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DESIGN AND ANALYSIS OF DISC BRAKE ROTOR FOR ALL TERRAIN VEHICLES

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Abstract: A disc brake is a device by means of which artificial frictional resistance is applied to the rotating member, in order to stop the motion of a machine or a vehicle. Brake fading and thermo-elastic deformation of brake rotor is frequently found problem in custom made rotors. Due to combined effect of mechanical forces and temperature rise, the rotor deforms. This problem is very severe in All Terrain Vehicles (ATVs). To avoid this problem, pre-manufacturing analysis of rotor is must. This project deals with the structural and thermal analysis of rotor disc of disc brake BAJA SAE CAR through finite element analysis approach using ANSYS software. The rotor discs are commonly manufactured of grey cast iron. The SAE also recommend grey cast iron for various applications. SS410 is also suitable for condition of parameters of rotor disc like Disc diameter, Pattern & Material composition. The modelling of rotor disc is done in Solidworks 2017 software. Analysis is done by using ANSYS 17.2

KEYWORDS: Structural and Thermal analysis, SAE BAJA, Rotor Disc, Grey Cast Iron, SS 410, Finite Element Analysis

INTRODUCTION

A brake is a mechanical device that inhibits motion by absorbing energy from a moving system. It is used for slowing or stopping a moving vehicle, wheel, axle, or to prevent its motion, most often accomplished by means of friction. It is an energy converting mechanism that converts vehicle movement into heat while stopping the rotation of the wheels. All braking systems are designed to reduce the speed and stop a moving. This is done by causing friction at the wheels. Friction converts the kinetic energy into heat. The greater the pressure applied to the objects, the more friction & heat produced, & the sooner the vehicle is brought to a stop. Kinetic friction acts in the brakes and static friction between the tire and road to slow the vehicle. Most brakes commonly use friction between two surfaces pressed together to convert the kinetic energy of the moving object into heat, though other methods of energy conversion may be employed. For example, regenerative braking converts much of the energy to electrical energy, which may be stored for later use. Other methods convert kinetic energy into potential energy in such stored forms as pressurized air or pressurized oil. Eddy current brakes use magnetic fields to convert kinetic energy into electric current in the brake disc, fin, or rail, which is converted into heat. Still other braking methods even transform kinetic energy into different forms, for example by transferring the energy to a rotating flywheel.

DISC BRAKE

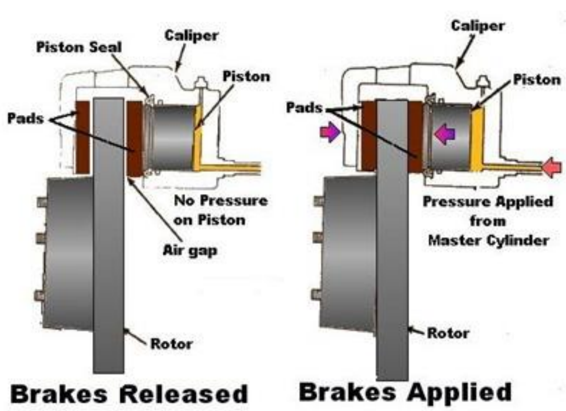
A disc brake is a type of brake that uses the calipers to squeeze pairs of pads against a disc or rotor to create friction. This action slows the rotation of a shaft, such as a vehicle axle, either to reduce its rotational speed or to hold it stationary. The energy of motion is converted into waste heat which must be dispersed.

Brake disc (Rotor)

The brake disc (rotor) is the rotating part of a wheel's disc brake assembly, against which the brake pads are applied. Some are simply solid, but others are hollowed out with fins or vanes joining together the disc's two contact surfaces. The weight and power of the vehicle determines the need for ventilated discs.

