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Numerical Investigation of Performance of Brake Shoe Liner to Fabricate and Test Modified Design

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***Abstract*—Braking of a vehicle involves conversion of kinetic energy of the vehicle into mechanical energy, which is required to be dissipated in the form of heat. During braking the frictional contact in the brake pads and disc can lead to high temperatures resulting in thermal and structural stresses in disc and pads. The resulting high stresses and frequent engagement and disengagement of brakes lead to higher rate of wear of disc and pads.**

An attempt has been made to increase the heat dissipation and reduce wear and fading of the brake pads by making geometrical modifications to the brake pads. The structural integrity of the proposed options was confirmed with ANSYS Workbench 2015. From the manufacturing point of view the brake pad with single slot was feasible to manufacture, as brake pad with two or four slot would require very small size cutter. For two slots we require a 2 mm diameter cutter, which is not easily available. Also such small size cutter has very less strength. If we make larger slots on the brake pad, the contact area is small. So for disc brakes of bike, we decided to go for brake pads with single slot.

Inertial dynamometer system (test rig) has been used to test disc brake pads at different operating conditions. Two sets of brake pads are tested. The results explains the temperature distribution obtained by experiments on two different shapes of brake discs pads affected by the types of shapes of brake discs pads, disc geometry and operating conditions.

Keywords— Brake pads, Finite element analysis, Heat generation, Inertia Dynamometer,DAQ.

I. INTRODUCTION

Motorcycles use disc brakes in place of drum brakes. They are preferred because of their higher braking efficiency. A disc brake of floating caliper design typically consists of pads, calliper, carrier, rotor (disc), piston, and guide pins. One of the major requirements of the calliper is to press the pads against the rotor to absorb the kinetic energy of the vehicle in form of heat and then rejecting this heat to the atmosphere.However, the major problem that occurs is their frictional area is small, heat generated gets concentrated in a confined space. Due to high temperature the friction material constantly gets deposited on the disc and the pads wear out quickly. To avoid loss of braking they need to be replaced frequently.

Number of people investigated the brake pads heat generation and rejection phenomenon. HuajiangOuyang[1] investigated disc brake systems by transient analysis and attempted to combine heat conduction analysis, contact analysis and transient analysis of disc brakes. He experimented on few geometrical modifications and found which had better fade resistance. Moses and John[2] modeled the heat generation and dissipation in a disc brake during braking and the following release period. The model simulates the braking action by investigating both the thermal and elastic actions occurring during the friction between the two sliding surfaces, represented by the maximum temperature on the contact surface. They modeled the finite element model of the brake disc and two pads to observe the thermo mechanical behavior developed between the pad and disc during the braking period.In another research by SaeedAlbatlan[3], he described application of inertial dynamometer to experimentally verify different operating conditions of a brake pad. Inertia dynamometer is a setup to simulate actual braking conditions in a laboratory and measure the effect of various parameters. Abu Bakar[4] investigated the contact (interface) pressure distributions at the rotor and piston-pad interface in response to several ideas of simulated structural (geometric or material) modifications.M. Timur[5] investigated the heat transfer mechanisms and studied them on for different pressure conditions. UN regulations (Addendum89: Regulation No. 90)[6] gives the amount of wear after which the brake pad becomes unsafe for use

To counter the problem of heat concentration some of the ways are to increase the contact area, increase heat dissipation or use materials that have high temperature resistance. Heat dissipation from the brake pads takes place by transfer of heat to the disc, the brake fluid and to the atmosphere. During actual braking the heat conducted to atmosphere is negligible because area exposed to atmospheric air is very less. To increase the area exposed to atmosphere, radial slots were made. To investigate the effect of making different number of radial slots on the brake pad, we found out the increase in heat dissipation theoretically and the same analysis was carried out in 'ANSYS simulation software'. Wear test was carried out based on the standard ST-1037 which is followed by BREMBO BRAKE INDIA PVT. LTD on the optimum modification found during the analysis. After the experimental tests, we compared the results of the modified brake pad with the existing model. The comparison was made based on both braking performance and fade rate.

