Mathematical Modeling & Analysis of Brake Pad for Wear Characteristics

S. R. Kakad[1], R.M. More[2], Dr. D. N. Kamble[3]
[1]*Prof. S. R. Kakad, Department of Mechanical Engineering, SKNCOE, Pune,*
sharad_kakad@yahoo.com
[2]*Prof. R.M. More, Department of Mechanical Engineering, SKNCOE, Pune,*
more.rms@gmail.com
[3]*Dr.D.N.Kamble, Department of Mechanical Engineering, SAOE, kondhwa Pune,* dnkamble81@gmail.com

ABSTRACT

Robust braking results in heat generation whose effects may have considerable impact on the parameters of the process such as wear rate and coefficient of friction. In this paper, the performance of the brake is studied for the different material. The simulation of wear and contact pressure is reported by using FEA software. The investigation can be extending for the verification of mathematical model. The FEA software is more effective as a tool for verification of analytical contribution. The friction and wear performances of brake material in dry condition are also reported in the paper. The thermal analysis and the structural analysis are found to be a core of investigation in study of disc brake. The collective dependent mathematical model is developed to study their collective effect on specific disc material. The application can be extend to validate a mathematical model and the predictions can be reported from the investigation.

In this paper, The Mathematical model for the disc brake for analysis of wear is developed. The equations are designed to study the effect of parameters like breaking energy, breaking power, contact pressure and actuating force on the wear of disc brake. The obtained mathematical model is studied to report the modification to improve the degree of acceptance. The customize software package like ANSYS is used to verify the modified analytical investigation.

Keywords*: Breaking power1, Mathematical model2, Wear3, Frictional force4, Brake pad5 etc.*

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1. INTRODUCTION

Passenger car disc brakes are safety-critical components whose performance depends strongly on the contact conditions at the pad-to-rotor interface. When the driver steps on the brake pedal, hydraulic fluid is pushed against the piston of the caliper, which in turn forces the brake pads into contact with the rotor. The frictional forces at the sliding interfaces between the pads and the rotor retard the rotational movement of the rotor and the axle on which it is mounted. The kinetic energy of the vehicle is transformed into heat that is mainly absorbed by the rotor and the brake pad. The pad-to-rotor interface can be classified as a conformal dry sliding contact. During braking a point on the brake pad is in constant contact with the rotor, whereas a point on the rotor experiences intermittent contact. A natural consequence of the sliding contact is that both the rotor and the pads are worn out, affecting the useful life of the brake as well as its behaviour.

The brake disc (or rotor) is usually made of cast iron, but may in some cases be made of composites such as reinforced carbon– carbon or ceramic matrix composites. This is connected to the wheel or the axle. To stop the wheel, friction material in the form of brake pads, mounted on a device called a brake caliper, is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. Friction causes the disc of wheel to slow or stop. Brakes convert motion into heat, and if the brakes get too hot, they become less effective, a phenomenon known as brake fade.[2]

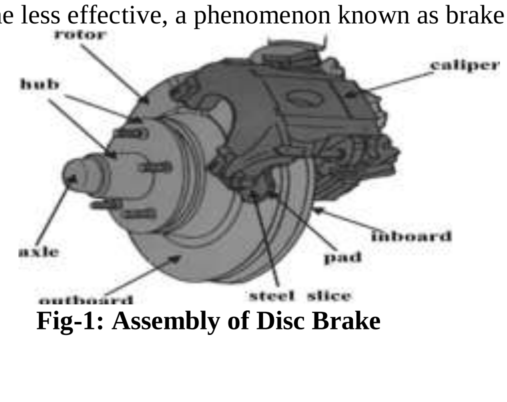

Fig-1: Assembly of Disc Brake

2. OBJECTIVE & SCOPE

1. Development of mathematical model for the disc brake for the wear analysis. The equations are designed to study the effect of parameters like breaking energy, breaking power, contact pressure and actuating force on the wear of disc brake.
2. The study of obtained mathematical model to report the modification to improve the degree of acceptance.
3. The customize software package like ANSYS is used to verify the modified analytical investigation.

