Design and Simulation of Disc Brake Rotor by Using Taguchi Analysis to Study the Major Influence on Mechanical Behaviour of Bajaj Pulsar 150cc

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ABSTRACT: Braking system is the major part of the vehicle in order to slow down and stop the vehicle. Disc brakes were most popular on sports cars when Disc brakes were first introduced, since these vehicles are more demanding about brake performance. Disc brakes are more common form in most passenger vehicles, although many (particularly light weight vehicles) use drum brakes on the rear wheels to keep costs and weight down as well as to simplify the provisions for a parking brake. As the front brakes required most of the braking effort, this can be a reasonable compromise. Many early implementations for automobiles located the brakes on the inboard side of the driveshaft, near the differential, while most brakes today are located inside the wheels. An inboard location reduces the unsparing weight and eliminates a source of heat transfer to the tires. Here we are studying the disc brake of BAJAJ pulsar by suing Taguchi analysis to optimize the disc for maximum brake condition. To optimize the disc brake we have taken the 3 * 3 TAGUCHI method. We have taken the outer radius and inner radius and thickness of the disc brake. We have optimized for better performance.

Key words: Design, Simulation, Taguchi, Mechanical Behaviour, Breaking system

I. INTRODUCTION

In recent years, brake systems have undergone tremendous changes in terms of performance, technology, design and safety.

A brake is device means of which artificial frictional resistance is applied to moving machine member, in order to stop the motion of a machine. In the process of performing this function, the brakes absorb either kinetic energy of the moving member or the potential energy given up by objects being lowered by hoists, elevators etc.

The energy absorbed by brake is dissipated in the form of heat. This heat is dissipated in the surrounding atmosphere to stop the vehicle, so the brake system should have following requirements:

- The brake must be strong enough to stop the vehicle with in minimum distance in an emergency.
- The driver must have proper control over the vehicle during braking and vehicle must not skid.
- The brakes must have well anti-fade characteristics i.e. their effectiveness should not decrease with Constant prolonged application.

1.1 PRINCIPLE OF BRAKING

Braking system is necessary in an automobile for stopping the vehicle. Brakes are applied on the wheels to stop or to slow down the vehicle.

"The kinetic energy due to motion of the vehicle is dissipated in the form of heat energy due to friction between moving parts (Wheel or Wheel drum) and stationary parts of vehicle (brake shoes)".

Brakes operate most effectively when they are applied in manner so that wheels do not lock completely but continue to roll without slipping on the surface of road.

1.2 FUNCTION OF VEHICLE BRAKING

- To slow down or stop the vehicle in the possible time at the time of need.
- To control the speed of vehicle at turns and also at the time of driving down on hill slope.

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1.3 CLASSIFICATION OF BRAKES

On the basis of method of Actuation,

- Foot brake (also called service brake) operated by foot pedal.
- Hand brake (also called parking brake) operated by hand.

On the basis of Mode operation,

- Mechanical brakes.
- Hydraulic brakes.
- Air Brakes.
- Vacuum brakes.
- Electric brakes.

On the basis of Action on Front or Rear Wheels,

- Front -wheel brakes
- Rear – wheel brakes

On the Basis of method of Application of Braking Contact,

- Internally – expending brakes
- externally – contracting brakes

Mechanical Brakes

Internal expending brake shoe commonly used in automobiles. In an automobile, the wheel is fitted on a wheel drum. The brake shoes come in contact with inner surface of this drum to apply brakes. The whole assembly consists of a pair of brake shoes along with brake linings, a retractor spring two anchor pins a cam and a brake drum. Brake linings are fitted on outer surface of each brake shoe. The brake shoes are hinged at one end by anchor pins. Other end of brake shoe is operated by a cam to expand it out against brake drum. A retracting spring brings back shoes in their original position when brakes are not applied . The brake drum closes inside it the whole mechanism to protect it from dust and first . A plate holds whole assembly and fits to car axle. It acts as a base to fasten the brake shoes and other operating mechanism.

Internal expanding shoe brakes are the generally used braking system in automobiles. In an automobile, the wheel is fitted on a wheel drum. The brake shoes are fitted in contact with inner surface of this drum to apply brakes.

The construction of mechanical disk brake is shown in picture. The whole assembly contains of a pair of brake shoes with brake linings, two anchor pins and retractor spring, a cam and a brake drum. Brake linings are attached on outer surface of each brake shoe.

