International Engineering Research Journal

Experimental stress, Thermal analysis and Topology optimization of Disc brake using strain gauging technique and FEA

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Abstract

Disc brakes are subjected to various thermal and structural loads during operation. Excess wear occurs due to thermal loadings and stress occurring due to application of brake pads onto disc. Optimization of disc brake after investigating of thermal and structural loads is done. Mathematical calculations are used to calculate pressure loading applied due to the brake pads on disc. A test rig simulating brake test environments is made. Similar testing are done using UTM and strain gauging method for experimental stress analysis. Non-contact type of thermocouple is used to measure the temperature. Thermo coupled structural analysis is performed on disc using FEA (ANSYS). Comparison of strain are done to provide validation between experimental and FEA results.

Keywords:Disc brake, Optimization, thermo coupled structural analysis.

1. Introduction

A brake is a device that uses frictional resistance to stop the movement of the machine or the vehicle. Brakes absorb kinetic energy and dissipate it as thermal energy. Brake systems must meet the following requirements:

• The vehicle must be stopped at a minimum distance in case of an emergency.
• Braking properties must not be attenuated with a constant extended application.
• Must have anti-wear properties

The disc brake is an important component of the vehicle's delay system. It is the type of brake that uses tweezers to tighten pairs of pads against a disc to create friction. That friction slows the rotation of an axis (axis of the vehicle) to keep it fixed or to slow down its speed of rotation. The disc brake is usually made of cast iron or ceramic composite such as carbon, aluminum, and silica. The friction material known as the brake pad is made to engage on either side either mechanically or pneumatically. This friction material causes the disc to slow down or stop

In addition to this, macro cracks may appear in the disc brake in the radial direction after some brake cycles. Therefore, the life and performance of the disc brake is affected.

Mainly three types of mechanical stresses are subjected to disk brake.

• The tensile force, caused by the centrifugal effect, occurs when the wheel is rotating and the braking force is applied to the disc.
• Compressive force, when the brake is connected due to the activity of drive, applied by squeezing the cushion opposite to the surface of the circle
•Due to the braking activity brought on by rubbing on the brake cushion against the surface of the circle. It acts the other way of the pivot of the circle.

2. Literature Review

Nakatsuji in 2000 conducted a study on the onset of hair-like cracks that formed around small holes in the flange of a one-piece disc during overload conditions. The study showed that the thermally induced cyclic stress strongly affects the start of the crack in the brake discs. Keeping in mind the end goal to demonstrate the split start instrument, the temperature appropriation in the rib must be measured. Utilizing the limited component technique, the temperature dispersion under over-burden was examined. Flimsy 3D warm exchange examinations were performed utilizing ANSYS. A 1/8 of the one-piece plate was isolated into limited components, and the model had a normal thickness because of symmetry in the thickness heading.

Dufreny and Weichert showed the existence of radial tensile stresses on the surface of the disk by means of perforation of holes. The effect of pressure distribution plays a vital role.Uniform pressure distribution between the pad and the rotor there is uniform wear of the pad and even a coefficient of friction. Then again, the uneven conveyance of the weight can prompt unpredictable wear and its purported plate brake cry. The warmth of rubbing created between two sliding bodies is in charge of the thermo flexible twisting which at last modifies the contact weight conveyance.In order to predict the temperature distribution, much research has been done on the

phenomenon of heat generation between the contact surfaces in the arm.

A.V.Chichinadze in 2001 proposed the three-dimensional computational model of EF for the analysis of temperature in a disc brake system, which explains the dependence of the coefficient of friction on temperature. A significant simplifying hypothesis of this model was the heat partition established as the input parameter in the calculations. In physical sense, it means that the condition of equal temperature of the pad and disc is replaced by the condition of equal or maximum temperatures. From the computational point of view, it leads to separate solutions of heat conduction limit value problems for the pad and the heated disk on the friction surfaces by the thermal flux densities proportional to the specific power of friction. In this case, the heat separation ratio is adjusted a priori. In addition, an integration of the equation of motion and, in turn, the time and the braking distance were performed to a constant (initial) value of the coefficient of friction.

A.A. Yevtushenko developed in 2013 an article on a 3D thermal mechanical contact FE model of a mine lift disk brake to analyse temperature and thermal stress in the coefficient of friction dependent on temperature and slip speed. Static friction coefficients (temperature dependence and contact pressure) and kinetics (temperature and velocity dependence) were applied to the FE contact model of a disc brake. The critical velocity was investigated, in which initiation of the TEI phenomenon is initiated. Separated and solid brake discs were studied to identify the maximum axial distortions and the Huber-Mises equivalent stress. For the purposes of validation, temperature measurements were performed on the contact surface using the infrared camera.

