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Transient Thermal Analysis of Disc Brake Using FEM



Ms.Pallavi Pedapenki
M.Tech.[Thermal] Student,
Department of Mechanical Engineering,
Adarsh College of Engineering,
Chebrolu, Kakinada.

Dr.T.Dharma Raju, Ph.D
Professor,
Department of Mechanical Engineering,
Adarsh College of Engineering,
Chebrolu, Kakinada.

Mr.A.Rupesh Venkata Ramana
Assistant Professor,
Department of Mechanical Engineering,
Adarsh College of Engineering,
Chebrolu, Kakinada.

ABSTRACT:

Sliding systems with frictional heating exhibit thermo elastic instability (TEI) when the sliding speed exceeds the critical value. TEI can lead to hot spots on contact surfaces and is generally of great practical importance in friction brakes and clutches. The phenomenon is well defined in terms of the theory of stability with a classic perturbation approach being commonly used. While the perturbation analysis determines the stability limit, recent interest extends further towards exploration of the unstable behaviour. This is motivated by practical reasons, namely by the fact that many common friction brakes and clutches operate instantaneously at speeds exceeding the critical speed for TEI, i.e. in the unstable regime.

Using the brake disk prepared with SiC network ceramic frame reinforced 6061 aluminum alloy composite (SiC/Al). The thermal and stress analyses of SiC/Al brake disk during emergency braking at a speed of 300 km/h considering airflow cooling were investigated using finite element (FE). All three modes of heat transfer (conduction, convection and radiation) were analyzed along with the design features of the brake assembly and their interfaces. The results suggested that the higher convection coefficients achieved with airflow cooling will not only reduce the maximum temperature in the braking but also reduce the thermal gradients, since heat will be removed faster from hotter parts of the disk. Airflow cooling should be effective to reduce the risk of hot spot formation and disc thermal distortion. The highest temperature after emergency braking was 461 °C and 359 °C without and with considering airflow cooling, respectively. However, the maximum surface stress may exceed the material yield strength during an emergency braking, which may cause a plastic damage accumulation in a brake disk without cooling.

1. INTRODUCTION:

1.1 DISK BRAKE:

A disc brake is a type of brake that uses calipers to squeeze pairs of pads against a disc in order to create friction that retards the rotation of a shaft, such as a vehicle axle, either to reduce its rotational speed or to hold it stationary. The energy of motion is converted into waste heat which must be dispersed. Hydraulic disc brakes are the most commonly used form of brake for motor vehicles but the principles of a disc brake are applicable to almost any rotating shaft. Compared to drum brakes, disc brakes offer better stopping performance because the disc is more readily cooled. As a consequence discs are less prone to the brake fade caused when brake components overheat. Disc brakes also recover more quickly from immersion (wet brakes are less effective than dry ones). Most drum brake designs have at least one leading shoe, which gives a servo-effect. By contrast, a disc brake has no self-servo effect and its braking force is always proportional to the pressure placed on the brake pad by the braking system via any

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brake servo, braking pedal, or lever. This tends to give the driver better "feel" and helps to avoid impending lockup. Drums are also prone to "bell mouthing" and trap worn lining material within the assembly, both causes of various braking problems.

Fig 1.1 : Close-up of a disc brake on a car

1.2 HISTORY

EARLY EXPERIMENTS

Development of disc brakes began in England in the 1890s.The first caliper-type automobile disc brake was patented by Frederick William Lanchester in his Birmingham factory in 1902 and used successfully on Lanchester cars. However, the limited choice of metals in this period meant that he had to use copper as the braking medium acting on the disc. The poor state of the roads at this time, no more than dusty, rough tracks, meant the copper wore quickly making the system impractical. Successful application began in airplanes and tanks during World War II. At Germany's Argus Motoren, Hermann Klaue (1912-2001) had patented disc brakes in 1940. Argus supplied wheels fitted with disc brakes e.g. for the Arado Ar 96. The German Tiger I heavy tank, introduced in 1942 with a 55 cm Argus-Werke disc on each drive shaft, can be considered the first mass produced vehicle with disc brakes.

