Design And Cad Model of Automotive Disc Brake System

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***Abstract* - Automotive disc brake is a device by means of which artificial frictional resistance is quick applied to rotating disc, in order to stop the motion of vehicle instantly. During the braking phase, the frictional heat generated at the interface of the disc and pads can lead to high temperature. For reduce this temperature improving design modelling of an automotive disc rotor. The objective of the current study is to design a Disc, Brake pads, Piston and also Core and Cavity of Disc Brake Rotor by using Creo parametric 7.0, To manufacturing of disc brake rotor using Aluminum Metal Matrix Composites by using stir casting method. AMMC is the combination of aluminum reinforced with fly ash along with small quantity of other material like magnesium, graphite which are added in precise quantity to enhance the chemical, mechanical, thermal strength of material and testing in real time application and to investigate and analyses the temperature distribution, also analysis design of disc plate of rotor and to identify critical temperature during operation using FEA analysis.**

***Index Terms* - Disc brake, Calliper, Master cylinder**

INTRODUCTION

The automotive disc brake is a device for slowing or stopping the rotation of a wheel instantly. A tandem type of master cylinder was selected so that independent to hydraulic circuit can be obtained and it can be obtained by single control from brake pedal. A two-wheeler vehicle disc rotor made of aluminium alloy and ceramic composite (including carbon and silica). These brakes offer better stopping performance than comparable drum brakes, including resistance to "brake fade" caused by the overheating of brake components, and are able to recover quickly from immersion (wet brakes are less effective).

In this automotive disc brake system advanced type calliper are used in which used two pistons in one side and other side flexible calliper used to reduce speed of vehicle, The stopping power or capacity of a friction brake depends on the area in contact and coefficient of friction of the working surfaces as well as on the actuation pressure applied. Wear occurs on the working surfaces, and the durability of a given brake (or service life between maintenance) depends on the type of friction material used for the replaceable surfaces of the brake. If drake disc is in solid body the Heat transfer rate is low. Time taken for cooling the disc is low. If brake disc is in solid body, the area of contact between Disc and Pads are more, so efficiency of brake is high. The variation in temperature between a full and ventilated disc having same material is about 65 degrees at the moment1.8839 s from application of brake The obtained results are very useful for the study of the thermo-mechanical behaviour of the disc brake (stress, deformations, efficiency, and wear).

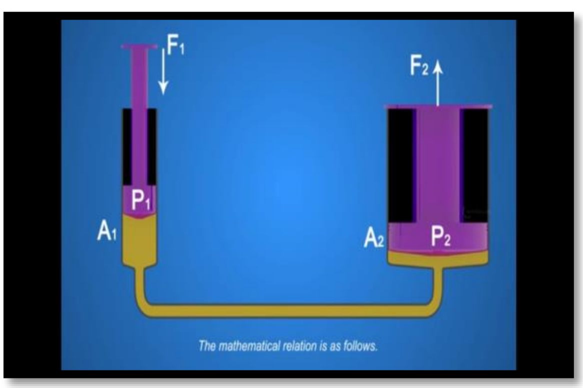
PROBLEM DEFINITION

- Before we used automotive disc brake, we are used drum brake system in two-wheeler.
- In drum brake system excessive wear and hauling happens due to this heat may produce. If we used continuously this brake system oversize of drums may occur replace the drum immediately.
- Due to above problem a control of vehicle may loosed, and accident may happen to overcome this problem we used disc brake system instead of drum brake system.
- Sometimes a loud noise or high-pitched squeal occurs when the brakes are applied. Most brake squeal is produced by vibration (resonance instability) of the brake components, especially the pads and discs (known as force-coupled excitation). This type of squeal should not negatively affect brake stopping performance.

- Techniques include adding chamfer pads to the contact points between calliper pistons and the pads, the bonding insulators (damping material) to pad back plate, the brake shims between the brake pad and pistons, etc. All should be coated with an extremely high temperature, high solids lubricant to help reduce squeal.

