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DETERMINATION OF HEAT TRANSFER COEFFICIENT OF BRAKE ROTOR DISC USING CFD SIMULATION

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ABSTRACT

Safety aspect in automotive engineering is of prime importance and given priority in design and development of vehicle. This vehicle development needs to meet safety requirement. Braking system along with good suspension systems, good handling and safe cornering, is one of the most critical system in the vehicle. The objective of this work is to design, analyze and investigate the temperature distribution of rotor disc during braking operation using suitable numerical tools. The numerical study uses the finite element analysis techniques to predict the temperature distribution on the full and ventilated brake disc. This analysis is further used to identify the critical temperature of the rotor by varying geometric design of the disc. The analysis also gives us, the heat flux distribution for the disc rotors with various groove patterns. Numerical analysis of the rotor disc of disc brake is aimed at evaluating the performance of disc brake rotor under severe braking conditions and there by assist in disc rotor design and analysis. Hence best suitable design is suggested based on the performance and temperature distribution criteria.

Key words: Braking System, Heat flux distribution, Rotor disc, Temperature distribution, CFD.

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1. INTRODUCTION

Braking system is an energy converting system that converts vehicle movement into heat while stopping the rotation of the wheels. This is done by causing friction at the wheels.

There are two basic types of friction that explain how brake systems work: kinetic or moving, and static or stationary. The amount of friction produced is proportional to the pressure applied between the two objects, the type of materials in contact & the smoothness of their rubbing surfaces. Friction converts the kinetic energy into heat. The greater the pressure applied to the objects, the more friction & heat produced, & the sooner the vehicle is brought to a stop [1].

Belhocine Ali and Bouchetara Mostefa (2013) [3], analyzed the thermo mechanical behavior of the dry contact between the brake disc and pads during the braking phase. The thermal-structural analyze is then used to determine the deformation and the Von Mises stress established in the disc, the contact pressure distribution in pads.

D. Murali Mohan Rao, Dr. C. L. V. R. S. V. Prasad, T. Ramakrishna (2013) [4], studied that the frictional heat generated during braking application can cause numerous negative effects on the brake assembly such as brake fade, premature wear, thermal cracks and Disc Thickness Variation (DTV). He stated that numerical simulation for the coupled transient thermal field and stress field is carried out sequentially by thermal-structural coupled method based on ANSYS to evaluate the stress fields and of deformations which are established in the disc which is another significant. He compared the results obtained by the simulation with those of the specialized literature.

Behnam Ghadimi, Fars Kowsary and M.Khorami (2012) [5], investigated the thermal analysis of the wheel-mounted brake disc R920K for the ER24PC locomotive. The brake disc and fluid zone were simulated as a 3D model with a thermal coupling boundary condition. The braking process was simulated in laboratory and the experimental data was used to verify the simulation results. During the braking, he observed that the maximum temperature is in the middle of braking process instead of the braking end point. Moreover, a large lagging was observed for fins temperature which renders no cooling at the beginning of the braking

2. DESIGN PARAMETERS AND CALCULATIONS

The calculation and verification of braking force is a crucial step in the design process of an automobile as the braking system directly factors as a good control and safety feature in the product. While designing, the main objective is to generate more braking force than ideally required to account for inefficiencies in mechanical linkages and hydraulic systems.

Considering an off-road vehicle, the maximum coefficient of friction between tire and road is 0.7 [1]. The maximum possible deceleration without wheel lockup is thus limited to the 0.7G. Stopping distance for a vehicle should be as minimum as possible without affecting the driver control. The stopping distance depends on the velocity at which vehicle is moving and the maximum possible deceleration.