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3. METHODOLOGY

3.1 Analytical Investigation

1. **Brake power**

Thermal modeling Regarding to the uniform pressure or the constant wear boundary condition at the contact surface, two methods are available for calculating the braking heat generation rate. Uniform pressure distribution in the contact region is often valid when the pad is new. However after braking for several times, assumption of uniform wear is more pragmatic. In this study, the pad is used several times and uniform wear between pad and brake disc is stabilized, hence the heat flux is just a function of time and it is independence of the spatial variables. For a vehicle which is decelerating on a level surface from a higher velocity V1 to a lower velocity V2 the braking energy Eb can be written as

$$E_b = \frac{1}{2} m(V_1^2 - V_2^2) + \frac{1}{2} I(\omega_1^2 - \omega_2^2)$$

Where, I is mass moment of inertia of the rotating parts, m locomotive mass and ω is the angular velocity of rotating parts. If the locomotive stops completely (V2 = ω2= 0) then all the rotating parts will be expressed relative to the revolutions of the wheel.

$$E_b = \frac{1}{2} m(1 + \frac{I}{R_w^2 m})V_1^2 = \frac{1}{2} K_{cf} m V_1^2$$

Where, Kcf is the correction factor for rotating masses and R_w is the wheel radius. Braking power P_b is equal to braking energy divided by the braking time t,

$$P_b = \frac{dE_b}{dt}$$

The brake power is given by,

$$P_b = K_{cf} ma(V_1 - at)$$

Where, a is the deceleration of the locomotive. The distribution of braking energy between pad and disc cannot be predicted readily. [5]

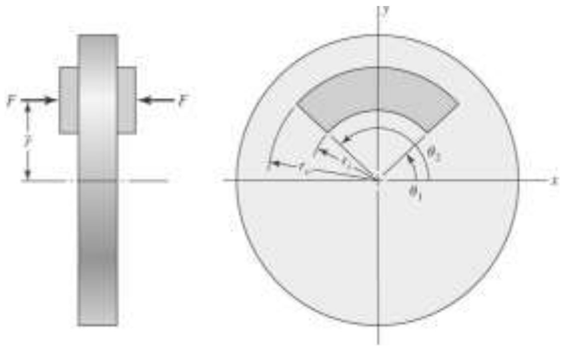

Fig-2: Parameter of Disc Brake

The model consist of various forces on the vehicle during braking and computes the time-dependent braking power dissipated, from the calculated brake torque Tb, and wheel speed ω, i.e. Pb =Tb ω =Tb (1-Tire Slip) wheel speed/Rolling radius.

2. **Linear Sliding wear**

Considering a block with surface area A, sliding over fixed surface with contact pressure P, where the co-efficient of sliding friction fs & define a linear wear measure w. [10]

The work done by frictional force fsPA during displacement S is fsPAS or fsPAVt (where V is velocity & & t is time).

The volumes of material removed because of wear wA is proportional to the work (wA α fsPAVt)

i.e. wA = KPAVt

Where, K is proportionality factor which includes the co-efficient of friction fs. The SI unit of K is m3.s / (N.m.s)

w = KPVt

Additional correction factors f1 & f2 can be included such that:

w = f1f2KPVt

Where, f1 correction factor for motion type

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f2 correction factor for environment