2.0 Description of model components

Physical brake system and its components

The brakes of UTA formula race cars, like every other brakes, serves two basic functions of a brake system i.e. to slow down a vehicle and to stop a vehicle to complete rest. As described earlier, disk brakes are used for their ease in application, reliability, better time response and their ability to withstand a higher temperature tolerance than their counterpart drum brakes. There are two types of disk brakes available in market. Those are floating type disk brake and fixed type disk brake. The description for both of them are given below:

(i) Floating type: In the floating type disk brake type, there is a hydraulic line coming from the master cylinder and entering only one side of the caliper.
(ii) Fixed type: In fixed type disk brake type, hydraulic line is present in both sides of the caliper. On application if brakes, the fluid exerts equal pressure on both the sides of the caliper equally, which in turn makes the piston push the brake pads at the same time, thus eliminating any delay, which is a major factor in formula racing cars.

2.1 Thermal dynamics of a disk brake and its components

As mentioned earlier, frictional heating is the major source of heat generation in any brake system. The kinetic energy of the vehicle is converted into frictional heat when a moving vehicle is brought to rest by disk- pad contact.

$$F = \frac{M}{3} \quad v = \frac{W}{3} g' S$$

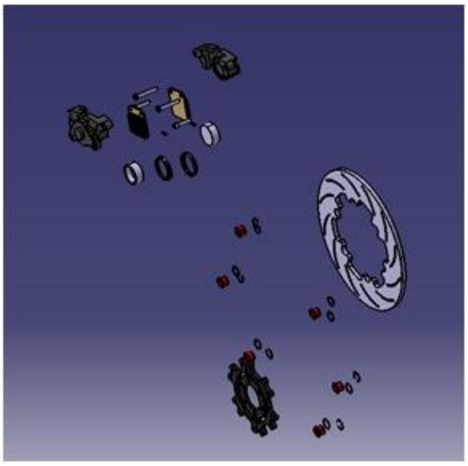

Fig 2.1: Solidworks model of the disk brake arrangement used in the system

The heat generation for each tire can be calculated by the equation

$$Q_{brake} = Fv$$

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2.2 Disk pad interaction and heat distribution:

Once the heat is generated by virtue of frictional contact between the brake pad and the brake rotor there is a rise in temperature of both the contacting surfaces.

$$\sigma = \frac{\xi_d S_d}{\xi_d S_d + \xi_p S_p}$$

Rotor disk thermal analysis:

The rotor disk or simply the rotor of a disk brake is the component is the rotating part of a brake system which is required to be stopped or slowed during a deceleration. The brake pads come in direct contact with the rotor to do the job.

Fig 2.2 : Solid works model of the disk used in simulation

$$MC_pT_r = Q_{brake} - {}_{conduction} - Q_{convection} - Q_{radiation}$$

$$Q_{convection} = hA(T_r - T_\infty)$$

$$Q_{conduction} = \frac{4kwt}{L}(T_r - T_\infty)$$

$$Q_{radiation} = f_v f_e \sigma A(T_r^4 - T_\infty^4)$$

Brake pad thermal analysis:

The heating of the brake pads will occur in a similar manner like the rotor disk. The brake pads are made of low thermal conductivity, insulating material so as to absorb most of the heat produced during braking and only a small portion of the heat is conducted through it to the piston.

$$M_{pad}C_{pad}T_{pad} = Q_{pad} - Q_{padconvection} - Q_{padconduction} - Q_{padradiation}$$

$$Q_{padconvection} = h\,A_{pad}(T_{pad} - T_\infty)$$

$$Q_{padconduction} = k_{pad}A_{pad}\,t_{pad}(T_{pad} - T_\infty)$$

$$Q_{padradiation} = f_{vpad}\,f_{epad}\,\sigma_{pad}\,A_{pad}\,(T_{pad}^4 - T_\infty^4)$$

Theoretical modelling of the problem and solution

Numerical solution for disk temperature

The following data were being fed into the final governing equation of disk Eq. 2.3.

Ambient temperature=25°C

Mass of vehicle (along with driver) = 290 Kg

Surface area of disk, A, = 0.05 square meter

g's (deceleration) = 1

g's (acceleration) = 0.5

Maximum velocity = 26.83 meters per second (60 miles per hour)

Minimum velocity = 13.41 meters per second (30 miles per hour)

Thermal conductivity of steel k, = 51.88 W/(mK)

Specific heat of steel C_p, = 504 Joules/kg

Thickness of disk, t= 0.0053 meter

Width of the mounting hub slot, w= 0.00635 meter

Length of the mounting slot, L= 0.042 meter

Mass of disk, M= 0.9906 Kilograms

Convection coefficient h= 150 W/ (m^2°K) (assumed)

View factor, f_v = 1.0

Emissivity, = 0.9 (assumed)

Stefan-Boltzmann constant = $5.67 \times 10^{-8} N/m^2/C^{-4}$

The MATLAB simulation has been done in two parts. The first one is considered that the vehicle was running continuously and there was no time lapse during the racing duration of Formula car.