3. Objective

The objective is to design a disc brake with CATIA V5 R20 for the solid modelling of disc brake and to perform finite element analysis (FEA) in the model prepared using ANSYS 17.0. Thus, the von-mises stress, total deformation, thermal stresses and temperature distribution in the disc brake were obtained.

4. Methodology

The CAD model is developed in CATIA V5 R20. The analysis is performed in ANSYS 17.0. Static structural analysis is performed to determine the stress and deformation of optimized model prepared. The best optimized is found compared to the existing model. The thermal analysis is performed to verify the thermal behaviour with change in the surface area.

An experimental analysis of the disc brake rotor will be performed by rotating at constant RPM due to the arrangement of the engine. Therefore, the disc temperature measured by the infrared sensor, which is the type of non-contact sensor, projects the laser beam into the diameter of the region of the original disc brake rotor disc disk and optimized that are decided by CAD and stable temperature distribution.

5. Finite Element Analysis and optimization of disc brake

5.1 Input parameters of disc brake

- Mass of the vehicle = 144 kg
- Initial velocity (u) = 27.7 m/s (100kmph)
- Vehicle speed at the end of the braking application (v) = 0 m/s
- Brake rotor diameter = 0.240 m
- Percentage of kinetic energy that disc absorbs (90%) k=0.9
- acceleration due to gravity g =9.81m/s2
- coefficient of friction for dry pavement μ=0.7

1) Energy generated during braking:

$$k.E = k \times \frac{1}{2} \times \gamma \times \frac{m(u-v)^2}{2}$$

$$= 0.9 \times \frac{1}{2} \times 0.3 \times \frac{264(27.7-0)^2}{2}$$

$$= 13673.10\ J$$

2) To calculate stopping distance:

$$d = \frac{u^2}{2\mu g}$$

$$= 55.86m$$

3) To calculate deceleration time:

$$v^2 = u^2 + 2as \qquad a = 6.86\ {}^{m}/_{s^2}$$

$$v = u + at \quad t = 4.03s$$

4) Braking power:

$$p_b = \frac{k.E}{t} = 3392.08\ W$$

5) Heat flux:

$$Q = \frac{P_b}{A} = 0.03074\ m^2$$

6) Temperature rise due to single braking condition:

$$= \frac{0.527 \times q \times \sqrt{t}}{\sqrt{\rho \times C \times k}} + T_{max}$$

$$= 34\ ℃$$

7) Restoration energy:

$$3\%\ of\ kinetic\ energy = 410.9\ J$$

8) Total energy:

$$k.E + R.E = 14083.19\ J$$

9) Area of contact surface:

$$total\ enrgy \times (R_2 - R_1) = 352.07\ m^2$$

10) Force on disc:

$$= \frac{30\% \times k.E}{2 \times \frac{R_r}{R_t} \times \left(V \times t_s \times \frac{1}{2} \times \frac{v}{t_s} \times t^2 \times s\right)}$$

$$= 3695.15\ N$$

11) External pressure between disc and pad:

$$= \frac{f_b}{\mu \times A} = 0.1709\ Mpa$$

5.2 Original disc brake

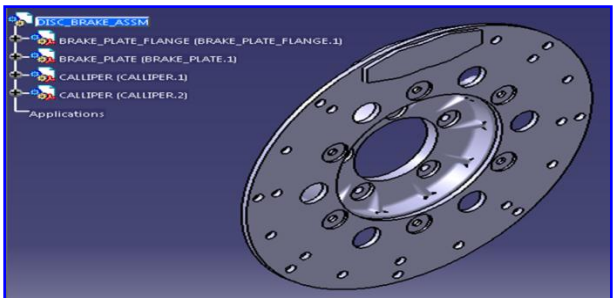

Fig.1 Solid model of disc

Solid model of the disc brake assembly is shown in fig.1, consisting of brake pad along with arrangement of circular holes, with mass of disc 862.29 gm.

