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Structural Analysis, Thermal Analysis and Design Optimization of Ventilated Brake Disc

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Abstract - Safety aspect in automotive engineering has been considered as a number one priority in development of new vehicle. Each single system has been studied and developed in order to meet safety requirement. Instead of having air bag, good suspension systems and good handling, there is one most critical system in the vehicle which is brake systems. Without brake system in the vehicle will put a passenger in unsafe position. Therefore, it is a must for all vehicles to have proper brake system. Due to critical system in the vehicle, many of researchers have conducted a study on brake system and its entire component. In this project,a ventilated brake disc for a commercial vehicle with gross vehicle weight of 9.6t is considered.

***KeyWords*:** Disc Brake, Ansys, Catia V5, Thermal Analysis

1. INTRODUCTION

This Disc type brake development and its use began in England in the 1890s. Disc brakes werepatented by Frederick William Lanchester in 1902 but the commercial use of these brakesstarted in the early 1950s [1] [3]. A brake disc rotor is the rotating part of a disc brake assembly normally located on the front axle. It consists of a rubbing surface, a top-hat and a neck section. The rubbing surface is where a tangential friction force between the rotor and the stationary pad is generated that gives rise to the brake force in the tire- ground plane which retards the vehicle. The top-hat section is mounted to the hub of the wheel. The connection between the rubbing surface and the hat is known as the neck. Brake discs are favored by most manufacturers of vehicles as the standard foundation brake at the front wheel. A drum brake, in contrast, is an internal expanding type of brakethat uses a device for expanding two shoes against a rotating drum when the brake pedal is pressed. A typical disc brake assembly is shown in the below figure 1[28].

At the present time, the disc brake used in most high specification cars has a vented rotor consisting of separate inboard and outboard rubbing surfaces that are connected by fins (or vanes). Thus, air can flow between the rubbing surfaces for cooling the brake disc both during and after braking. In practice, the neck of the brake disc can be connected to either the outboard or inboard rubbing surface. The outboard connection design, known as a front-vented disc, is the most common. On the other hand, the inboard connection is used in a back-vented disc where assembly is most critical with its cooling. Both brake disc designs may use the same calipers to house the pistons that arc used to press the pads of friction material against the rubbing surfaces of the brake disc and so generate the friction forces.

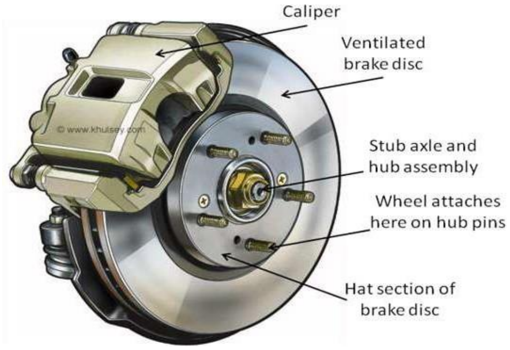

Fig -1: Disc brake assembly

1.1 Types of brake discs

The brake disc is a component of a disc brake assembly, in which the brake pad applies the brake force to stop or decelerate the vehicle. The design of the disc varies somewhat, some are simply solid, but others are ventilated with fins or vanes joining together the disc's two contact surfaces.

1. Solid brake disc

In this type, the braking surfaces are not separated by ventilation patterns. This is also called as one type of simple design where weight is not concern. Generally solid type brake discs are installed on rear wheels in the vehicles. Because, the brake force acting on these disc is less than the front discs when vehicle travelling in forward direction. Solid type design of brake disc is shown in the below figure 2[4].

Fig -2: Solid brake disc

2. Ventilate d brake disc

When braking, brake pads and caliper are very hot, this may lead to reduced braking efficiency. For better heat dissipation in the disc, sometimes cross drilled holes are provided, and the disc brake mechanism operates with a ventilated inner surface. The ventilated disc design helps to dissipate the generated heat and is commonly used on the more-heavily-loaded front discs. Ventilated type design of brake disc is as shown in below figure 3.

Fig -3: Ventilated brake disc

a. Simple ventilate d disc

Sometimes it is also called as front ventilated brake disc. In this type of disc, the hat portion is connected with outboard braking surface as shown in the below figure 4.

