ISSN Print: 2394-7500
ISSN Online: 2394-5869
Impact Factor: 5.2
IJAR 2016; 2(8): 250-254
www.allresearchjournal.com
Received: 08-06-2016
Accepted: 09-07-2016

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Structural thermal coupled field analysis of disc brakes by finite element method

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Abstract
The frictional heat generated in the disc brake when the rotating disc of an automobile is stopped by the action of the rubber pads affects the temperature profile and stress distribution all along the surface. The widely used disc brake material is Cast Iron, but due to its high specific density, it consumes a lot of fuel during the braking process. The present work is to develop the three dimensional finite element model of a disc brake in order to suggest a suitable alternative for the traditional Cast iron. Three different materials were considered for the present analysis namely, SS 420 Annealed, Titanium alloy and Aluminium alloy. The temperature profile, mechanical deformation and thermal stresses induced are studied for each of the materials. Furthermore, the output of the transient thermal analysis has been coupled with the static structural analysis to study their combined effects on the disc brake geometry. The result from the study reveals that the Aluminium alloy is the best among the three materials considered as it shows the least values of thermal and mechanical stresses induced as a result of the braking. The finite results are validated with the analytical results and found to be close correlation between them.

Keywords: Disc brake, finite element method, coupled field analysis

1. Introduction
The disc brakes, during the course of their operation are subjected to varying environmental and stress conditions. Studies have been made to study the wear mechanisms in the disc and the shoe pads, after running the vehicle under extremely warm conditions [5] and the SEM photographs have shown that the resulting wear differs from those studied under normal operating conditions. Thus, it is required that the disc brakes should have high reliability and high durability. Studies on temperature discrepancies on the contact surfaces of the disc for different combinations of materials of disc brake and pads shows that the largest discrepancy occurred in the case of Aluminium alloy series [6]. Thus, it is of importance to maintain the appropriate levels of friction interface temperature during the braking process to ensure the overall operating effectiveness of the disc brakes. There are different techniques which can be used to study the interface friction temperature distributions [7] and the finite element analysis is one such technique. Disc brakes are generally made up of Cast Iron. Thus, it means that the discs can be damaged in any of the three ways: *scarring, cracking, warping or excessive rusting* [1]. Comparing with various materials such as stainless steel, it has been shown that Cast Iron shows the best results in terms of deformation and stress distributions [2]. But Cast iron, owing to its high specific density, consumes a lot of fuel during the braking process [3], which totally goes against the goal of today's automotive sector, which is fuel consumption. Nowadays, composite materials find application in building of various engineering structures from airplanes to bridges, owing to their enhanced properties [4]. These have excellent specific hardness and good specific strength. Thus, these can be considered as a suitable alternative to the traditional Cast Iron disc brakes. It is to be noted that apart from weight, manufacturing ability and cost, there are several factors to be considered in the design phase of the disc brake. Cast Iron is advantageous in a way that it can be recycled but it offers a serious disadvantage of CO_2 evolution. Also, the brake disc must have enough thermal storage capacity to prevent distortion or cracking from thermal stresses [3]. It is required that the load due to the disc brakes are minimum and the rotor material chosen must have higher resistance to road-salt corrosion. Titanium alloys and their composites serve the purpose as they can potentially reduce the load by 37% to that of the

traditional Cast iron [8]. Commercial Ti alloys and experimental Ti-based hard particle composites were studied for their wear characteristics and resistance to road-salt corrosion. Thus, the use of Cast Iron to make disc brakes is not desirable as it violates the goal of limited greenhouse gases emission and fuel consumption. Hence, the need of the hour is to find suitable alternatives that meet today's need for *Design for Environment (DFE),* i.e., a design process that incorporates both sustainable development and enterprise integration. Hence, three alternatives to Cast Iron have been preferred and checked for their thermal and structural rigidity in this paper.

2. Pressure and power calculation for the disc brake

In order to calculate brake line pressure and the brake force on the pad the following assumptions have been made.

2.1 Assumptions

- The thermal conductivity of the material used for the analysis is uniform throughout.
- The specific heat of the material used is constant throughout and does not change with the temperature.
- The kinetic energy of the vehicle is lost through the brake discs i.e. no heat loss between the tyres and the road surface and the deceleration is uniform.
- Convection is neglected.
- Material should have constant properties throughout.
- It should have the property of material linear expansion coefficient.

2.2 Brake line pressure

Master cylinder is the compartment for the hydraulic oil storage. Human effort is required for pumping the oil. According to Pascal the required pressure in the brake tube is obtained.