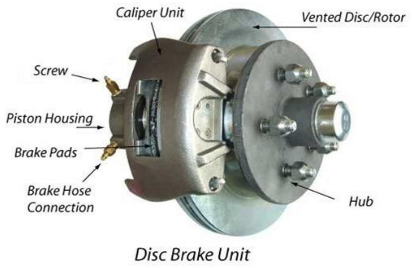

Figure 1.1 Mechanical Brake

The brake shoes are hinged at one end by means of anchor pins. Last end of brake shoe is functioned by a cam to expand it out counter direction to brake drum. Retracting springs provided are used for bringing the shoes to their original position when brakes are not applied. The brake drum closes inside it the entire mechanism to protect it from dust and sand. A plate holds the total assembly and fits to car axle. It also acts as a base to fasten the brake shoes and other operating mechanism.

Hydraulic Brakes

The brakes which are actuated by the hydraulic pressure are called hydraulic brakes. Hydraulic brakes are commonly used in the automobiles. Hydraulic brakes work on the principle of Pascal's law which states that

"Pressure at a point in a fluid is equal in all directions in space". According to this law when pressure is applied on a fluid it travels equally in all directions so that uniform braking action is applied on all four wheels.

A hydraulic braking system transmits brake-pedal force to the wheel brakes through pressurized fluid, converting the fluid pressure into useful work of braking at the wheels. A simple, single-line hydraulic layout used to operate a drum and disc brake system is illustrated. The brake pedal relays the driver's foot effort to the master-cylinder piston, which compresses the brake fluid. This fluid pressure is equally transmitted throughout the fluid to the front disc-caliper pistons and to the rear wheel-cylinder pistons.

As per the regulations a separate mechanical parking brake must be incorporated with at least two wheels. This provision also allows the driver to stop the vehicle in the event of failure of the hydraulic brake system.

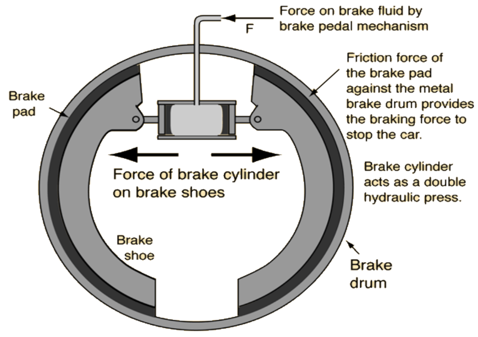

Figure 1.2 Hydraulic Brakes

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Disc brake

Modern motor bikes are fitted with disc brakes instead of conventional drum type brakes. Discbrake consists of a rotating disc and two friction pads which are actuated by hydraulic braking system as described earlier. The friction pads remain free on each side of disc when brakes are no applied. They rub against disc when brakes are applied to stop the vehicle. These brakes are applied in the same manner as that of hydraulic. But mechanism of stopping vehicle isdifferent than that of drum brakes.

The most common type of disc brake on modern bikes is the single-piston floating calliper. In this article, we will learn all about this type of disc brake design.

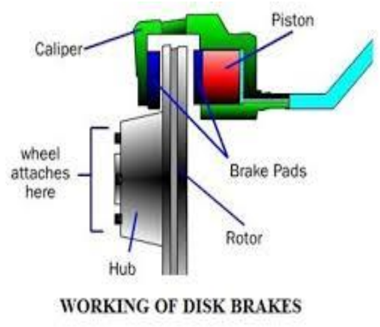

Figure 1.3 Disc Brake

1.4 PRINCIPLE OF FRICTION RELATED OF THE DISC BRAKES

Friction is a force that resists the movement of one surface over another. In some instances it can be desirable; but more often is not desirable. It is caused by surface rough spots that lock together. These spots can be microscopically small which is why even surfaces that seem to be smooth can experience friction.

Friction is a measure of how hard it is to slide one object over another. Take a look at the figure below. Both of the blocks are made from the same material, but one is heavier. I think we all know which one will be harder for the bulldozer to push.

Different materials have different microscopic structures; for instance, it is harder to slide rubber against rubber than it is to slide steel against steel. The type of material determines the coefficient of friction, the ratio of the force required to slide the block to the block's weight.

(a) Low coefficient of friction for a pair of surface means they can move easily over each other.

(b) High coefficient of frictionfor pair of surface means they cannot move easily over each other.

1.5 TYPES OF DISC BRAKE

SOLID DISC BRAKE

One slightly tarnished, solid disc of metal. Vented discs, on the other hand more like two discs of metal with ribs in between, allowing air to flow through and provide a cooling effect. These are consequently generally much thicker than solid discs.

Figure 1.4Solid Disc

VENTILATED DISC BRAKE

Both will be able to have the same amount of braking force applied to them, but the vented discs are able to shed the heat build-up more quickly than solid discs which leads to a longer period of time before brake fade becomes an issue and more consistent braking accordingly.