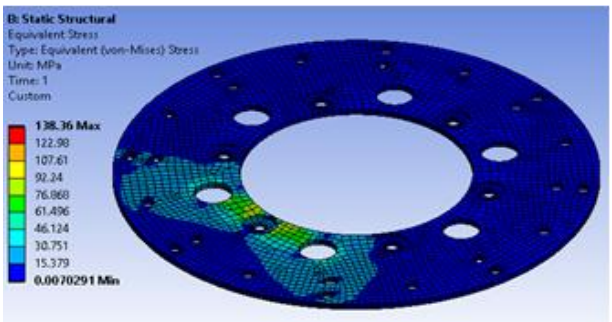

Fig.2 Von-mises stress

Stress generated due to static structural analysis is shown in fig.2 having maximum stress of 138.36 Mpa and minimum stress 0.007029 Mpa.

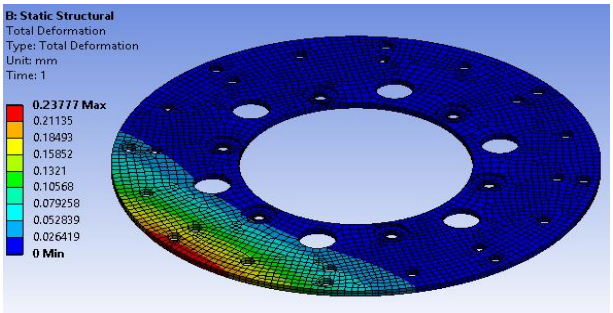

Fig.3 Deformation

Deformation is the result of increasing stress as well as temperature. As shown in fig.3 total deformation is 0.2377 mm and minimum deformation near to 0 mm.

Steady state thermal analysis of original disc

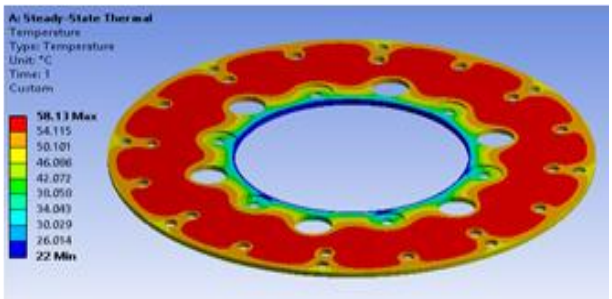

Fig.4 Temperature

Steady state thermal analysis of disc for temperature is shown in fig.4, with Max temperature 58.13℃ and min temperature 22 ℃.

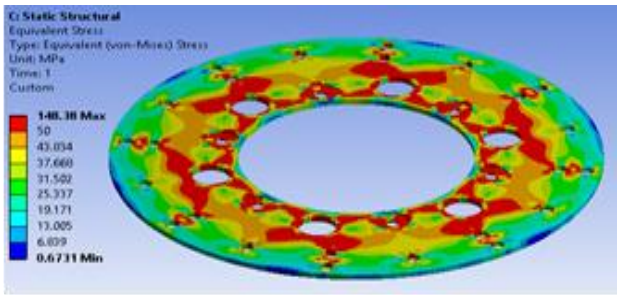

Fig.5 Von-mises stress

Steady state thermal stress is shown in fig.5, with max thermal stress 148.38 Mpa and min thermalstress 0.6731 Mpa.

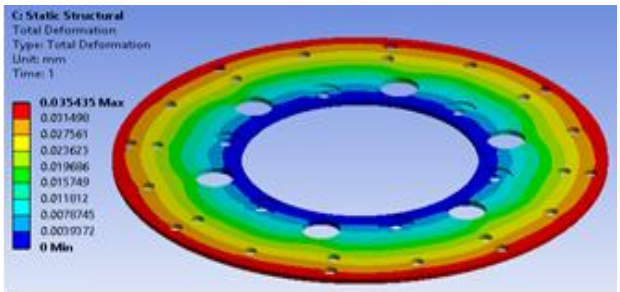

Fig.6 Deformation

Total deformation of disc is mainly on the outer edge of disc as shown in fig.6 with max deformation of 0.03543 mm and min deformation of 0 mm

5.3 Optimized disc brake

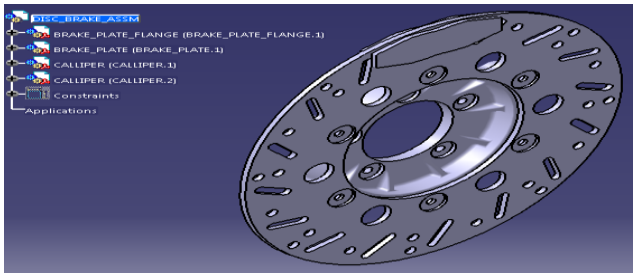

Fig.7 Solid model

Solid Model: Fig.7 is the solid model of optimized disc brake analysis, consisting of additional kidney shaped holes with mass of disc 798.1g

