Thermal Analysis of Disc Brake For Grey Cast Iron Using Ansys

Mr. Satish Lekkala[1], Mr. Anil Kumar Godea[2], Mr.Rajasekhar Naviri[3]
[1, 2, 3] Cyient LTD, Hyderabad

Abstract- *Brakes are most important safety parts in the vehicles. Generally, all of the vehicles have their own safety devices to stop their car. Brakes function to slow and stop the rotation of the wheel. To stop the wheel, braking pads are forced mechanically against the rotor or disc on both surfaces. They are compulsory for all of the modern vehicles and the safe operation of vehicles. In short, brakes transform the kinetic energy of the car into heat energy, thus slowing its speed. Existing literature revealed some of the gaps in works on the analysis of disc brakes to know the effect of friction, thermal stresses and their distribution with different materials. Work done particularly in the area of structural and thermal analysis of different shapes of ventilated disc brake in view of weight reduction without affecting the life of disc brake. Chosen different materials for analysis of thermal and structural analysis of disk brakes and also planned to study the influence of. different shapes of ventilated holes for the disc brakes to know the effect of cooling rate and stress distributions.*

I. INTRODUCTION

Brakes are most important safety parts in the vehicles. Generally all of the vehicles have their own safety devices to stop their car. Brakes function to slow and stop the rotation of the wheel. To stop the wheel, braking pads are forced mechanically against the rotor or disc on both surfaces. They are compulsory for all of the modern vehicles and the safe operation of vehicles. In short, brakes transform the kinetic energy of the car into heat energy, thus slowing its speed.

II. CLASSIFICATION OF BRAKES

1) Purpose:

- Primary brakes
- Secondary brakes

2) Construction:

- Drum brakes
- Disc brakes

3) Method of Actuation:

- Mechanical brakes
- Vacuum brakes
- Hydraulic brakes
- Air brakes
- Electric brakes
- By-wire brakes

4) Extra braking effort:

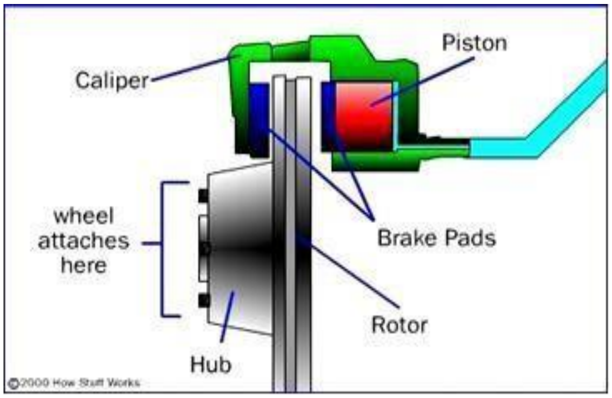
- Power assisted brakes
- Power operated brakes

III. APPLICATIONS

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 calipers. The brake disc (or rotor in American English) is usually made of cast iron, but may in some cases be made of composites such as reinforced carbon–carbon or ceramic matrix composites. This is connected to the wheel and/or the axle.

To stop the wheel, friction material in the form of brake pads, mounted on a device called a brake caliper, is forced mechanically, hydraulically, or electromagnetically against both sides of the disc. Friction causes the disc and attached wheel to slow or stop. Brakes convert motion to heat, and if the brakes get too hot, they become less effective, a phenomenon known as brake fade.

IV. CONSTRUCTION



TYPES: Drilled Discs: The faces of these discs are drilled all the way through mainly to increase surface area so that they can get rid of heat quickly. The holes also go a little way to stopping the gas build up that causes brake fade. They also speed the clearance of water in wet conditions. The problem with drilled discs is that the holes can have a tendency to start cracking and collect dust and debris.

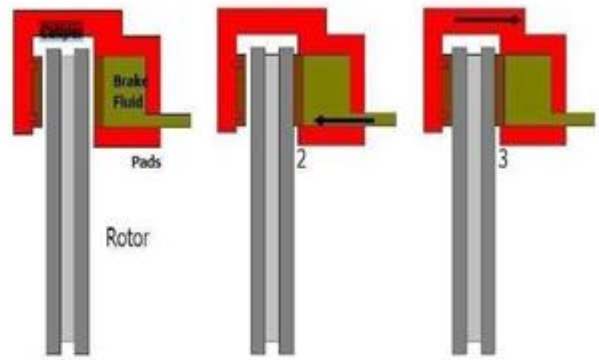
Grooved Discs: The faces of these discs have diagonal lines cut into them, there are two reasons for this. Firstly they allow the venting of brake pad gases, thus eliminating brake fade. They also eject brake pad dust to stop glazing of the pad. This keeps the pad face fresh allowing better braking. The problem is that grooved discs have a tendency to be louder when the brakes are applied due to the scrubbing of the pads.