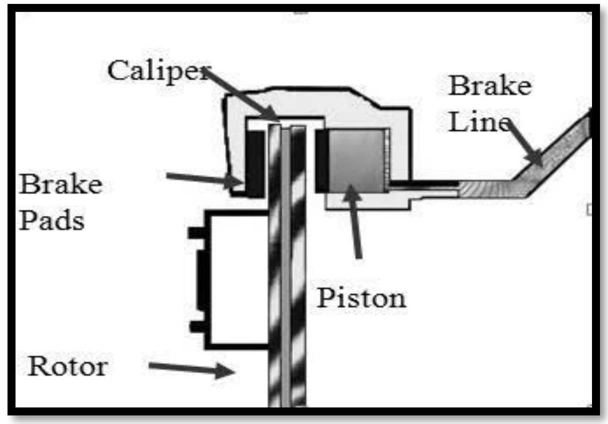
WORKING PRINCIPLE

The brake system works based on Pascal's Law.
"It state that pressure exerted anywhere in a container incompressible fluid is distributed equally in all direction throughout the fluid"
According to Pascal's law pressure on both pistons are equal.
$P1=F1/A1$
$P2=F2/A2$
$P1=P2$
So, when force is applied to the left piston, the fluid will transmit the force to the right piston surface.
$F1/A1=F2/A2$
But it will be a factor of the ratio of two piston surface.
$F2=A2/A1 * F1$
This concludes that with a small force F1 at left piston will give a higher force F2 at right piston.
Provided A2 is greater than A1.
With this, left piston can act at the pedal side and the right piston can act at the wheel.
Transferring energy from pedal to the brakes.



PROPOSED WORKING

- As the brake pedal is pressed, a pushrod exerted force on the piston in the master cylinder.
- This forces fluid through the hydraulic lines toward caliper.
- This brake caliper piston(s) then apply force to the brake pads.
- This causes them to be pushed against the spinning rotor, and then friction between the pads and the rotor causes a braking torque to be generated , then slowing the vehicles.



DESIGN CALCULATION

Selection of components
1) Brake rotor and calliper: -A bike's front rotor with diameter of 160 m was selected for 10-inch wheel rim.
 - Thickness of the disc was 4 mm and suitable callipers with dual pistons were selected.
2) Master cylinder: A tandem type of master cylinder was selected, so one hydraulic circuit can be obtained, and it can be obtained by a single control from brake pedal.
3) Brake pedal: The brake pedal was machined from checker from Aluminium plate having thickness 5.2 mm. It was designed to withstand a force of 1900N at the footrest.
The leverage of the pedal was set to 2.6:1.
Note: that all the relationships assume 100 % efficiency in the whole system. All dimensions are in S.I units.
Pedal ratio = $l2/l1 = 0.100/0.040 = 2.6$
$Fbp = Fd * (L2 / L1) = 250 * 2.6 = 650N$
Where,
Fbp = force output of the assembly
Fd = force applied to the pedal pad by the driver.
L1 =distance from the brake pedal arm pivot to the
 Output rod clevis attachment.
L2 =distance from the brake pedal arm pivot to the brake pedal pad.

A. The Master Cylinder:
Diameter of master cylinder piston =Dmc
= 0.0205 m.
Area of master cylinder piston = Amc
= (π/4) x Dmc2
= 3.3006×10^{-4} m^2
Pressure generated by the master cylinder
= Pmc = Fbp / Amc
= 650/ 3.3006×10^{-4}
= 1.969 MPa
Where,
Pmc = hydraulic pressure generated by the master cylinder
Amc = effective area of the master cylinder hydraulic piston.

B. The Calliper:
Diameter of calliper piston = Dcal = 0.024 m
Area of calliper piston = Acal = (π/4) x $Dcal^2$
= 4.523×10^{-4} m^2
Pressure transmitted to calliper = Pcal = Pmc
= 1.969 MPa
One sided linear mechanical force generated by the calliper will be equal to:
Fcal = Pcal x Acal = 1.969 x 10^6 x 4.523×10^{-4}
= 890.5787 N
Where, Acal = the effective area of the calliper
The clamping force will be equal to twice the linear mechanical force as follows:
Fclamp = 2 x Fcal = 2 x 890.5787 = 1781.15 N