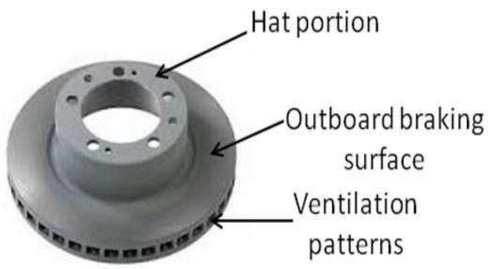

Fig -4: Simple ventilated brake disc

b. Back ventilate disc

This type of design in brake disc is preferred, when assembly conditions are more complicated and space constraints for ventilation. In this type of disc, the hat portion is connected with inboard braking surface as shown in the figure 5[4].

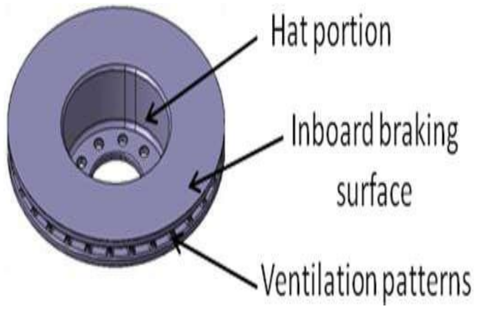

Fig -5: Back ventilated brake disc

c. Crossed drilled face design

Cross drilling has been used in the automotive industry for quite a while and the traditional belief is that it helped with degassing the brake pads as improve the initial bite of the brake pad into the rotor. Gas would develop under extreme braking as brake pad binders would burn off and this gas would act as a lubricant or cushion between the pad and the rotor, hence decreasing the efficiency of the brakes. The holes in the rotor provide escape routes to help get rid of the gas that builds up.

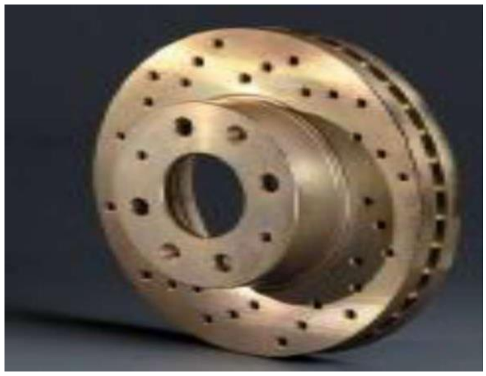

Fig -6: Cross drilled face design rotor

Objectives

This project concerns of the temperature distribution and constraint of the disc brake rotor along with structural analysis. Most of the commercial vehicles today have disc brake rotors that are made of grey cast iron. High temperature during braking will caused to: Brake fade, premature wear, Brake fluid vaporization, Bearing failure, Thermal cracks, Thermally-excited vibration. Therefore, it is important to study and predict the temperature rise of a given brake component and assess its thermal performance in the early design stage. Finite element analysis (FEA) is to be preferred to investigate some of the above concerns such as disc brake rotor temperature rise and thermal cracks. Due to the application of brakes on the car disc brake rotor, heat generation takes place due to friction and this temperature so generated has to be conducted and dispersed across the disc rotor cross section. The condition of braking is very much severe and thus the thermal analysis has to be carried out. The thermal loading as well as structure is axis symmetric. Hence axis-symmetric analysis is performed which is an exact representation for this thermal analysis and structural analysis.

2. Calculation of brake torque, heat flux and temperature rise

For calculation of braking requirements, vehicle parameters or specification are required. In this project, a commercial vehicle is discussed in detail. The vehicle parameters are required to calculate the braking terms like braking torque, heat flux and temperature rise. The effects on these braking terms are analyzed in this project Table 2.1 shows the parameters of the vehicle in consideration for calculating the braking torque for structural analysis.

Table -1: Vehicle parameters

Sr.No	Parameters	Values	Units
1	Gross Vehicle weight	9.6	Ton
2	Maximum speed	100	Km/h
3	Wheel base	5.22	m
4	Tire specification	215/75R17.5	
5	Tire dynamic radius	0.445	m
6	Centre of gravity height	1.2	m
7	Dist. Of CG from front axle	2.45	m
8	Dist. Of CG from rear axle	2.77	m

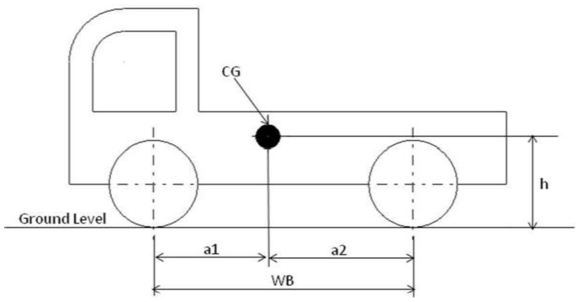

Fig -7: Vehicle dimensions

2.1 Brake Torque Calculation

As the gross vehicle weight is 9.6 t and it falls under N2 category as stated in table 3.1. For brake torque calculation following methodology is implemented.