Drilled Disc Brake

Grooved Disc Brake

OPERATION OF DISC PAD

Brake pads are designed for high friction with brake pad material embedded in the disc in the process of bedding while wearing evenly. Friction can be divided into two parts: Adhesive and abrasive. Depending on the properties of the material of both the pad and the disc and the usage, pad and disc wear rates will vary considerably.



Operation of a Disc Brake

When hydraulic pressure is applied to the caliper piston, it forces the inside pad to contact the disc. As pressure increases the caliper moves to the right and causes the outside pad to contact the disc. Braking force is generated by friction between the disc pads as they are squeezed against the disc rotor. Since disc brakes do not use friction between the lining and rotor to increase braking power as drum brakes do, they are less likely to cause a pull. The friction surface is constantly exposed to the air, ensuring good heat dissipation, minimizing brake fade. It also allows for self- cleaning as dust and water are thrown off, reducing friction differences.

THERMAL CONSIDERATIONS

The energy absorbed by brake is converted into heat, which increases the temperature at the rubbing surfaces. When the temperature increases, the coefficient of friction decreases, adversely affecting the torque capacity of the brake. At high temperature, there is rapid wear of the friction lining, which reduce the life of the lining. Therefore, the temperature rise should be kept within permissible range.

It is very difficult to precisely calculate temperature rise. In preliminary design analysis, very often the product (pv) is considered in place of temperature rise. When the coefficient of friction is constant, the rate of heat generated is proportional to the product (pv) where p is the intensity of normal pressure (N/mm^2) and v is the rubbing speed (m/min). The temperature rise depends upon the mass of the brake drum assembly, the ratio of the braking period to the rest period and the specific heat of the material. For peak short-time requirements, it is assumed that all the heat generated during the braking period is absorbed by the brake drum assembly. In that case, the temperature rise is given by

$$\Delta t = E/mc$$

Where,

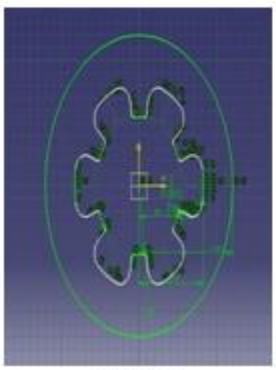
Δt = temperature rise of the brake drum assembly (°C)

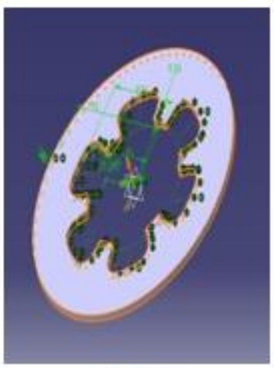
E = total energy absorbed by the brake
m = mass of the brake drum assembly (kg)
c = specific heat of the brake drum material (J/kg°C)

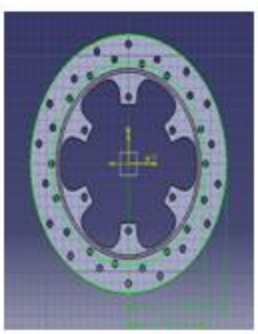
The actual temperature rise will be less than that calculated from above equation. Some heat will be radiated to the atmosphere and some carried away by the air flow. The equation gives approximate value and the actual temperature rise is obtained byexperiments.

Existing Disc Model

The below figures 3.1, 3.2, 3.3, 3.4 shows the geometry, padding, creating holes and final model of the existing disc model which are designed in CATIA part modeling and further imported to ANSYS workbench.
Coupled-field analysis depends on which fields are being coupled, but two distinct methods can be identified: sequential and direct.


Geometry


Padding


Creating Holes


Final model

Modified Disc Model

The below figures 3.5, 3.6, 3.7, 3.8 shows the geometry, padding, rotation and final model of the Modified disc model which are designed in CATIA part modeling and further imported to ANSYS workbench.