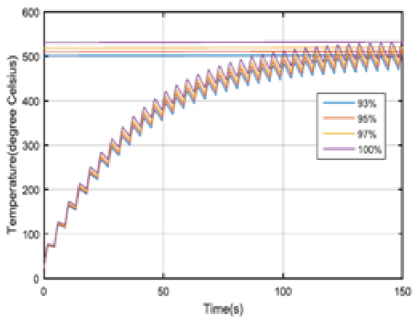

Fig 2.3 : Temperature of rotor without time lapse

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Table 2.1 : Temperature of rotor at different heat distribution factor after 10 braking cycles

Heat Distribution	93%	95%	97%	100%
Temperature°C	359	366	373	384

Table 2.2 : Temperature of rotor after 20 braking cycle without any time lapse for various heat distribution factor

Heat Distribution	93%	95%	97%	100%
Temperature°C	468	476	485	498

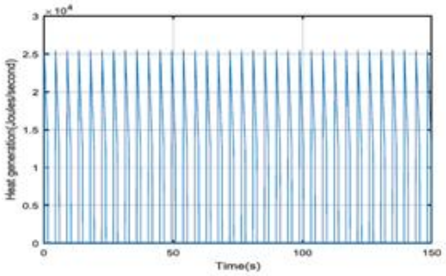

Figure 2.4 : Heat generation in rotor without any time lapse

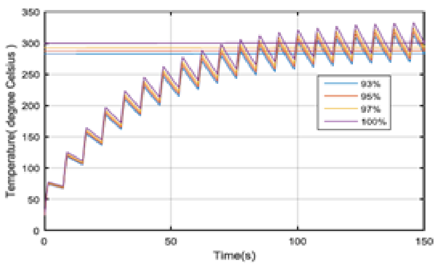

Fig 2.5 : Temperature of rotor at a time interval of 3 seconds

Table 2.3 : Temperature of rotor after 10 braking with time lapse of 3 seconds

Heat Distribution	93%	95%	97%	100%
Temperature°C	278	283	288	296

Table 2.4 : Temperature of rotor after 20 braking with time lapse of 3 seconds

Heat Distribution	93%	95%	97%	100%
Temperature°C	278	283	288	296

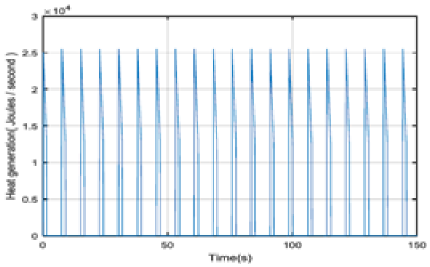

Fig 2.6: Heat generation in rotor during braking

Numerical solution for brake pad material

Mass of pad material, M_{pad}= .042 kg
Width of pad material = 0.038 meter
Length of pad material = 0.038 meter
Area of pad material, A_{pad} = .00145 square meter
Thickness of pad material, t_{pad}= 0.00877 meter
Thermal conductivity of brake pad material k_{pad}= 5 W/m/˚K
Specific heat of brake pad material C_{ppad} = 1000 J/kg/˚K
Convection coefficient h = 150 W/m^2/K (assumed)
View factor, f_v = 1.0 (assumed)
Emissivity, f_e = 0.2 (assumed)
Stefan-Boltzmann constant= $5.67 \times 10^{-8}\text{N/m}^2/\text{K}^{-4}$

The assumed view factor for the pad material is lesser than that for the steel rotor because emissivity is a factor of surface polish along with temperature. Due to a poorer surface finish and lesser temperature rise than the steel rotor, it's being assumed as 0.2.