a. Static load distribution

Static load distribution describes the weight distribution according to horizontal position of centre of gravity. Static load distribution on front axle can be calculated with the following equation 1.

$$R_f = M \left[\frac{a_2}{a_1+a_2}\right] \quad (1)$$

Static load distribution R_f is 5094 kg.

b. Mean fully developed deceleration (MFDD)

Value of MFDD can be calculated by using equation (2) and calculated deceleration is 4.42 m/s^2 i.e. 0.45g units distribution.

$$MFDD = \frac{V_b^2 - V_e^2}{25.92(s_e - s_b)} \; m/s^2 \quad (2)$$

c. Dynamic load distribution after braking

The vehicle is considered as one rigid body which moves along an ideally even and horizontal road. At each axle the forces in the wheel contact points are combined in one normal and one longitudinal force.

The dynamic load on front axle is calculated with the help of below equation 3.

$$F_{zf} = M.g \frac{h}{a1+a2} \frac{v}{g} \quad (3)$$

The calculated value is 9753 kg.

The dynamic loading of the vehicle during braking is the function of mass, CG height, wheel base and deceleration.

d. Total load while braking

While braking or decelerating, the total load is the sum of static load and dynamic load. The load acting on front axle is calculated with the help of below equation 4.

$$T_f = R_f + F_{zf} \quad (4)$$

The calculated value is 14847kg.

e. Brake force on front axle

After deciding the static, dynamic and total load on front axle, the brake force acting on front the axle can be calculated with the help of below equation 5.

$$B_f = T_f \, \mu_r \quad (5)$$

The calculated value is 5940 kg. Then the brake force on per wheel is 2970 kg.

f. Brake torque acting on front axle

The brake torque acting on front axle is the function of brake force and tire dynamic radius. The equation 6 is given as follows

$$T_{bf} = B_f \, R_{dyn} \quad (6)$$

The calculated value for the brake torque acting on front axle is 1321 kg-m.

Below figure 3.2 shows the relationship between the values of brake torque with respective values of deceleration.

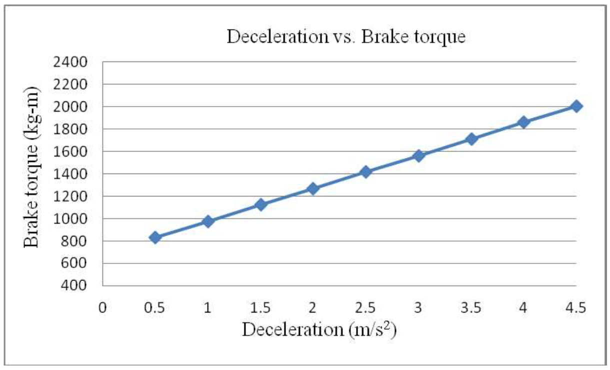

Fig -7: Deceleration vs. brake torque relation

3. Calculation of temperature rise based on total energy

When the vehicle is decelerating, the total energy is converted into heat energy. That heat energy must be transferred to atmosphere. Equating this heat energy to total energy and can be expressed with the help of equation 7[19].

$$TE = m \, C_p \Delta t \quad (7)$$

Table -2: Temperature rise based on equation

Materials / Parameters	AISI-4140	EN-GJL-200 (GG20HC)	FG220MoCr	FG260cr
Total Energy (J)	38148.15			
Density of disc material ρ (kg/m^3)	7625	7150	7800	7600
Mass of disc m (kg)	15.75	14.77	16.11	15.71
Specific heat Cp (J/kg K)	420	540	480	435
Temperature (K)	577	480	494	560
Temperature (°C)	304	207	221	287

3.1Temperature Calculation for Repeated Braking

During repeated brake applications, the vehicle is decelerated at a given deceleration from 100 km/h to a lower or zero speed, after which the vehicles accelerated again to test speed and the next braking cycle is carried out. Brake temperature attained during brake application will be less than those achieved during repeated braking because the braking power is low.