The Force and the torque equations are given as,

w = f1f2KPVt

$$F = \int_{\theta_1}^{\theta_2}\int_{ri}^{ro} pr.dr.d\theta$$

$$= (\theta_2 - \theta_1)\int_{ri}^{ro} pr.dr$$

$$T = \int_{\theta_1}^{\theta_2}\int_{ri}^{ro} f.pr^2.dr.d\theta$$

$$= (\theta_2 - \theta_1) f \int_{ri}^{ro} pr^2.dr$$

$$r_e = \frac{T}{fF} = \frac{\int_{ri}^{ro} pr^2.dr}{\int_{ri}^{ro} pr.dr}$$

$$M_x = F\bar{r} = \int_{\theta_1}^{\theta_2}\int_{ri}^{ro} pr(r.\sin\theta).dr.d\theta$$

$$= (\cos\theta_1 - \cos\theta_2)\int_{ri}^{ro} pr^2.dr$$

$$\bar{r} = \frac{M_x}{F} = \frac{(\cos\theta_1 - \cos\theta_2)}{\theta_2 - \theta_1} r_e$$

These equations are applied for uniform wear and uniform pressure conditions:

A. Uniform Wear Condition

$$F = (\theta_2 - \theta_1) P_a.r_i(r_o - r_i)$$

$$T = (\theta_2 - \theta_1) f.P_a r_i \int_{ri}^{ro} r.dr$$

$$= \frac{1}{2}(\theta_2 - \theta_1) f.P_a r_i (r_o^{\,2} - r_i^{\,2})$$

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$$r_e = \frac{P_a r_i \int_{ri}^{ro} r.dr}{P_a r_i \int_{ri}^{ro} dr}$$

$$= \frac{r_o^{\ 2} - r_i^{\ 2}}{2} \frac{1}{r_o - r_i} = \frac{r_o + r_i}{2}$$

$$\bar{r} = \frac{\cos\theta_1 - \cos\theta_2}{\theta_2 - \theta_1} \frac{r_o + r_i}{2}$$

B. Uniform Pressure Condition

$$F = (\theta_2 - \theta_1) P_a \int_{ri}^{ro} r.dr$$

$$= \frac{1}{2}(\theta_2 - \theta_1) P_a (r_o^{\ 2} - r_i^{\ 2})$$

$$T = (\theta_2 - \theta_1) f.P_a \int_{ri}^{ro} r^2.dr$$

$$= \frac{1}{3}(\theta_2 - \theta_1) f.P_a (r_o^{\ 3} - r_i^{\ 3})$$

$$r_e = \frac{P_a \int_{ri}^{ro} r^2.dr}{P_a \int_{ri}^{ro} r.dr}$$

$$= \frac{r_o^{\ 3} - r_i^{\ 3}}{3} \frac{2}{r_o^{\ 2} - r_i^{\ 2}} = \frac{2}{3} \frac{r_o^{\ 3} - r_i^{\ 3}}{r_o^{\ 2} - r_i^{\ 2}}$$

$$\bar{r} = \frac{\cos\theta_1 - \cos\theta_2}{\theta_2 - \theta_1} \frac{2}{3} \frac{r_o^{\ 3} - r_i^{\ 3}}{r_o^{\ 2} - r_i^{\ 2}}$$

$$= \frac{2}{3} \frac{r_o^{\ 3} - r_i^{\ 3}}{r_o^{\ 2} - r_i^{\ 2}} \frac{\cos\theta_1 - \cos\theta_2}{\theta_2 - \theta_1}$$

3. Force system for car model

The Normal force FT on the inner and outer faces can be obtained from the Torque

T = FT.R

FT= T/Rµ = Coefficient of friction

FT = Total normal forces on disc brake = FTRI + FTRO

FTRI=FT/2

FTRI = µ1.FRI

FRI= FTRI/ µ1

FRI = P max / 2 × A pad brake area

µ1 = Coefficient of friction = 0.5

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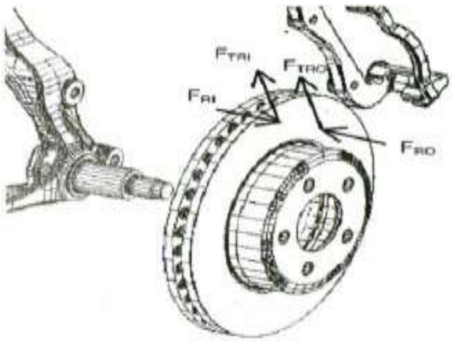