Figure 1.5Ventilated Disc

Early brake shoes contained asbestos. When working on brake systems of older cars, care must be taken not to inhale any dust present in the brake assembly. The United States Federal Government began to regulate asbestos production, and brake manufacturers had to switch to non-asbestos linings. Owners initially complained of poor braking with the replacements; however, technology eventually advanced to compensate. A majority of daily-driven older vehicles have been fitter with asbestos-free linings. Many other countries also limit the use of asbestos in brakes.

II. MATERIAL DEFINITION

All elements are defined by nodes, which have only their location defined. In the case of plate and shell elements there is no indication of thickness. This thickness can be given as element property. Property tables for a particular property set 1-D have to input. Different types of elements have different properties.

Engineering Application

1. Design of aircraft and aerospace structures for minimum weight.

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2. Finding the optimal trajectories of space vehicles.

3. Optimal production planning, controlling, and scheduling.

4. Design of optimization pipeline networks for process industries. Etc.,

V. M. Phalle, S. S. Mantha et.al (2010) Presented a paper, Comparative Frictional Analysis of Automobile Drum and Disc Brakes In this paper Todays technological developments in the vehicle technology seek better control on active safety by addressing the need for better and safer braking systems. The braking system is integral and vital part of the active safety control of vehicles. The selection of drum brakes for rear wheels and disc brakes for front wheels has been the trend in the earlier vehicles. Better heat dissipation achieved by disc brakes is responsible for the wide use of disc braking system in the modern vehicles. The functional dependence of the friction coefficient upon a large variety of parameters including sliding speed, acceleration, critical sliding distance, temperature, normal load, humidity, surface preparation and of course material combination. The variations in the friction coefficient are highly dependent on the frictional materials and The study presents an estimation methodology of the friction coefficient for drum-shoe and rotor disc pad interface in the automobiles. The output from deduced equations was compared with the similar obtained from virtual Simulink models for drum and disc brakes. Guru Murthy Nathi, T.N. Charyulu, K. Gowtham, p. Sathish Reddy(2012) The motive of coupled structural and thermal analysis is to study and evaluate the performance under severe braking conditions and thereby assist in disc rotor design and analysis. A transient thermal analysis has been carried out to investigate the temperature variation across the disc using axisymmetric elements. As a future work, a complicated model of Ventilated disc brake can be taken and there by forced convection is to be considered in the analysis is complicated by considering variable thermal conductivity, variable specific heatand non-uniform deceleration of the vehicle. Chetan T. Jadav, K.R. Gawandeet.al(2014)have worked on optimization of disc brakes. Their study showed that by keeping the braking torque constant if we reduce the diameter of disc rotor and increase the friction pad area then we can reduce the cost and weight of disc assembly up to some extent. Cost of disc is depends on so many factors some factor like transportation cost, material handling cost, different kinds of taxes etc. So if we reduce the diameter of disc we can reduce the materials consumption and ultimately we can reduce the cost of disc. A disc brake is a wheel brake which slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of callipers. To stop the vehicle, friction material in the form of brake pads is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. Friction causes the disc brake and attached wheel to slow or stop. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled and disc brakes recover more quickly from immersion. The brake disc is the disc component of a disc brake against which the brake pads are applied. Generally, the disc rotor is made of grey cast iron and is either solid or ventilated. The ventilated type rotor consists of a wider with cooling fins cast through the middle to ensure good cooling. Some ventilated rotors have spiral fins which creates more air flow and better cooling. BorchateSourabh Shivaji, N.S. Hanamapure and Swapnil S. Kulkarni et.al (2014) have reduced the thermal stress disc when it is working at high temperature. In this dissertation work it is proposed to consider any two/three/four wheels vehicle which is used with disc brake now- a-days on the road. For that vehicle it is proposed to carry out Reduction of weight and increase cooling effect of disc rotor So as it will enhance the performance of disc brake. For this dissertation work following proposed methodology is adopted. A suitable Solver would be employed for securing the simulated result. Performed using the finite element analysis. To analyse the thermoelastic phenomenon occurring in the disk brakes , the occupied heat conduction is solved with contact problem. Also, thermoelastic instability (TIE) phenomenon is investigated in the present study, and the influence of the materials properties on the thermoelastic behaviour is investigated to facilities the conceptual design of the disc brake system. R. J. Patil, P.R. Sonawaneet.al(2014) is to investigate and analyse the temperature distribution of rotor disc during braking operation. The work uses the finite element analysis techniques to predict the temperature distribution on the full and ventilated brake disc and to identify the critical temperature of the rotor by holding account certain parameter such as the materials used, the geometric design of the disc and the mode of braking. The analysis also gives us, the heat flux distribution for the two discs. The initial heat flux q0 into the rotor face is directly calculated using the formula. A. Belhcoine, A.R. Abu Baker, M. Bouchetraet.al (2014) This works on Design Modification & Optimization of Disc Brake Rotor deals with the study of disc brake rotor by modelling & analysis of different shapes of slots of different vehicles disc brake rotor with same outer diameter &inner mounting position of holes on wheel hub. Therefore, it gives optimal stress, deformation & weight of the modified disc brake rotor & also good heat dissipation.

