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Design of Rotor Disc Brake using Structural & Thermal Analysis

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Abstract

These day technologies go beyond us. On comparison of vehicle production before 20-25 years ago & later, we find abundant difference in aspect of comfort, economy, function & particularly in safety. Instead of having air bag, good suspension systems, good handling and safe cornering, there is one most critical system in the vehicle which is brake systems. In old days, brake system is very simple and less efficient.Even various brake systems are existed now days, disc brake system has also got crucial attention with progressive development in its performance.This reserach studies the problem existed in Yamaha Fz-25 disc brake and its rectification with the improved design of disc brake that delivered superior mechanical and thermal performances than original disc brake.The current disc rotor using in Yamaha Fz-25 has a load of 1.05 kg which is heavier than other's company bike of same power. So, 4 numbers of different disc rotor models with same geometric dimension with original disc rotor are designed with different ventilation shape to reduce the weight than original disc rotor. The 3D solid modelling of CAD is designed in Autocad 2017 software whereas the finite modelling of CADs and simulation on structural and thermal strategies are based on ANSYS17 software. From the analysis, among all disc rotor models, disc rotor model 4 has highest bearing stress of 15.752 MPa, disc rotor model 2 has least deformation of 0.00426 mm, disc rotor model 1 can sustain high temperature upto 95℃ and disc rotor 4 has high heat flux of 9.79*10^5 W/m^2 where as original disc rotor has bearing stress of 13.7 MPa, deformation of 0.0072 mm, sustainable temperature of 53.8℃ and heat flux of 1.63*10^5 W/m^2.As the main purpose of this research is to select the best design that rectifies the existing problem in original disc rotor,so, disc rotor model 4 had succeeded to deliver its superior performance against the original disc rotor with increased stress by 2.05 MPa, decreased deformation by 0.0002 mm, increase of sustaining temperature upto 15.73℃ more, increasing of heat transfer by 8.16*10^5W/m^2, decreasing weight by 0.05 kg and decreasing production cost by Rs 26.9.

Keywords
Disc Brake, Yamaha Fz-25, Stress, Deformation, Temperature, Heat flux

1. Introduction

A brake is a device used in the machines to inhibit the motion by providing artificial frictional resistance to a moving member. Brake plate absorbs the kinetic energy and dissipates the heat energy. There are different types of brakes among which disc brakes, drum brakes, air brakes, and vacuum brakes are prominent. The disk brake is a round, flat piece of metal, made usually of cast iron that is attached to the wheel. When braking, the brake discs are squeezed against the wheel on either side by brake pads. Disk brakes are more effective than drum brakes.

Disc-style brakes development and use began in England in the 1890s. The first calliper-type automobile disc brake was patented by Frederick in his Birmingham, UK factory in 1902 but it was failed due to use of copper disc. The first motorcycles to use disc brakes were racing machines. The first mass-produced road-going motorcycle to support a disc-brake was on 1969 by Honda CB750. and used successfully on Lanchester cars. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled [1].

Ali Belhocine studied the thermal behavior of the full and ventilated brake discs of the vehicles using computing code ANSYS. The numerical simulation for the coupled transient thermal field and stress field is carried out by sequentially thermal-structural coupled method based on ANSYS to evaluate the stress fields and deformations which are established in the disc with the pressure on the pads The thermo-structural analysis is then used with coupling to determine the deformation established and the Von Mises stresses in the disc, the contact pressure in brake pads[2].

Lee. K. & Barber studied the thermo elastic phenomenon occurring in the disk brakes, the occupied heat conduction and elastic equations are solved with contact problems.The computational results are presented for the distribution of heat flux and temperature on each friction surface between the contacting bodies [3]. Hartsock studied thermo-elastic phenomenon of disc brake rotor & pad occurring while braking & numerical analysis of result get. In this paper they studied thermo-elastic instability occurs due to torque variation during initial rotor run out because differences in rotor thickness or coefficient of friction [4]. Dow studied analysis of load variation on disc rotor & optimization of rotor thickness. In this they analyze magnitude of speed require to cause TEI & experimentally analyze automotive disc brake system[5]. Segal research on considerable effect of temperature on brakes. In this research they found at the higher temperature braking system & properties of brake material become worse [6]. Garcia-Pozuclo Ramos studied deformation disc brake while braking a vehicle. Deformation of brake disc is one major negative factor occurs in disc brake rotor while braking [7]. Alyazeed Albatlan study effect of brake pipes thickness on brake efficiency & disc brake components.At lower load it will reduces & at higher load it will increase up to considerable amount [8].

