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Comparative Analysis of Disc Brake Model for Different Materials Investigated Under Tragic Situations

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Abstract* - The paper focuses on the modelling, analysis and design of disc brake under tragic situations for maximizing the heat dissipation and minimizing the brake release period. The main objective of this paper is to examine and analyze the temperature distribution in disc brake during tragic running condition. Finite Element Analysis (FEA) techniques are used to envisage the temperature distribution and to spot the critical temperature on the disc for diverse materials using analysis software COMSOL MULTIPHYSICS. The brake disc model is analysed for four different materials with simple geometry. The basic features of the disc brake are designed based on the three modes of heat transfer i.e. conduction, convection and radiation and are analyzed using COMSOL MULTIPHYSICS. The results obtained from the analysis illustrate diverse temperature characteristics for each material. Finally, a comparison is made for four different materials and the best material is selected under given constraints by optimizing the rate of heat dissipation and brake release time.

***Keywords*: disc brake, finite element analysis, tragic braking**.

I.INTRODUCTION

A brake system must be tested under tragic situations prior to actual manufacturing. Under testing conditions, the brake system should not only stop the vehicle within the smallest possible distance/time with maximum safety and minimum wear but also facilitate the driver to achieve proper control over the vehicle. Thus, the brakes must have effective and prolonged anti-fade characteristics which can be achieved through maximum heat dissipation rate and minimum brake release time/distance.

When the brake is applied, a friction force is generated which at its turn generates a friction torque on the disc. The sliding friction force (on the turning disc surface) generates a work which is converted in heat. On one side the torque brakes the moving mass and on the other side the heat enters the disc and increases its temperature. Usually pads are made of materials which have a low difusibilty so it is assumed that most of the energy enters the disc. So, the disc must have made of the material so that it can absorb almost all energy without a temperature rise in the pad surface (increase excessive of wear). The disk material must have a difusibilty high enough in order to avoid thermal stresses which can crack the surface. So selection of the disc material plays an important role in designing the braking system. A number of investigations have been performed on disc brakes for maximizing the heat dissipation rate. Transient temperature field analysis has been offered for a three-dimensional model of a brake with non-axisymmetric geometry [1]. Mathematical results demonstrate that the functional characteristics of the disc brake mainly affect the surface temperature distribution and the maximum temperature which leads to fake thermal elastic strains and impractical contact situations. Investigation of automotive disc brake has been executed for cooling characteristics [2]. The disc brake cooling characteristics have been considered and analysed experimentally and numerically using a particularly developed spin rig and Finite Element (FE) methods and Computational Fluid Dynamics (CFD) methods respectively. A process has been developed on finite element analysis of transient thermo-elastic behaviours in the various models of disk brakes [3]. The numerical simulation for the thermo-elastic behaviour of different disk brakes has been obtained in the frequent tragic condition. The computational results have been demonstrated for the distributions of pressure and temperature on every friction surface between the contacting parts [4]. The parameters affecting the boundary temperature, including the number of braking applications, sliding speed, load and type of friction materials have been studied. An experiment regarding transient analysis of the thermo-elastic contact problem for disk brakes with frictional heat generation has been performed using the finite element analysis (FEA) method [5]. The affect of the material properties on the thermo-elastic behaviour, represented by the maximum temperature on the contact surface has been compared for different types of disk materials found in the literature, such as grey cast iron (grey iron grade 250, high-carbon grade iron, titanium alloyed grey iron, and compact graphite iron (CGI)), Aluminium metal matrix composites (Al-MMC's), namely Al2O3 Al-MMC \and Sic Al-MMC (Ceramic brakes). An investigation of disc brake rotor using finite element analysis has been described [6]. The study has more likely concern of heat and temperature distribution on disc brake rotor. Due to overheating during tragic braking, the brake shoes stop working and in a worst-case scenario, can melt and thus, may result a disaster. Thus, the factors like selection of material, heat dissipation rate and brake release time are important criterions to optimize the performance of disc brakes.

In view of above discussion, the current work aims at analyzing and designing the disc brake of an ordinary car for four chosen materials under panic braking for minimizing the brake release period and maximizing the heat dissipation rate. The specifications of automobile, input parameters and material properties for designing the disc brake are defined in Section II. Section III illustrates the modeling and meshing of disk brake using COMSOL software. Section IV elaborates the mathematical model representing heat generation and heat dissipation rates. Simulation and analysis is performed to study heat generation and dissipation rate and brake release time for different materials, see Section V. Finally, the results are compared and paper is concluded with future scope in Section IV.