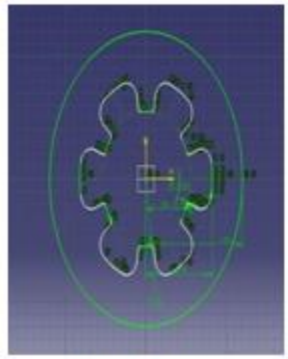


Fig 3.5 Geometry


Fig 3.6 Padding

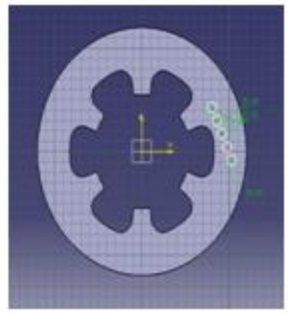


Fig 3.7 Rotation

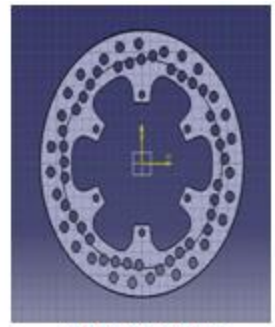
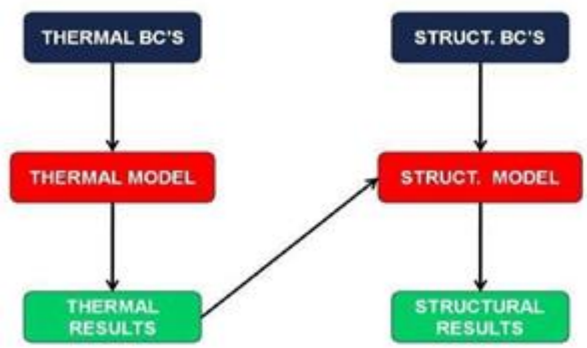
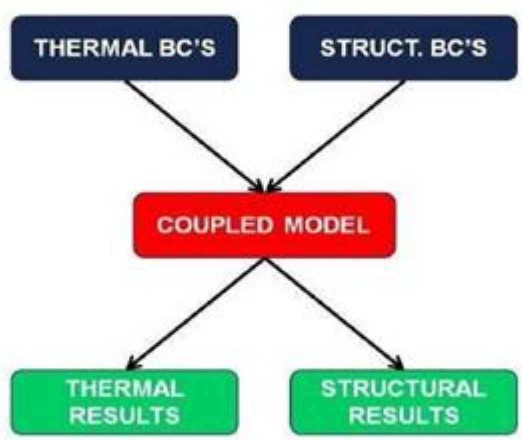


Fig 3.8 Final model

After creating the final model we have to convert the part into STP format to import the Model into ANSYS workbench.



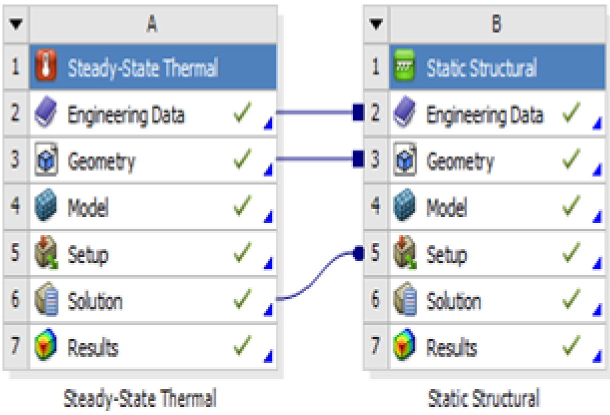
Flowchart of sequential analysis



Flowchart of Direct analysis

Thermal-Structural Coupling

Thermal-stress analysis involves two sequential analyses. Fig. 4.4 shows a Thermal-Structural Coupling in ANSYS Workbench.



Thermal-Structural Coupling in ANSYS Workbench

V. MODELING AND ANALYSIS

It is very difficult to exactly model the brake disc, in which there are still researches are going on to find out transient thermo elastic behavior of disc brake during braking applications. There is always a need of some assumptions to model any complex geometry. These assumptions are made, keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those which are ignored. In modeling we always ignore the things that are of less importance and have little impact on the analysis. The assumptions are always made depending upon the details and accuracy required in modeling. The Fig. 4.5 shows a Thermal-Structural Coupling and Inputting properties of the required material in workbench

The assumptions which are made while modeling the disc brake are given below:-

1. The disc material is considered as homogenous and isotropic.
2. Inertia and body force effects are negligible during the analysis.
3. The disc is stress free before the application of brake.
4. Only ambient air-cooling is taken into account and no forced convection is taken.
5. The kinetic energy of the vehicle is lost through the brake discs i.e, no heat loss between the tyre and the road surface and deceleration is uniform.
6. The thermal conductivity and specific heat of the material used for the analysis is uniform throughout.