2.1. Braking Force (F_b)

The braking force is directly proportional to the maximum possible deceleration and the gross weight of the vehicle. The braking force equation can be given as [1]:

$$F_b = 0.7 * \text{Gross weight of vehicle} \tag{1}$$

2.2. Braking Force on Single Wheel (F_{b1})

Dynamic weight transfer in wheeled vehicles is the measurable change of load borne by different wheels during acceleration (both longitudinal and lateral). This includes braking, and deceleration. No motion of the center of mass relative to the wheels is necessary. Hence, the braking force on single wheel is directly proportional to the dynamic weight transfer and the total braking force divided by two since in a four wheel vehicle there are two discs each at the front wheels [1].

$$F_{b1} = \text{Dynamic weight transfer} * \frac{F_b}{2} \quad (2)$$

2.3. Minimum Size of Disc:

The minimum size of disc is calculated by using the following relation [1]:

$$F_{b1} = 2 \times \mu \times \frac{r}{R} \times \frac{A_w}{A_m} \times R_p \times f \quad (3)$$

Where A_m and A_w are area of master cylinder and caliper cylinders respectively,

$$A_m = \frac{\pi}{4} {d_m}^2 \quad (4)$$

$$A_w = \frac{\pi}{4} {d_w}^2 \quad (5)$$

Table 1 Initial Design Parameters

NO	Parameter	Value
1	Gross weight of Vehicle	220 kg
2	Dynamic weight transfer on front	0.65
3	Diameter of master cylinder (d_m)	19 mm
4	Diameter of caliper cylinder (d_w)	28 mm
5	Mechanical leverage (R_p)	4
6	Force exerted on pedal (f)	300N

Substituting the initial design parameters in the above relations, we get

Table 2 Calculated Design Parameters

NO	Parameter	Value
1	Braking force (F_b)	1510.74 N
2	Braking force on front wheel (F_{b1})	491 N
3	Area of master cylinder (A_m)	283.5 mm^2
4	Area of caliper cylinder (A_w)	981.7 mm^2
5	Minimum outer radius of disc (r)	41.88mm

Further, it is necessary to determine and decide the inner and outer diameter of the rotor disc. The inner and outer diameter of the rotor disc depends on the minimum swept area required for applying brakes and the dimensions of the brake pad. It also depends on the assembly constraints of the wheel assembly.

Considering the case of a vehicle having following assembly specifications:

Diameter of the wheel = 22”

For the above specifications and calculated values the rotor dimensions selected are as follows.

Outer diameter of disc (Do) = 160 mm
Inner diameter of disc (Di) = 110 mm

2.4. Required Actuating Force:

Actuating force required to be generated by the brake caliper is the brake force on the single assembly divide by the coefficient of friction between brake pad and disc [1].

$$W_{braked} = \frac{F_{b1}}{\mu} \tag{6}$$

W_{braked} = 491/0.4
W_{braked} = 1227.5 N
As the force is exerted by both the brake pads,
W_{braked} = 2455 N

2.5. Rotor Material

Stainless steel although a little more expensive has a lot more positives. It doesn't rust, or at least not to any great extent. It is very robust; it is tolerant to almost all brake pads and particularly to sintered brake pads. It is highly resistant to wear, it doesn't shatter and it resists heat very well. Hence the material selected for disc is SS410.

Table 3 Properties of SS410

NO	Parameter	Value
1	Density	7.845 g/cc
2	Tensile Strength	1475 MPa
3	Yield Strength	1005 MPa
4	Thermal Conductivity	24.9 W/ m-K
5	Coefficient of Thermal Expansion	9.9 µm/m/°C
6	Specific Heat Capacity	460 J/ Kg-k

3. HEAT TRANSFER COEFFICIENTS

It is used in calculating the heat transfer, typically by convection or phase transition between a fluid and a solid. The heat transfer coefficient has SI units in watts per square meter Kelvin: W/ (m2K).

3.1. Analytical Calculations

Since the forced convection takes place on the contact surface during every rotation of a disc (out of pad area on the rubbing path) as well as on the cylindrical external and internal surface. The convective heat transfer coefficient h is of the form [1]:

$$h = 0.04 * \frac{k_a}{D_o} * Re^{0.8} \tag{7}$$

Where Reynolds number is given by Re

$$Re = {}^{\rho v D_0}/_{\mu} \tag{8}$$

ρ = Density of Material.

