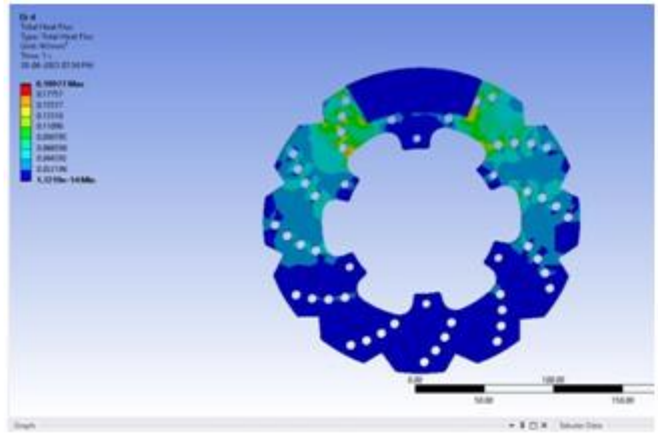
Standard Brake Disc (MODEL-A)



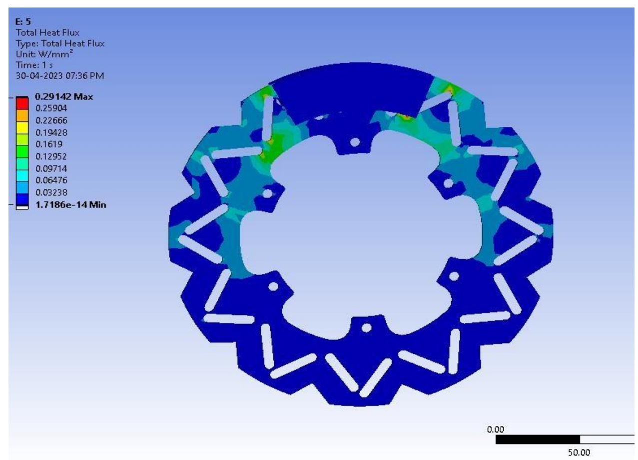
Alternative Brake Disc (MODEL-B)



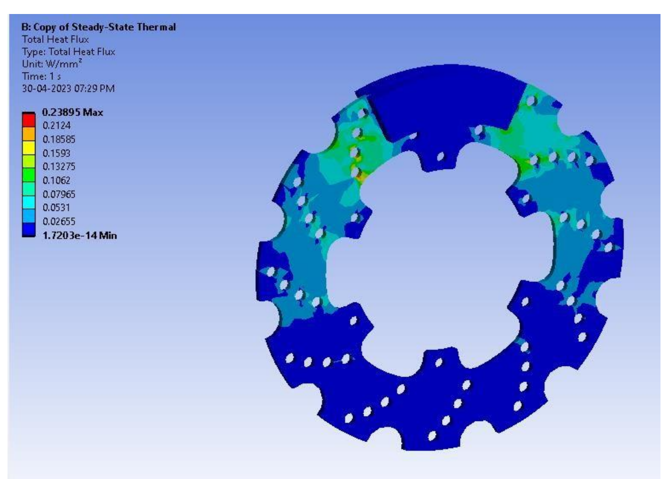
Alternative Brake Disc (MODEL-C)



Alternative Brake Disc (MODEL-D)



Alternative Brake Disc (MODEL-E)



IV. RESULTS

MODEL-A

Parameters	Values
TOTAL DEFORMATION (mm}	0.0025844
EQUALANT ELASTIC STRAIN	0.00016491
EQUALANT STRESS (MPa)	17.848
TEMPERATURE (°C)	250.579
TOTAL HEAT FLUX (W/mm^2)	0.17474

MODEL-B

Parameters	Values
TOTAL DEFORMATION (mm}	0.0028991
EQUALANT ELASTIC STRAIN	0.000163309
EQUALANT STRESS (MPa)	17.85
TEMPERATURE (°C)	251.53
TOTAL HEAT FLUX (W/mm^2)	0.18482

