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Thermo-mechanical performance of automotive disc brakes

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Abstract

The kinetic energy of the vehicle is converted into mechanical energy while braking which leads to heat dissipation and temperature rise of the disc and the disc-pads. The aim of this investigation was to study the rise in temperature of an automotive disc brake at the time of braking and its effect on disc durability using finite element method. Application of a specified braking torque on the rotor led to generation of the heat flux. The heat flux generated and the heat transfer coefficient taken into consideration were numerically analyzed, which were then used to calculate the rotor rigidity, maximum temperature rise on the disc rotor. The rotor was further loaded with thermo-mechanical cyclic stresses which were used to analyze the durability and fatigue factor of safety of disc. The influence of variations in disc rotor geometry i.e. holes and airfoil vents in comparison to a simple flange type disc were studied and their effect on maximum temperature rise and disc durability has been investigated by modeling and conducting FEM techniques in Solid works and ANSYS respectively.

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Keywords: Braking torque; disc brake; heat flux; thermo-mechanical analysis; thermal fatigue

1. Introduction

A braking system is one of the most important safety components of an automobile. It is mainly used to decelerate vehicles from an actual speed to a desired speed. Friction based braking systems are still the common device to convert kinetic energy into thermal energy, through friction between the brake pads and the rotor faces [3-7].Disc brakes operate with less fade as compared to drum brakes under the same conditions. An additional advantage of disc brakes is their linear relationship between braking torque and pad/rotor friction coefficient [3].

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Advantages of disc brakes over drum brakes have led to their universal use passenger-cars and light-trucks. Brake friction materials i.e. brake pads and linings are made from materials which have a high coefficient of friction. The choice of material depends on the braking application, but it needs to absorb and disperse large amount of heat without the braking performance being adversely affected [3]. Thermal analysis is a primordial stage in the study of the brake systems because the temperature determines thermo-mechanical behaviour of the structure [2]. Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperatures during braking may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks and thermally-excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage [1].

During stop braking, the temperature does not have time to be stabilized in the disc. It is essential to evaluate the thermal gradients which require a three-dimensional modelling of the problem. The thermal loading is calculated by a heat flux entering the disc through the brake pads. The large amount of heat generated at the pad/disc interface during emergency braking indisputably evokes non-uniform temperature distributions in the domain of the rotor [2].The energy dissipated in the form of heat can generate rises in temperature ranging from 300˚C to 800˚C [2,7]. The heat quantity in the contact area is the result of plastic micro-deformations generated by the friction forces. Generally, the thermal conductivity of material of the brake pads is smaller than that of the disc. The heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated from this surface is equal to the power friction [2].

The large temperature excursions in disc brakes results in thermal shocks that generate surface cracks and/or large amount of plastic deformation. In absence of thermal shock, a relatively small number of high-g braking cycles create macroscopic cracks running through rotor thickness along the radius of the disk [4]. The disc rotor is subjected to cyclic loads consisting of 2 simultaneous components during braking. There are various studies which can be used to trace the mechanism of failure in a rotor subjected to thermal and mechanical loading. Thermo-mechanical fatigue is a prior mechanism to calculate mode of disc failure [9].

Nomenclature

$q_{\circ}$	average braking power in Watt
$q_{(\circ)}$	average braking power absorbed per hour by one half or one side of one of the front brake
a	vehicle deceleration =7.84 m/s^2
s	tire slip which is defined as the ratio of the difference between vehicle forward speed and circumferential speed to vehicle forward speed = 0.08 [7]
w	weight of the vehicle = (285 x 9.81) N
u	initial velocity of the vehicle =16.67 m/s
A_{m_c}	Area of master cylinder = 0.004906 m²
l_p	Pedal lever ratio = 6:1
η_p	Pedal lever efficiency = 0.8
R_m	radius at which force F_c will be applied [6]
f	Coefficient of friction between the 2 friction surfaces
k_a	Thermal conductivity of air, BTU/h˚f ft
D	outer diameter of disc swept area = 0.160 m
d	inner diameter of disc swept area = 0.116 m
M_f	braking torque
P	pressure produced by pedal in each brake [6]
R_e	Reynolds number
h	heat flux
F_c	Force on each cylinder
F_p	force on brake pedal

2. Methodology

2.1. Numerical Method

2.1.1. Assumptions

- The dynamic weight transfer during braking on the front and rear axle is 60:40 percent of the total vehicle weight respectively.
- Brakes are applied and all 4 wheels come to standstill without brake lock (wheel slip is taken into account).
- The calculations done are for an all-terrain vehicle weighing 285 kg and not for a commercial vehicle.
- Ambient air-cooling is taken in to account along with forced convection where applicable with ambient temperature being constant.
- The kinetic energy of the vehicle is lost in the form of thermal energy dissipated by the disc rotors.
- The tyres come to rest with a consistent deceleration.
- The thermal conductivity of the material used for the analysis is uniform throughout [5].
- Heat flux on each front wheel is applied on one side of the disc on the swept area, in a direction normal to the friction surface [5].
- Radiative heat transfer is included in terms of an equivalent radiative heat transfer coefficient [7].