v = velocity of air

When Re > 2.4 * 105 the convective heat transfer coefficient h is of the form,

K_a = Thermal conductivity of air

D_o = Outer diameter of disc

3.2. Numerical Analysis

The finite volume commercial CFD software fluent version 15 is used for analysis. This commercial CFD tool is widely used in the numerical simulation of different flow conditions with various complexities. It is chosen in this study because of its proven capability in flows similar to those investigated here. In present simulation the flow fields are calculated by solving the Realizable k-" turbulence model. This involves splitting the geometry into many sub volumes and then integrating the differential equations over these volumes to produce a set of coupled algebraic equations for the velocity components, and pressure at the centroid of each volume. The solver guesses the pressure field and then solves the discretized form of momentum equations to find new values of the pressure and velocity components. This process carries on, in an iterative manner, until the convergence criterion is satisfied. Following methodology is used for CFD analysis

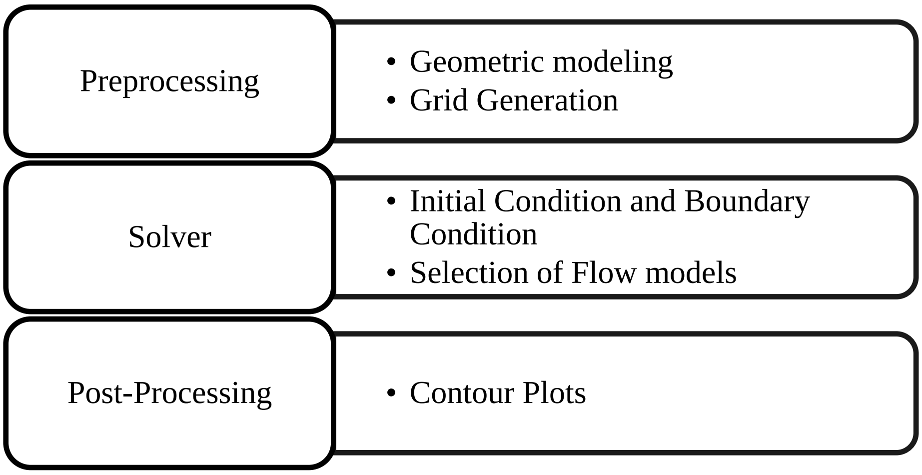

Figure 1 Methodology for CFD Simulation

3.2.1. Assumptions made for CFD analysis

The assumptions made for CFD analysis are as follows.

1. The flow medium is air.
2. Flow is steady and turbulent.
3. Properties of air are taken at standard temperature pressure conditions.
4. 90% of frictional heat transferred to disc.
5. Heat Flux is uniform over pad area.
6. Radiation effects are neglected.
7. Constant properties (density, Conductivity, specific heat, and viscosity)

3.2.3. Modeling of brake disc for CFD simulation

A quarter disc models are created for CFD simulation. As the disc is symmetric about both the axis the quarter disc model provides reasonably correct solution with decrease in the solver time. The enclosure was made around the quarter model taking into consideration that it may not affect the solution. Hence, the grid was sufficient large. The disc cross section does not change along its thickness. Hence, disc was modeled with half thickness. Symmetry boundary condition was used across the thickness so the grid size was reduced to half. At the interface of the disc and surrounding air boundary layer is generated by means of Prisms. Fine mesh is obtained by use of proximity and curvature.