Below table 3 shows the calculated temperature for each nth application in 10 intervals in °K and converted into °C keeping the cooling cycle time same as 120 sec. For this calculation FG260Cr is selected with heat transfer coefficient (hR) as 368000 Nm/hkm2. Further these values are compared with brake testing dynamometer.

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Table - 3: Disc temperature for n^{th} application

Analytical Calculation					
Sr.No	**N^{th} Application**	**Cooling Cycle Time (Sec)**	**Ambient Temperature (^{0}K)**	**Final Temp (^{0}K)**	**Final Temp. (^{0}C)**
1	10	120	323	383.36	110.36
2	20	120	323	403	130
3	30	120	323	409	136
4	40	120	323	411.9	138.9
5	50	120	323	413	140
6	60	120	323	415	142
7	70	120	323	416.5	143.5
8	80	120	323	418.2	145.2
9	90	120	323	421.7	148.7
10	100	120	323	423.1	150.1
11	110	120	323	424.8	151.8
12	120	120	323	426.7	153.7
13	130	120	323	427.5	154.5
14	140	120	323	429.4	156.4
15	150	120	323	430.1	157.1
16	160	120	323	431.8	158.8
17	170	120	323	433.2	160.2
18	180	120	323	434.6	161.6
19	180	120	323	433.3	160.3
20	200	120	323	436.7	163.7

4. Modeling and analysis of ventilated brake disc

4.1 Modeling of ventilated brake disc in CATIA V5

CATIA V5 R22 is mechanical design software, addressing advanced process centric design requirements of the mechanical industry. With its feature based design solutions, CATIA proved to be highly productive for mechanical assemblies and drawing generation. CATIA, with its broad range of integrated solutions for all manufacturing organization

Modeling of project brake discs.

For this project two types of disc are to be analyzed for structural, thermal and CFD analysis for their design optimization. The two types of disc model are created in CATIA V5 R22 having different ventilation patterns.

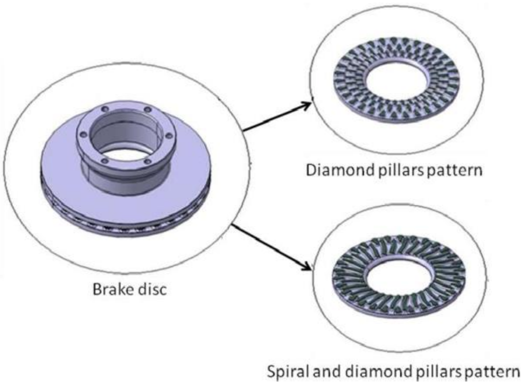

Fig -8: Types of discs modeled for analysis.

4.2ANSYS result discussion for structural analysis

To carry out the structural analysis for four different materials in ANSYS, figure 9 shows meshed model. For structural analysis, tetra element is used for meshing.

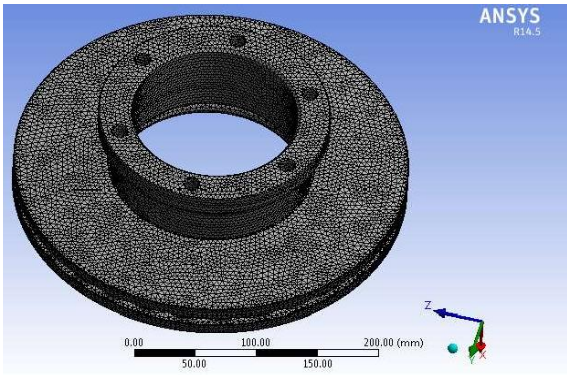

Fig -9: Meshed model for structural analysis

Boundary conditions

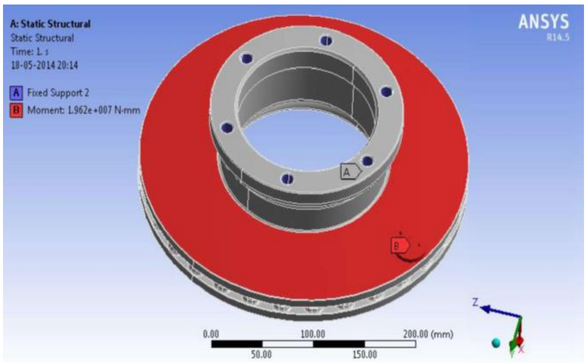

Fig -10: Boundary conditions for structural analysis

Figure 10 shows the boundary conditions which are applied for structural analysis. The boundary conditions are as follows:-

A. Fixed Support: The notation in the figure 4.4 shows the points where brake disc fixed i.e. all degrees of freedom are locked.