Fig-3: Forces System for car model

4. Braking Distance(x)

The tangential braking force acting at the point of contact of the brake and the work done relation is given by,
Work done = F_T. x
Where FT = FTRI + FTRO
x = Distance travelled (in meter) by the vehicle before it come to rest.
The kinetic energy of the vehicle, K.E. = $(mv^2) / 2$
Where, m = Mass of vehicle & v = Velocity of vehicle
In order to bring the vehicle to rest, the work done against friction must be equal to kinetic energy of the vehicle.
FT. x = $(mv^2) / 2$
Assumption v = 100 km/h = 27.77 m/s
m = Dry weight of Vehicle
Therefore, x = $(mv^2) / 2 F_T$

5. Thermal Model

The thermal conductivity of the brake pads is smaller than the disc (Kp < Kd), so one can consider that the total amount of the braking heat will be completely absorbed by the brake disc. This assumption leads to higher temperature estimation for brake disc. To avoid this issue, suppose that the braking operation time is short, hence the pad and brake can be considered as semi-infinite solids and the heat generation ratio can be calculated as follows[6]

$$\lambda = \frac{q_d''}{q_p''} = \left(\frac{\rho_d . C_d . K_d}{\rho_p . C_p . K_p} \right)$$

where λ is the heat generation ratio, q_d” and q_p” are the heat flux absorbed by the brake disc, and pad, ρ represents density, C is the specific heat, K the thermal conductivity and the index d and p indicate disc and pad, respectively. The heat flux on the braking surface can be found

$$q_d'' = \frac{\lambda}{A(\lambda + 1)} K_{cf} . m . a (V_1 - at)$$

Where, A is the disc and pad contact area.

Heat Generated (Q_d) = m .Cp. ΔT

ΔT= Q_d/m Cp

3.2 Software verification

A. Background

Linear static analysis is carried out to get the desired results.

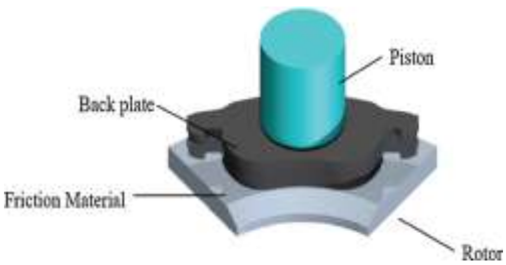

Fig 4 Simplified CAD model of the studied subsystem

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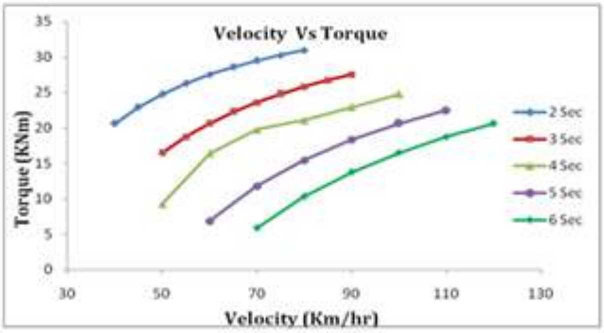
4. RESULT & DISCUSSION

The investigation of wear of brake pad is done and categorized based on analytical investigation & software verification. The results are studied, compared and are reported.

4.1 ANALYTICAL RESULT

The studies are performed for braking time of 2-6 Sec. It is observed that for a min. braking time the torque reports the higher values. As the velocity of vehicle increases, the time required for braking is also increases. Since braking time vary with velocity.