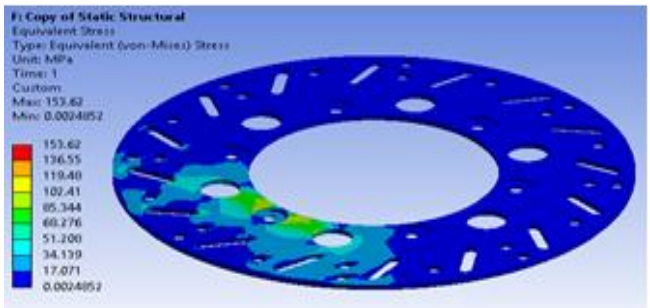

Fig.8 Von-mises stress

Static structural analysis of optimized disc brake is shown in fig.8, having max equivalent stress of 153.62Mpa and min stress of 0.002485 Mpa.

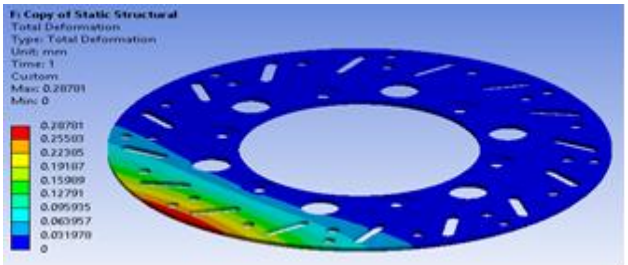

Fig9. Deformation

Total deformation for optimized disc is shown in fig.9, with max deformation of 0.28781mm and minimum deformation of 0 mm.

Steady state thermal analysis of optimized disc

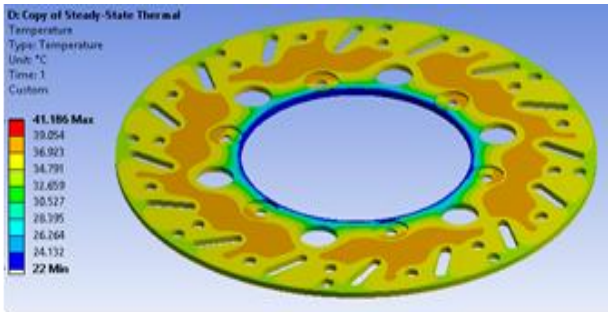

Fig.10 Temperature

Steady state thermal analysis of optimized disc for temperature is shown in fig10, with Max temperature 41.186℃ and min temperature 22 ℃.

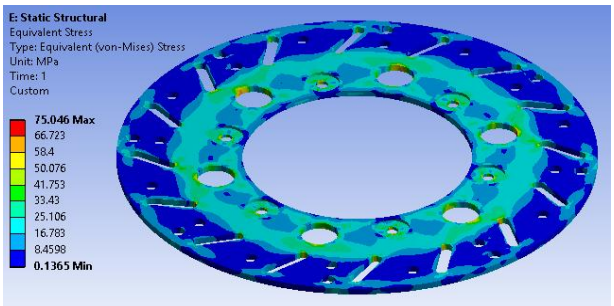

Fig.11 Von- Mises stress

Thermal stress of optimized disc is shown in fig.11, with max stress of 75.046 Mpa and min stress of 0.136 Mpa.

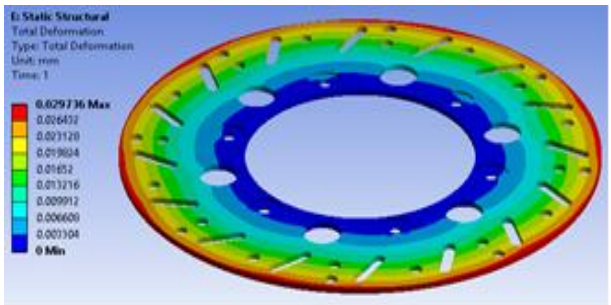

Fig.12 Deformation

Total deformation of optimized disc for steady state thermal analysis is shown in fig.12, with max deformation of 0.28781mm and minimum deformation of 0 mm.