III. METHODOLOGY

3.1GENERAL OBJECTIVES

- Surface Wear analysis of disc brake and drum brake

3.2 THE SPECIFIC OBJECTIVES OF THIS THESIS WORK

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I. Force analysis.

II. Stress analysis.

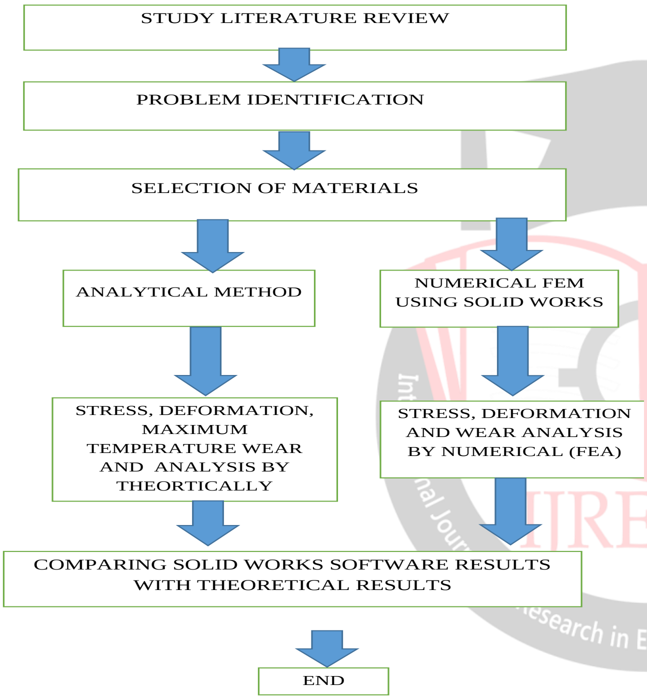
III. Identifying the critical point of Disc brake and Drum brake wear.

a. Modelling of disc brake and drum brake system to study contact using solid works software to study the stress analysis.

IV. To estimate of stress in the contact condition in disc brake and drum brake by finite element method using SOLID WORKS finite element software.

V. Comparing the solid works software results with theoretical formulation.

3.3 METHODOLOGY OF WORK DONE RELATED TO THIS WORK



DESIGN AND ANALYSIS OF DISC BRAKE

The calliper disc brake and drum brake is operated hydraulically. Two pads are pressed against opposite side of the disc brakes to provide a braking torque. In this chapter to design the dimension of disc brakes and drum brake such as outside and inside diameter, thickness,etc. and also determine the analysis parameter such maximum temperature, compressive stress, heat flux and deformation etc.

INPUT PARAMETER

Brake disc

Mass of vehicle = 150 kg
Mass of disc = 2.483 kg
Initial velocity = 20 m/sec
Radius of wheel = 350 mm

Drum brake

Mass of vehicle = 150 kg
Mass of disc = 4.23 kg
Initial velocity = 20 m/sec
Radius of wheel = 420 mm

4.1 DETERMINATION OF DISC BRAKING TORQUE

Braking distance, d = $\frac{v^2}{2\mu g}$

(4.1)

$= \frac{20^2}{(2\times0.7\times9.81)}$

d= 4.04 m

After finding out braking distance we must calculate deceleration,

Deceleration, a $= \frac{v^2}{2d} (or) \mu g$

(4.2)

$= \frac{(20^2)}{(2\times14.04)}$

a = 6.867m/s²

Braking force:

Already we know that newton's second law,

Braking force, F_b = ma

(4.3)

= 150×6.867

F_b = 1030.05N

Braking Torque:

Braking torque, T= force × radius of wheel

= 1030.05 × 0.35

T=360.517 N-m

Torque equation from reference (9),

Torque = μ P R_f

(4.4)

T = μ × p_{avg} × pad area × R_f

$= \mu \times 0.89 \times p_{max} \times \frac{1}{2}\ \theta\ (R_o^2 - R_i^2) \times \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right)$

360.517 = 0.4 × 0.89 × 0.65 X 10^6 × 1/2 × 54 × 2/3 × ($R_o^3 - R_i^3$)