WORKING PRINCIPLE OF DISC BRAKE

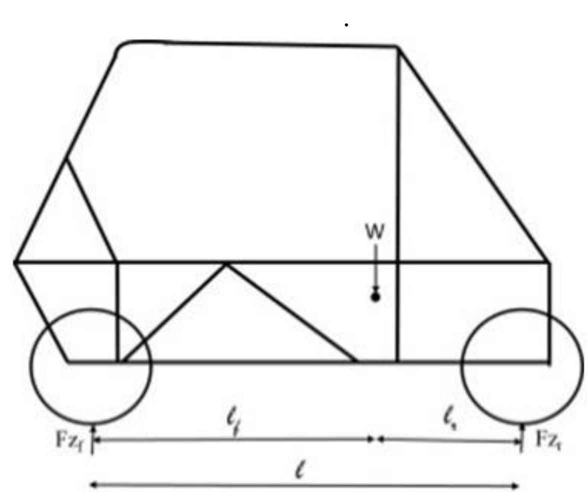
When a brake lever or pedal is pressed, the push rod which is connected to lever or pedal and master cylinder piston pushes the master cylinder piston. This movement allows the master cylinder piston to slide and push the return spring inside the bore of master cylinder, which generates pressure in reservoir tank. At this moment a primary seal allows the brake fluid of reservoir tank to flow over it into the brake hosepipes. A secondary seal ensures that the brake fluid does not go other side.Then the fluid enters in to cylinder bore of caliper assembly via brake hosepipes and pushes the caliper piston or pistons. At this time the piston ring moves in rolling shape with piston. Then the caliper piston pushes brake pad. This movement causes brake pads to stick with brake disc which creates friction and stops the brake disc/rotor to rotate. This way disk brake system stops or slows down the vehicle.



When the brake lever or pedal is released the piston ring pushes the calliper piston back to cylinder bore of calliper till both, calliper piston and piston ring come into their original shape. At this time retraction spring pushes the brake 5 pads to their original position. The return spring in master cylinder assembly pushes the master cylinder piston back into its original position and allows the fluid to flow back to reservoir via hosepipe and master cylinder bore.

CALCULATION OF BRAKE DISC

Data required for calculation			
Parameter	Symbol	Value	Unit
Pedal Ratio	PR	6:1	
Master cylinder Bore Diameter	d_b	19.05	mm
Calliper piston Diameter	d_c	34	mm
Area of Master cylinder	a_b	2.85×10^{-4}	m^2
Area of caliper piston	a_c	9.08×10^{-4}	m^2
Pedal force(assumed)	PF	250	N
Total weight of vehicle	W	200+60	Kg
Wheel base (l)	l	142.2	mm
Distance of rear wheel centre from Center of gravity	l_r	58	mm
Center of gravity height	h	75	mm
Radius of wheel	r	0.2794	m
Coefficient of friction between tyre and road	μ_r	.65	
Coefficient of friction between pad and disc	μ_d	.40	
Swept area of pad for 160 mm disc	A	0.0205	m^2
Swept area of pad for 170 mm disc	A	0.0221	m^2



Static Weight Distribution :

Taking moment about rear axle,

Fzf x l = W x lr

Fzf = (W x lr) / l = (2600 x 58)/142.2

Fzf = 106.05 Kg

(Therefore, static weight distribution is in the ratio 40:60)

Dynamic Weight Distribution:

Consider Δdyn as dynamic weight distribution, this dynamic weight is loaded to front axle by unloading from rear axle during braking. Taking moment of the forces acting on the vehicle about the centre of gravity.

We get dynamic weight equation as follows,

$\Delta dyn = (h/l) \times (a/g) \times W$

$= (h/l) \times \mu r \times W$

Since Traction force (F= μrmg) and Force produced by the vehicle (F=ma) should be equal.

$\Delta dyn = (75/142.2) \times 0.65 \times 260$

$\Delta dyn = 89.14$ Kg

Front Axle Reaction Force:

It is equal to normal force acting on the front axle between road and vehicle.

$Fzf + \Delta dyn = 106.05 + 89.14$

$= 195.2$ Kg (1914.81 N)

Front Axle braking Force:

It is the frictional force acting on the front axle between road and tyre.

$Fxf = Fzf, \Delta dyn \times \mu r = 1914.81 \times 0.65$

$Fxf = 1244.62$ N

Torque generated by the disc:

$Txf = Fxf \times r = 1244.62 \times 0.2794$

$= 347.75 \sim 350$ Nm

Torque produced by the single front wheel (or disc)

$Txf = 175$ Nm

Validation of Brake Torque Requirement

The braking torque should be more than the torque produed in order for braking.