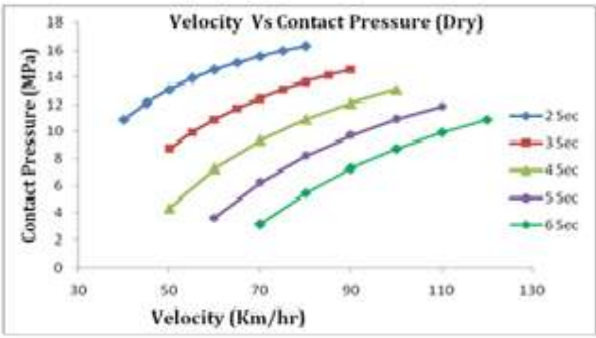
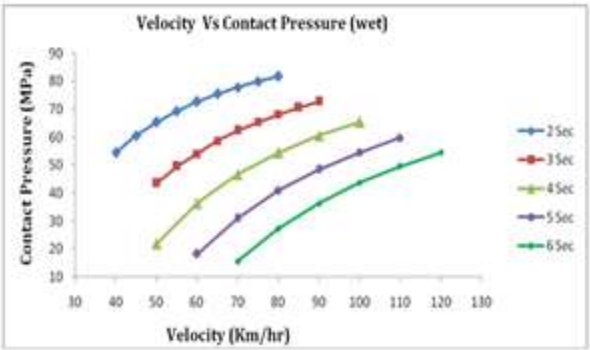


Fig.5 Velo. Vs Torque for braking time of 2-6 Sec.

Fig.6 Velo. Vs Contact Pressure (Dry) for braking time of 2-6 Sec.



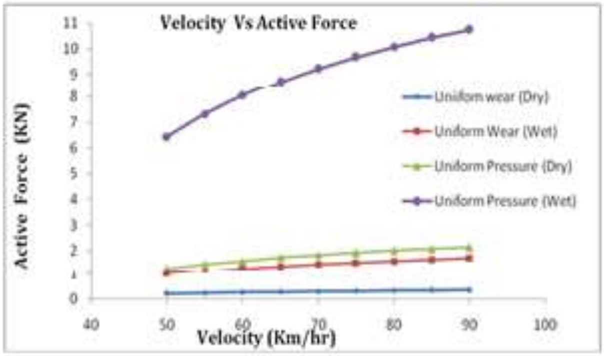
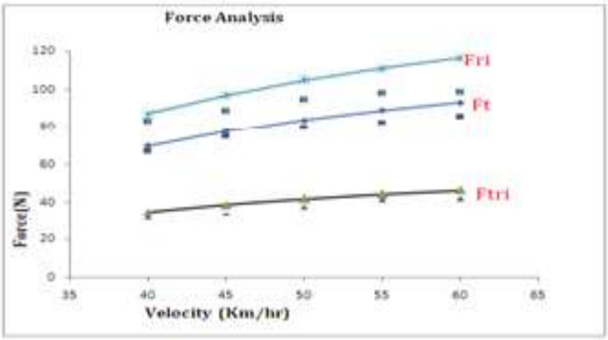


Fig. 7 Velo. Vs Contact Pressure (Wet) for of 2-6 Sec.

Fig. 8 Velo. Vs Active Force (Dry & Wet) for uniform braking time Wear & pressure condition



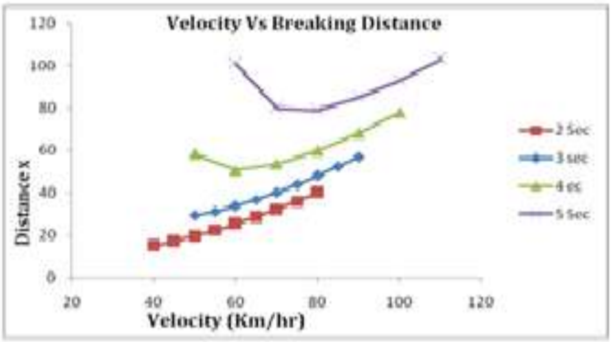


Fig. 9 Velocity Vs Forces

Fig. 10 Velo. Vs Braking Distance for braking time of 2-5 Sec.