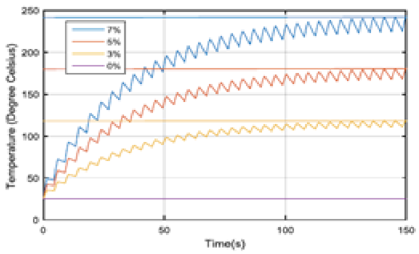

Fig 2.7 : Temperature of brake pad material without time lapse

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Table 2.5 : Temperature of brake pad material with time lapse of 3 seconds after 20 braking

Heat Distribution	7%	5%	3%	0%
Temperature°C	174	131	88	25

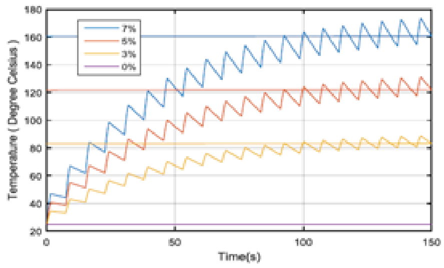


Fig 2.8 : Temperature of brake pad material with time lapse

3.0 RESULT:

Properties of disc plate:

The following data were being fed into the final governing equation of disk.

Ambient temperature=25°C

Mass of vehicle (along with driver) = 290 Kg

Surface area of disk, A, = 0.05 square meter

g's (deceleration) = 1

g's (acceleration) = 0.5

Maximum velocity = 26.83 meters per second (60 miles per hour)

Minimum velocity = 13.41 meters per second (30 miles per hour)

Thermal conductivity of k, = 51.88 W/(mK)

Specific heat of , = 504 Joules/kg

Thickness of disk, t= 0.0053 meter

Width of the mounting hub slot, w= 0.00635 meter

Length of the mounting slot, L= 0.042 meter

Mass of disk, M= 0.9906 Kilograms

Convection coefficient h= 150 W/ (m^2°K) (assumed)

View factor, f_v = 1.0

Emissivity,   = 0.9 (assumed)

Stefan-Boltzmann constant = $5.67 \times 10^{-8} N/m^2/C^{-4}$

Numerical solution for brake pad material

Mass of pad material, M_{pad}= .042 kg

Width of pad material = 0.038 meter

Length of pad material = 0.038 meter

Area of pad material, A_{pad} = .00145 square meter

Thickness of pad material, t_{pad}= 0.00877 meter

Thermal conductivity of brake pad material k_{pad}= 5 W/m/°K

Specific heat of brake pad material C_{ppad} = 1000 J/kg/°K

Convection coefficient h = 150 W/m^2/K (assumed)

View factor, f_v = 1.0 (assumed)

Emissivity, f_e = 0.2 (assumed)

Stefan-Boltzmann constant= $5.67 \times 10^{-8} N/m^2/K^{-4}$

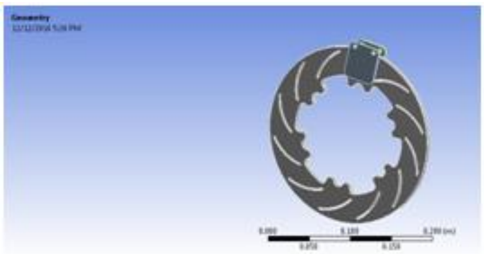
Table 3.1

S.NO.	MATERIAL PROPERTIES FOR DISC	STEEL	Cast Iron
1	Thermal conductivity	51.88 W/(mK)	45 W/(mK)
2	Specific heat	504 Joules/kg	0.49 kJ/(kg K
3	Tensile strength	620MPA	414MPA
4	Yield strength	415MPA	276MPA
5	Density	7.8g/cm3	7.2g/cm3
6	Young's Modules	210MPA	140GPA
7	Poisson Ratio	0.3	0.26
8	Thermal Expansion	2.10E-06	1.10E-06

Table 3.2:

S.NO.	MATERIAL PROPERTIES FOR brake pad	VALUE
1	Thermal conductivity	5 W/m/°K
2	Specific heat	1000 J/kg/°K
3	View factor	1
4	Emissivity	0.2

Model & Mesh of disc with pad plate:



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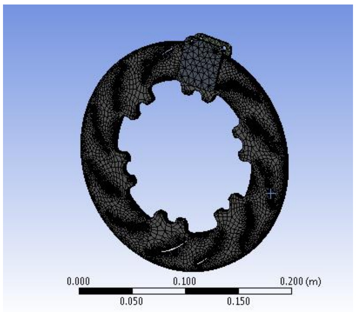
Model of disc and pad is imported into ansys