Force at master cylinder:

The applied force is transferred through the push rod to master cylinder

$Fmc = PF \times PR = 250 \times 6$

$= 1500$ N.

Force at Calliper Piston:

It is the normal force acting on the disc. It is called as clamping force. The force applied traveled to caliper through brake fluid. According to Bernoulli's theorem pressure at master cylinder and calliper piston is equal.

$Fc = Pmc \times ac = (Fmc \times ac) / amc$

$= (1500 \times 9.08 \times 10\text{-}4) / 2.85 \times 10\text{ -}4$

$= 4778.94$ N

Considering Brake fluid loss, leverage loss, assume efficiency as 80%.

$Fc = 3822.68$ N

Braking Force on Discs:

It is the frictional force between pad and disc. This force is acting on both side of the disc

$Fc(disc) = 2 (\mu d \times Fc) = 2 (0.40 \times 3822.68)$

$= 3058.144$ N.

Torque Generated:

$Tfront = Fc(disc) \times Reffective$

Where,

Reffective = (Disk diameter/2) - (Calliper diameter/2)

$= (160/2) – (34/2)$

= 63 mm

Tfront = 3058.144 x 0.063

= 192 Nm ~ 200 Nm

Thus, Braking torque required > Braking torque (front). Thus, the expected disc diameter is validated for design..

Heat Flux Entering the Rotor

The following assumptions are made before calculations:

1. Whole kinetic energy transferred into heat energy through rotor

2. Heat transfer through the tyre is neglected

3. Heat transfer through the pad is neglected

4. Approximated swept area is used to calculate heat flux

Abbreviations:

E = Heat Energy

u = Initial Velocity

v = Final Velocity

t = Braking Time

P = Power (Heat Flow)

A = Swept area of pad on the disc

Φ = Heat Flux

Maximum Heat Energy:

The kinetic energy produced is wholly transferred to disc.

Maximum heat energy(E) = maximum kinetic energy

E = 0 .5 x mv 2 = 0.5 x (260 x 16.672)

= 36125.557 J

Braking Time:

Using velocity as a function of acceleration equation v = u + at

Deceleration = μr g = 0.65 x 9.81

= 6.3765 m/s 2

Acceleration = -6.3765 m/s 2

Hence,

0 = 16.67 + (-6.3765 x t)

t = 2.615 s

Power generated on a disc:

Heat generated per time is called Power (Heat flow).

P = E / t = 36125.557/2.615

= 13814.75 J/s

As 75% of the kinetic energy is been distributed to the front wheels power generated in one rotor

= (P x 0.75) / 2 = (36125.557 x 0.75)/2

=5180.53 J

Heat flux generated:

Flow of energy per unit of area per unit of time is called Heat flux (Heat flux density).

Heat flux entering disc is through the swept area of pad.

Φ = P/2A

= 13547.08/ (2 x 0.0221)

=117467.82 w/m2 (for 170 mm)

Design of Rotor (Brake disc)

The design part of the project was done in Solidworks 2017.

The design is done with following specification

Specifications

Disc Inner Diameter = 60 mm

Disc Outer Diameter = 160 mm

Thickness = 4 mm

Diameter of hole for bolt = 9 mm

With the above specification , the rotor is designed along with the two different patterns

Slot 1 = Forward Curved Fin

Slot 2 = fin with small hole in outer range

In both the slots the fins are ranged around 16-17 in such a way that the surface area is approximately equal between different patterns

170 mm – Slot 1

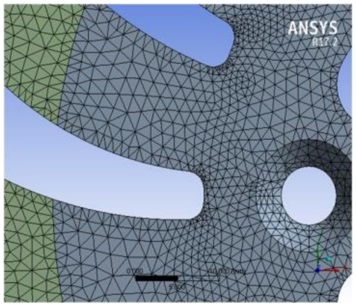
170 mm – Slot 2

Meshing

Element size - 2mm

Span angle center - fine

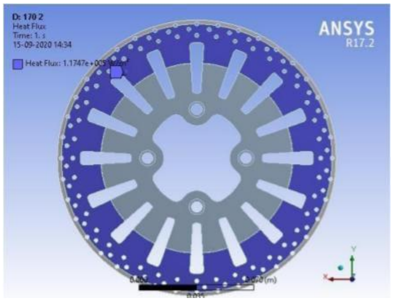
Smoothing – high



THERMAL ANALYSIS

Heat flux

The heat flux is enter into disc through the area swept by the pad which was imprinted in the disc. Heat flux is applied to both side of the disc.