B. Moment: This means the calculated brake torque for required deceleration is applied on both the braking surface of the disc.

4.3 Structural analysis

Structural analysis of brake disc is carried out for four different materials. For each disc maximum principle, middle principle and minimum principle stress are calculated in ANSYS and the results of each brake disc are discussed.

1. Mate rial: AISI-4140 (High tensile steel)

a. Maximum Principle Stress (σ_1)

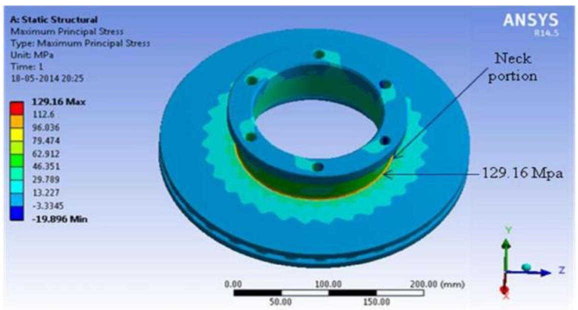

Fig -11: Maximum principle stresses for AISI-4140

Figure 11 shows the results of analysis of brake disc with material AISI-4140 which is high tensile steel. The maximum principle stress observed during analysis is 129.16 Mpa for

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the calculated brake torque which is observed at the neck portion of disc.

b. Total deformation

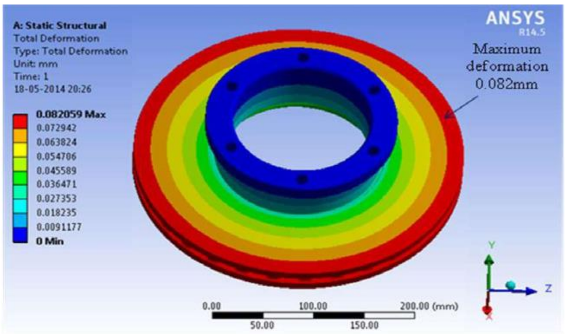

Fig -12: Total deformation for AISI-4140

Figure 12 shows the total deformation of the disc due to brake torque and value is 0.082059mm which is observed on the braking surface of the disc.

2. Material: - EN-GJL-200 (GG200HC)

a. Maximum Principle Stress (σ₁)

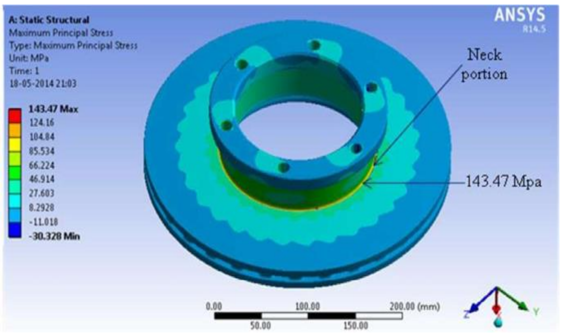

Fig -13: Maximum principle stress for EN-GJL-200 (GG200HC)

Figure 13 shows the results of analysis of brake disc with material EN-GJL-200. The maximum principle stress observed during analysis is 143.47 MPa for the calculated brake torque which occurs at the neck portion of disc.

b. Total de formation

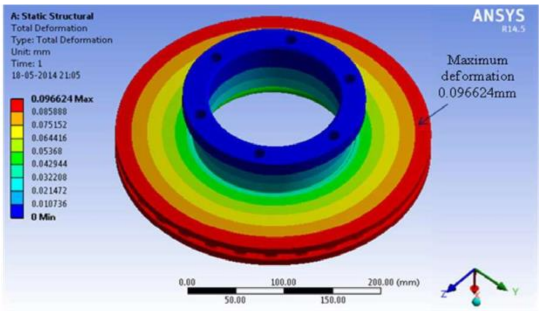

Fig -14: Total deformation for EN-GJL-200 (GG200HC)

Figure 14 shows the total deformation of the disc due to brake torque and the maximum value is 0.096624mm which is observed on the braking surface of the disc.

3. Material: - FG220MoCr

a. Maximum Principle Stress (σ₁)

Figure 15 shows the results of analysis of brake disc with material FG220MoCr. The maximum principle stress observed during analysis is 129.24 MPa for the calculated brake torque which occurs at the neck portion of disc.