Mesh model

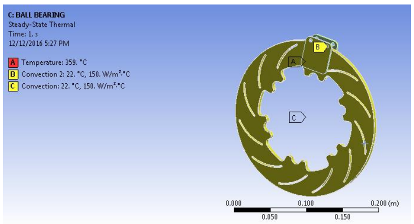
Table:3.3

Heat Distribution	93%	95%	97%	100%
Temperature°C	359	366	373	384

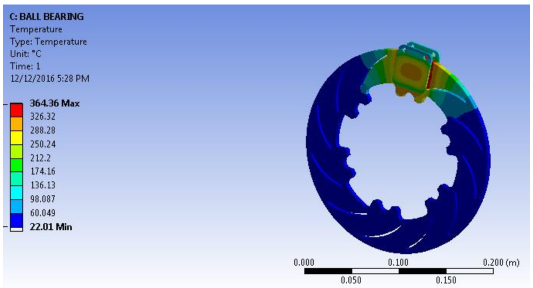
Heat Distribution	93%	95%	97%	100%
Temperature°C	468	476	485	498

Case1:

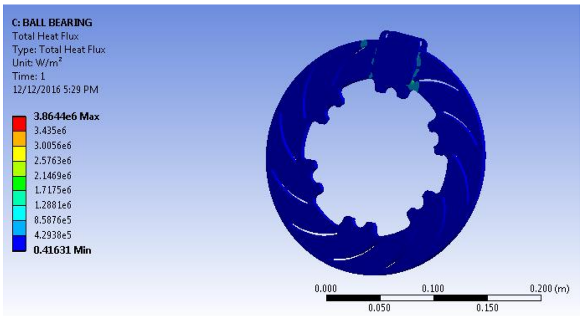
Temperature of 359 °C



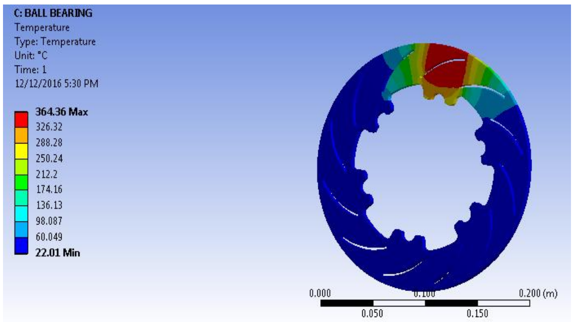
Load and BC applied to model



Temperature distribution various from 22.01 to 364.36°C



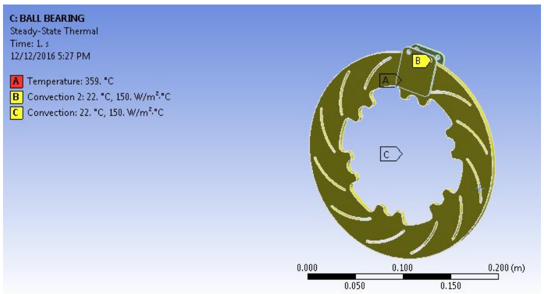
Heat flux destruction ranges from 0.416 to 3.0644e6 (W/M^2)



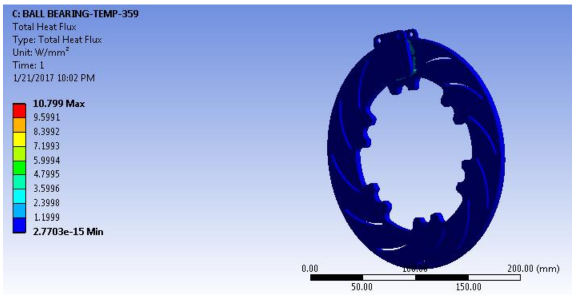
Temperature distribution for disc is shown

Case2-CI MATERIAL FOR DISC:

Temperature of 359 °C



Load and BC applied to model



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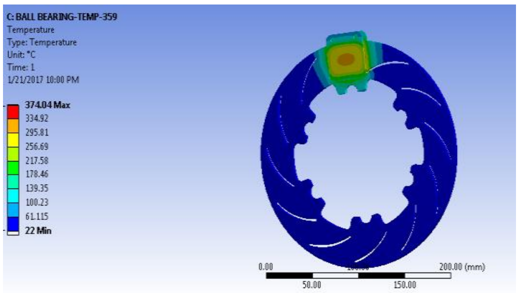
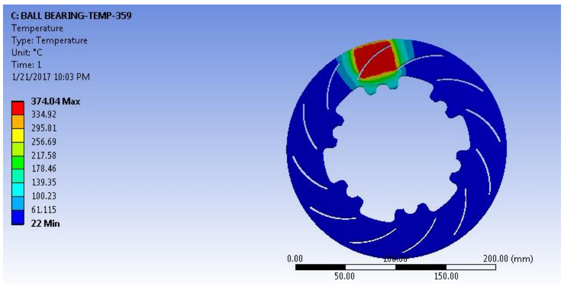
Heat flux destruction ranges from 2.77E-15 to 10.79 (W/MM^2)