C. The Brake Pad:
The frictional force is related to the calliper clamp force as follows:
Ffriction = Fclamp x μbp = 1781.15 x 0.4
=712.4629 N
Where, μbp = the coefficient of friction between the brake pad and the rotor

D. The Rotor:
The torque is related to the brake pad frictional force as follows:
Tr = Ffriction x Reff = 712.4629 x 0.07875
= 712.4629 Nm
Where, Tr = torque generated by the rotor
Reff = the effective radius of the rotor (measured from the rotor centre of rotation to the centre of pressure of the calliper pistons)

As the rotor is mechanically coupled to the hub and wheel assembly and the tyre is assumed to be rigidly attached to the wheel, the torque will be constant throughout the entire rotating assembly as follows:
Torque on Tyre (Tt) = Torque on wheel (Tw) = Torque on rotor (Tr)

E. The Tyre:
The force reacted at the ground will be equal to:
Force on the front tyre,
Ffront = Tt / Rfront = 712.4629 / 0.2032
= 350.62 N
Where, Rfront = effective rolling radius of front tyre

Force on the rear tyre,
Frear = Tt / Rrear = 712.4629 / 0.2032
= 350.62 N
Where, Rrear = effective rolling radius of rear tyre
The total braking force reacted between the vehicle and the ground,
Ftotal = (2 x Ffront) + (Frear)
= (2 x 350.62) + (350.62)
= 703.24 N.

F. Deceleration of vehicle in motion:
The deceleration of the vehicle will be equal to:
av = Ftotal / mv = 703.24 / 220
= 3.196 m/s^2
Where, av = deceleration of vehicle
Mv = mass of vehicle

G. Braking distance of vehicle:
The theoretical braking distance of a vehicle in motion can be calculated as follows:
d = v2 / 2av = $(10.111)^2$ / (2 x 3.196)
= 15.993 m
Where, v = velocity of vehicle
d = braking distance

H. Braking time of vehicle:
The theoretical braking time of a vehicle in motion can be calculated as follows:
Tstop = (v x mv) / Ftotal = (10.111 x 220) / 703.24
= 3.163 sec

I. Weight distribution:
From the vehicle's centre of gravity,
Vt = Ftotal = 703.24 N
Vf = (Vt x CGr) / WB

= (703.24 x 0.716) / 1.644
= 306.27 N

Where,
V_f = front axle vertical force
CG_r = distance from the rear axle to the CG
WB = wheelbase (distance from the front axle to the rear axle)
V_t = Total vertical force of vehicle

$V_r = (V_t \times CG_f) / WB$
= (703.24 x 0.928) / 1.644
= 396.96 N
Where, V_r = rear axle vertical force
CG_f = distance from front axle to the CG

Now,
Percentage front weight = $(V_f / V_t) \times 100$
= (306.27 / 703.24) x 100
= 43.55 %
Percentage rear weight = $(V_r / V_t) \times 100$
= (396.96 / 703.24) x 100
= 56.44 %

J. Dynamic impacts of vehicle:
Absolute weight transferred from the rear axle to the front axle,
$WT = (a_v / g) \times (h_{cg} / WB) \times V_t$
= (3.196 / 9.81) x (0.350 / 1.644) x 703.24
= 48.77 N

Where, WT = absolute weight transferred from rear axle to front axle
g = acceleration due to gravity
h_{cg} = height of CG from the ground
In order to calculate the steady-state vehicle axle vertical forces during a given stopping event, the weight transferred must be added to the front axle static weight and subtracted from the rear axle static weight as follows:
$V_{f,d} = V_f + WT$ = 306.27 + 48.77
= 355.04 N
Where, $V_{f,d}$ = the front axle dynamic vertical force for a given deceleration
$V_{r,d} = V_r - WT$ = 396.96 - 48.77
= 352.19 N
Where, $V_{r,d}$ = the rear axle dynamic vertical force for a given deceleration