II. EQUATIONS FOR HEAT GENERATION

Amount of heat generated is proportional to loss of kinetic energy of the vehicle during braking. The disc absorbs part of the heat and rest is absorbed by the two brake pads. The frictional force per unit area $F_f$can be calculated by equating retarding power P derived in two different sets of parameters. Retarding power based on kinetic energy of the vehicle

$$P = -\frac{d}{dt}\left(\frac{mv^2}{2}\right) = mR^2 w(t).\alpha \quad (1)$$

Retarding power based on caliper pressure

$$P = -2F_f w(t) \iint r da = 2F_f(t).w(t)\, r_m A \quad (2)$$

From this the frictional force per unit area is

$$\therefore F_f = -\frac{mR^2\alpha}{2r_m A} \quad (3)$$

Heat generated per unit area can be derived from (1), (2) and (3)

$$q_{generated} = -F_f.V_d(r,t)$$

$$\therefore q_{generated} = -\frac{mR^2\alpha}{2r_m A} \times r(w_0 + \alpha t)$$

The using energy balance

$$q_{stored} = q_{generated} - q_{dissipated}$$

Heat dissipated includes heat rejection due to convection and radiation.

$$\therefore mC_p(T_{x+1} - T_x) = \frac{mR^2\alpha}{2r_m A}.r(w_0 + \alpha t) - \{h(T_x - T_{atm}) + \varepsilon\sigma(T_x^4 - T_{atm}^4)\} \quad (4)$$

From (4) temperature $(T_{x+1})$of brake pad is found.

The heat generated during braking is distributed between the brake pads and the disc in the ratio given by

$$\sigma = \frac{\varepsilon_d S_d}{\varepsilon_d S_d + \varepsilon_p S_p}$$

whereε is the thermal effusivity

TABLE I.VALUE OF INPUT PARAMETERS

Property	Value
Mass of the vehicle(m)	300kg
Radius of the disc (R)	320 mm
Deceleration (a)	1 m/s^2

Substituting these values in the above equations, also considering total heat rejected through all surfaces we calculated average temperature attained by the body.

Also the heat generated during a braking cycle was calculated so that it can be used in the analysis using ANSYS WORKBENCH 2015.

TABLE II.TEMPERATURES OBTAINED FOR THE MODIFICATIONS

Brake pad	Temperature
Original	205.17
1 Slot	195.48
2 Slots	194.3
4 Slots	198.3

III. FINITE LEMENT ANALYSIS

Finite element analysis (FEA) involves certain steps, of which first is creating a 3Dimensional model of the brake pad assembly.

Figure.1 shows the actual unmodified brake pad. This model was created using the AUTODESK INVENTORsoftware, which was then imported to the ANSYSWORKBENCH 2015.

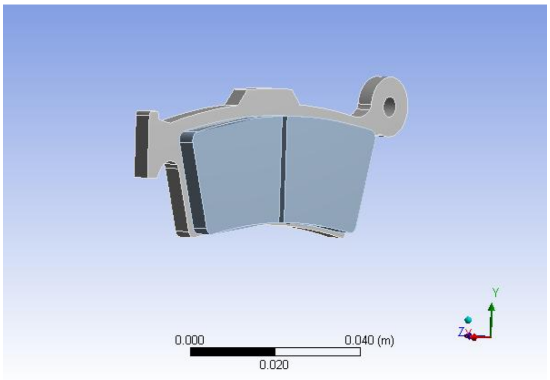

Fig.1.Model of existing pad

Then thermal analysis was carried out to find out the temperature distribution through out the pad for one braking cycle. The model was divided into 2932 number of elements, and 5994 number of nodes. Fig.2 shows the temperature distribution on the brake pad generated due to braking from 60km/hr for 4sec. The temperature results are compared with calculations to verify the correctness of the analysis procedure.

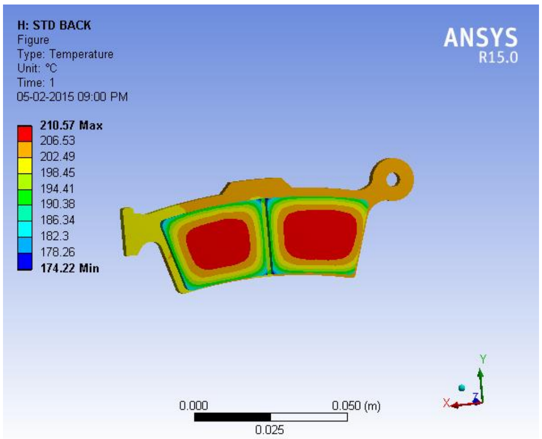

Fig.2. Temperature for existing pad

It is seen that the temperature obtained from the analysis of the existing brake pad is almost equal to that calculated theoretically. The error present is because we considered the brake pad as a lumped body.

Fig.3 shows temp distribution on brake pad with a single slot. It is observed that heat concentration as observed in the original is reduced. Instead of a single high temperature spot, the high temperature regions are distributed and are small in size. Also the maximum temperature reached is also reduced.