$360.517 = \frac{12.495\times10^6}{3} \times R_o^3 - R_i^3$

$R_o^3 - R_i^3 = 8.655\times10^{-4}$

From Reference (7),

We know the following condition,

Ri/Ro = 0.80

if $\frac{p_{avg}}{p_{max}}$ = 0.89 , so

$R_i = 0.80\ R_o$

$R_o^3 - R_i^3 = 8.655\text{x}\ 10^{-4}$

Substitute 'R_i' value in above equation , we get

$R_o^3 - 0.80^3\ R_o^3 = 8.655 \text{ x } 10^{-4}$

$R_o^3\ (1-0.80^3) = 8.655 \text{ x } 10^{-4}$

$R_o^3 = \frac{8.655\times10^{-4}}{(1-0.80^3)}$

$R_o^3 = \frac{8.655\times10^{-4}}{(0.488)}$

R_o= 0.135 m

$R_i = 0.80 \times R_o$

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= 0.80 × 0.135

R_i = 0.108 m

Outside radius = 0.135 m

Inner radius = 0.108 m

Thickness of Disc

Mass of the disc = 2.48kg

Density, $\rho = \frac{m}{v}$

$7100 = \frac{1.43}{Area \times t}$

Area of disc, $A = \frac{\pi}{4}(D^2 - d^2)$ (4.5)

$= \frac{\pi}{4}(0.270^2 - 0.216^2)$

$A = 0.02061\ m^2$

$7100 = \frac{1.43}{0.02061 \times t}$

$t = \frac{1.43}{0.02061 \times 7100}$

$t = 7.995 \times 10^{-3}$m

$t = 4.2 \times 10^{-3}$m

- Outside Radius= 0 .135 m
- Inner Radius= 0.108 m
- Thickness of disc = 4.4 mm

Determination	**Brake disc**
Inner diameter	**270mm**
Outer diameter	**216mm**
Thickness	**3.8mm**

Determination	**Brake disc**
Braking distance	**4.04 m**
Braking force	**1050N**
Braking torque	**360.517 N-m**
Heat flux	**4.5x106**

DETERMINATION OF CONTACT STRESS

From following Motion equation braking time should be calculated, that follows

$S = ut + \frac{1}{2} g t^2$ (4.6)

$14.04 = 13.88 + \frac{1}{2} \times 9.81\, t^2$

t = 3.65 s

CONTACT STRESS FOR GREY CAST IRON

From tribology journal we can take this heat flux formula,

Heat flux of the disc,

$$q = \frac{1-\varphi}{2} \cdot \frac{mgz}{2} \cdot v_o / A_d \varepsilon_p \quad (4.7)$$

$$= \frac{1-0.7}{2} \cdot \frac{(1.43 \times 9.81 \times \frac{6.867}{9.81})}{2 \times 0.0206} \cdot (25/0.5)$$

q = 1.356x 10^6 W/m^2

Also consider the temperature distribution among the rotor disc following Formula can be taken from reference journal,

Maximum Temperature,

$$T_{max} = \frac{0.527 \times q \times \sqrt{t}}{\sqrt{\rho c p k}} + T_{amb} \quad (4.8)$$

$$T_{max} = \frac{0.527 \times 1.356 \times 10^6 \times \sqrt{2.45}}{\sqrt{7100 \times 586 \times 54}} + 30$$

T_{max}= 136.208 °C

Maximum temperature, **T_{max}= 136.208 °C**

Compressive stress can be found by following formula,

$$\sigma = \frac{E}{1-\mu} \times \alpha \times \Delta T \quad (4.9)$$

$$\Delta T = \frac{E}{cm} = \frac{125 \times 10^5}{380 \times 1.17}$$

= 47.4134

$$\sigma = \frac{125 \times 10^9}{(1-0.27)} \times 0.12 \times 10^{-6} \times 47.4134$$

= 64.28Mpa

Compressive stress, **σ = 64.28Mpa**

Deformation can be found by following formula,

$$\text{Youngs modulus, } E = \frac{\sigma}{e} = \frac{108.6 \times 10^6}{\frac{\delta d}{0.27}}$$

$$110 \times 10^9 = \frac{108.6 \times 10^6}{\frac{\delta d}{0.27}}$$

$$\frac{\delta d}{0.27} = \frac{108.6 \times 10^6}{125 \times 10^9}$$

Deformation, **δd= 0.342 mm**

CONTACT STRESS FOR ALUMINIUMALLOY

Also consider the temperature distribution among the rotor disc following Formula can be taken from reference journal,

Maximum Temperature,

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

$$T_{max} = \frac{0.527 \times 1.356 \times 10^6 \times \sqrt{2.45}}{\sqrt{2700 \times 900 \times 155}} + 30$$