MATERIALS USED FOR DISC BREAK

- Stainless steel
- Grey cast iron

THEORETICAL CALCULATIONS

i) Specifications of engine 150cc pulsar

Engine type : 4-stroke, DTS-i, air cooled single cylinder

Displacement	:	149cc
Maximum power	:	15.06 @ 9000 (Ps @ RPM)
Maximum torque	:	12.5 @ 6500 (Nm @ RPM)
Length (mm)	:	2055
Width (mm)	:	755
Height (mm)	:	1060
Ground clearance (mm)	:	165
Wheelbase (mm)	:	1320
Kerb weight (kg)	:	144
Front	:	240mm Disc
Rear	:	130mm Drum

ii) Calculations

Speed of the vehicle=60kmph = $60\times\frac{5}{18}$ = 16.67m/sec

Stopping time = u/4.6 = 16.67/4.6 = 3.62

Velocity V= u+at (Final velocity = 0) 0 =16.67+a (4)

a = -4.16 m/s^2;

Distance $x = ut + \frac{1}{2}at^2$

Distance $x = 16.67\times4+\frac{1}{2}\times(-4.16)\times4^2$

Distance x= 33.4m;

Kinetic energy $=\frac{1}{2}mv^2 = \frac{1}{2}\times145\times16.67^2 = 20139$ J

Rubbing area $= 2r^2 = 2\times3.14\times(115^2 - 89^2) = 0.0333\ m^2$

Heat Generated at disc brake = Kinetic energy – ((Drag) + (Friction))

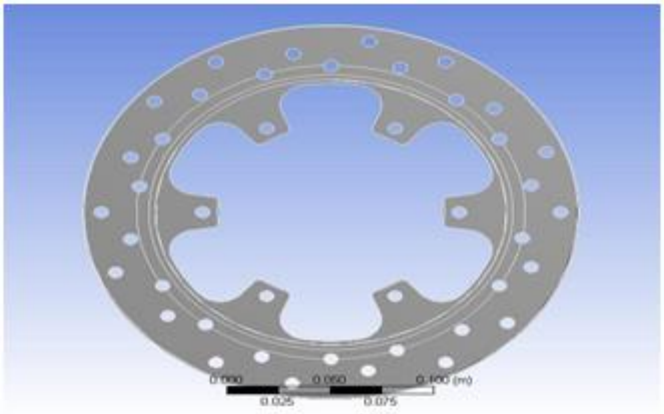
For calculating energy carried out by disc brake, the energy carried out by Drag, Friction and rear brake is taken as about 85 percentage.

Thus, Heat generated = 20139×0.15 = 3020.8 w

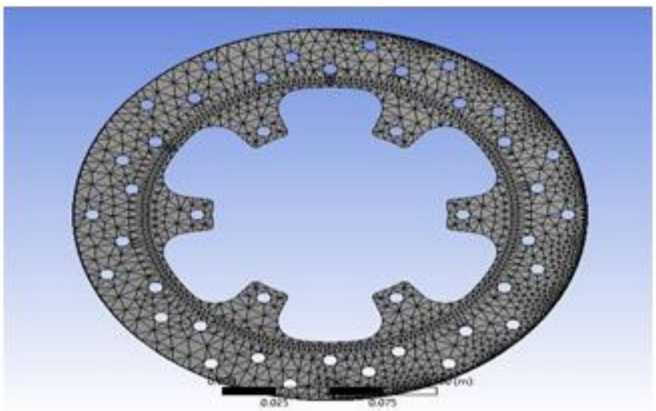
$$\text{Heat flux } \emptyset = \frac{\text{Heat Generated}}{\text{time}\times\text{rubbing area}}$$