Nagumas study on relative axial braking force on front axle of vehicle. In research they found relative force on front axle is considerably bigger than rear axle [9]. Xiaojing study experimentally on braking process.The result of test affects by friction coefficient, duration between repetitively application of brake while running & structure of stand also [10].

Aggrawal studies Disc brake basics like, Disc brake, are often used in automobile transmission system to stop moving machine. This research paper explains the design and finite element analysis (FEA) model of brake disc by which deflections in X, Y, Z direction and Von-mises stress can be calculated by applying boundary conditions.The developed method improves the understanding of the structural failure, modal prediction, operating conditions, and reduces product development time and cost [11]. The detailed drawings of all parts are to be furnished by Daniel[12] & Oder[13] present paper shows a thermal and stress analysis of a brake disc for railway vehicles using the finite element method (FEM). Performed analysis deals with two cases of braking; the first case considers braking to a standstill; the second case considers braking on a hill and maintaining a constant speed. In both cases the main boundary condition is the heat flux [14].

Friction between disc and pads always opposes motion and the heat is generated due to conversion of the kinetic energy [15]. The three-dimensional simulation of thermo-mechanical interactions on the automotive brake, showing the transient thermoelastic instability phenomenon, is presented for the first time in this academic community [16]. In this work, we will make a model of the thermomechanical behavior of the dry contact between the discs of brake pads at the time of braking phase; the strategy of calculation is based on the software Ansys [17].

Numerious of anlaysis has been already performed for its performance efficiency under different circumstances. The disc rotor in Yamaha Fz-25 has the simple designed disc rotor which is shown in figure 2. The amount of material using in current disc rotor can be minimized by redesign of disc brake rotor which will save a lots of material & cost as well. According to the study done by ANNA University[18], the overall cost of the manufacturing the disc brake for grey cast iron as given below.

S.N	Description	Cost(Rs)/kg
1	Sand Casting Cost	320
2	Machining cost	120.51
3	Material cost	132
	Total Cost	572.51

Table 1: Cost of Manufacturing of brake rotor

The details of the disc rotor considering for this research work are given below:

The initial heat flux q_o entering the disc is calculated by the following formula:

$$q_o = \frac{(1-\phi)}{2} * \frac{m*g*v_o*z}{2*A_d*e_p} \quad (1)$$

Where,m= mass of vehicle, v_o= velocity, g= acceleration due to gravity, A_d = area where brake

S.N	Description	
1	Rotor outer diameter	280 mm
2	Rotor inner diameter	226 mm
3	Rotor thickness	4 mm
4	Rotor disc material	Grey cast iron
5	Pad Brake area	2000 sq mm
6	Pad Brake material	Asbestos
7	Coefficient of friction (Wet)	0.08-0.12
8	Coefficient of friction (Dry)	0.3-0.5
9	Maximum temperature	250 C
10	Maximum pressure	1 Mpa

Table 2: : Information of disc rotor of Yamaha Fz-25

experiences e_p=Factor load distribution on the disc surface, ϕ=rate distribution of the braking forces, z= Braking effectiveness
Swapnil has provided the followng expression for the calculation of brake force.[19]

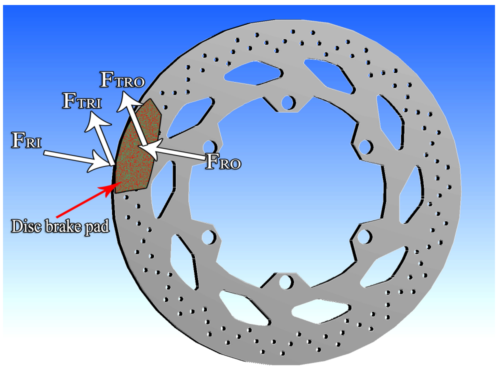

Figure 1: Forces action on disc rotor sources

Inner Tangential force between pad and rotor (Inner face), FTRI

$$FTRI = \mu 1 * FRI \tag{2}$$

Then,

$$FRI = \frac{P_{max}}{2} * A_p \tag{3}$$

In this case, FTRO equal FTRI because same normal force and same material.

$$FTRO = FTRI \tag{4}$$

Here, the assumption is made that frictional force is applied by brake pad on the disc rotor with the equal coefficient of friction Then,

$$Total force(FT) = FTRI + FTRO \tag{5}$$

$$BrakeTorque(BT) = FT * R \tag{6}$$

Where FRI = Normal force between pad brake and rotor (inner) , μ1= Coefficient of friction, TB = Brake torque,R = Radius of rotor disc, P_{max} is pressure applied on disc, A_p = area of pad brake area.