II.SPECIFICATIONS AND INPUT PARAMETERS FOR DISK BRAKE

The specifications, input parameters and material properties are defined to study and analyze the performance of disc brakes, as given in Table I, Table II and Table III, respectively. For the analysis, it is assumed that the wheels will not skid during operation.

TABLE I SPECIFICATIONS

Sl. No.	***Parameter***	***Value***
1.	Automobile weight	1800 kg
2.	Speed	25 m/s
3.	Initial brake applying time	2 sec
4.	Number of materials used	Four
5.	Number of brake shoe	Eight
6.	Retardation rate	10 m/s^2
7.	Speed of vehicle after initial applied brakes	5 m/s
8.	Time taken to stop the vehicle after applying brakes	8 sec

TABLE II INPUT PARAMETERS

Sl. No.	*Description*	*Symbol*	*Expression / values*
1.	Initial vehicle speed	v_0	25 m/s
2.	Vehicle acceleration	a_0	-10 m/s^2
3.	Wheel radius	r_{wheel}	0.25 m
4.	Initial angular velocity, disc	w_0	v_0/r_{wheel}
5.	Angular acceleration, disc	A	a_0/r_{wheel}
6.	Vehicle mass	m_{car}	1800 kg
7.	Area of pad	A_{pad}	4770e-6 m^2
8.	Pad's centre of mass radius	r_m	0.1025m
9.	Friction	f_f	$m_{car} * r^2_{wheel}\alpha/(4*r_m*A_{pad})$
10.	Braking time	t_{brake}	2 s

TABLE IIIPROPERTIES OF PAD AND DISC BRAKE MATERIALS

	Disc				*Pad*
Material Properties	*Cast Iron*	*Mild Steel*	*Al Alloy*	*Ceramic*	*Asbestos*
Density (kg/m^3)	7200	7850	2700	3800	2000
Heat capacity (J/(kg*K))	460	560	840	600	935
Th. conductivity (W/(m*K))	50	48	140	27.5	8.7
Emissivity	0.44	0.28	0.40	0.80	0.80

III. GEOMETRIC MODELING AND MESH GENERATION

Assembly of disc brake involves a number of parts, mainly: disc rotor which rotates with the wheel, calliper assembled with steering knuckle and disc pads fixed to the calliper assembly. Modelling of the whole disc brake assembly is performed using COMSOL MULTIPHYSICS. 2D geometric model of disc brake is modelled as shown in Fig. 1. The disc has a radius of 0.13 m and a thickness of 0.020 m. Finite element mesh is generated using tetrahedral elements (22593 elements), see Fig. 2. An automatic method is used to generate the mesh in the present work.

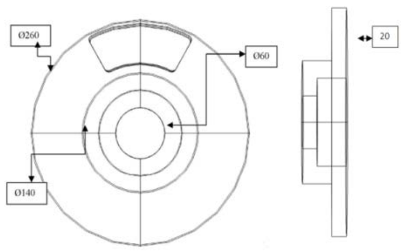


Fig.1 2D Drawing of Disc Brake

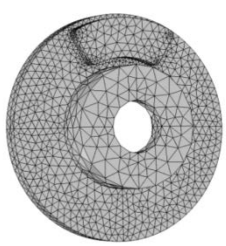


Fig. 2 Meshed Model of Disc Brake

IV. MATHEMATICAL MODEL FOR HEAT TRANSFER IN A DISC BRAKE

All the kinetic energy is assumed to be transformed to thermal energy, apart some energy may be spent as elastic and plastic energy within the brake system. In such cases, the computation of the maximum (allowable) temperature reached by the braking elements requires a complex modeling of the braking system and information about the deceleration rate which obviously is an experimental determination. Thus, scope of the paper is limited to kinetic energy conversion only. However, the complex braking system including elastic and plastic energy transformation may be explored in future.

A mathematical model is presented for effective heat dissipation in the case of disc brakes. The heat power generated per unit contact area is defined. Heat lost in conduction, convection and radiation during braking is defined.