$$= \frac{3020.8}{3.6\times0.0333} = 25031\ w/m^2 = 0.025031\ w/mm^2$$

Thermal Analysis

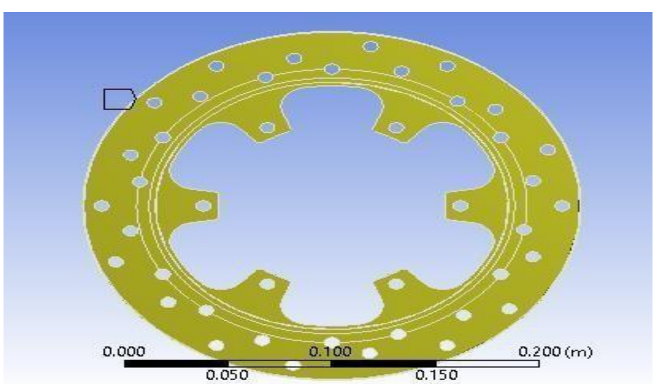


Geometry Model

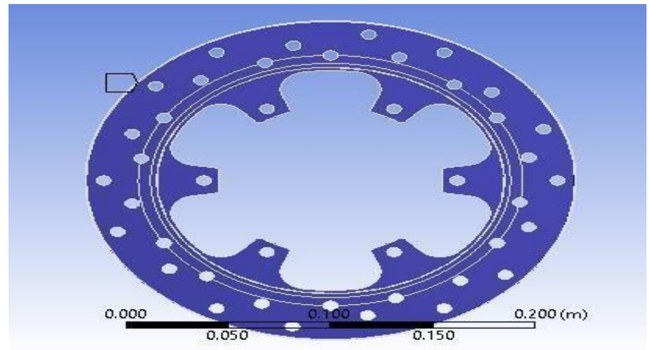


Meshing Model

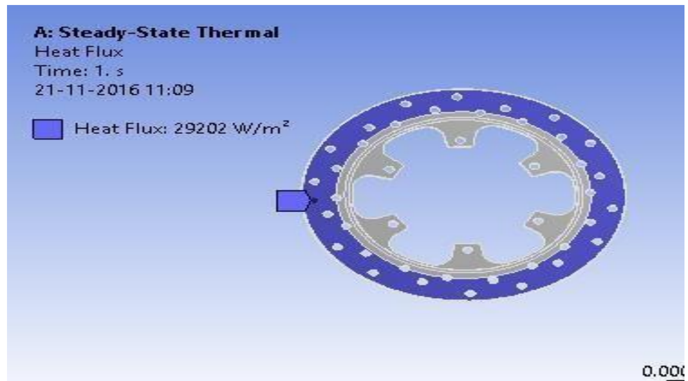
the input boundary conditions i.e we are giving Convection, Radiation and Heat Flux as input conditions for thermal analysis.



Convection Model

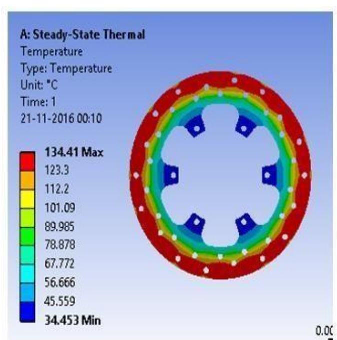


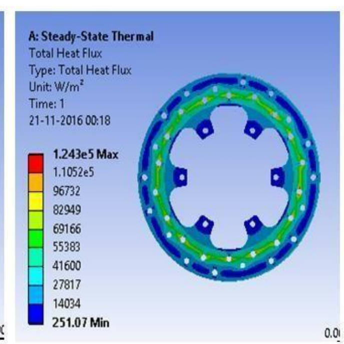
Radiation Model



Heat Flux model

the Temperature and Total heat flux of Stainless Steel Existing Model. The maximum temperature and the maximum total heat flux are represented by red color.

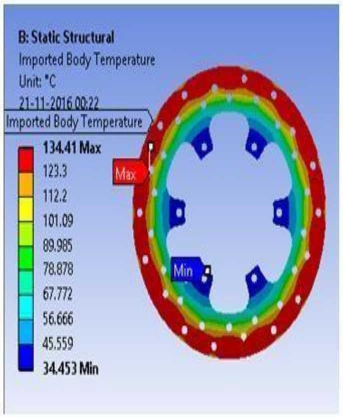




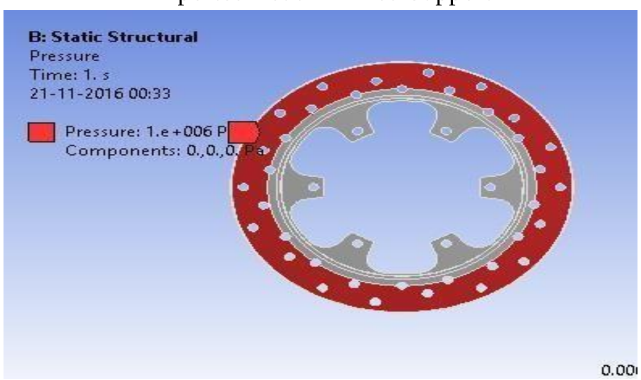
Temperature Heat Flux

Stress Analysis

the boundary condition of Stress analysis. I.e. imported load from thermal analysis, Fixed Support and pressure as input boundary conditions for Stress analysis.