2. Methodology

The other brand of bikes like pulsar, apache, honda CBR have been using low weight disc rotor than Yamaha Fz-25 (figure 2). The problem of higher weight on Yamaha Fz-25 disc rotor can be eliminated by modifying the design of disc brake rotor that has low weight but delivering the same or superior results in performance. For this, literature review was done to gain knowledge about the disc brake, ventilation holes relating to mechanical and thermal performances. So, 4 different models of disc brake rotor have been designed for the analysis excluding the original disc. These four models have different ventlation holes pattern and also have less weight than original disc rotor. Firstly, 2D cads models was drafted and 2D was extruded to 3D models in Autocad17 to carry out the Finite Element Analysis in ANSYS17 software. 3D cads models have divided into numbers of the small elements of tetrahedron shape with the size of 4 mm. This division of the whole 3D cad models in numbers of Finite Element Modelling (FEMod) helps to analyze the affected parts during the braking activities in details. The FEMod models are subjected to the 500 N of compressive braking forces with the application of 1 MPa pressure and 140 Nm torque for structural analysis and for thermal analysis, the same FEmods are subjected to the initial heat flux of 261294.3 W/m^2 and assumed at 26°C as the initial boundary condition. The results from the analysis was interpreted in terms of stress, deformation, temperature and heat flux to select the best final disc rotor model that exhibit less deformation, bear high stress, should sustain high temperature range and maximum heat flow rate than the original disc rotor of Yamaha. The different models of disc brake rotor are as follows.