Neglecting drag and other losses outside the brakes, the brakes' retardation power (P) is given by the negative time derivative of the car's kinetic energy, as in (1).

$$P = -\frac{d}{dt}\left(\frac{mv^2}{2}\right) = -mv\frac{dv}{dt} = -mR^2\omega(t)\propto \quad (1)$$

where 'm' is the car's mass, 'v' denotes its speed, 'R' equals the wheel radius, $'\omega'$ is the angular velocity, and $'\propto'$ is the angular acceleration. The angular acceleration is constant in this case and angular velocity is defined in (2).

$$\omega(t) = \omega_o + \propto t \quad (2)$$

where ω_o is the initial angular velocity. By definition, the retardation power equals the negative of the work per unit time done by the friction forces on the discs at the interfaces between the pads and the discs for the eight brakes. Thus, the resultant work is defined in (3).

$$P = -4\iint f_f dA \,.\, v_D = 4f_f(t)\omega(t)\iint r dA \quad (3)$$

$$v_D = v_d e_\emptyset \quad (4)$$

where Velocity vector $'v_D'$ is defined in (4). Symbol '$e_\emptyset$' denotes a unit vector in the azimuthal (angular) direction and the magnitude of 'v_d' at the distance 'r' from the centre is given in (5).

$$v_d(r,t) = \omega(t)\,.\,r \quad (5)$$

The friction force per unit area 'f_f' is approximately constant over the surface and is directed opposite to the disc velocity vector. One can approximate the last integral with the pad's area (A) multiplied by the distance from the centre of the disc to the pad's centre of mass (r_m). Combining the two expressions for P, the magnitude of the friction force can be calculated as given in (6).

$$f_f = -\frac{mR^2\propto}{4r_m A} \quad (6)$$

Under the previously stated idealization that retardation is due entirely to friction in the brakes. The heat power generated per unit contact area at time 't' and the distance 'r' from the centre can be calculated using (7).

$$q(r,t) = -f_f . v_d(r,t) = -\frac{mR^2\propto}{4r_m A}(\omega_o + \propto t) \quad (7)$$

The disc and pad dissipate the heat produced at the boundary between the brake pad and the disc by convection and radiation. Rotation is modeled as convection in the disc. The local disc velocity vector is provided by (8).

$$v_d = \omega(t)(-y, x) \tag{8}$$

This model also includes heat conduction in the disc and the pad through the transient heat transfer expression as given in (9).

$$\rho C_p \frac{\partial T}{\partial t} + \nabla.(-k\nabla \mathrm{T}) = Q - \rho C_p u \nabla \mathrm{T} \tag{9}$$

where 'k' represents the thermal conductivity (W/(m·K)), 'Cp' is the specific heat capacity (J/(kg·K)), and 'Q' is the heating power per unit volume (W/m^3), which in this case is set to zero. At the boundary between the disc and the pad, the brake produces heat according to the expression for q given earlier. The heat dissipation from the disc and pad surfaces to the surrounding air is described by both convection and radiation, see (10).

$$q_{diss} = -h(T - T_{ref}) - \varepsilon\sigma(T^4 - T_{ref}^4) \tag{10}$$

where 'h' equals the convective film coefficient (W/(m^2·K)), 'ε' is the material's emissivity, and 'σ' is the Stefan-Boltzmann constant ($5.67 \cdot 10^{-8}$ W/(m^2·K^4)). To calculate the convective film coefficient as a function of the vehicle speed 'v', (11) is used.

$$h = \frac{0.037k}{l} Re^{0.8} Pr^{0.33} = \frac{0.037k}{l} (\frac{\rho l v}{\mu})^{0.8} (\frac{C_p \mu}{k})^{0.33} \tag{11}$$

where 'l' is the disc's diameter. The material properties i.e. the thermal conductivity (k), the density (ρ), the viscosity (μ), and the specific heat capacity (Cp) are defined for air as the medium.

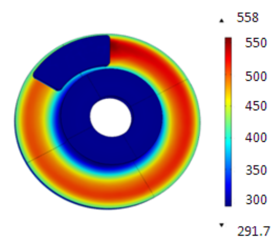
V. FINITE ELEMENT ANALYSIS (FEA) OF DISC BRAKE

Finite element analysis is performed for four models of disc brake, each made of different materials. Analysis illustrates the surface temperature distribution for the disc brake and asbestos pad at the time instantly previous to releasing the brake, temperature distribution profile along the indicated line at the disc brake surface as a function of time, and comparison of total heat produced (solid line) and dissipated (dashed) for each of the four models. All the cases are presented in following sections.