2.1.2 Average braking power

$$q_{\circ} = \frac{\mathrm{K}(1-\mathrm{s}).\mathrm{u.a.w}.3600}{2(778)} \tag{1}$$

Therefore, $q_{\circ} = \frac{1(1-0.08)\times 16.67\times 7.84\times 275\times 9.81}{2\times 9.81} = 16532.63$ Watt

$q_{(\circ)} = q_{\circ} \times 0.60 \times 0.50 \times 0.50 \times 0.90 = 2231.90$ Watt

Where,
0.60 = weight distribution on front wheels
0.50 = since one front brake rotor is considered
0.50 = since one side of the rotor is considered
0.90 = 10% heat lost (heat lost coefficient)

$$\text{Swept Area of Disc Rotor} = \frac{\pi}{4}(D^2 - d^2) \tag{2}$$

$= \frac{\pi}{4}[(0.16)^2 - (0.116)^2] = 0.0184\ \mathrm{m}^2$

Stopping Time, $t = \frac{\mathrm{u}}{\mathrm{a}} = \frac{16.67}{7.84} = 2.12$ sec.

Final Braking Power, $P = \frac{q(\circ)}{t} = \frac{2231.90}{2.12} = 1052.78$ Nm/sec.

Heat Flux = $\frac{\text{Power}}{\text{Area}} = \frac{1052.78}{0.0184} = 57216.62\ \mathrm{W}/m^2$

2.1.3. Braking force

$$P = \frac{1}{A_{m_c}}\left[\frac{F_p}{2} l_p \eta_p\right] \tag{3}$$

Therefore, $P = \frac{1}{490.625}\,[250 \times 6 \times 0.8] = \frac{1200}{490.625} = 2.445$ MPa

Considering 2 cylinders in parallel, so force on each cylinder, $F_c = \frac{1200}{2} = 600$ N

2.1.4. Braking torque

$$R_m = \frac{2}{3}\,\frac{D_1^3 - D_2^3}{D_1^2 - D_2^2} \tag{4}$$

$R_m = 0.1413$ ($D_1 = 0.164$ m; $D_2 = 0.112$m)

Braking Torque, M_f[6] $= 2\,F_c\,.f\,R_m$ (5)

$= 2 \times 600 \times 0.6 \times 0.1413 = 101.76$ Nm

2.1.5. Convective heat transfer coefficient

Heat transfer coefficient for a disc rotor having laminar type heat flow (Re> 2.4×10e5) is given by [7]:

$H_r = 0.04(k_a/D)R_e^{0.8}$, (BTU/h˚f ft^2) (6)

$= 0.04(0.0148/.525)(2.4)^{0.8}$

$= 106.18$ W/m^{2}K

2.2. Disc rotor design

Disc 1 is a solid disc with pitch circle diameter 0.1 m and a disc pad swept area of 0.0184 m^2 at 0.058 m from the disc centre .The thickness of disc is 0.007 m and the disc is made of gray cast iron [3,7,9] weighing 1.824 kg. Discs 2 and 3 are made from disc 1 with same dimensions but altering its geometry with have holes and airfoil pattern slots. The airfoil blades have the maximum mechanical efficiency in displacing the largest volume of air [3,7]. The weight of disc 2 and 3 is 1.728 kg and 1.574 kg respectively.