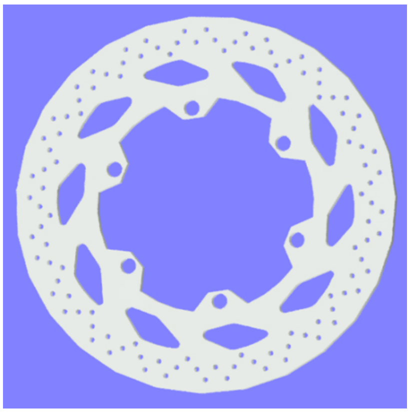

Figure 2: Original Baseline Disc Rotor

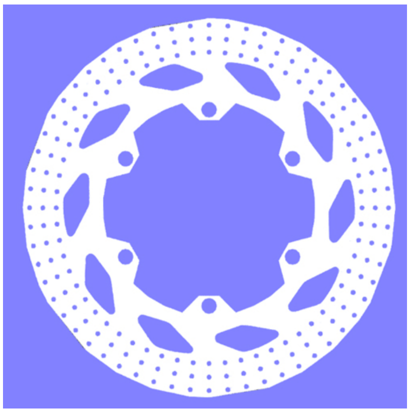

Figure 3: Disc Rotor Model 1

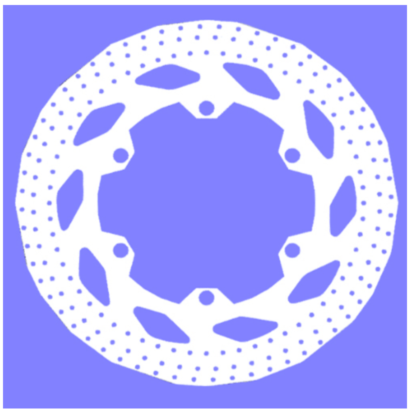

Figure 4: Disc Rotor Model 2

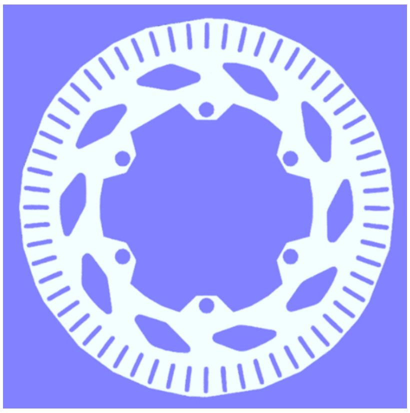

Figure 5: Disc Rotor Model 3

Figure 6: Disc Rotor Model 4

3. Mathematical Calculation

3.1 Calculation of Disc Brake Force

For the structural analysis, equal compressive force are given by disc pads on disc rotor.The disc rotor is fixed to the wheel with the help of six holes located around inner side of disc rotor. Then,
The pressure applied during brake (P_{max}) = 1 Mpa
The radius of rotor (R) = 140 mm
The coefficient of friction is assumed (μ1)= 0.5
From equation 2, 3, 4, 5, & 6, we get, **FTRI**=500 N, **FTRO**=500 N, **FT**=1000 N & **BT**=140 Nm.

3.2 Calculation of Heat Flux entering the disc

For the calculation of initial heat flux, we assume T(x, y, z) =26 °C as initial boundary condition.
Assumption:
Rate distribution of the braking forces ϕ=0.5
Factor of charge distribution of the disc e_p=0.5
Surface disc swept by the pad A_d =0.0214 m^2
Initial Speed v_o =30 km/hr
Mass of vehicle m =152 kg

Braking effectiveness, z= 0.5
From equation 1, we get the flux entering the disc q_o=261294.3 W/m^2

4. Result & Discussion

After implication of the boundary condition to the ANSYS software,the results obtained on original disc rotor and disc rotor model 4 are as follows. However, the stress, deformation, temperature, and heat flux analysis have been done on every disc rotor model which can be seen on figure 11 & figure 16. The results of all 4 models of disc rotors via analyis will be interpreted in terms of both structural and thermal analysis and will be compared to the orignal disc rotor.

4.1 Stress & Deformation Analysis

Figure 7 shows stress analysis on original disc rotor, the maximum value of stress obtained against the compressive forces of 500 N and torque of 140 Nm on original disc rotor is 13.7 MPa and the lower value of stress is 0.03 MPa. The stress is higher near the fixed support holes and lower in outer periphery of the disc rotor.

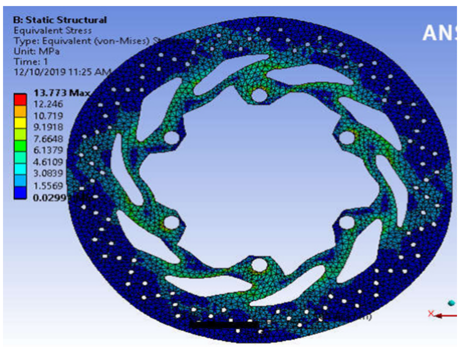

Figure 7: Stress analysis on Original Disc

The maximum value of deformation on original disc rotor is 0.0073 mm at the outer pheriphery of disc rotor as in below figiure.

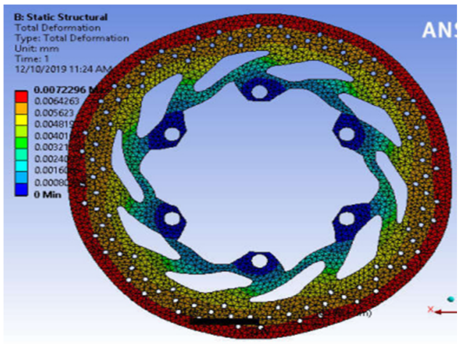

Figure 8: Deformation analysis on Original Disc

Figure 9 shows the stress analysis on disc rotor model 4 upon braking action, disc rotor nodel 4 can go upto 15.75 MPa and the minimum value of stress is 0.0253 MPa. The stress level is high on fixed support holes and low on the disc brake pad area.

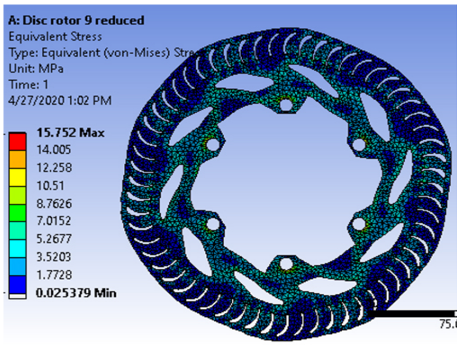

Figure 9: Stress analysis on Disc Model 4

Figure 10 shows the deformarmation analysis on disc rotor model 4 upon braking action, it shows the maximum deformation of 0.007 mm. The deformation has been increased from inner section to the outer section.