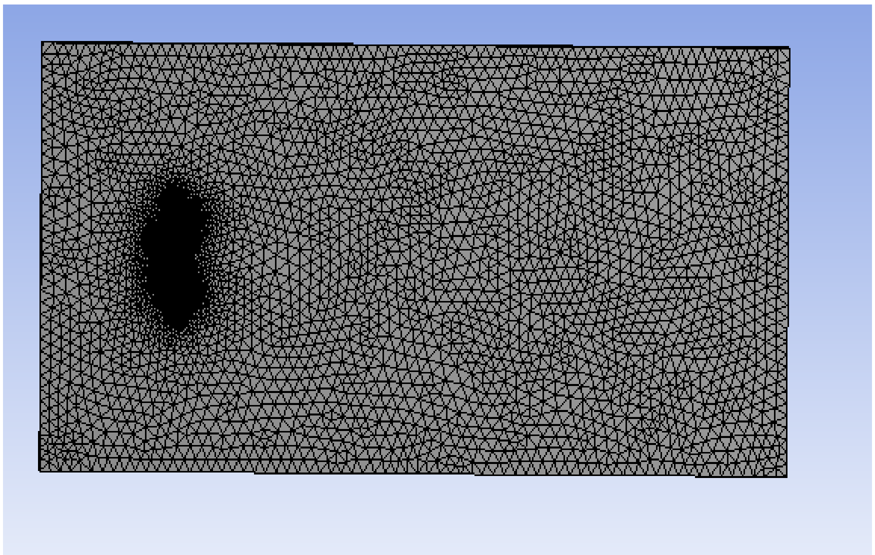

Figure 2 Grid Generation

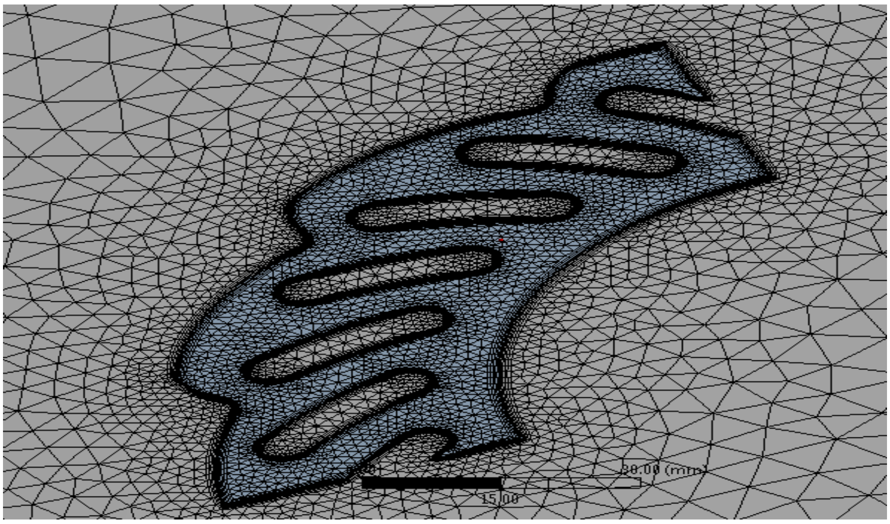

Figure 3 Boundary Layer Generations at Interface

3.2.4. Initial conditions

Considering the standard ambient conditions for temperature as

T (x, y, z) = 27°C at time t = 0

3.2.5. Boundary conditions

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterized by a total heating of the disc and pads during the braking phase. 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 of the disc. We consider that the heat quantity produced will be completely absorbed by the brake disc. The heat flux evacuated of this surface is equal to the power friction. The initial heat flux entering the disc is calculated by the following formula [2]:

$$\mathbf{q} = \frac{1-\emptyset}{2} \times \frac{m * g * v * z}{2 * A * \varepsilon} \tag{9}$$