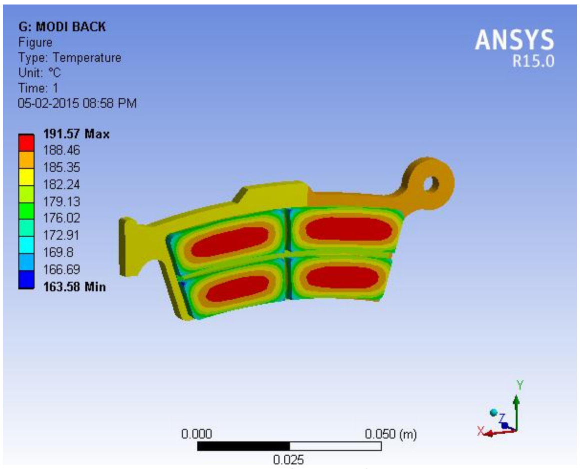

Fig.3. Temperature for modification 1

With the increasing number of slots, in further 3 cases, the amount of heat removal increases, but along with that the energy generation also is high due to smaller area. Also as the size of slot becomes smaller the value of convective coefficient decreases.Fig.4 and Fig.5 shows the effect on temperature distribution due to 2 and 4 slots.

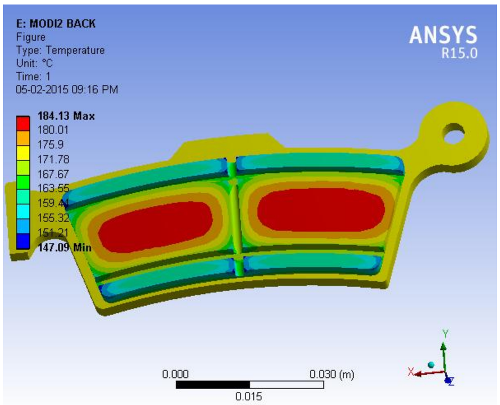

Fig.4. Temperature for modification 2

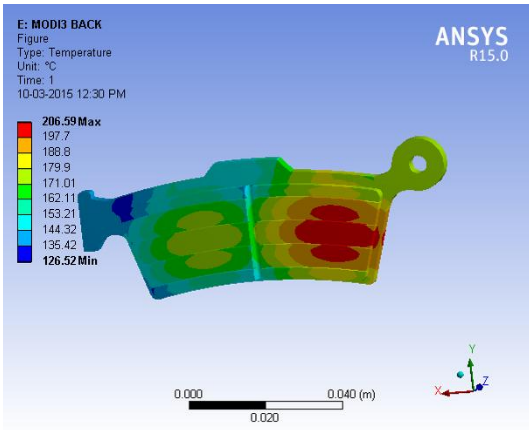

Fig.5. Temperature for modification 3

IV. EXPERIMENTAL RESULTS

Theoretically analysis was done on the existing brake pads and found out the relation between heat generated due to loss of kinetic energy to amount heat lost and stored inside the pad. Based on this relation we found the temperature that is achieved due to brake application to reduce the vehicle speed. Then we made three modifications to the existing brake pads and also found temperature that is reached for the same set of braking conditions as used for the existing brake pad.To compare the results for existing and other modifications we calculated temperature achieved when the vehicle brakes from 60 km/hr to dead stop at constant deceleration. Table III.shows the results obtained for the calculations.

TABLE III. CALCULATED VALUES

Brake pad	Temperature achieved(°C)	Heat generated (kW/m^2)
Standard	205.17	1327.97
With single slot	195.48	1486.29
With two slots	194.3	1578.51
With four slots	198.3	1591.79

Now a 3-D model of the brake pad of PULSAR-220 was prepared in AUTODESK INVENTOR. It was analyzed in ANSYS WORKBENCH 2015 to obtain the temperature distribution and stresses induced in the pads. Maximum temperature obtained in the analysis was noted and compared with the theoretical values as shown in Table IV.

TABLE IV.COMPARISON OF VALUES

Brake pad	Theoretical value (°C)	Numerical investigation(°C)
Standard	205.17	210.5
With single slot	195.48	191.5
With two slots	194.3	194.13
With four slots	198.3	206.59

As far as the manufacturing point of view is concerned, the brake pad with single slot was feasible to manufacture, as brake pad with two or four slot would require very small size cutter. For two slots we require a 2 mm diameter cutter, which is not easily available. Also such small size cutter has very less strength. If we make larger slots on the brake pad, the contact area is small. So for disc brakes of bike, we decided to go for brake pads with single slot. To investigate experimentally, lab tests were carried out on an inertia dynamometer. The existing and the brake pad with single slot were tested according to the standard ST-1037. The test measured braking torque, coefficient of friction, initial and final temperature after every braking step. Brake pads were subjected to 800 brake applications. These brake application consisted of groups of 6 continuous braking action (1 cycle) followed by a 180 sec cooling period.The Variation of torque and Coefficient of friction for 800 stops is plotted on graph for both existing and modified brake pads. Fig 7 shows variation of Average coefficient of friction v/s cycle. Fig 6 shows variation of braking torque v/s cycle.