Master cylinder diameter (d) = 0.019 m

Area of the master cylinder $A = \frac{\pi \times (d)^2}{4}$. (1)

$A = \frac{\pi \times (.019)^2}{4} = 0.000283385\ m^2$

Line pressure $(\frac{F}{A} \times L.R \times P) = \eta p$. (2)

$P = \frac{445}{0.0002833} \times 3.5 \times 0.8 = 4.39\ MPa$

Hence the line pressure created from the master cylinder is 4.39 MPa

2.3 Brake force on pad

Brake pad is a friction material which offers a dissipation of heat when the pad comes in contact with the rotating disc.

Slave cylinder efficiency (ηc) = 0.98

Brake force on pad F = P × A. (3)

Brake force on pad F= $4.4 \times 10^6 \times 0.0006387 \times 0.98$ =2754.07N

Brake force on wheel = $2 \times F \times (\frac{r}{R})$. (4)

$= 2 \times 2754.07 \times (\frac{0.08}{0.02921})$.

=1508.56 N

2.4 Braking pressure

Brake swept area = 0.1m

$P = \frac{2 \times 1508.56}{0.01} = 301712 = 0.3MPa$

2.5 Generated brake torque

Torque is developed due to the braking force applied on the disc. To obtain the torque value mean effective radius of the disc is considered.

Obtained brake torque per wheel (T) = F× rm. (5)

T = 1508.56 × 0.08 = 120.66 Nm

2.6 Generated brake power

The kinetic energy of the rotating disc is converted in to heat flow.

$BP = \frac{2\pi N T}{60000}$ (6)

Brake power = $2\pi \times 435.78 \times 120.66/60{,}000$ = 4.5KW

3. Development of finite element disc brake

3.1 Development of the geometry

The geometry of the disc brake is developed using CATIA V5. The figure below shows the developed geometry. The developed model is imported to the finite element software and it is discretized into large number of elements. An important requirement is the need to obtain convergence, a point where the accuracy is saturated. Thus the optimum number of elements for convergence comes around 45000-55000 elements.

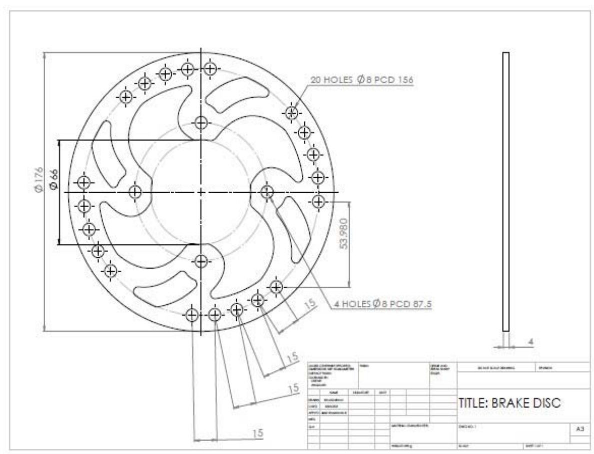


Fig 1: Brake disc Geometry

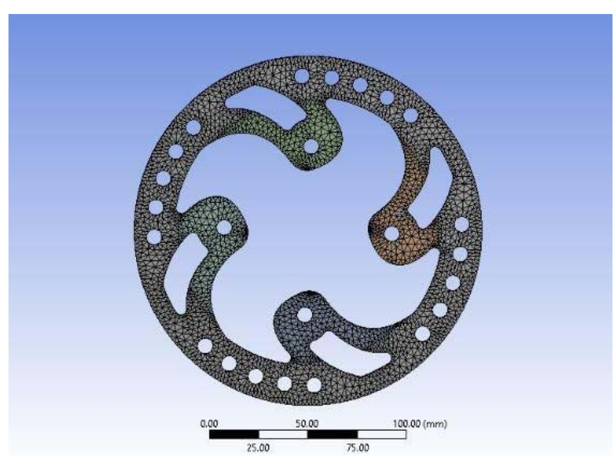


Fig 2: Finite element meshed model of the disc brake

4. Loading Conditions

The rotor is fixed at the bolted ends and breaking pad pressure is applied on both sides of the rotor when breaking force is actually present and the wheel torque is applied to all parts of the rotor since the whole rotor rotates with the wheel.

