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Research Article

Simulation and Experimental Investigation of Automotive Disc Brakes for 150CC Pulsar Bike

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Abstract

Various parameters affect performance of disc, out of those three main parameters are pressure applied on pad, coefficient of friction at pad-disc interface and temperature generated due to friction at interface. It was decided to check these parameter effect by simulating them using Explicit Dynamics code LS-Dyna. Simulation is effective in a way as the cycle time for final product design and its validation can be reduced significantly. Also it aids to reduced prototyping cost. Simulation and Experimentation was carried out to monitor the effect of these parameters for structural and thermal analysis of disc brakes. The selected disc brake under study is of Pulsar 150 Bike. The Simulation and Analytical outcomes of the project are presented in the paper along with experimental validation planned in course.

Keywords: *Thermal Analysis, Structural Analysis, Disc brake.*

1. Introduction

Nowadays bikes and cars uses a combination of disc brakes and drum brakes. For car, normally disc brakes are on the front two wheels and drum brakes on the rear wheels and in case of bikes disc brake on front and drum brake on rear. However, the performance of braking system depends on the design and selection of material. Three functions that must be complied by the brakes for all the time are:

1) The braking system should reduce the speed of a vehicle in a controlled fashion and when appropriate brings the vehicle to stop.
2) The braking must allow the vehicle to maintain a constant speed when travelling downhill.
3) The braking system should hold the vehicle stationary when on the flat or on a gradient.

Study of moment of inertia is done to simulate the realistic brake process instead of theoretically predefines the train deceleration rate, nonlinear deceleration rate and thermo-mechanical behavior has been revealed (Jiguang Chen, 2014). A coupled numerical-experimental approach was studied to determine the critical thermo mechanical loadings associated with braking induced metallurgical phase transformations (M. Collignon *et al*, 2013). The results of the braking numerical simulations on a pegs-wing ventilated disk brake rotor, obtained by performing a fading braking procedure and the results are compared with the experimental ones obtained in the same brake, mounted on a work bench, in the same fading braking procedure (Pier Francesco Gotowicki *et al*, 2005). The generated methodology adopted for determining the thermal and structural boundary conditions which is determined experimentally and validated by FEA for a two wheeler drum brake. (O.P.Singh *et al*, 2010). 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). In the past, surface roughness and wear at the pad interface have rarely been considered in studies of thermal analysis of a disc brake assembly using finite element method. The ventilated pad-disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique.

The numerical simulation for the coupled transient thermal field and stress field is carried out by sequentially thermal-structural coupled method based on ANSYS to evaluate the stress fields and of deformations (D. Murali Mohan Rao *et al*, 2013). Temperature fields and structural fields of the solid disc brake during short and emergency braking with four different materials was Investigated. The distribution of the temperature depends on the various factors such as friction, surface roughness and speed. The effect of the angular velocity and the contact

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pressure induces the temperature rise of disc brake. The finite element simulation for two-dimensional model was preferred due to the heat flux ratio constantly distributed in circumferential direction. The values of temperature, friction contact power, nodal displacement and deformation for different pressure conditions were evaluated using analysis software with four materials namely cast iron, cast steel, aluminum and carbon fiber reinforced plastic. Presently the Disc brakes are made up of cast iron and cast steel. With the value at the hand best suitable material for the brake drum with higher life span can be determined (Daniel Das.A *et al*, 2013). A new method to evaluate thermal fatigue by a simulating high-speed braking test using an actual disc brake rotor was studied. Thermal fatigue strength is confirmed to be improved with increasing graphite number in the microstructure. It is also confirmed that the graphite number increases in proportion to the amount of nickel added, and that the inoculation of cerium, a rare earth element, produces an effect similar to that of adding nickel. Based on this approach, a new, low cost material for disc brake rotors for heavy- and medium-duty trucks is developed using both nickel and cerium(Junichiro Yamabe *et al*, 2003). Hence it was decided to carry out analysis to study the effect of pad pressure, coefficient of friction at pad-disc interface and temperature effect due to friction on disc performance.