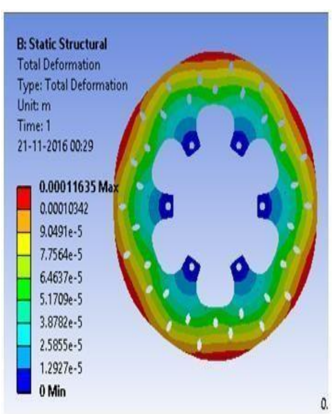


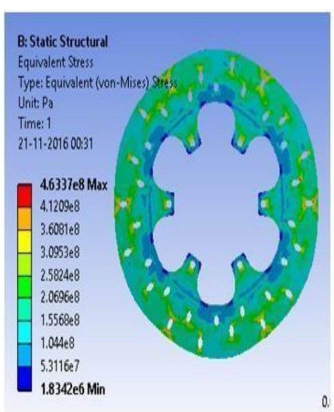
Imported Load Fixed Support



Applying Pressure

the Total deformation and Equivalent (von-Mises) stress of Stainless Steel Existing Model. As it is evident from the figures, the maximum total deformation and equivalent (von-Mises) stress are represented by red color.





Total Deformation Equivalent Stress

Stainless steel Existing model results at different speeds

speed(m/s)	Temperature(C)	Heat Flux(W m2)	Deformation(mm)	Equivalent Stress(Mpa)
0	30.000	0	0	0.00
5	37.575	8975	0.010	84.82
10	45.146	15472	0.027	106.53
15	52.719	23208	0.035	129.13
20	60.174	30854	0.042	151.45
25	67.674	38535	0.049	182.48
30	75.162	46207	0.057	213.86
35	82.631	53866	0.064	245.18
40	90.083	61512	0.072	276.45
45	97.516	69144	0.079	307.62
50	104.93	76761	0.086	338.73
55	112.32	84364	0.094	368.78
60	119.70	91952	0.101	400.70
65	127.08	115530	0.109	432.50
70	134.41	124300	0.116	463.37

Grey cast iron Existing model results at different speeds

speed(m /s)	Temperature (C)	Heat Flux(W/m 2)	Deformation (mm)	Equivalent Stress(Mpa)
0	30	0	0	0
5	36.489	14733	0.012	34.60
10	42.977	29466	0.017	47.90
15	49.466	44200	0.022	58.90
20	55.955	58933	0.027	69.97
25	62.307	73485	0.032	80.70
30	68.735	88138	0.037	91.65
35	75.151	102770	0.041	102.55
40	81.555	117390	0.047	113.43
45	87.945	132000	0.051	124.28
50	94.322	146580	0.056	135.11
55	100.690	161140	0.061	145.91
60	107.040	175690	0.065	156.69
65	113.370	190210	0.070	167.45
70	119.690	204710	0.075	178.17

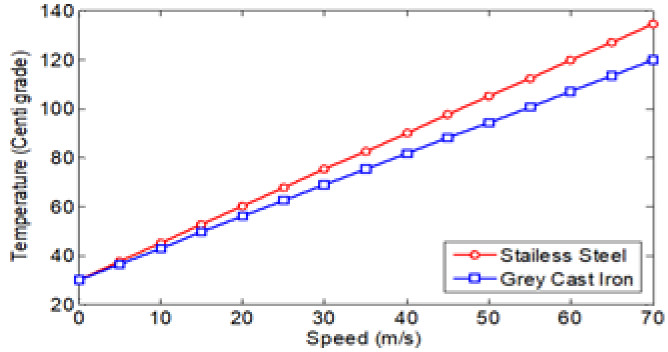
Modified Design results for Grey Cast Iron HT250

speed(m/s)	Temperature(C)	Heat Flux(W m2)	Deformation(mm)	Equivalent Stress(Mpa)
0	30.000	0	0	0.00
5	36.233	16600	0.012	29.45
10	42.466	33200	0.017	36.68
15	48.699	49800	0.022	43.97
20	54.933	66400	0.026	51.29
25	61.040	82762	0.031	58.47
30	67.217	99256	0.035	65.76
35	73.382	115730	0.04	73.11
40	79.537	132180	0.044	82.31
45	85.679	148610	0.049	91.50
50	91.809	165020	0.054	100.67
55	97.927	181400	0.058	109.82
60	104.03	197750	0.063	118.95
65	110.120	214080	0.067	128.06
70	116.200	230380	0.072	137.15