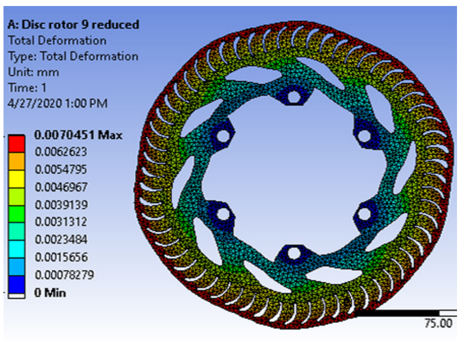

Figure 10: Deformation analysis on Disc Model 4

Similarly, disc rotor 1 can sustain the stress upto 11.064 MPa with deformation of 0.00427 mm. The stress & deformation of disc rotor 2 can go up to

11.507 MPa & 0.004269 mm. Likewise, the maximum value of stress obtained of disc rotor 3 is 11.209 MPa along with deformation of 0.0053 mm. In all disc rotor models, mostly, the area of the disc rotor where disc pad imposed its force and pressure is on lower stress value. The high value of stress can be found near the area where fixed support lies also it can be said that the deformation is continuously rising from inside to outside periphery of disc brake rotor.

In order to make comparative analysis between stress and deformation at one figure, the stress has been divided by 1000. From the below figure 11, it is found that the disc rotor model 1 and 3 behave similar characteristics in terms of stress with value about 11 MPa and disc rotor model 1 and 2 behave similar characteristics in terms of deformation with value about 0.0043 mm.

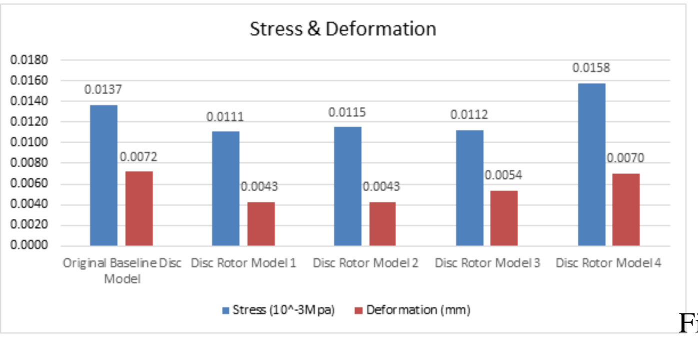

Figure 11: Combined Stress & Deformation

The disc rotor model 4 has highest stress of 15.8 MPa than all disc rotor model.The original disc rotor has highest deformation of 0.0072 mm among all of the models and similar characteristics can be observed in the deformation of disc rotor model 4 as well. Hence, only disc rotor model 4 exhibits the superior performance in terms of stress and deformation against the original disc rotor.

4.2 Heat Flux & Temperature Analysis

Figure 12 shows heat flux analysis on original disc rotor, the maximum heat flux value of the original disc rotor model is 1.63*10^5 W/m^2 and the minimum value is 0.00026 W/m^2. It shows the heat flux is maximum at disc pad area.

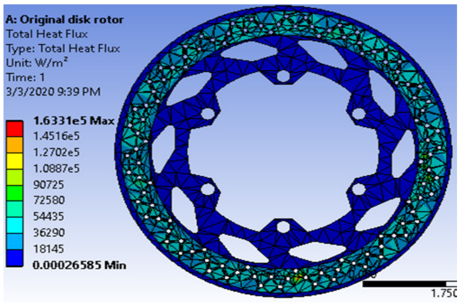

Figure 12: Heat flux analysis on Original Disc

The temperature ranges from minimum 12.98°C to maximum 53.83° on original disc rotor. Likewise, heat flux the temperature value is also high on disc pad area.

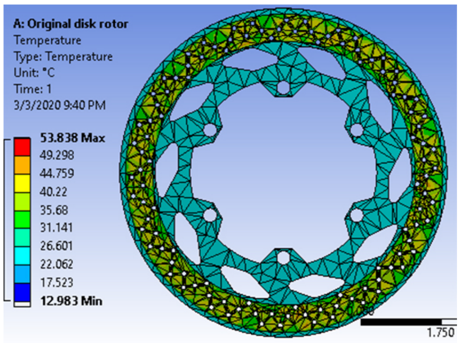

Figure 13: Temperature analysis on Original Disc

Figure 14 shows heat flux analysis on disc rotor model 4 upon braking action, disc rotor 4 can go upto 9.79*10^5 W/m^2 and the minimum value is 0.045 W/m^2. The value of heat flux is quite high than original disc rotor.