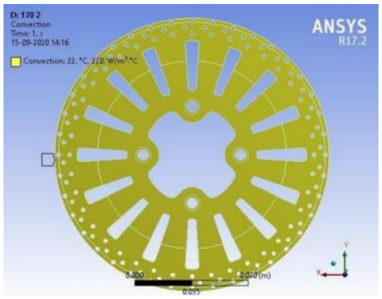


CONVECTION

we assumed heat transfer coefficient as

220 w/moC for 170 mm

Convection takes place at all the surface of the rotor, hence whole body is selected

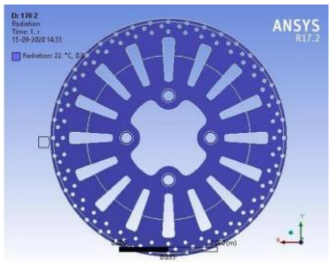


Radiation

Thermal radiation generates from the emission of electromagnetic waves..

It depends on the parameter called emmisivity taken as 0.8 for both material .

The whole body Is taken for radiation.



RESULTS FOR STEADY STATE THERMAL ANALYSIS

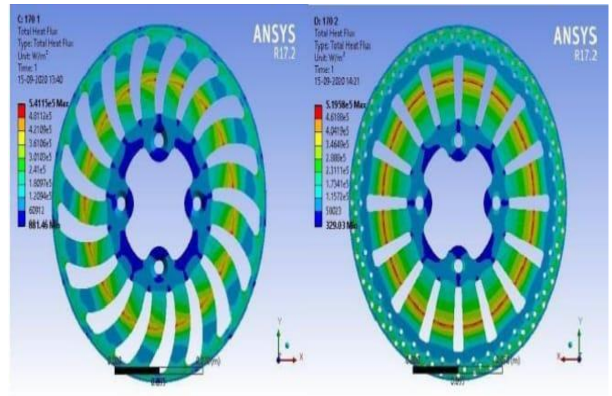
Steady-state thermal analyses to calculate the thermal response to heat loads depending on the prescribed temperatures or applied convection conditions or both. Steady-state thermal analyses assume a steady-state for all thermal loads and boundary conditions. This form of analysis does not evaluate changes over time, where heat storage effects varying over a period of time can be ignored.

Heat transfer is the physical act of thermal energy being exchanged between two systems by dissipating heat. Temperature and the flow of heat are the basic principles of heat transfer. The amount of thermal energy available is determined by the temperature, and the heat flow represents movement of thermal energy.

Temperature variation and heat flux throughout the geometry of the rotor are calculated and analysed here

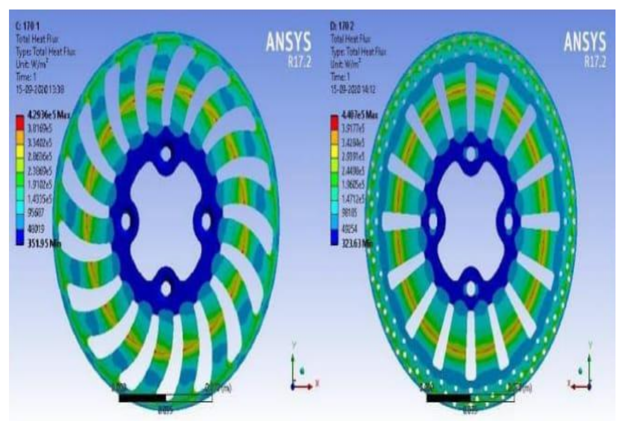
MAXIMUM HEAT FLUX

GREY CAST IRON



170 mm(slot 1) 170 mm(slot 2)