MODEL-C

Parameters	Values
TOTAL DEFORMATION (mm}	0.0028594
EQUALANT ELASTIC STRAIN	0.00018339
EQUALANT STRESS (MPa)	20.044
TEMPERATURE (°C)	251.41
TOTAL HEAT FLUX (W/mm^2)	0.19977

MODEL-D

Parameters	Values
TOTAL DEFORMATION (mm}	|0.0049337
EQUALANT ELASTIC STRAIN	0.00020527
EQUALANT STRESS (MPa)	22.541
TEMPERATURE (°C)	250.41
TOTAL HEAT FLUX (W/mm^2)	0.29142

V. CONCLUSION

Based on the results of the brake disc analysis, we can conclude that Model E is the best option for this application.

Firstly, Model E has a low maximum total deformation of 0.003007, indicating good structural stability and resistance to deformation under load. Its equivalent elastic strain of 0.00016418 and equivalent stress of 17.952 also demonstrate its durability and ability to withstand high stress without undergoing plastic deformation.

Additionally, Model E has a low maximum total heat lux of 0.16764, which suggests good heat dissipation capabilities. This is essential for brake discs, as they are subjected to high levels of heat during operation and need to effectively dissipate it to prevent overheating and subsequent failure.

Finally, the temperature of Model E is relatively high at 252.19, indicating good thermal conductivity and the ability to dissipate heat effectively.

In conclusion, Model E provides a balanced set of results across different parameters, making it a suitable option for brake disc design and optimization. It offers good structural stability, durability, and heat dissipation capabilities, which are all essential for effective brake disc performance.

REFERENCES

[1] Prashant C. Jadhav and Sandip. H. Deshmukh. Simulation and Experimental Investigation of Automotive Disc Brakes for 150CC Pulsar Bike, International Journal of Current Engineering and Technology, E-ISSN 2277 – 4106, P-ISSN 2347 – 5161.

[2] Manjunath T V , Dr Suresh P M. Structural and Thermal Analysis of Rotor Disc of Disc Brake, International Journal of Innovative Research in Science, Engineering and Technology, ISSN ONLINE(2319-8753), PRINT(2347-6710).

[3] Swapnil R. Umale, Dheeraj Varma. Analysis And Optimization of Disc Brake Rotor, International Research Journal of Engineering and Technology, Volume: 03 Issue: 11 | Nov - 2016, e-ISSN: 2395 -0056, p-ISSN: 2395- 007SSS2.

[4] R.S.Kajabe,R.R.Navthar,S.P.NeharkarDesign & Implementation of Disc Brake Rotor By using Modified Shapes. International Journal of Innovative Science, Engineering & Technology, SSSSSSVol. 2 Issue 3, March 2015, ISSN 2348 – 7968.

[5] Ali Belhocine. Mostefa Bouchetara. Thermomechanical modeling of dry contacts in automotive disc brake‖ at International Journal of Thermal science 60 (2012) 161 el 70, 2012 Published by Elsevier Masson SAS.

[6] Jiguang Chen, Fei Gao (2014), Thermo- Mechanical Simulation of Brake Disc Frictional Character by Moment of Inertia, Research Journal of Applied Science Engineering and Technology 7(2), 227-232.

[7] M. Collignon, L.Cristol, P.Dufrenoy, Y.Desplanques, D.Balloy (2013), “Failure of truck brake discs: A coupled numerical– experimental approach to identifying critical thermomechanical loadings, Tribology International 59, 114 –120.

[8] Pier Francesco Gotowicki, Vinzenco Nigrelli, Gabriele Virzi Mariotti, Dr. Cedomir Duboka (2005), Numerical And Experimental Analysis Of A Pegs-Wing Ventilated Disk Brake Rotor, With Pads And Cylinders, 10th EAEC European Automotive Congress.

[9] Adam Adamowicz, Piotr Grzes (2011), Analyzed disc brake temperature distribution during single braking

under nonaxissymmetric load, Applied Thermal Engineering 31, 1003- 1012.
[10] O.P. Singh et al (2010), Thermal seizures in automotive drum brakes, Engineering Failure Analysis 17, 1155–1172.