T_{max}= 154.929 °C

Maximum temperature, **T_{max}= 154.929 °C**

Compressive stress can be found by following formula,

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

$$\sigma = \frac{72 \times 10^9}{(1-0.33)} \times 0.24 \times 10^{-6} \times 47.4134$$

= 63.82 Mpa

Compressive stress, **σ = 63.82 Mpa**

Deformation can be found by following formula,

$$\text{Young's modulus, } E = \frac{\sigma}{e} = \frac{122.3 \times 10^6}{\frac{\delta d}{0.27}}$$

$$72 \times 10^9 = \frac{122.3 \times 10^6}{\frac{\delta d}{0.27}}$$

$$\frac{\delta d}{0.27} = \frac{122.3 \times 10^6}{72 \times 10^9}$$

Deformation, **δ D= 0.323 mm**

CONTACT STRESS FOR CHROME STAINLESS STEEL

Also consider the temperature distribution among the rotor disc following Formula can be taken from reference journal,

Maximum Temperature,

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

$$T_{max} = \frac{0.527 \times 1.356 \times 10^6 \times \sqrt{3.65}}{\sqrt{7800 \times 460 \times 18}} + 30$$

T_{max}= 209.07 °C

Maximum temperature, **T_{max}= 209.07 °C**

Compressive stress can be found by following formula,

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

$$\sigma = \frac{193 \times 10^9}{(1-0.28)} \times 0.11 \times 10^{-6} \times 47.4134$$

$$= 62.32\text{Mpa}$$

Compressive stress, **σ = 62.32Mpa**

Deformation can be found by following formula,

Youngs modulus, $E = \frac{\sigma}{e} = \frac{139.8 \times 10^6}{\frac{\delta d}{0.27}}$

$$193 \times 10^9 = \frac{139.8 \times 10^6}{\frac{\delta d}{0.27}}$$

$$\frac{\delta d}{0.27} = \frac{139.8 \times 10^6}{193 \times 10^9}$$

Deformation, **δd= 0.132mm**

CONTACT STRESS FOR TITANIUM 6AL-4V

Also consider the temperature distribution among the rotor disc following

Formulacan be taken from reference journal,

Maximum Temperature,

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

$$T_{max} = \frac{0.527 \times 4.5 \times 10^6 \times \sqrt{2.65}}{\sqrt{7850 \times 475 \times 44.5 \times 0.65}} + 30$$

T_{max}= 128.26°C

Maximum temperature,**T_{max}= 128.26 ° C**

Compressive stress can be found by following formula,

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

$$\sigma = \frac{2.1 \times 10^5}{(1-0.29)} \times 1.23 \times 10^{-5} \times 47.4134$$

$$= 61.78\text{Mpa}$$

Compressive stress, **σ = 61.78Mpa**

Deformation can be found by following formula,

Youngs modulus, $E = \frac{\sigma}{e} = \frac{102.4 \times 10^6}{\frac{\delta d}{0.27}}$

$$2.1 \times 10^5 = \frac{102.4 \times 10^6}{\frac{\delta d}{0.27}}$$

$$\frac{\delta d}{0.27} = \frac{172.4 \times 10^6}{2.1 \times 10^9}$$

Deformation, **δd= 0.22 mm**

CONTACT STRESS FOR COPPER ALLOY

Also consider the temperature distribution among the rotor disc following

formula can be taken from reference journal,

Maximum Temperature,

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

$$T_{max} = \frac{0.527 \times 1.356 \times 10^6 \times \sqrt{3.65}}{\sqrt{7750 \times 486 \times 0.65 \times 52}} + 30$$

T_{max}=122°C

Maximum temperature, **T_{max} =122°C**

Compressive stress can be found by following formula,

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

$$\sigma = \frac{2.1 \times 10^5}{(1-0.29)} \times 1.25 \times 10^{-5} \times 47.4134$$

$$= 58.21\text{Mpa}$$

Compressive stress, **σ = 58.21Mpa**

Deformation can be found by following formula,

Youngs modulus, $E = \frac{\sigma}{e} = \frac{286.2 \times 10^6}{\frac{\delta d}{0.27}}$

$$2.1 \times 10^5 = \frac{286.2 \times 10^6}{\frac{\delta d}{0.27}}$$

$$\frac{\delta d}{0.27} = \frac{286.2 \times 10^6}{2.1 \times 10^5}$$

Deformation, **δd= 0.20 mm**

Materials	Compressive stress		Deformation		Maximum temperatur (°C)
	Theoretical (Mpa)	Analysis (Mpa)	Theoretical (mm)	Analysis (mm)	
Greycast iron	64.28	62.14	0.34	0.337	136.208
Aluminium alloy	63.82	61.39	0.32	0.3108	154.929
Chrome stainless steel	62.32	61.75	0.13	0.1108	209.07
Ti-6Al-4v	61.78	61.56	0.22	0.2079	128.26
copper	58.21	60.89	0.20	0.1893	138.42

In table 4.3 the modelled disc brake is analysed in Finite element software solid works v 2016 and their results are tabulated. The simulation is done with grey cast iron, aluminium alloy, chrome stainless steel, titanium alloy and copper alloy.