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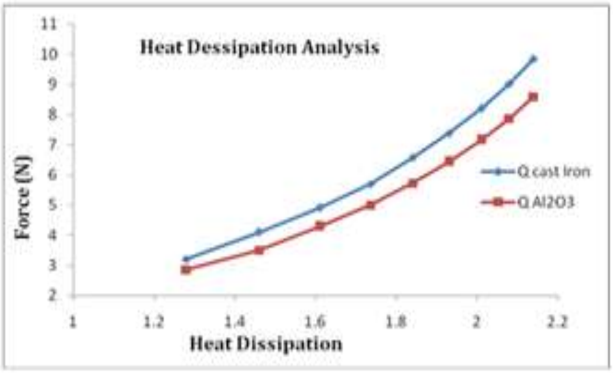

Fig. 11 Heat Dissipation Vs Force for Cast iron & AL_2O_3

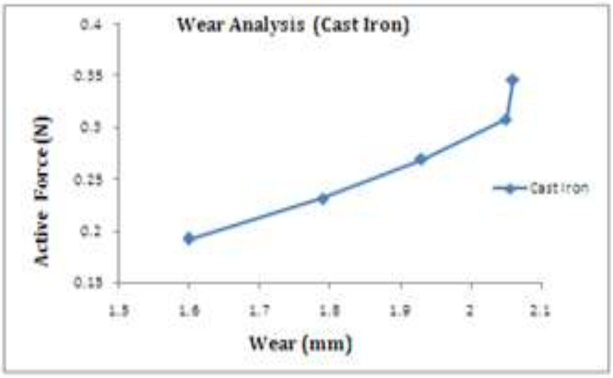

Fig. 12 Wear Vs Active Force for Cast Iron

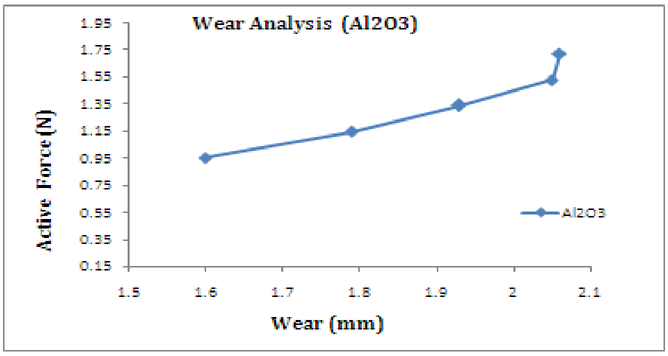

Fig. 13 Wear Vs Active Force for Al_2O_3

5.2 SOFTWARE VERIFICATION

The wear of the brake pad material is analyzed by using software package like ANSYS. During the variation of velocity, braking time is kept constant & corresponding wear of cast iron brake pad material is analyzed. There is variation of wear of brake pad material with respect to velocity of the vehicle & its corresponding braking time. The static structural analysis is performed analysis of total deformation of cast iron brake pad material.

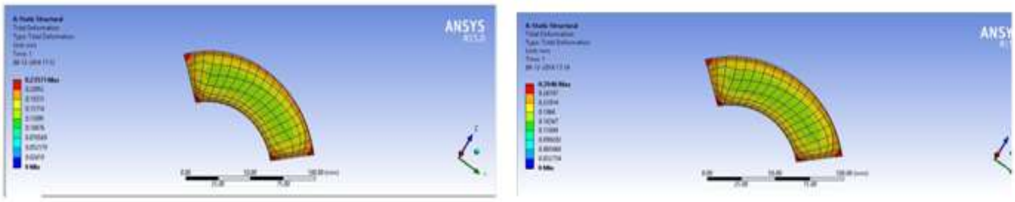

Fig. 14 Wear deformation of brake pad at velocity 40km/hr & 45 km/hr with 2 seconds braking time