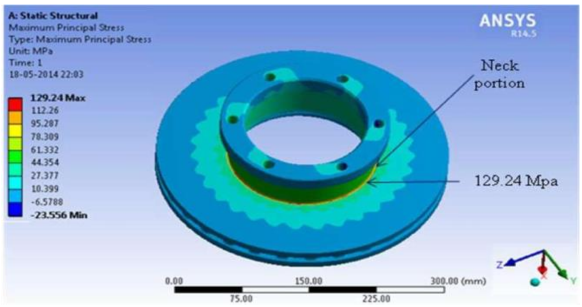

Fig -15: Maximum principle stress for FG220MoCr

b. Total de formation

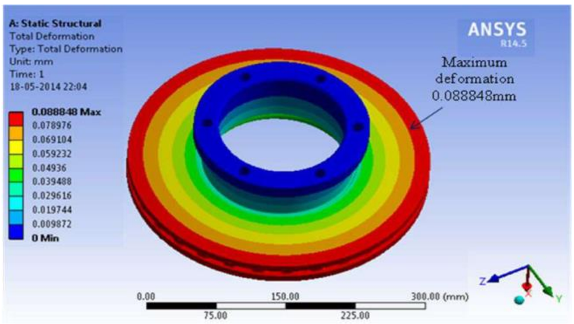

Fig -16: Total deformation for FG220MoCr

Figure 16 shows the total deformation of the disc due to brake torque for the material FG220MoCr and the maximum value of total deformation observed on braking surface is 0.088848 mm.

4.4 Thermal analysis

This section concentrates on a thermal analysis of a new design of ventilated brake disc. The purpose is to investigate disc performance and verify whether the maximum temperature plot under severe brake conditions To investigate the thermal stress behavior of brake discs under cyclic thermal load, it is necessary to obtain typical temperature distributions in these brake discs as a function of time. Therefore, the objective of this section is to predict the temperature response of the back-vented brake disc design.

A. Heat flux and temperature distribution for material AISI-4140

a. Heat flux analysis

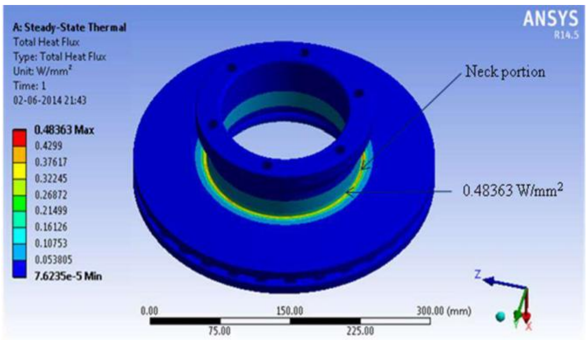

Fig -16: Heat flux analysis AISI-4140

Figure 16 shows the heat flux analysis for the brake disc with material AISI-4140. Maximum heat flux value is observed at the neck portion of disc is 0.48363 W/mm2.This is due to less wall thickness at the neck portion. The maximum heat flux

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value observed in thermal analysis is less than the calculated value 2.7 W/mm2

b. Temperature distribution plot

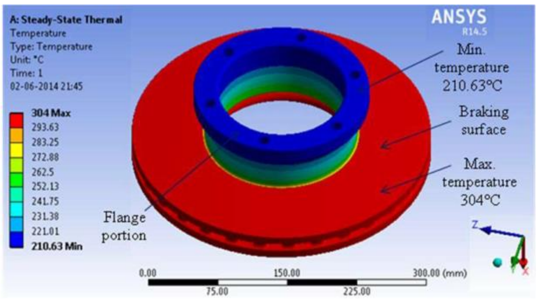

Fig -17: Temperature distribution plot for AISI-4140

Figure 17 shows the temperature plot for the brake disc with material AISI-4140. The calculated temperature 304°C is distributed on the braking surface of disc. Minimum temperature 210.63°C is observed on the flange portion of disc.