Figure 4.1 shows the bar chart representation of deformation in modelled disc brake with respective to various materials.

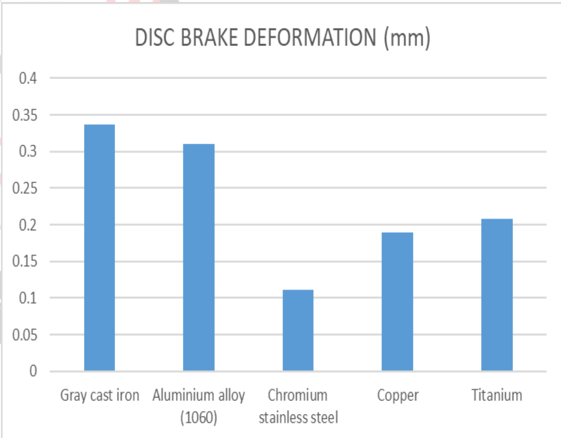

Figure 4.1 Comparison of disc brake deformation

4.3 WEAR ANALYSIS OF DISC BRAKE

Wear Analysis of Grey cast iron Brake disc as given below,

Archards wear law equation,

$$V = k \times \frac{F_n}{H} \times S$$

Where, v = wear volume

F_n = normal force

H = hardness of the material

The sliding distance S can be given as

$$ds = r \times d\theta$$

$$V = \frac{K \times F_n \times r\theta}{H}$$

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The angular displacement θ can be expressed as:

$$\theta = \omega t$$

$$V = \frac{K \times F_n \times r\omega t}{H}$$

The rate of wear is

$$Q = \frac{K \times F_n \times r\omega}{H}$$

Where, r is basic circle radius

θ is brake disc angle

V = 0.234μm/sec.

MODELLING OF DISC BRAKE USING SOLIDWORKS SOFTWARE

The inner radius, outer radius and thickness of disc are as 0.108m, 0.135m, and 0.08m, respectively. The hydraulic pressure is applied to the boundary along radius of the piston side pad and the immobility condition in the axial direction is applied to the boundary along the radius of the finger side one. The heat finite element model of disc brakes with boundary condition are shown. The convective boundary condition are imposed on all boundaries to consider more realistic heat conditions. The initial temperature is T=27^0 C in the study.

Figure 5.1 Isometric view of Disc brake model

5.2 ANALYSIS RESULTS OF DISC BRAKE

Material properties of Grey cast iron

Young's modulus, E = 1.25 × 10^9 N/m^2

Density, ρ =7200 kg/m^3

Specific heat, c = 510J/kg K

Thermal conductivity, k =45 w/m K

Thermal co-efficient of expansion, α = 1.2 × 10^{-5}/ ^{o}C

Poisson ratio, μ = 0.27

ANSYS analysis

CASE 1: Deformation in grey cast iron disc.

CASE 2: Stress in grey cast iron disc.

CASE 1: Deformation in grey cast iron disc.

Total deformation of Grey cast iron brake disc for applied force of 1030.4 N has been shown in figure 5.2,

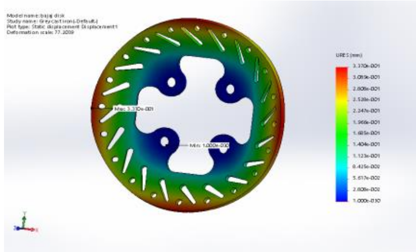

Figure 5.2 Deformation in grey cast iron disc

CASE 2: Stress in grey cast iron disc.

The von missesdistribution for disc brake is shown in figure 5.3,

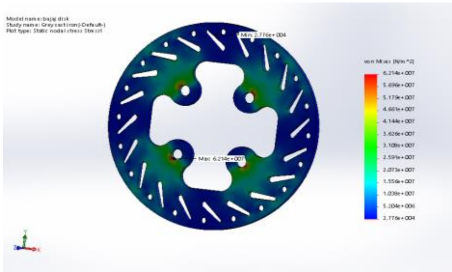

Figure 5.3 Stress in grey cast iron disc

The disc brake surface von misses stress is Minimum 60.24 Mpa and Maximum 62.14 Mpa.