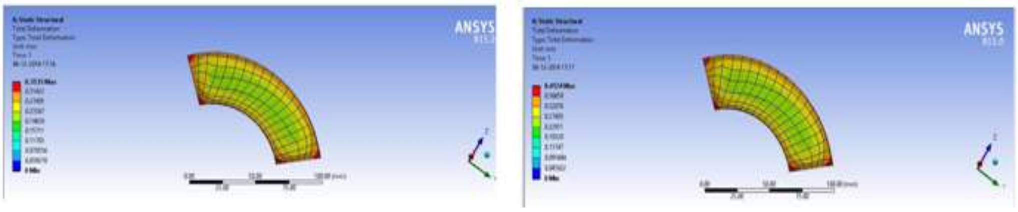

Fig. 15 Wear deformation of brake pad at velocity 50km/hr & 55km/hr with 2 seconds braking time

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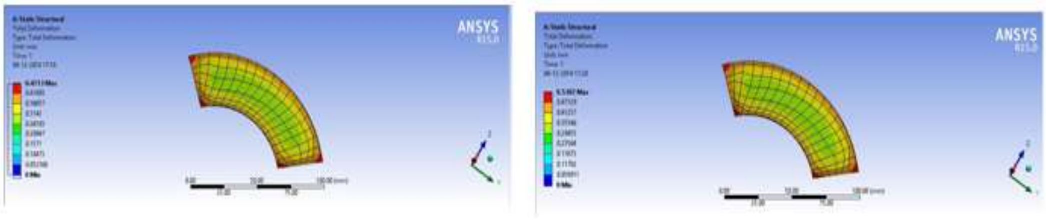

Fig. 16 Wear deformation of brake pad at velocity 60km/hr & 65km/hr with 2 seconds braking time

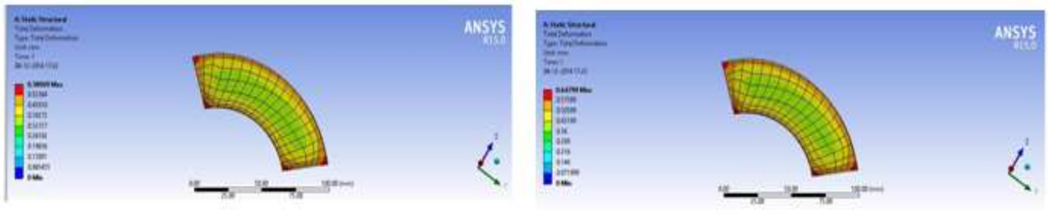

Fig. 17 Wear deformation of brake pad at velocity 70km/hr & 75 km/hr with 2 seconds braking time

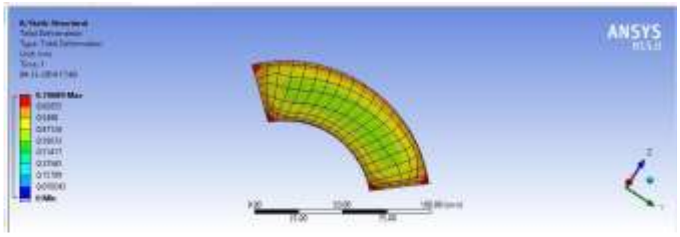

Fig. 18 Wear deformation of brake pad at velocity 80km/hr with 2 seconds braking time

6. CONCLUSION

1. The mathematical Equations are developed for investigation of wear of brake pad for various materials. The obtained analytical results from mathematical equations are studied and verified by using ANSYS. The equations are designed to study the effect of parameters like breaking energy, breaking power, contact pressure and actuating force on the wear of disc brake.
2. The Investigation is performed on brake pad material for uniform pressure condition & uniform wear condition under dry & wet condition.
3. The results are compared on common scale and the acceptance of modification shows good agreement in the range of 5 to 8 %.

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