B. Heat flux and thermal analysis for material EN-GJL-200

a. Heat flux analysis

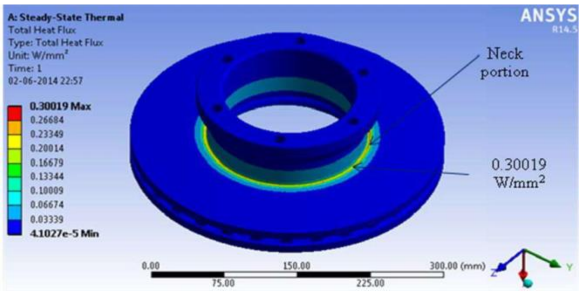

Fig -18: Heat flux analysis for EN-GJL-200

Figure 18 shows the heat flux analysis for the brake disc with material EN-GJL-200. Maximum heat flux value is observed at the neck portion of disc is 0.30019 W/mm2.This is due to less wall thickness at the neck portion. The maximum heat flux value observed in thermal analysis, which is less than the calculated value 2.7 W/mm2.

b. Temperature distribution plot

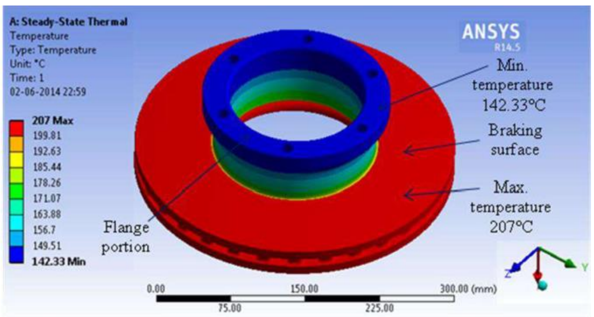

Fig -19: Temperature distribution plot for EN-GJL-200

Figure 19 shows the temperature plot for the brake disc with material EN-GJL-200. The calculated temperature 207°C is distributed on the braking surface of disc. Minimum temperature 142.33°C is observed on the flange portion of disc.

C. Heat flux and thermal analysis for material FG220MoCr

a. Heat flux analysis

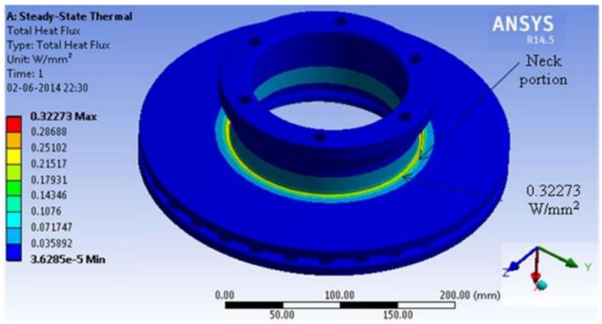

Fig -20: Heat flux analysis FG220MoCr

Figure 20 shows the heat flux analysis for the brake disc with material FG220MoCr. Maximum heat flux value is observed at the neck portion of disc is 0.32273 W/mm2.This is due to less wall thickness at the neck portion. The maximum heat flux value observed in thermal analysis, which is less than the calculated value 2.7 W/mm2.

b. Temperature distribution plot

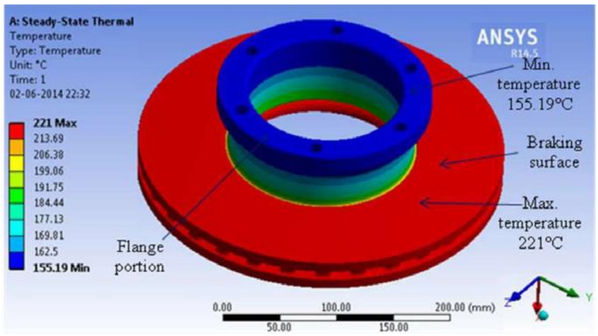

Fig -21: Temperature distribution plot FG220MoCr

Figure 21 shows the temperature plot for the brake disc with material FG220MoCr. The calculated temperature 221°C is distributed on the braking surface of disc. Minimum temperature 155.19°C is observed on the flange portion of disc.

4.5 Ventilation pattern analysis

The main aim of this CFD analysis is to study and predict the effect of various design parameters on the aero-thermal performance of a disc brake rotor.

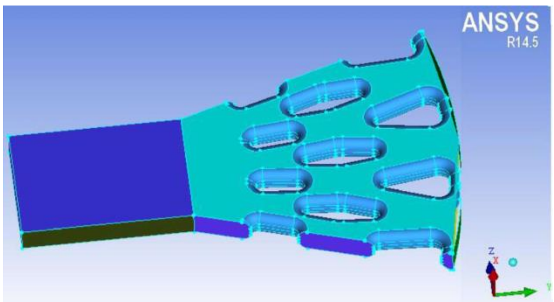

Fig -22: CAD model repair for ventilation pattern

Figure 22 shows the CAD model repaired for ventilation pattern. The CAD model repair methodology is implemented to create closed surface for mesh generation.