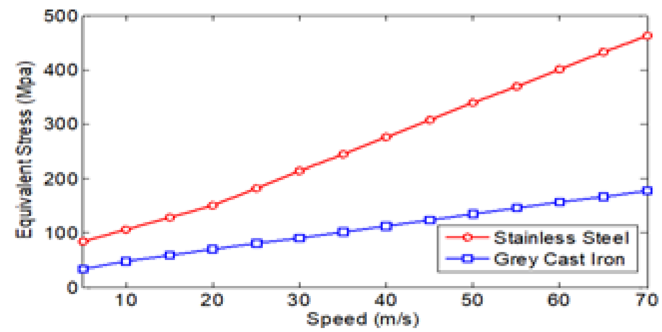
Modified Design results with Grey Cast Iron HT300

speed(m/s)	Temperature(C)	Heat Flux(W m2)	Deformation(mm)	Equivalent Stress(Mpa)
0	30.000	0	0.000	0.00
5	36.233	16600	0.017	5.35
10	42.466	33200	0.024	10.70
15	48.699	49800	0.030	16.07
20	54.933	66400	0.037	21.40
25	61.040	82762	0.044	26.70
30	67.210	99256	0.050	32.02
35	73.380	115730	0.057	37.34
40	79.530	132180	0.064	42.60
45	85.670	148610	0.070	47.90
50	91.800	165020	0.077	53.20
55	94.920	181400	0.083	58.50
60	104.030	197750	0.090	63.70
65	110.120	214080	0.097	69.03
70	116.200	230380	0.100	74.20

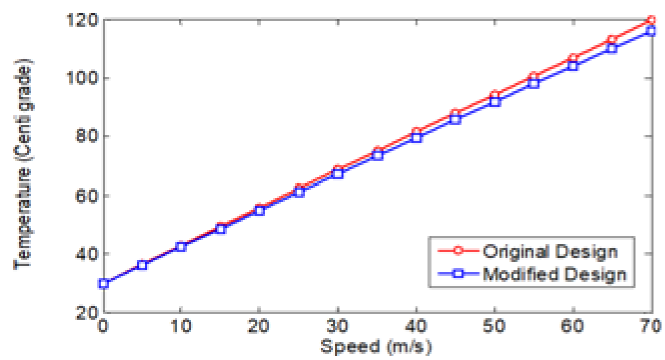
VI. GRAPHS



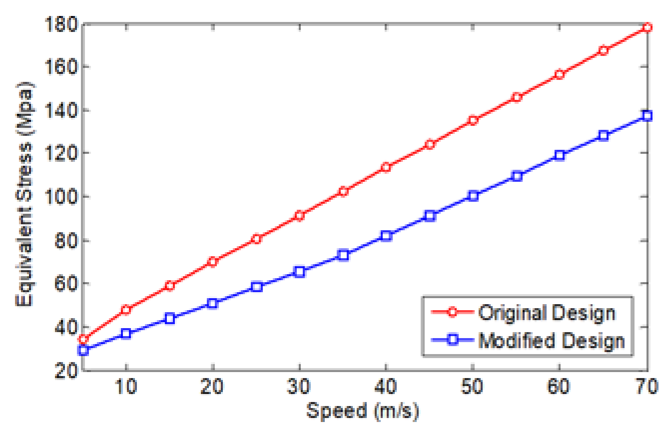
Temperature (Stainless Steel vs Grey Cast Iron)



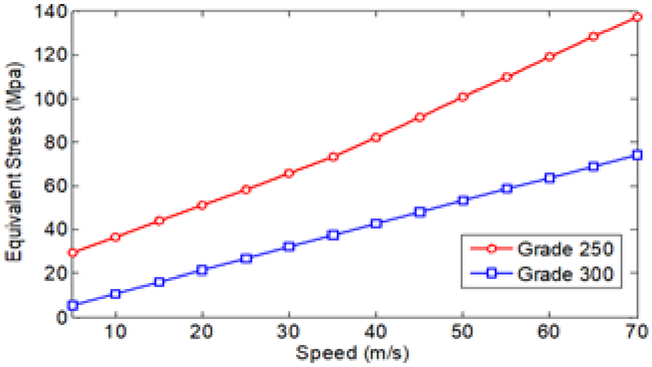
Equivalent Stress (Stainless Steel vs Grey Cast Iron)



Temperature (Existing Design vs Modified Design)



Equivalent Stress (Existing Design vs Modified Design)



Equivalent Stress vs Speed for (HT250 vs HT300)

VII. CONCLUSION

The Conclusions drawn about material and Design Modification:

- Grey cast iron gives better results when compared to Stainless steel for Existing model.
- The Modified Design gives better results in comparison to Existing Design with Grey Cast iron.

