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Design and Analysis of Disc Brake for a Tadpole Hybrid Trike

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ABSTRACT

Every single system has been studied and developed to meet safety requirements .One most critical system in the vehicle in context of safety is brake systems other than other expensive systems like air bag, good handling and good suspension systems. If an effective brake system is absent it puts the passengers in great danger. Therefore it's a must requirement for all vehicles to have a proper brake system. In this paper, we describe the design and analysis of a mechanically actuated disc brake for a hybrid trike in tadpole configuration. In context of design of brakes the formulae used are in reference to practical applications and analysis is done replicating practical physical boundary conditions.

Keywords

Vehicle Dynamics, Foundation Brake, Generating Braking, Real life Deceleration & Stopping Distance, Brake Heating, Parking on an Incline, Thermal Analysis.

1. INTRODUCTION

Disc brakes development and use began in England in the 1890s. Compare to drum brakes, disc brakes offer stopping performance, because the disc is more readily cooled. The disc brake is a wheel brake which slows rotation of the wheel by friction caused by pushing brake pads against a brake disc with a set of calipers. The disc brake may range from simple mechanical (cable) systems to expensive and powerful multi-piston hydraulic disc systems.

To stop the wheel, friction material in form of brake pads, mounted on a device called brake caliper. The caliper is connected to some stationary part of the vehicle like axle casing or the stub axle. Friction causes the disc and attached wheel to slow or stop the vehicle. Brakes convert motion to heat due to friction, and if the brakes get too hot, they become less effective, and this phenomenon is known as brake fade. Improved technology has seen creation of vented discsfor mountain bikes similar to those on cars, introduced to help avoid brake fade. Most bicycles brake discs are made of steel. Stainless Steel is preferred due to its anti-rust properties. In context to lightweight some disc brakes are made of titanium or aluminium. Discs are thin often about 2mm.

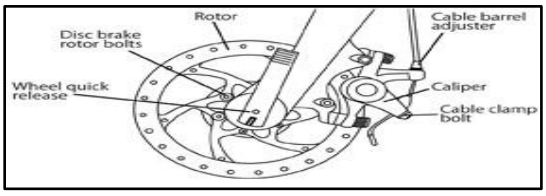

Fig 1: Disc Brake – Mechanically Actuated

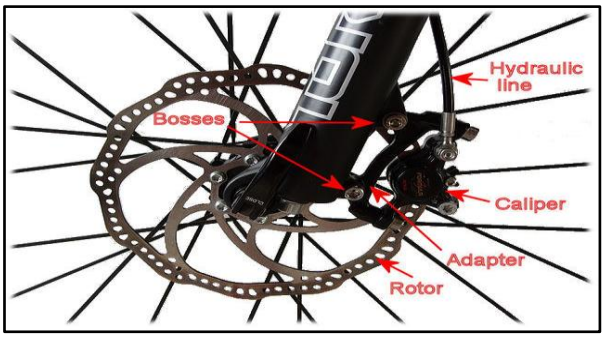

Fig 2: Disc Brake – Hydraulically Actuated

2. Calculation of Disc Brake

There are many books on brake systems but if you need to find formulae for design of brake in particular you never can. These formulae will work for any two axle vehicle.

Disc brake standard details:

Brake disc radius = 0.08 m

Brake disc material = Stainless steel

Pad brake material = Asbestos

Coefficient of friction μ= 0.8

Vehicle test speed = 8.6 m/sec

=0.8682 m (from rear axle)

2.1 Vehicle Dynamics

2.1.1 Static Axle Load Distribution

$$\Psi = M_r / M$$

where,

M_r = Static rear axle load(Kg)

M = Total vehicle mass(Kg)

Ψ = Static axle load distribution

So, M = 240 Kg, M_r = 0.4*240 = 96 Kg

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$\Psi = 96 / 240 = 0.4$

2.1.2 Relative Center of Gravity

$$X = h / W_b$$

where,

h = Vertical distance from C.G. to ground on the level (m)

W_b = Wheelbase (m)

X = Relative Centre of gravity height

So, h = 0.434 m, W_b = 0.9652 m

X = 0.434 / 0.9652 = 0.449647741

2.1.3 Dynamic Axle Load

$$M_{fdyn} = M.((1- \Psi)+(X.a))$$

where,

a = Deceleration (g units)

M_{fdyn} = Dynamic front axle load (Kg)

So, a = 1.272792206 g units

M_{fdyn} = 240 * ((1 – 0.4) + (0.449647741 * 1.272792206))

= 281.3539538 N

2.2 Stopping the Vehicle

2.2.1 Braking Force

$$B_F = Mag$$

where,

B_F = Total braking force (N)

g = Acceleration due to gravity (m/sec^2)

So, g = 9.81 m/sec^2

B_F = 240 * 1.272792206 * 9.81 = 2996.66197 N

2.2.2 Wheel Lock

$$F_A = M_{fdyn} .g . \mu$$

where ,

F_A = Total possible braking force on the axle (N)