Considering the following initial conditions, for structural analysis

Pressure: 0.3 MPa

Moment: 1.2×10^5 N-mm

5. Static Structural Analysis

Total Deformation

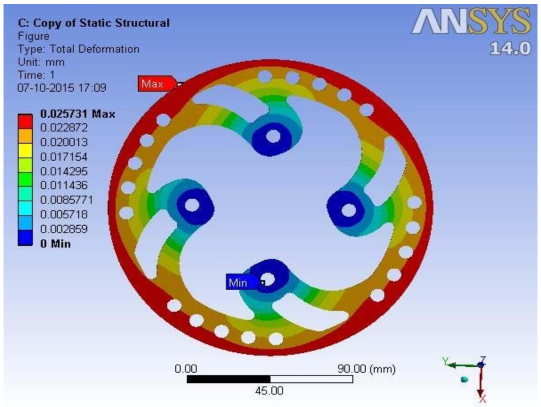


Fig 6: Aluminium alloy Structural Analysis – Deformation

Stress Distribution

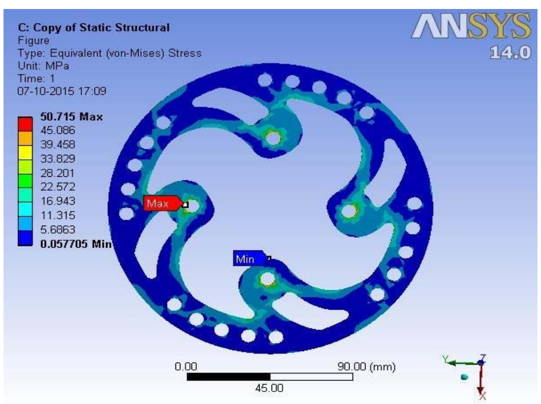


Fig 7: Aluminium alloy Structural Analysis – Stress

The analysis shows that the maximum deformation occurs at the periphery with a value of 0.025731mm and minimum at the bolted ends. The analysis obtained shows that the maximum stress acts at the bolted ends with a calculated value of 50.715 Mpa and the stress is minimum at the regions in the periphery.

5.2 Thermal Analysis

For single stop, the heat generated is 4500 Watts. The heat flow is obtained from the kinetic energy that is possessed by the vehicle at its maximum speed. When braked hard, all the kinetic energy is dissipated in the form of friction. Increase in temperature is sufficient to cause brake fade as the Coefficient of friction decreases with increasing temperature. The heat lost by air convection is neglected.

In case of thermal problems heat flow is considered as the boundary condition Heat flow: 4.5KW

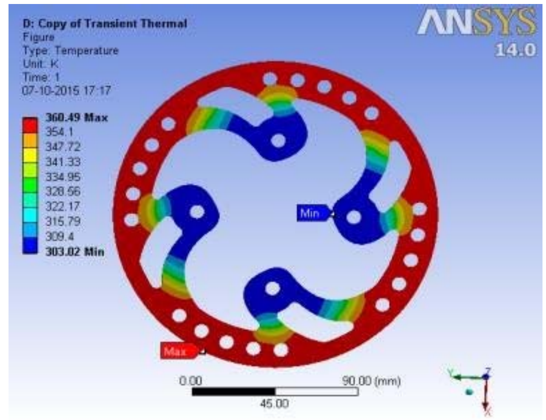


Fig 8: Thermal Analysis – Aluminium alloy

The analysis made shows that the temperature distribution is maximum around the periphery of the disc brake with a value of 360.49K (87.490C).

5.3 Coupled Structural Thermal Analysis

The thermal and the structural analysis are linked together to obtain the coupled structural thermal analysis. The result of the thermal analysis are fed to the structural analysis. Like in FEA, the deformation (strain) are given to the structural system. It will convert that into thermal stress using the material linear expansion coefficient and yield strength value.

Coupled Analysis Showing Deformation and Stress

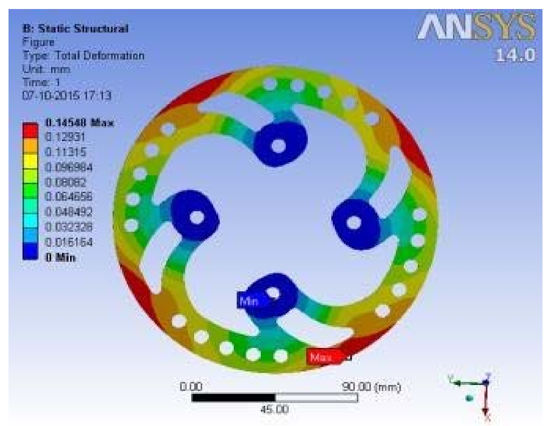


Fig 9: Coupled Deformation -Al Alloy

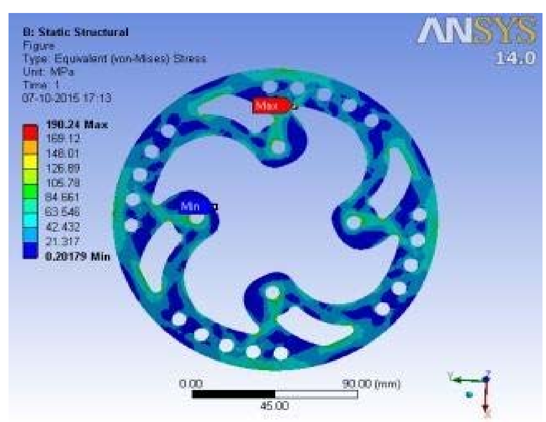


Fig 10: Stress analysis -Al Alloy

The figure shows that the maximum deformation occurs at some regions in the periphery of the disc brake with a value of 0.14548 mm that is much greater than 0.025731 mm obtained for static structural deformation. The value is much less than the clearance between the brake pads and the calliper and hence the brakes will not lock up. It can be inferred from the above diagram that the maximum stress acts at the bolted ends, like in the case of static structural analysis. But the value obtained for the coupled analysis (190.24Mpa) is higher than the value obtained for the structural analysis, which was found out to 50.715 MPa.