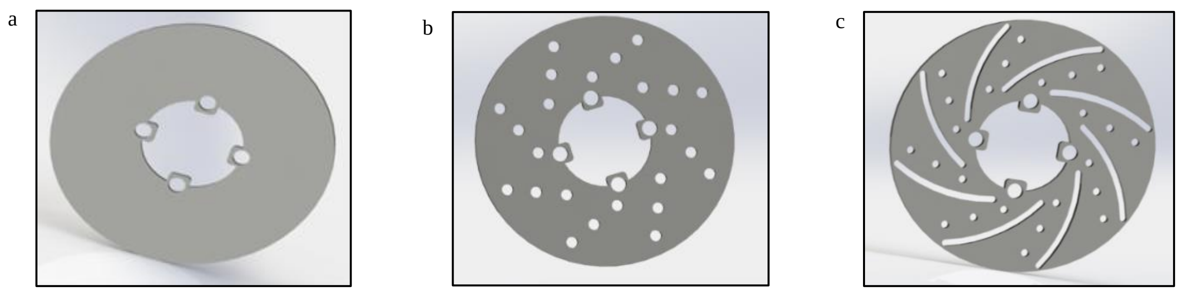


Fig 1. (a) Schematic model of disc rotor 1; (b) Schematic model of disc rotor 2; (c) Schematic model of disc rotor 3

2.3. Finite element method

2.3.1. Structural analysis

The following conditions were applied to determine rotor rigidity at ambient temperature (22˚C):

- Natural boundary conditions: The load applied on the disc rotor included a moment of 101 N-m on one face of the disc i.e. the circular cross-section on which the brake pads are in contact with disc [6].
- Essential boundary conditions: This includes fixture that constraint the bolt holes which fix the disc to the drive hub from moving or rotating, thus limiting the degree of freedom of the nodes in this area to zero.

2.3.2. Steady state thermal analysis

Steady State thermal analysis were performed on all three discs with same boundary conditions. The heat flux equal to 57216.62 W/m^2 was applied on the circular cross-section of the disc in contact with the piston cylinder. The modes of heat transfer i.e. conduction, convection and radiation were taken into account for cooling of the disc brake. The coefficient of conductivity was taken as 52 W/mK and the film coefficient for convection equal to 106.18 W/m^{2}k. The ambient temperature for both convection and radiation was taken as 295 K. While modeling this phenomenon to calculate temperatures, the following heat exchanges were taken into account [7]:

- Convection between disc and ambient air
- Convection between caliper and ambient air
- Conduction between disc hub and disc
- Radiation from the disc

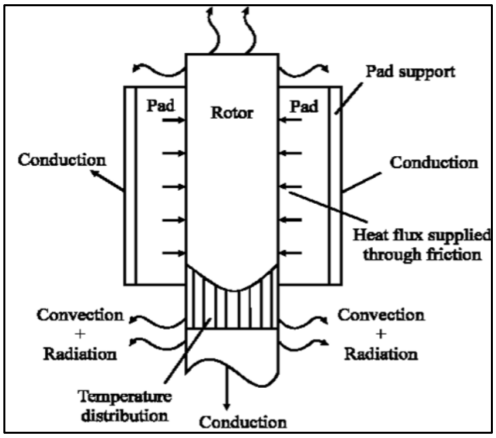


Fig 2.Modes of heat transfer in disc rotor [4]

2.3.3. Thermo-mechanical fatigue analysis

Using Thermal-Structural coupling in the simulation tool, the alternating stresses induced in the rotor due to cyclic thermal and mechanical loads were determined. This was used to evaluate durability and fatigue factor of safety of each disc rotor. The angular velocity of disc rotor as derived from maximum vehicle speed was 57.9 rad/sec. Same braking torque was applied on the swept area in contact with the disc pad. A cut off limit of 10^7 cycles was used for infinite lifecycles [10, 12]. The endurance limit of the material was taken to be 110 MPa [11].

3. Result analysis and discussions

3.1. Influence of geometry change on disc weight

Figure 1(a) shows the solid disc model weighing 1.824 kg, out of which other two disc models have been evolved by making the respective alterations in the geometry. Each hole and airfoil vent has resulted in mass and volume reduction of 0.21% and 1.8% respectively of rotor 1 geometry and hence an overall weight reduction of 5.26 % and 15.18% in disc 2 and 3 respectively as compared to disc 1 due to material removal.