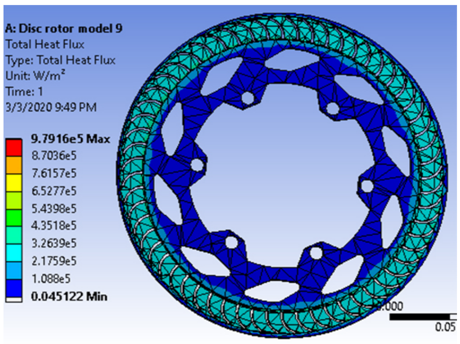

Figure 14: Heat flux analysis on Disc Model 4

Below figure indicates the thermal analysis of disc rotor model 4 in terms of temperature which ranges from minimum 24.95°C to maximum 69.53°C. The temperature level is high on disc pad area and low on the fixed support area.

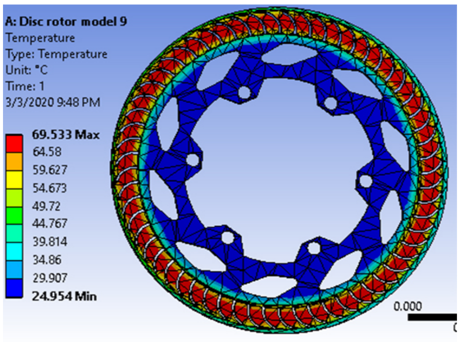

Figure 15: Temperature analysis on Disc Model 4

From the analyis in terms of temeperature & heat flux, disc rotor 1 has maximum heat flux of $6.179*10^5$ W/m^2 with temperature ranges from minimum 24.8°C to maximum 95.371°C. The maximum heat flux & temperature ranges of disc rotor 2 can go up to $5.279*10^5$ W/m^2 & minimum 24.6°C to maximum 93.371°C. Likewise, the maximum value of heat flux obtained of disc rotor 3 is $4.051*10^5$ W/m^2 along with temperature ranges from from minimum 24.7°C to maximum 70.04°C.

In order to find to make comparative analysis of all the disc rotor at one figure, the heat flux has been multiplied by 10 and the combined temperature and heat flux has made. From the above figure, disc rotor model 1 has highest sustaining temperature of 95°C and disc rotor model 4 has highest heat flow rate of $9.79*10^5$ W/m^2. The original disc rotor has lowest heat flow rate of $1.63*10^5$ W/m^2 with the sustaining temperature of 53.8°C against the braking action.

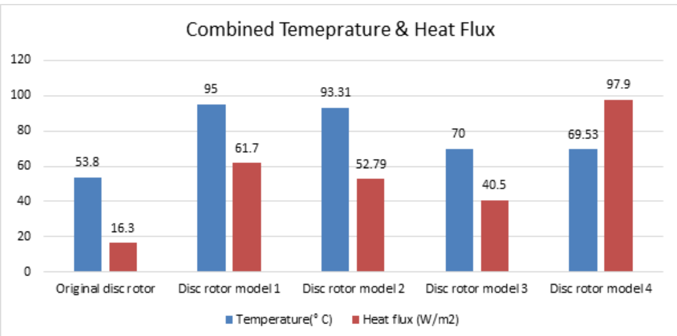

Figure 16: Combined Heat Flux and Temperature

Disc rotor model 2 shows nearly supportive result with disc rotor model 1, it has heat flux value of $5.279*10^5$ W/m^2 and sustaining temperature upto 93.371°C. Disc rotor model 3 shows poor performance among 4 improved design models with heat flux value of $4.051*10^5$ W/m^2 and sustaining temperature upto 70.04°C only. In all disc rotor models, the area of the disc rotor where disc pad imposed its force and pressure has higher heat flux value. The low value of heat flow rate can be found near the area where fixed support lies, also it can be said that the temperature is continuously rising from inside to outside periphery of disc brake rotor.It seems that the higher value of heat flux of disc rotor model may be due to the curve ventilation holes. That's why most of the disc brake of two wheelers have curved ventilation holes.