K. Effects of weight transfer on tyre output:
Under static conditions, the maximum braking force that an axle is capable of producing is defined by the following relationships:
$F_{tyres,f} = \mu_f \times V_f$ = 0.7 x 306.27
= 214.389 N
Where, $F_{tyres,f}$ = the combined front tyre braking force
μ_f = coefficient of friction between front tyre and road
$F_{tyres,r} = \mu_r \times V_r$ = 0.7 x 396.96
= 277.87 N
Where, $F_{tyres,r}$ = the combined rear tyre braking force
μ_r = coefficient of friction between rear tyre and road
However, as a result of weight transfer during a deceleration event the maximum braking force that an axle is capable of producing is modified as follows:
$F_{tyres,f,d} = \mu_f \times V_{f,d}$ = 0.7 x 355.04
= 248.528 N
Where, $F_{tyres,f,d}$ = dynamic force on front tyre
$F_{tyres,r,d} = \mu_r \times V_{r,d}$ = 0.7 x 352.19
= 246.53 N
Where, $F_{tyres,r,d}$ = dynamic force on rear tyre

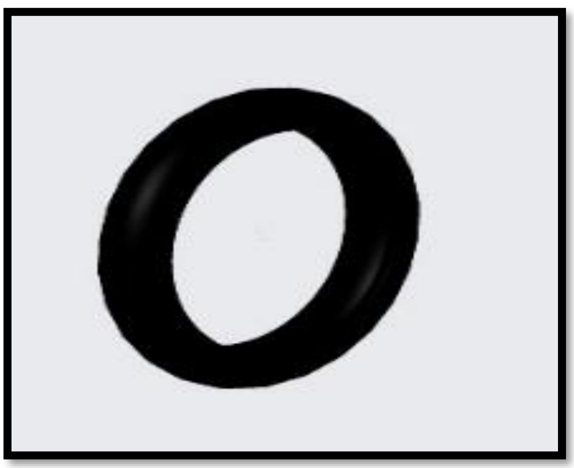
L. Calculating optimum brake balance:
Under static conditions,
$(F_{tyre,f} / V_f) = (F_{tyre,r} / V_r)$
However, as the brakes are applied the effects of weight transfer must be considered, as the ratio of front and rear vertical forces will change as follows:
$(F_{tyre,f,d} / V_{f,d}) = (F_{tyre,r,d} / V_{r,d})$

M. Thermal Calculations:
Kinetic Energy (KE) = $(1/2) \times m_v \times V_v^2$
= (1/2) x 220 x (10.111)2
= 11245.555 J
Where, m_v = mass of the vehicle
V_v = velocity of vehicle
For the braking kinetic energy is converted into thermal energy,
$KE = m_b \times C_p \times \Delta T_b$
$\Delta T_b = (KE) / (m_b \times C_p)$
= 11245.555 / (0.683 x 450).
= 36 °C
Where,
m_b = mass of braking system components which absorbs energy
C_p = specific heat of braking system components which absorbs energy

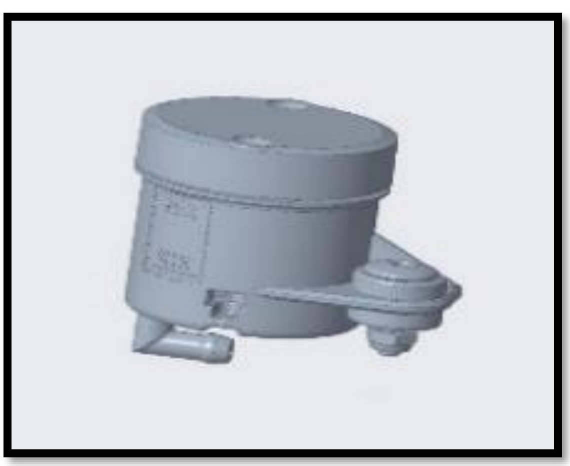
ΔTb = Temperature increase in the braking system components which absorbs energy.