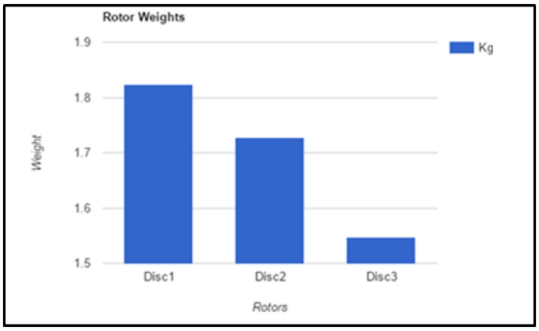
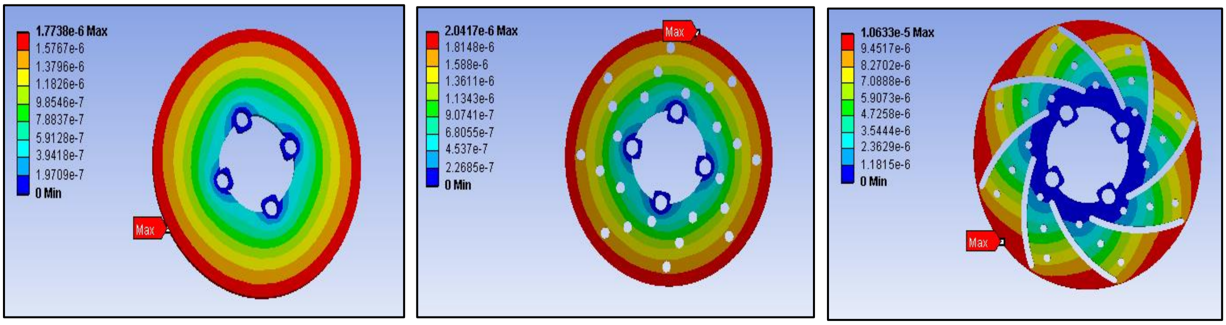


Fig 3.Comparison between rotor weights

3.2. Influence of geometry change on rotor rigidity

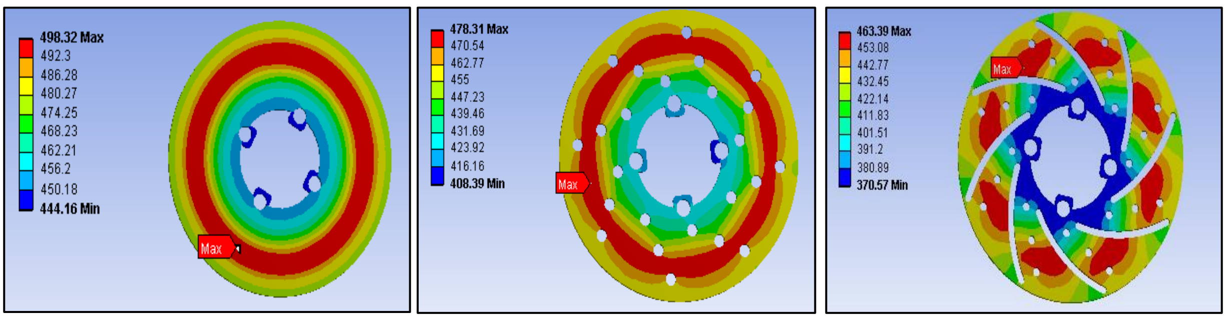


(a) (b) (c)

Fig 4. Total deformation plot in three rotors (in m); (a) Total deformation plot in rotor 1; (b) Total deformation plot in rotor 2; (c) Total deformation plot in rotor 3

A highest deformation is observed in disc 3 scaling from 0 m to 1.0633 e-5 m as depicted in Fig. 4 (c), compared to the other two. The loss in rigidity of rotor accounts for increased deformation due to material removal [5].The maximum deformation among all three rotors being 1.0633e-5 m is below the maximum allowable deformation of 5.0e-4 m for a gray cast iron disc.

3.3. Influence of geometry change on maximum temperature rise



(a) (b) (c)

Fig 5. Temperature contours obtained in three rotors (in K); (a) Temperature contour on rotor 1; (b) Temperature contour on disc rotor 2; (c) Temperature contour on rotor 3

The maximum rise is temperature among all three discs is in case of disc 1 rising from an ambient temperature of 295 K to a maximum of 498.32 K at the pad swept area as shown in Fig. 5(a). The maximum attained temperature in the other two discs is less than the first rotor due to better cooling and heat dissipating characteristics. Disc 2 and 3 have drilled holes and airfoil vents which increase the contact surface area of the disc with the air, thus increasing the rate of heat dissipation through convection [3, 7].

3.4. Influence of geometry change on fatigue

The value of alternating stress induced in each rotor can be seen in the graph below.