The Conclusions Drawn with Reference to Coupled field Analysis:

- The disc brake made up of grey cast iron exhibited more heat flux and capacity to with stand high thermal stresses.
- The couple field analysis of different materials shows that the grey cast iron gives reliable results as it has good cooling rate efficiency when compared to stainless steel.

VIII. FUTURE SCOPE

In the present investigation of Thermal analysis of disc brake, a simplified model of the disc brake without any vents with only ambient air cooling is analyzed by FEM package ANSYS. 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. The analysis still becomes complicated by considering variable thermal conductivity, variable specific heat and non-uniform deceleration of the vehicle. This can be considered for the future work.

A full 3D analysis of the brake disc including the pads should be considered in order to investigate the effects of rotating heat source and the non-uniform heat flux over the rubbing surfaces due to non- uniform pressure distributions.

A programe of experimental work needs to be undertaken using a full size dynamometer since it can subject the brake to the same sequence of high energy stops that has been modeled in the numerical situation. This will provide the necessary data to validate the model and provide an indication of the location of possible fracture sites.

IX. ACKNOWLEDGEMENT

This case study and required models created by own without using any customer related technical data at CYIENT Ltd.

REFERENCES

[1] Floquet, A. and Dubourg, M.C. Non axis symmetric effects for three dimensional Analyses of a Brake, ASME J. Tribology, vol. 116, page 401-407, (1994).
[2] Ali.Belhocine and Mostefa.Bouchetara,Thermo mechanical Behavior Of Dry Contacts In Disc Brake Rotor With a grey cast Iron Composition, laboratory of mechanics applied, faculty of mechanical engineering, university of sciences and the technology of oran, l.p 1505 el-mnaouer, us to 31000 Oran (Algeria).
[3] KengLengKhong, Simulation of Temperature Distribution in Brake Discs, Submitted, Department of Mechanical Engineering.
[4] Kennedy, F.E., Colin, F.Floquet, A. and Glovsky, R. Improved Techniques for Finite Element Analysis of Sliding Surface Temperatures. Westbury House page 138-150, (1984).
[5] Lin, J.Y. and Chen, H.T. Radial Axis symmetric Transient Heat Conduction in Composite Hollow Cylinders with Variable Thermal Conductivity, vol. 10, page 2-33, (1992).
[6] Brilla, J. Laplace Transform and New Mathematical Theory of Viscoelasticity, vol. 32, page 187- 195, (1997).
[7] Tsinopoulos, S. V, Agnantiaris, J. P. and Polyzos, D.An Advanced Boundary Element/Fast Fourier Transform Axis symmetric Formulation for Acoustic Radiation and Wave Scattering Problems, J.ACOUST. SOC. AMER., vol 105, page1517-1526, (1999).
[8] Blot, B., 'Computation of a thermally stressed brake disc', SAE Technical Paper Series: 890089, 1989
[9] Sheridan, D.C., 'Approaches to the thermal modeling of Disc Brakes', SAE Technical Paper Series: 880256, 1988
[10] Yano, M. and Murata, M., 'Heat flow on Disc Brakes', SAE Technical Paper Medenos, S., 'Study of structural behavior of Ventilated disc brake', IMechEConference Transactions No C521/009/98, 1998

[11] Fukano, A. and Matsui, H., 'Development of Disc Brake Method using computer simulation of heat phenomena' SAE Technical Paper Series: 860634, 1986
[12] Wang, H.C. and Banerjee, P.K Generalized Axis symmetric Elastodynamic Analysis by Boundary Element Method, vol. 30, page 115-131, (1990).
[13] Burton, R. A. Thermal Deformation in Frictionally Heated Contact, Wear, vol. 59, page 1- 20, (1980).
[14] Anderson, A.E. and Knapp, R. A. Hot Spotting in Automotive Friction System Wear, vol. 135, page 319-337, (1990).