CAD MODEL

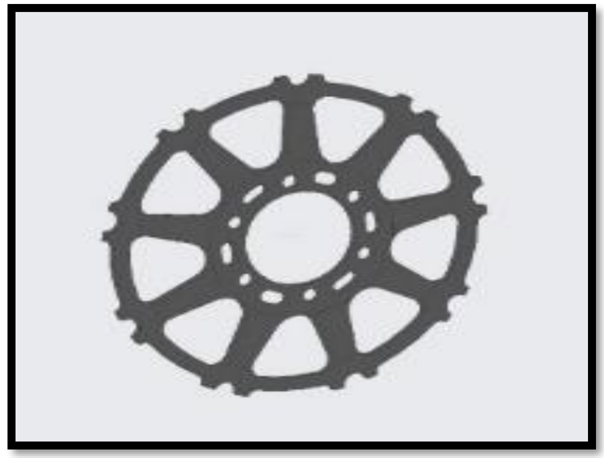
1.1 Drum of Wheel



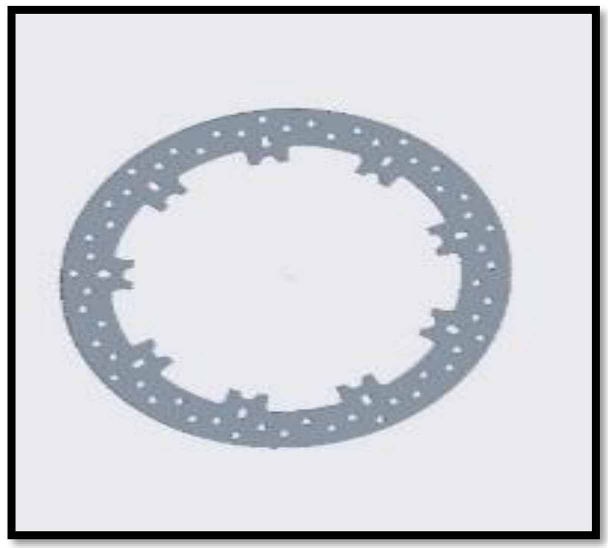
1.2 Drum tube



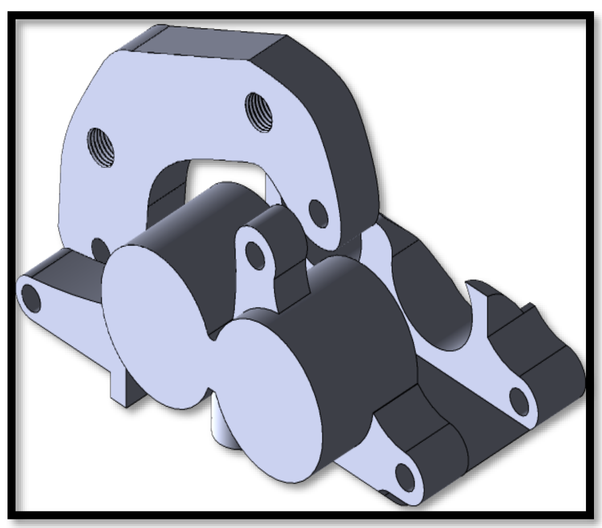
1.3 Brake pad piston



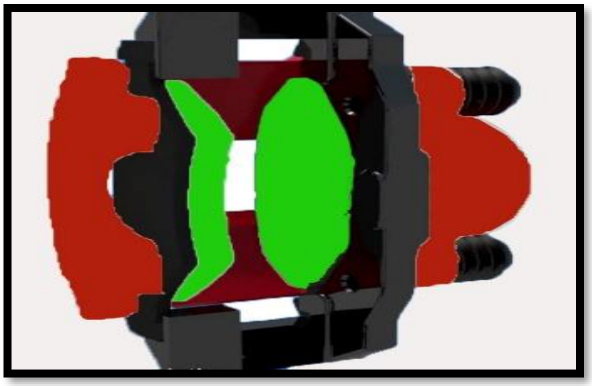
1.4 Disc rotor plate



1.4 disc plate



1.5 Dual piston



1.6 Flexible Caliper Assembly

1.7 Assembly part of automotive disk brake system

CONCLUSION

Now a days automotive disc brakes are mostly used in vehicles to reduce the speed quickly.
Automotive disc brakes are huge successful for the automotive industry.
With this project we achieved a safe durable and variable design for a rotor component in a automotive disc brake system taking in consideration the forces exerted for all the components in the brake system.
We are concerned with the points of safety, we calculated forces and analysis exerted on this using material.

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