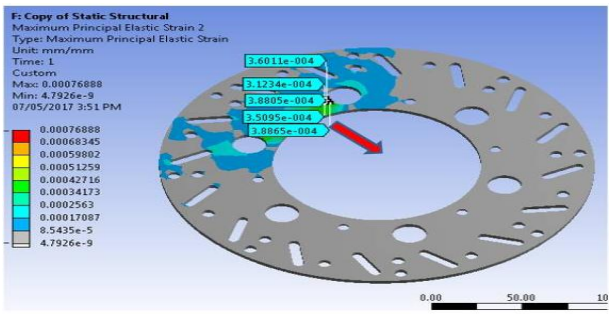

Fig.13 Maximum principal elastic strain

The principal elastic strain for the optimized disc is shown in the above fig. with maximum strain 00.388mS and measured strain 00.370mS.

Fig.14 Actual strain of optimized disc

Table 1 Results of Stress and Deformation fororiginal disc and optimized disc

S. No	Parameters	Original disc		Optimized disc
1	Von-mises stress (Mpa)	Max	138.36	153.62
		Min	0.007	0.002
2	Deformation (mm)	Max	0.2377	0.287
		Min	0	0

Table 2 Results of steady state thermal analysisfororiginal disc and optimized disc

S.No	Parameters	Original disc		Optimized disc
1	Temperature (℃)	Max	58.13	41.186
		Min	22.00	22.00
2	Von-mises stress (Mpa)	Max	148.38	75.046
		Min	0.6731	0.136
3	Deformation (mm)	Max	0.035	0.29
		Min	0	0

From the finite element analysis shown above:

1. The stress in the optimized disc is increased as compared to original disc but it is below the yield strength of the material shown in table 1
2. The thermal stresses of optimized disc are greatly reduced as compared to original disc shown in table 2.
3. In case of principal elastic strain, strain value matches with experimental and FEA.

Whereas the temperature of optimized disc is compared with the experimental testing value shown below.

6. Experimental setup

The trial Setup comprises of various parts

1. Motor-Three Phase, 1 HP, 1440 Rpm.
2. Shaft & Bearings
3. Caliper and hydraulic brake arrangement
4. Non-contact temperature measurement gun
5. Base Frame
6. Optimum Disc Brake Rotor
7. Handle arrangement

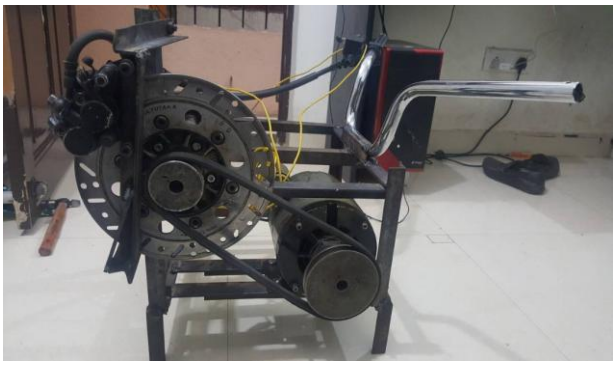

Fig.15 Experimental setup

6.1 working:

The disc is rotating at constant RPM with the help motor arrangement. Brake is applied periodically to reduce or to stop the disc. While applying brake the friction takes place between disc and pad. These friction forces to resist the motion of the disc, due to friction between disc and pad heat is generated in the disc and distributed over the disc. Heat generated in the disc is dissipated by the conduction as well as convection mode of heat transfer. Temperature measuring set-up mounts the disc on the frame and gives it rotation by the motor with 1440 rpm with constant speed.

Fig.16 Temperature reading using temperature sensor

Experiment test is taken on the optimized disc and the temperature readings are taken.

During experiment maximum temperature for optimized disc brake could not go beyond 41.2 ℃ or matches FEA results as shown below, because the frictional heat escapes in the air by convection as well as radiation. Temperature of disc is measured by using infrared sensor, which is a non-contact type of sensor, projecting laser beam on optimized disc.

Table 3 Comparison of software and experimental results for original and optimized disc

S.No	Parameters	FEA Results	Experimental Results
		Temperature ℃	
1	Original disc	58.13	-
2	Optimized disc	41.18	41.2

7. Conclusion

By comparing the different results obtained from FEA and experimental setup, it can be concluded that,

1. The stresses are within the yield limit of material
2. Percentage reduction in mass is about 8%
3. Thermal stresses are reduced form 148.38 Mpa to 75.046 Mpa.
4. Strain value matches with the experimental and FEA
5. The steady state thermal analysis temperature value matches with the experimental value.

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