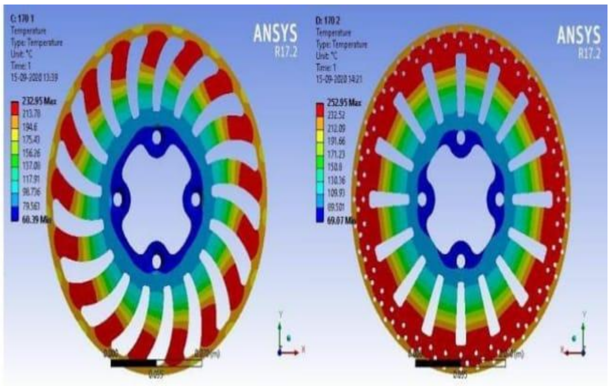
STAINLESS STEEL 410



170 mm(slot 1) 170 mm(slot 2)

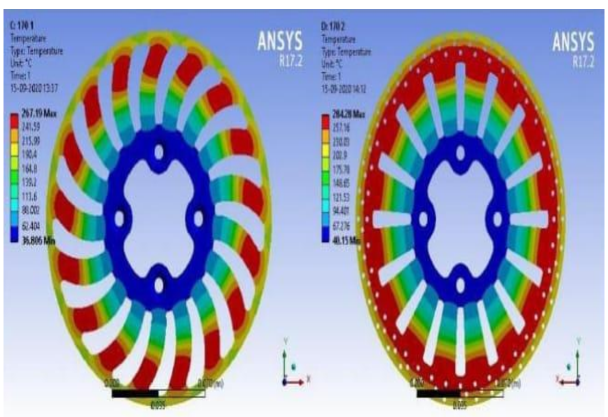
MAXIMUM TEMPERATURE

GREY CAST IRON



170 mm (slot 1) 170mm (slot 2)

STAINLESS STEEL 410



170 mm (slot 1) 170 mm (slot 2)

MAXIMUM HEAT FLUX RESULTS

GREY CAST IRON

SLOT 1

MAXIMUM HEAT FLUX: 5.4115e 5 w/m2

MINIMUM HEAT FLUX: 881.46 w/m2

SLOT 2

MAXIMUM HEAT FLUX: 5.1958e 5 w/m2

MINIMUM HEAT FLUX: 329.03 w/m2

STAINLESS STEEL 410

SLOT 1

MAXIMUM HEAT FLUX: 4.2936e 5 w/m2

MINIMUM HEAT FLUX: 351.95 w/m2

SLOT 2

MAXIMUM HEAT FLUX: 4.407e 5 w/m2

MINIMUM HEAT FLUX: 323.63 w/m2

MAXIMUM TEMPERATURE RESULTS

GREY CAST IRON

SLOT 1

MAXIMUM TEMPERATURE: 232.95 ℃

MINIMUM TEMPERATURE: 60.39 ℃

SLOT 2

MAXIMUM TEMPERATURE: 252.95 ℃

MINIMUM TEMPERATURE: 69.07 ℃

STAINLESS STEEL 410

SLOT 1

MAXIMUM TEMPERATURE: 267.19 ℃

MINIMUM TEMPERATURE: 36.806 ℃

SLOT 2

MAXIMUM TEMPERATURE: 284.28 ℃

MINIMUM TEMPERATURE: 40.15 ℃

INFERENCE AND CONCLUSION

1. Comparing material, it has been found that maximum stress produced is equal to each other and below its yield strength. But for SS410 maximum stress very much lower than its yield strength which results less deformation than gray cast iron and it can further increase its surface area to increase heat dissipation.

2. Maximum temperature of SS410 is little more than gray cast iron. And SS410 temperature range is lesser than grey cast iron, which shows poor conduction.

3. Comparing pattern design slot 2 has less stress than slot 1 as well as its deformation

4. So, it has been concluded

I) If corrosion resistance is the priority, SS410 of diameter 170 mm with slot 1 is best selection though it has high stress which is far less than yield strength, it has less possibility of cracking and wear off at bolted place. And also, the cost of machining is also comparatively less.

ii) If reducing brake fade is priority, gray cast iron of diameter 170 mm with slot 2 is best selection.

iii) Hence SS410 of diameter 170mm with slot1 is the best choose. Further it can be optimized more by increasing width of the slot so as to reduce maximum temperature without compromising with stress control since it has more yield strength.

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