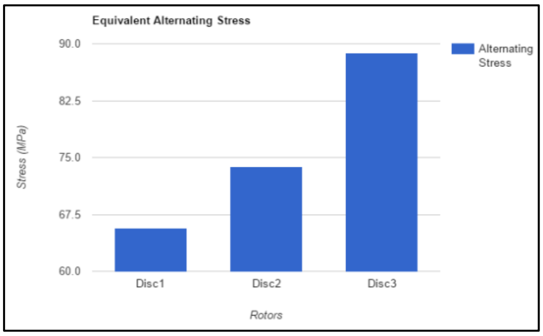
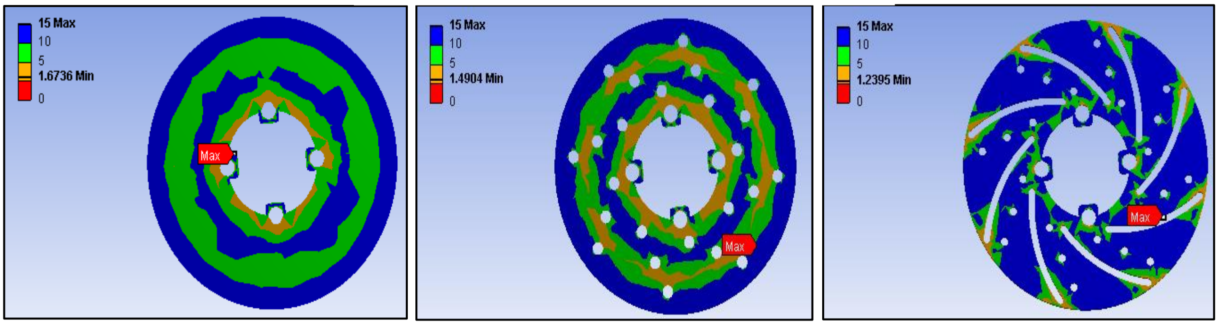


Fig 6. Equivalent alternate stresses induced in rotors



(a) (b) (c)

Fig 7. Fatigue factor of safety plot obtained in three rotors; (a) Fatigue factor of safety plot in rotor 1; (b) Fatigue factor of safety plot in rotor 2; (c) Fatigue factor of safety plot in rotor 3

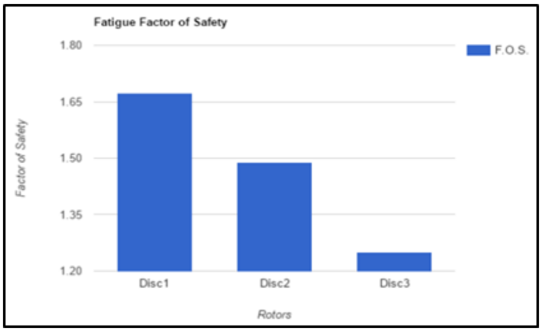


Fig 8. Comparison of fatigue factor of safety

The values of equivalent alternating stress induced in the rotors at given cyclic loading conditions as seen in Fig.6 are less than the endurance limit of material indicating a lifespan of greater than 10^7 cycles (infinite life) [12] in all rotors. The increase in equivalent alternating stress in rotors 2 and 3 can be accounted for the reason that holes and vents are also subjected to heating-cooling cycles at each rotation, this produces thermal differential strains around the holes and the areas around the holes are subjected to thermal fatigue. Therefore, higher values of alternating stresses (tension to compression) are induced in these regions. Also, the minimum value of factor of safety among all three discs happen to be around holes or airfoil vents because in traction situation, the stress intensification factor on the holes is about 3 times the nominal stress in flange [9], augmenting the process of crack formation and expansion. Thus the fatigue factor of safety of rotor 2 and 3 is 10.9% and 25.9 % less than that of rotor 1 respectively as shown in Fig.8.

4. Conclusive summary

Thus from the above results we can say that geometric design of disc is an essential factor in deciding its thermo-mechanical characteristics. The study suggests and justifies the application of disc flange rotors in areas of heavy braking where a larger braking force is required. Discs with geometric patterns (holes and air foil vents) can be used where faster cooling and lightness in weight is preferable, as desired in the racing automobiles. We can conclude:

- The variability in geometric design i.e. holes and airfoil ventilation plays a very significant role in cooling of the disc in the braking phase. Disc rotor 1 has the maximum weight i.e. 15.18 % greater than the lightest rotor and has maximum rise in temperature i.e. 12.32 % greater than the temperature of disc 3. The less rise in maximum temperature in disc 2 and 3 can be reasoned to maximum increase in area of disc exposed to the atmosphere.
- The geometric variations also effect the total life span of disc brake. A simple flange type disc rotor has a maximum life span and can sustain under greater values of cyclic loads up to 35 % more as compared to rotor 3 with holes and airfoil vents under similar loading conditions. These geometric patterns account for greater values of alternating stress induced in the rotor, causing failure due to crack start and reducing crack proliferation period.

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