6. Validation of analytical model:

6.1 Generated Heat Flux

Brake swept area = 0.01m^2

HEAT FLUX=$\frac{\text{BRAKING POWER}}{\text{BRAKE SWEPT AREA}} = \frac{4500}{0.01} = 450000$Nm/sm^2

In terms of Nm/hm^2

= 450000×60×60

= 1620000000Nm/hm^2

6.2 Simplified Temperature Analysis in a Single Stop

In a single stop with high heat generation, (i.e.) high deceleration levels, the braking time may be less than the time required for the heat to penetrate through the rotor material. Under these conditions no convective brake cooling occurs and all the brake energy is assumed to be absorbed by brake and lining.

For smaller ventilated disc brakes with smaller wall thickness, without ambient cooling.

6.3 Maximum Temperature for Aluminium Alloy

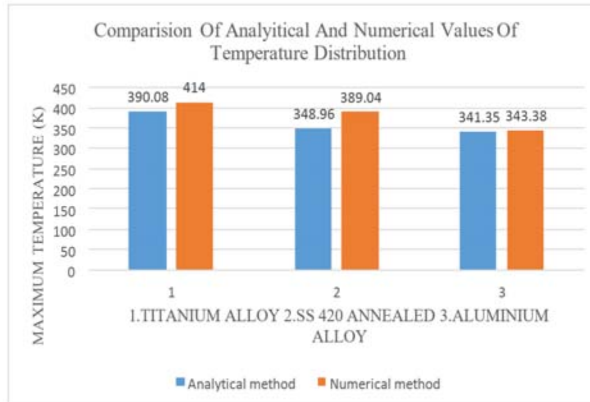
$$T_{max} = T_i + q'' \times (5/18)^{1/2} \times \left(\frac{T}{\rho \times C \times K}\right)^{½}$$

$$T_{max} = 303 + 1620000000 \times (5/18)^{1/2} \times \left(\frac{0.00322}{2770 \times 875 \times 594000}\right)^{½}$$

T_{max}=303+40.38

T_{max}= 343.38K

It can be thus inferred that the value of T_{max} for SS 420 ANNEALED obtained from ANSYS (348.96K) lies within the theoretical value of 389.04K. Similarly, for Titanium alloy, the analytically obtained value of 390.08K lies under the theoretical value obtained for Titanium alloy (414K). In case of Aluminium alloy, the maximum temperature obtained theoretically (343.38 K) is slightly higher than the maximum value obtained analytically (341.35 K).



6.4 Calculation of the percentage discrepancies in the three materials

It has been inferred that Aluminium and Titanium alloys show deviation less than ±10%. The largest discrepancy occurs in the case of SS 420 Annealed. However, it exceeds the tolerance by a very small fraction.

7. Conclusion

The results from the analysis made for the disc brake with four different materials (SS 420 Annealed, Aluminium alloy, Titanium alloy and Cast iron) lead us to make the following conclusions.

Result	Grey Cast Iron	SS420 Annealed	Titanium Alloy	Aluminium Alloy
Total Deformation (mm)	0.061376	0.025731	0.019	0.025730
Equivalent Stress (MPa)	50.66	50.715	50.80	50.71
Max. Temperature (K)	348.96	346.96	368.08	341.35
Coupled Equivalent Stress (MPa)	132.48	224.12	143.01	190.24
Coupled Total Deformation (mm)	0.016376	0.058207	0.06955	0.1458

- The Aluminium Alloys most preferred of all since its stress induced for similar conditions lies within its failure limit (yield value).
- Total deformation of the Aluminium Alloy disc is within the clearance limit, so there is less chance for self-locking.
- The minimum temperature is developed in Aluminium Alloy as compared all three material.
- Also, Aluminium Alloy is readily available in the market and is easily machinable comparatively making it the most preferred of all the three materials considered.
- More over the brake design is safe based on strength and rigidity criteria.
- Cast iron shows the best result as compared but due to its high value of specific density it's not preferred. Now a day's optimization and weight reduction plays a key role in industries.
- Titanium is not preferred due to its expensive material cost.

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