2. Scope

1) Thermal and Structural Analysis of Disc Brake Rotor.
2) Mathematical model generation for Brake Calculation.
3) Comparison of Mathematical and Simulation model.

3. Modeling and Analysis

It is not easy to exactly model the brake disc, in which still researches are going on to find out brake behavior during braking applications. In modeling we always ignore the things that are of less importance and have little impact on the analysis. The assumptions which are made are given below:

1) Rotor Disc material is assumed as homogeneous and isotropic.
2) The system is assumed as axis-symmetric.
3) Inertia and body force effects are bare minimum during the analysis.
4) The disc is assumed to be stress free before the application of brake.
5) The analysis does not determine the life of the disc brake.
6) Only ambient air-cooling is taken into account and no forced Convection is taken.
7) The kinetic energy of the vehicle is lost through the brake discs i.e. no heat loss between the tyre and the road surface and deceleration is uniform.
8) The disc brake model used is of solid type and not ventilated one.
9) The thermal conductivity and specific heat of the material used for the analysis is uniform throughout and does not change with temperature.
10) No disc wear is considered in analysis.

3.1 CAD Details

The CAD model is created using PRO-E 4.0. The dimensions are measured from the existing disc brake of Bajaj pulsar 150. Pro/ENGINEER Wildfire is a standard 3D product design software, featuring industry-leading productivity tools that promote best practices in design while ensuring compliance with industry standards.

The Disc Brake details are as follows

Material	:	Stainless Steel
Outer Diameter	:	240 mm
Inner Diameter	:	110 mm
Thickness	:	5 mm
Mean Contact Radius	:	107.5 mm
Hole Diameter	:	7 mm

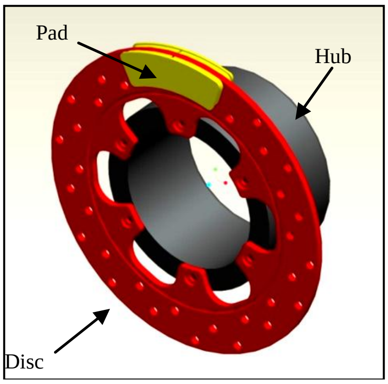

Fig.1 Disc Brake Assembly

3.2 Meshing Details

Mesh generation was done using Hypermesh 11.0. The element types chosen are Hexahedron (brick) element and Tetrahedron element.

Table 1 Mesh Details

	No. of Elements	No. of Nodes
Meshed model	18134	15702

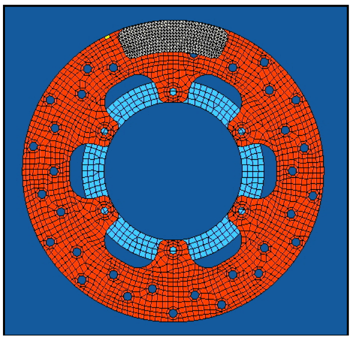

Fig.2 Meshed model of Assembly

3.3 Material Properties

The material properties considered for pad and disc are:

Table 2 Disc Pad Material properties

Density	2500 Kg/m^3
Thermal conductivity	12 W/m.K
Specific Heat Capacity	900 J/Kg.K
Poisson's ratio	0.3

Table 3 Disc Material properties

Property	Material 410
Density	7750 Kg/m^3
Thermal conductivity	43 W/m.K
Specific Heat Capacity	460 J/Kg.K
Coefficient of Thermal Expansion 20 to 200°C 200 to 600°C	 10.5 x 10-6 /°C 11.6 x 10-6 /°C
Modulus of Elasticity	200 GPa
Poisson's ratio	0.3
Shear Modulus	80 GPa
Tensile Strength Ultimate	450 MPa
Tensile Strength Yield	415 MPa

3.4 Evaluation of Input parameters

For evaluation of Input parameters following empirical relationships were used. As per Bureau of Indian standards (IS-13453.1994) the purpose of the Brake Inertia Dynamometer is to conduct various tests for performance and durability for the Two Wheeler Disc Brake System like:

Type D1 Test (Performance Test)
Type D2 Test (Fade Test)
Type D3 Test (Wet / Dry Test)

Type D2 Test:

Requirement of Brake Dynamometer the brake dynamometer shall be capable of adjustment to the inertia settings specified below, the inertia setting for the front and rear wheel shall be obtained by the formula

$$I = \frac{WR^2}{g} \times \frac{1}{N}$$

where,
I = Inertia in kg.m.s^2
W = the mass of vehicle to be simulated in kg = 205 Kg
R = static load radius of tyre of the wheel where the brake
under test is normally fitted in m = 0.3m
g = acceleration due to gravity = 9.81 m/s2
N = number of brakes normally fitted on the wheel = 1

$$I = \frac{205 \times 0.3^2}{9.81} \times \frac{1}{1}$$

$$I = 1.88 \text{ Kg.m.s}^2$$

$$I = 18.44 \text{ Kg.m}^2$$

To simulate the inertia of flywheel the hub density (ρ) was calculated to be 8821.8E+03 Kg/m^3

The units considered are as per the formulation and Simulation software requirement. The simulation parameters were -

Table 4 Simulation Parameters

Description	Notation	Value	Unit
Flywheel Inertia	I	18.44	Kg.m^2
Pressure in fluid system	P	35	Bar
Coefficient of Friction	ν	0.35	-
Radius of wheel	R	0.3	m
Initial velocity of vehicle	V	60	Kmph
Angular velocity	α	55.5	rad/s
Pressure Applied on pad	P_{pad}	2	MPa
Assigned Hub density	ρ	8821.8E+03	Kg/m^3

4. Calculation

For Velocity of vehicle = 60 kmph = 16.67 m/s
Considering deceleration of , g = 4.6 m/s^2
Mass = 205 Kg
1) Time for stopping vehicle

$$t = \frac{v}{g} = \frac{16.67}{4.6} = 3.62\ sec$$

2) Work done against friction or the heat available for dissipation can be calculated by

$$E_{diss} = 0.5 \times I \times \omega^2$$
$$= 0.5 \times 18.44 \times 55.5^2$$
$$= 28.4 \text{ KJ}$$

3) The area of rubbing surface can be calculated as

$$A = \pi\,(R_2^2 - R_1^2)$$
$$= \pi\,(118.5^2 - 93.5^2)$$
$$= 16650.44 \text{ mm}^2$$
$$= 0.01665 \text{ m}^2$$

The rubbing surface is on both sides of brake disc, so

$$A = 2 \times 0.01665$$
$$= 0.03333 \text{ m}^2$$

4) Heat Flux (Q_f)

$$Q_f = \frac{E_{diss}}{t} \times \frac{1}{A}$$

$$Q_f = \frac{28400}{3.62} \times \frac{1}{0.0333}$$

= 235.594 KW/m^2

5) Single stop temperature rise can be calculated as

$$Ts = \left(\frac{0.527 \times Q_f \times \sqrt{t}}{\sqrt{(\rho \times C_p \times k)}} \right)$$

$$Ts = \left(\frac{0.527 \times 235594 \times \sqrt{3.62}}{\sqrt{(7750 \times 460 \times 43)}} \right)$$

$$Ts = 19.08°K$$

Final temperature can be calculated as

$$T = T_s + T_{amb}$$

$$T = 19.08 + 300 = 319.08°K = 46.08°C$$

6) Compressive stress developed in surface due to sudden temperature increase

$$\sigma = \frac{E}{1 - v} \times \alpha \times \Delta T$$

$$\sigma = \frac{200000}{1 - 0.3} \times 10.5 \times 10\text{^} - 6 \times 19.08$$

$$\sigma = 57.24\ MPa$$

5. Results and Discussion

A node was selected at mean radius of contact zone of bake pad with disc (Fig.4) and temperature contour plot was generated (Fig.5)