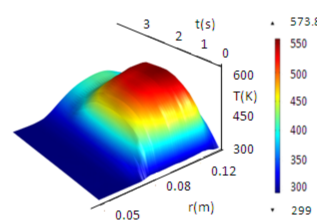
A. Case-I (Mild Steel Disc)

The surface temperatures of disc and pad vary with both time and position. At the contact surface between pad and disc, the temperature increases when the brake is engaged and then, decreases as the brake is released. We can best see these results in COMSOL Multiphysics by generating an animation. Fig. 3(a) displays the surface temperature distribution just before the end of the braking. A "hot spot" is visible at the contact surface between the brake pad and disc, just at the pad's edge. The temperature could become critical during braking at this spot. The temperature decreases along the rotational trace. During rest, the temperature becomes significantly lower and more uniform in the disc and pad.

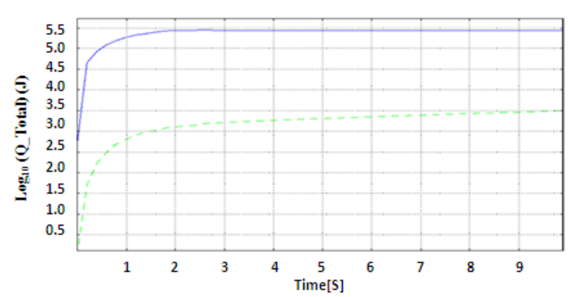
To find the location of the hot spot and the time of the occurrence of maximum temperature, it is necessary to plot temperature versus time graph along a line from the centre to the pad's edge as shown in Fig. 3(b) Maximum temperature spotted in this case is approximately 575.6 K. The hot spot is located near to the radial outer edge of the brake pad. The highest temperature occur approximately 1.2 s after applying the brake. Fig.3(c) shows a plot of the total heat produced and heat dissipated as a function of time. After 8 second of disengagement, the brake has dissipated only a portion of the produced heat. The plot specifies that the resting time must be extended significantly in order to dissipate all the generated heat.



(a)



(b)



(c)

Fig.3 Temperature Distributions for Mild Steel Disc (a) surface temperature of the mild steel brake disc and asbestos pad just before releasing the brake (b) temperature profile along the indicated line at the mild steel disc surface as a function of time and (c) comparison of total heat produced (solid line) and dissipated (dashed)

B. Case-II (Cast Iron Disc)

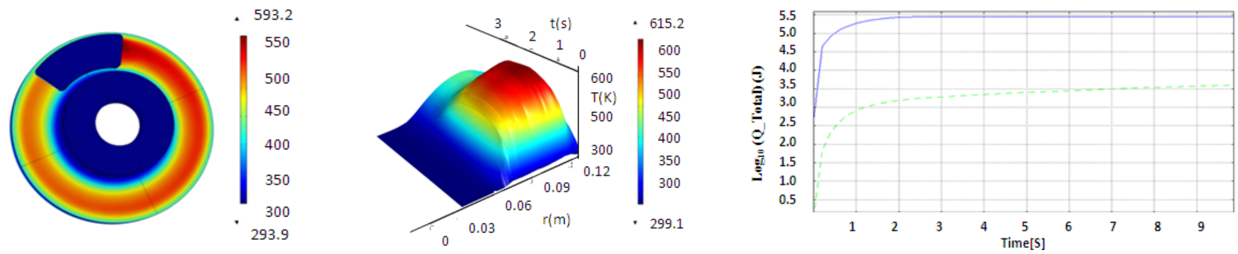
Similarly all the results for cast iron disc are shown in Fig. 4. The maximum temperature is 617.4 K. The heat generated and dissipated is compared in Fig. 4(c).

C. Case-III (Aluminum Alloy Disc)

Results for aluminium alloy disc are shown in Figure 5. The maximum temperature is approximately 569.2 K. The heat generated and dissipated is compared in Fig. 5(c).

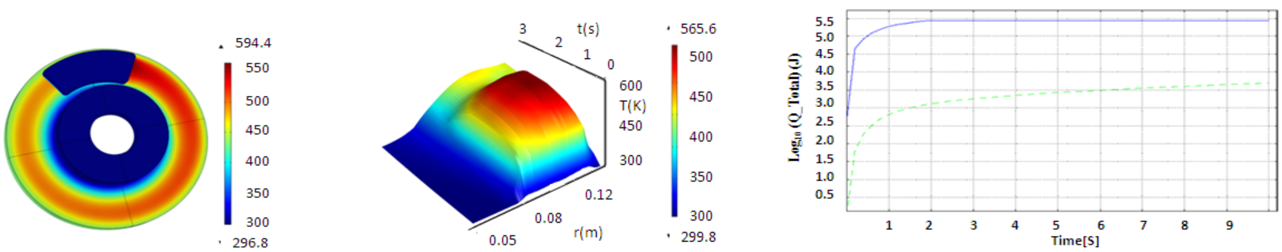
D. Case-IV (Ceramic Al_2O_3 (99.5%) Disc)

Similarly, we can see all the result for Ceramic Al_2O_3 (99.5%) disc in Fig. 6. The maximum temperature is approximately 788.9 K.