Temperature distribution various from 22 to 374.04°C



Temperature distribution for disc is shown

RESULTS TABLE 3.5
COMPOSITE MATERIAL BRAKE DISC AND PAD:

S.NO.	TEMPERATURE	COMP-TEMP(oC)	COMP-HEAT FLUX *E6(W/M2)	CI-TEMP(oC)	CI-HEAT FLUX *E6(W/M2)
1	T-359	364.53	3.864	374.04	10.79
2	T-366	371.47	3.944	381.35	11.02
3	T-373	378.58	4.024	388.66	11.24
4	T-384	389.75	4.151	400.15	11.6
5	T-468	475.09	5.114	487.9	14.29
6	T-476	483.22	5.206	496.26	14.54
7	T-485	492.36	5.309	505.66	14.83
8	T-498	505.57	5.458	519.26	15.23

AS PER TEMPERATURE LOAD FROM DISC TO PAD :

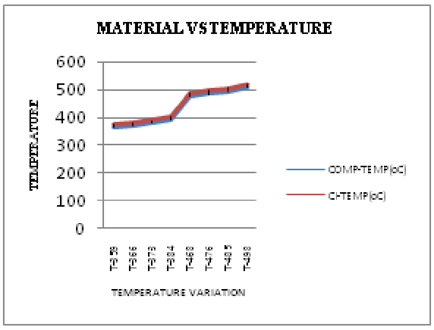


FIG3.1: PLOTED TEMPERATURE VARIATION VS MATERIAL

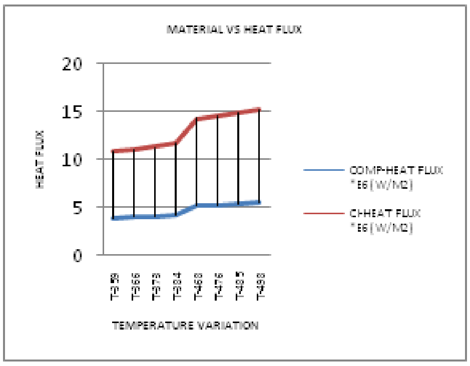


FIG3.2: PLOTED HEAT FLUX VS MATERIAL

CONSLUSION:

The model developed to determine the brake fluid vaporization temperature showed some really interesting patterns. The heat generation and temperature rise of the rotor was well. The governing equations used for the calculation of rotor temperature individually worked because the thermal mass capacitance was not required to be coupled to be with any other body. This provided the rotor the flexibility to reach a temperature irrespective of the temperature of the other components in the brake system. Temperature distribution and temperature output calculated from Matlab program. Same temperature output used for Ansys input load to disc and pad. Then calculated heat distribution and Heat flex are shown in results. From Result table it come concluded that Temperature distribution faster rate as per heat input. With the availability of more accurate data for the physical and thermal property of the components of the brake system, the above model can be used reliably to predict the brake fluid temperature using a much simpler approach as discussed in this report. The same modeling approach can be used to find the temperature response of backing plate and piston too. This small step towards the development of a more thermally stable brake system will give design engineers a hands on quick on the go option during the initial stages of prototype designing for the dimension and material selection which could be further modified during the later stages of product development. From basic material Cast Iron and Composite material is compared temperature flow and Heat Flux checked.

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From result table it is shows that heat flow is decrease using composite materials and temperature flow is decrease using composite materials.

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AUTHORS:

Student:

Ms.Pallavi Pedapenki M.Tech.[thermal] student, Department of Mechanical Engineering, Adarsh college of Engg Chebrolu Kakinada.

Guide:

Dr.T.Dharma Raju was born in Andhra Pradesh, India. he has received P.hD from JNTU Hyderabad,Telangana, India. He is working as Professor in Mechanical Engineering Department, Adarsh college of Engineering, Chebrolu, Kakinada. India.

Mentor:

Mr. A.Rupesh Venkata Ramana was born in Andhra Pradesh, India. He has received M.Tech. [CAD/CAM] from SRKR Engineering College, Bhimavaram. AP, India. He is working as Assistant professor in Mechanical Engineering dept, Adarsh college of engineering ,chebrolu, Kakinada.