5.3 WEAR ANALYSIS OF DISC BRAKE USING SOLID WORKS DESIGN

Stress Formula as given below,

$$\sigma = \frac{-Pmax}{\sqrt{1+\frac{z^2}{b^2}}}$$

Where, P_{max}= maximum pressure

z = module

b = contact width

Contact Width, b = $2\sqrt{\frac{2F}{l\pi}\frac{P(\frac{(1-\mu_1)}{E1}+\frac{(1-\mu_2)}{E2})}{(\frac{1}{d1}+\frac{1}{d2})}}$

Section Module, z = (1/12) (2.5t(6t)^3 – 1.5t(4t)^3) / (6t/2)

Where, t = thickness

The Compressive Stress from Solid works value is 82.24Mpa

Then, the **contact pressure is 72.86N/mm^2**

The contact pressure formula is,

$$P_0 = \frac{2Fn}{\pi bl}$$

Where, F_n= normal force

b = contact width

l = Contact length of the Brake disc

From equation (5.4),

Braking Normal Force, F_n = 981.2 N.

The Von- Misses Stress are given by equations 5.5, 5.6, 5.7,

$$\sigma_x = -2.v.Pmax\,[\sqrt{1} + \frac{z^2}{b^2} - \{\frac{z}{b}\}]$$

$$\sigma_y = -Pmax\,[\frac{1+2\frac{z^2}{b^2}}{\sqrt{1+\frac{z^2}{b^2}}} - 2\{\frac{z}{b}\}]$$

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Vonmises stress = 30.26 MPa

Archards wear law equation,

$$V = k \times \frac{F_n}{H} \times S$$

Where, v = wear volume

F_n = normal force

H = hardness of the material

The sliding distance S can be given as

$$ds = r \times d\theta$$

$$V = \frac{K \times F_n \times r\theta}{H}$$

The angular displacement θ can be expressed as:

$$\theta = \omega t$$

$$V = \frac{K \times F_n \times r\omega t}{H}$$

The rate of wear is

$$Q = \frac{K \times F_n \times r\omega}{H}$$

Where, r is basic circle radius

$\theta$is Brake disc Angle

V = 0.023 μm/sec.

The detailed summary of taguchi analysis is given below

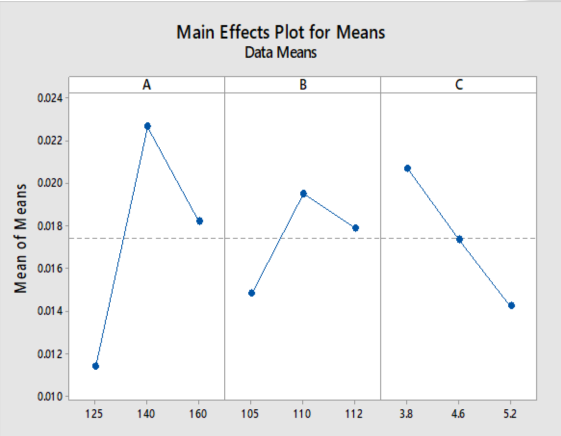

Figure 7.19Main effect plot for means

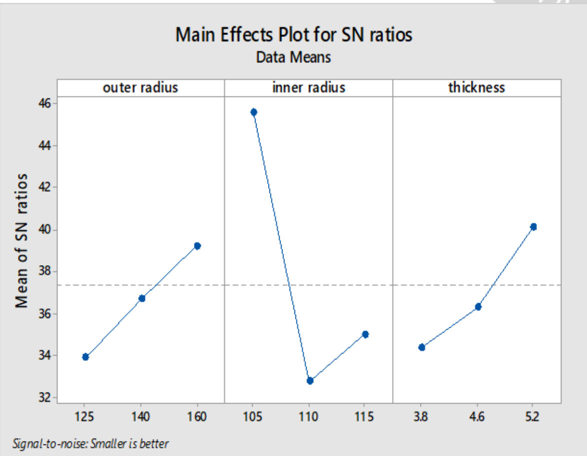

Figure 7.6 Main effect for SN ratios

From the results, the evaluation of wear rates under various width and radius , the contact pressure and wear rate decreases with increase in inner radius wear occur mainly on the plate surface when it is subjected to enormous load when the thickness of plate is low. It is concluded that the wear rate can be reduced by having a disc plate with its film thickness can be increased and also by increasing the inner radius.

IV. CONCLUSION

- In this paper, the disc brake for the Bajaj pulsar 150cc has been designed and it is analysed using theoretical and finite element method.
- Grey cast iron disc brake is suggested for this bike based on the Von misses stress, deformation, maximum temperature and wear of the disc and drum.
- In addition to verify the virtual simulation were performed and compared with theoretical calculations.

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