q = 36504 W/m2

Inlet velocity (v) = 8.33 m/s

Heat flux into disc surface = 36504 W/m2

4. RESULTS AND DISCUSSIONS

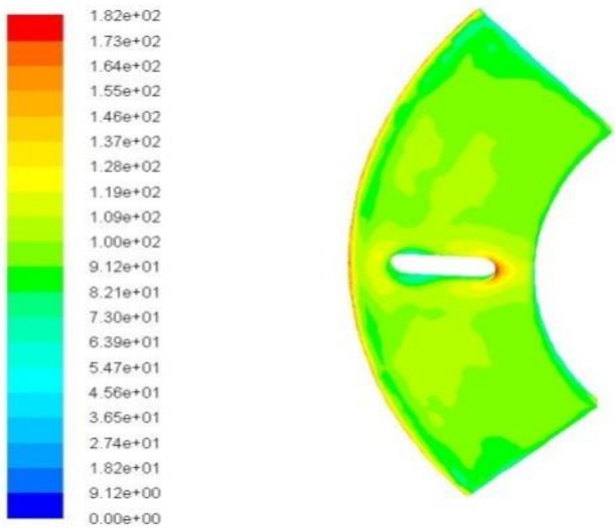

Figure 4 Contour Plots for Disc with 4 Grooves

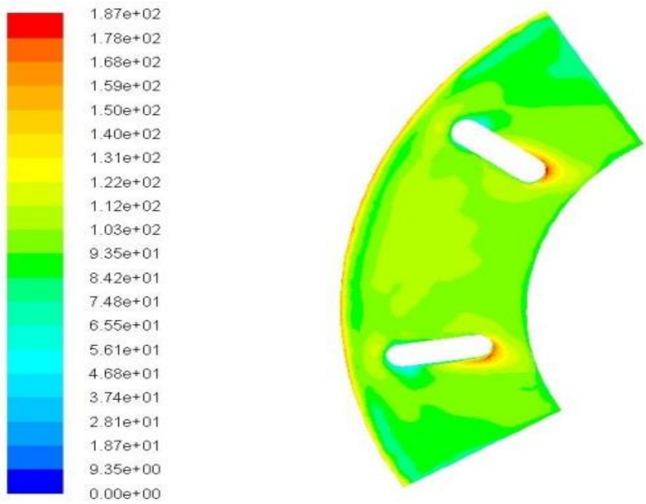

Figure 5 Contour Plots for Disc with 8 Grooves

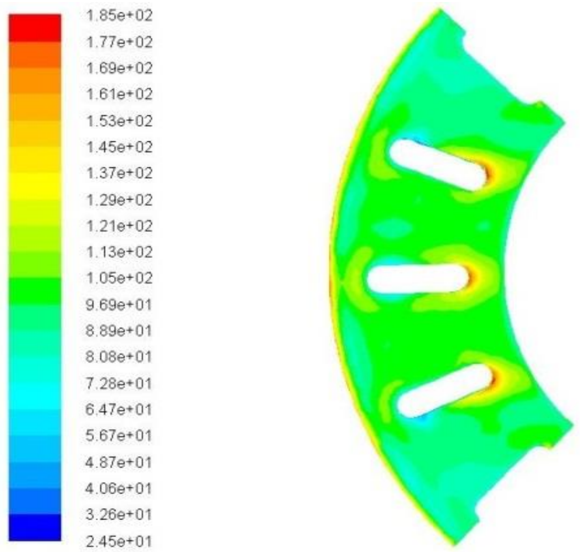

Figure 6 Contour Plots for Disc with 16 Grooves

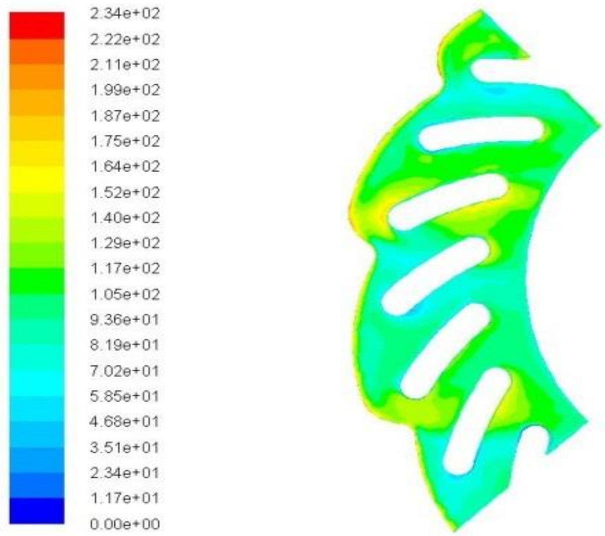

Figure 7 Contour Plots for Disc with 24 Grooves

Table 4 Variation of Heat Transfer Coefficient

Disc with 4 grooves	182 W/m^2K
Disc with 8 grooves	187 W/m^2K
Disc with 16 grooves	187 W/m^2K
Disc with 24 grooves	234 W/m^2K

The above results show that with the provision of grooves improves the heat transfer coefficient at the vicinity of the grooves. Hence, further increasing the number of the grooves enhances the heat dissipation rate. Further, provision of petals over the outer edge of the disc increase the surface area as well improves the heat transfer coefficient.

5. CONCLUSION

The following comments could be concluded:-

1. CFD simulation provides an insight to flow physics at vicinity at effect of the grooves on the heat transfer coefficient
2. Due to provision of grooves there is considerable increase in heat transfer coefficient at vicinity of grooves, increased heat transfer coefficient increases the heat dissipation rate.
3. The analytical solution of heat transfer coefficient and Numerical Solution are Close, the difference in the values is due to provision of grooves which is not taken into consideration while analytical calculation.

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