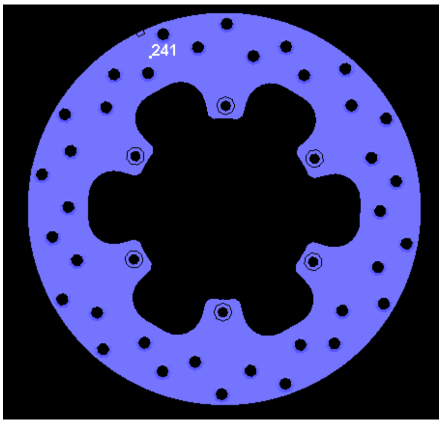

Fig.4 Node location on disc

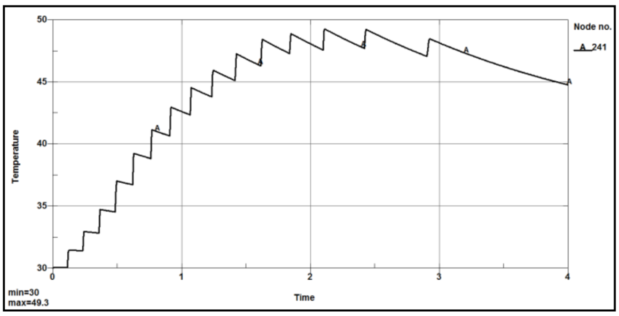

Fig.5 Nodal Temperature Plot

The temperature curves have a saw like shape which starts from the mutual rotational motion of the disc with respect to fixed pad. The presented temperature plot is obtained for certain fixed spot on the circumference of the disc, therefore periods of heating and cooling can be distinguished. The increase of temperature is because of accumulation of the frictional heat. On the other hand when the pad is out of considered spot on the rubbing path the cooling condition is established and the temperature decreases.

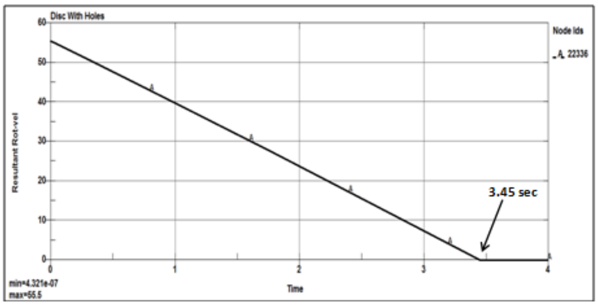

Fig.6 Resultant Rotational velocity of Disc

It can be observed that with application of constant pressure of 2 MPa on brake pad the resultant rotational velocity reduces gradually from 55.5 rad/s to 0 rad/s and stopping time observed is 3.45 sec (Fig.6).

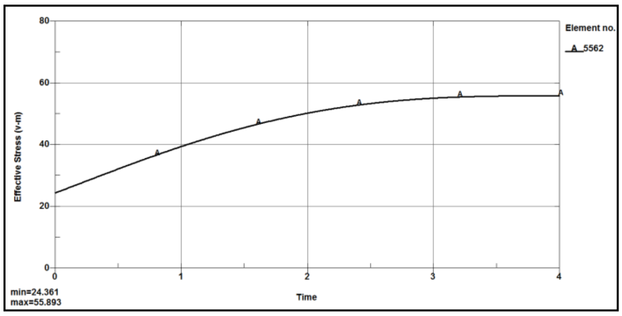

Fig.7 Stress developed due to temperature rise

The stress developed due to sudden temperature increase was measured at max. temperature location across the contact area and the stress value observed was 55.9 MPa (Fig.7).

Table 5 Analysis Result

	Analysis Data	Calculation	Unit
Avg. Stopping Time	3.45	3.62	Sec
Avg. Deceleration	4.83	4.6	m/s^2
Max. Temperature Rise	49.3	46.08	°C
Stress	55.89	57.24	MPa

6. Further work

Testing of disc brake will be performed to study the parameters like stopping time, deceleration, max. temperature at pad-disc interface and these parameters will be compared with analysis results so that analysis methodology can be validated.

Conclusion

Disc brake model Analysis was performed using LS-Dyna solver code for stopping time of disc, deceleration, Max. Temperature at contact surface and stress due to sudden temperature rise. The temperature obtained at contact is computed by the specified angular velocity, pressure applied on brake pad and coefficient of friction at contact interface. The study was performed by considering mentioned loads and boundary conditions and by changing these parameters we can further study its effect on disc brake.

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