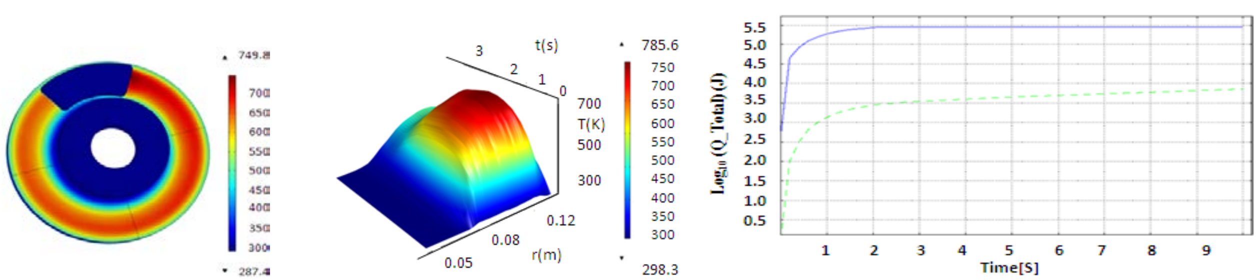
(a) (b) (c)

Fig. 4 Temperature Distributions for Cast Iron (a) surface temperature of the cast iron brake disc and asbestos pad just before releasing the brake (b) temperature profile along the indicated line at the cast iron disc surface as a function of time and (c) comparison of total heat produced (solid line) and dissipated (dashed)



(a) (b) (c)

Fig. 5 Temperature Distributions for Aluminium Alloy Disc (a) surface temperature of the aluminium alloy brake disc and asbestos pad just before releasing the brake (b) temperature profile along the indicated line at the aluminium alloy disc surface as a function of time and (c) comparison of total heat produced (solid line) and dissipated (dashed)



(a) (b) (c)

Fig. 6 Temperature Distributions for Ceramic (a) surface temperature of the ceramic brake disc and asbestos pad just before releasing the brake (b) temperature profile along the indicated line at the ceramic disc surface as a function of time and (c) comparison of total heat produced (solid line) and dissipated (dashed)

VI.COMPARISON OF RESULTS AND CONCLUSION

With reference to above cases, the results of disc brakes for different materials are compared. The critical temperatures for different material disc brakes at a number of time intervals during emergency braking are analyzed. The resultant data is shown in Table IV.

Table IV Critical Temperature of Brake Disc for Different Material at Various Time Intervals

Time in seconds	*1.2 s*	*1.8 s*	*10 s*
Materials	*Temperature in K*		
Mild steel	575.6	558	392.7
Cast iron	617.4	593.2	415.7
Aluminum alloy	569.2	564.4	448.1
Ceramic Al_2O_3 (99.5%)	788.9	749.8	471.3

On the basis of the above results and comparison, it is concluded that:

a) The mild steel has maximum weight among other material and the maximum temperature noted is about 575.6 K, which is significantly low as evaluated and compared in case of cast iron and ceramic but the rate of heat dissipation is less in mild steel.
b) The cast iron has the maximum temperature about 617.4 K, which is low as evaluated and compared in case of ceramic but the rate of heat dissipation is better as evaluated and compared in case of mild steel and aluminium alloy.
c) The aluminium alloy has the maximum temperature about 569.2 K, which is least as evaluated and compared in case of others but the rate of heat dissipation is least as evaluated and compared in case of other materials.
d) The ceramic has minimum weight among other material and the maximum temperature noted is about 788.9 K, which is highest as evaluated and compared in case of other materials but the rate of heat dissipation is greatest in ceramic as evaluated and compared to other materials.

So the cast iron material may be used in brake disc which will give moderate cooling at low temperature as compare to other material. Ceramic has good cooling characteristics but it is costly then the other materials and thus, has applications in sensitive applications i.e. defence, aircrafts, racing cars etc.

In this work a simple model of disc brake is analyzed with simple geometry. Disc brake having complex geometry with different type of designs i.e. ventilated disc, cross drilled disc, slotted disc, spiral finned disc can be analyzed. New materials like composite, heterogeneous, dynamic etc. can be tested to see the effect on the temperature distribution in the brake disc. The potential, elastic and plastic energy transformation may be included for more precise analysis.

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