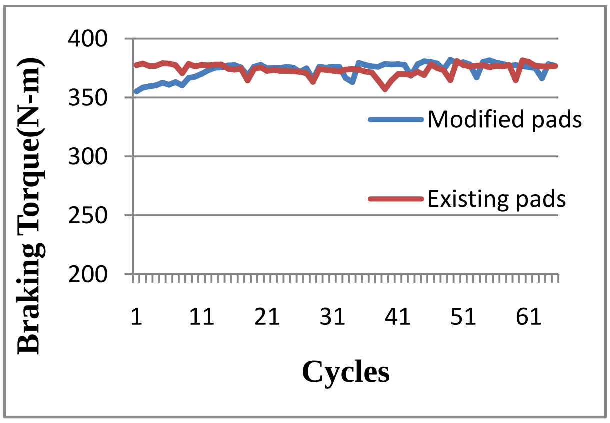

Fig. 6. Graph of braking torque v/s cycles

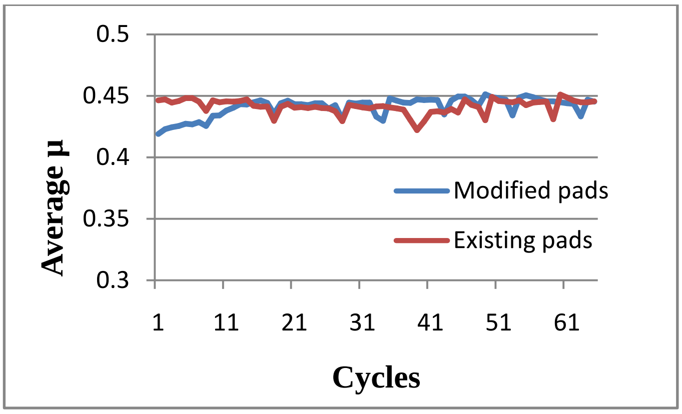

Fig7.Graph of average μ v/s cycles

In every stop the initial temperature is not same. So to make a comparison we found the average temperature rise in every stop in a cycle. Fig shows a graph of temperature rise v/s stop number.

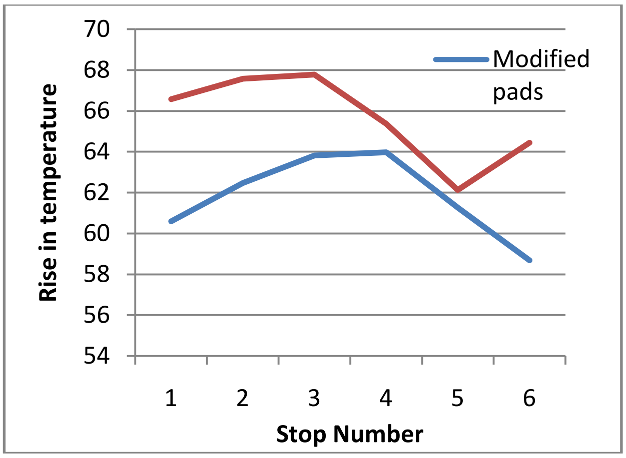

Fig. 8.Graph of rise in temperature v/s cycles

The weight and thickness of the brake pad was measured before the tests for wear calculation. The thickness of the brake pad before test was 4.5mm in case of both brake pads.After the test thickness and weight was measured. The thickness of existing brake pads was 4.1mm and that of modified pad was 4.17mm. The weight reduction was 1.5 g and 1.2 g. Thus wear rate in thickness loss was found to be 0.4mm/ 800 stops and 0.33mm/ 800 stops respectively i.e 0.0005mm/stop and 0.0004125mm/ stop. Similarly wear in terms of weight reduction is 0.001875g/stop and 0.0015g/stop.

To summarize all these results, the maximum temperature attained for modified brake pad is lesser then the existing brake pad. This temperature characteristic is due to the slot in the modified brake pad, which acts as a heat rejection zone even during braking. More number of slots will increase heat rejection but the area of contact is decreased which adversely affects the braking efficiency. As seen in the test the braking torque of both modified and existing pad is almost the same and also the coefficient of friction is not affected due to the slot. However the temperature rise in modified brake pads is lower. The observed wear rate is lesser than the existing. Hence the modification has a positive effect on the wear characteristic of the brake pad.

V. CONCLUSION

By observing the numerical analysis thermal efficiency of the disc brake system can be increased without compromising the structural strength of the brake pad. From experimental test carried out, modifications do not affect the braking torque and the coefficient of friction. The slot in the modification increases amount of heat rejectedthus controlling the surface temperature of brake pad. The modification reduces the wear rate and increases the fade resistance of the disc and brake pads.

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