4.3 Cost Basis Analysis

This analysis shows the cost comparison of all the disc rotor models that would incurred during the production of unit disc rotor models. It also assist to figure out how much cost will be saved by them.

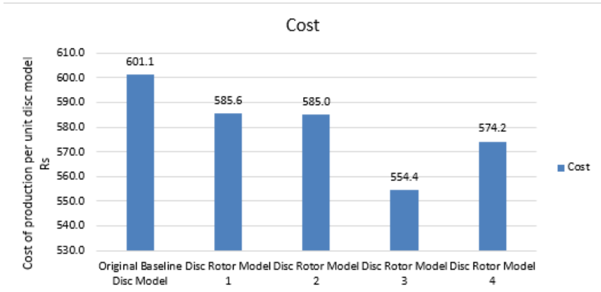

Figure 17: Analysis by cost basis

From the above figure it has been shown that, the cost of production per unit of original disc rotor is Rs 601.1 which is highest value among all models. The disc rotor model 1 & 2 have shown similar cost of production of about Rs 585.6 whereas cost of production is lowest for disc rotor 3 of Rs 554.4. The disc rotor model 4 has incurred cost of Rs 574.2 per unit of production.

S.N	Description	Cost(Rs)	Cost saved Rs
1	Original disc	601.1	
2	Disc Model 1	585.6	15.6
3	Disc Model 2	585.0	16.2
4	Disc Model 3	554.4	46.7
5	Disc Model 4	574.2	26.9

Table 3: : Cost saving of diffrent disc models

So, Disc rotor model 3 has saved maximum cost of Rs 46.7 on the unit production of material than the original disc rotor. Similarly, disc rotor model 1&2 save almost equal amounts of Rs 16 where as disc rotor model 4 becomes second place in saving the cost of production upto Rs 26.9. Hence, if Yamaha implemented these disc rotor, it can save upto Rs 26.3 on average and cost saving leads to maximization of the profit to the organisation aswell.

4.4 Weight Basis Analysis

This analysis shows weightage of different disc rotor models. It also helps to analyze how much material is saved by improved design of disc rotor models against original disc rotor.

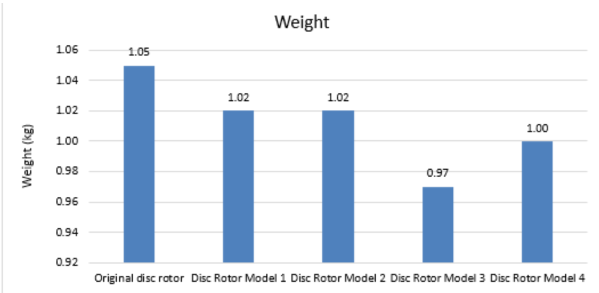

Figure 18: Analysis on weight basis

From the above figure, the original disc rotor has high value of weight of 1.05 kg. Similarly, the disc rotor model 1 and 2 weights same 1.02 kg material where as the disc rotor model 3 has the lowest weight occupying 0.97 kg of grey cast iron material and at last disc rotor model 4 weights 1 kg which is heavier than other all 3 rotor models but lighter than original disc rotor.

4.5 Selection of Final Design

From reviewing all the disc rotor models in terms of manufacturing cost, weight, stress, deformation, temperature and heat flux basis, the disc rotor model 4 is selected as the final improved design against the original baseline disc rotor model. In order to summarize the results of simulation analysis, all the analysis criteria have been compared between original disc rotor and disc rotor model 4. The comparative results are presented in below table.

Criteria	Orig. Disc	Disc Mod. 4
Weight (kg)	1.05	1.0
Cost (Rs)	601.1	574.2
Stress(MPa)	13.7	15.8
Deformation(mm)	0.0072	0.0070
Temperature (°C)	53.8	69.5
Heat Flux(10^5 W/m^2)	1.6	9.8

Table 4: Comparison analysis

5. Conclusion

Finally, the weight optimized design of disc rotor models have done using structural and thermal analysis after imposing of defined boundary condition and we have found that the disc rotor model 4 found superior in every criteria of analysis that has been selected as the final improved disc rotor to replace the current disc rotor using in Yamaha Fz-25. It may show inferior performance among all 3 improved design disc rotor models in some criteria; however, it shows superior performances than original disc rotor model. The possible reason behind the high temperature sustaining and higher heat flux than the original disc rotor could be the maximum ventilation area around the brake disk.

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