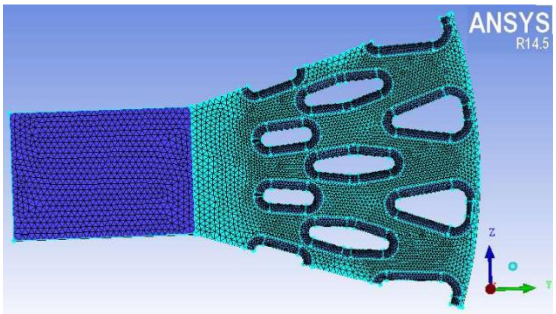

Fig -23: Surface mesh for ventilation pattern

The surface mesh generation process includes a mechanism as required near to the wall region. Inflation is used to resolve and to capture the flow effects for viscous problems. Surface mesh generation on disc model is shown in the above figure 23

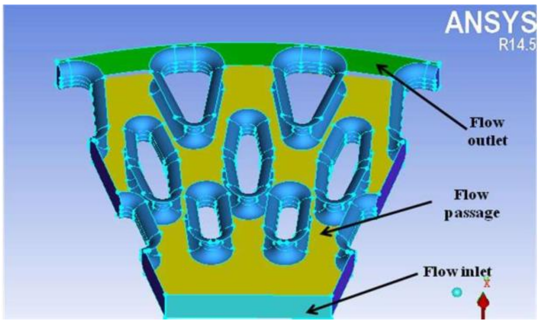

Fig -24: Attribute definition for CFD of ventilation pattern

The simulation was performed at various rotational speed varies from 463 rpm to 1643 rpm in order to get results of mass flow rate that would be required for running at higher speeds. Figure 24 shows the attribute definition for ventilation pattern which is required for mass flow rate simulation.

a. Mass flow rate and streamline plot

The mass flow rate was analyzed through CFD for ventilation patter is tabulated in table 4.

Table -4: Mass flow rate through ventilation pattern

RPM	Pattern A mass flow rate (kg/s)
493	0.00186166
657	0.002560938
822	0.00326553
986	0.003987126
1150	0.004690663

5. Result discussion and Conclusion

The Material FG260Cr is having lowest total deformation i.e. 0.065818mm for the applied brake torque 1962000N-mm as compared with other three materials. Allowable stress 136.69 Mpa which less than the ultimate strength calculated by applying "Maximu principle stress theory". For brittle material Also the calculated factor of safety is 2, which is optimum for using the brake disc material for commercial with 9.6t gross vehicle weight.

After selection of suitable material, the disc is tested on dynamometer for temperature rise. And it is found that the material FG260Cr is suitable for brake disc. The same temperature values verified with analytically as shown in figure 6.6

Table -5: Summary of structural and thermal analysis

Parameters \ Materials		AISI-4140	EN-GJL-200 (GG20HC)	FG220MoCr	FG260cr
Principle stress (Mpa)	Max(σ1)	129.16	143.47	129.24	129.13
	Middle (σ2)	24.136	36.192	27.118	23.696
	Min (σ3)	15.148	32.788	16.988	14.895
Ultimate strength (Mpa)		1020	200	220	260
Allowable stress (Mpa)		136.36	134.22	133.85	136.69
Total Deformation (mm)		0.082059	0.096624	0.088848	0.065818
Factor of safety		7.5	4.5	1.7	2
Single stop temp. rise (ºC)		69	56	57	85
Temp. rise (ºC)		304	207	221	287

6. Conclusions

In this project work, an attempt has been made to design a brake disc for front axle, which is suitable for 9.6t gross vehicle weight. Indian braking standard IS: 11852 is used to determine the category of vehicle. A 9.6t commercial vehicle falls under N3 category and calculated mean fully developed deceleration (MFDD) is 4.42m/s2 i.e. 0.45g units. Using vehicle dynamics, below braking requirements were calculated such as:-

1. Brake force on each front wheel as 2970 kg.
2. Brake torque on each front disc 2000 kg-m i.e. 1962000 N-mm

Finally, design optimization method is performed. This includes selection of candidate material and selection of ventilation pattern for better heat dissipation. As stated earlier, four different materials were analyzed for structural and thermal analysis using ANSYS. From structural and thermal analysis it has been found that the material FG260Cr is optimum material for the brake disc as its total deformation is less as compared to other materials. As its allowable stress is less than ultimate tensile stress.

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