M_{fdyn} = Dynamic axle mass (Kg)

So , F_A = 281.3539538 * 9.81 * 0.8 = 2208.065829 N

2.2.3 Brake Torque

$$T = B_F. R / r$$

where,

T = Brake torque (Nm)

R = Rotor Disc radius (m)

r = Speed ratio between the wheel and the brake

So, r = 1 (since both wheel and disc are rotating at same angular velocity), R = 0.08 m

T = 2996.66197 * 0.08 / 1 = 239.7329576 Nm

2.3 Foundation Brake

2.3.1 Disc Effective Radius

$$r_e = (D + d)/4$$

where,

r_e = Effective radius (m)

D = Disc useable outside diameter (m)

d = Disc useable inside diameter (m)

So, D = 0.16 m, d = 0.1092 m

r_e = (0.16 + 0.1092)/4 = 0.0673 m

2.3.2 Clamp Load

$$C = T/(r_e . \mu . n)$$

where,

C = Brake clamp load (N)

n = Number of friction faces

So, n = 2

C = 239.7329576 / (0.0673*0.8*2) = 2226.346 N

2.4 Real Life Deceleration & Stopping Distance

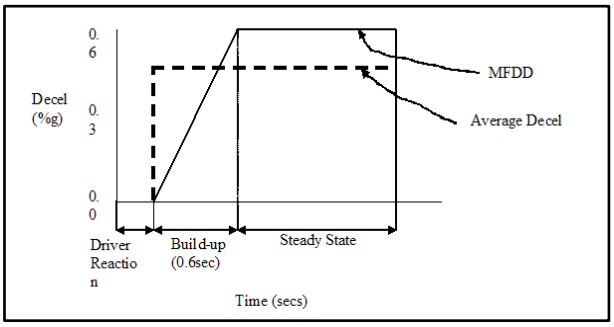

Fig 3: Mean Fully Developed Deceleration Graph

$$a_{ave} = v / ((v/a) + 0.3*g)$$

where,

a_{ave} = Average deceleration for the whole stop (g units)

v = Test Speed (m/sec)

So, a_{ave} = 8.6 / ((8.6/1.272792206) + 0.3 * 9.81)

= 0.88661639g units

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$s = v^2 / (2g \cdot a_{ave})$

where,

s = Stopping distance (m)

So, $s = 8.6^2 / (2*9.81*0.88661639) = 4.251695409$ m

2.5 Brake Heating

2.5.1 Stop Energy

The energy dissipated in a stop is the sum of energy from three sources, kinetic, rotational and potential.

2.5.2 Kinetic Energy

$KE = (1/2).M.v^2$

So, $KE = 0.5*240*8.6^2 = 8875.2$ J

2.5.3 Rotational Energy

3% of the KE is a reasonable assumption for RE.

So, RE = 0.03*8875.2 = 266.256 J

2.5.4 Potential Energy

$PE = (M.g.S)/ (1 + S^2)^{1/2}$

where,

S = Slope (tan of incline angle ɵ)

So, S = tan 45˚ = 1

$PE = (240*9.81*1)/ (1+1^2)^{1/2} = 1664.812206$ J

2.5.5 Braking Power

$t = v/ (a \cdot g)$

where,

t = brake on time (secs)

So, t = 8.6 / (0.88661639*9.81) = 0.988766374 sec

$P = E / t$

where,

P = Average power (Watts)

E = Energy (Joules)

So, E = 8875.2+266.256+1664.812206 = 10806.26821 J

P = 10806.26821 / 0.988766374 = 10929.04096 W

2.6 Dry Disc Temperature Rise

2.6.1 Heat Flux into one side of the disc

$q = 4P/ (\pi. (D^2-d^2))$

where,

q = heat flux (Watts/m^2)

So, $q = 4*10929.04096 / (\pi*(0.16^2-0.1092^2) = 1017544.484$ W/m^2

2.6.2 Single Stop Temperature Rise

$T_{max} =(0.527.q. (t)^{1/2}/ (\rho.c.k)^{1/2}) + T_{amb}$

where,

T_{max} = Maximum disc temperature (˚C)

ρ = Density of disc material (Kg/m^3) = 7700 Kg/m^3

c = Brake disc sp. heat capacity (J/Kg/K) = 500 J/Kg/K

k = Brake disc thermal conductivity (W/m/K) = 16.7 W/m/K

T_{amb}= Ambienttemperature (˚C) = 29 ˚C

So, $T_{max} = (0.527*1017544.484*(0.988766374)^{1/2}/ (7700. 500. 16.7)^{1/2}) + 29 = 95.50009469$ ˚C

2.6.3 Fade Stop Temperature Rise

$\Delta T = P.t / (\rho.c.V)$

where,

ΔT = Fade stop temperature rise (˚C)

V = Disc Volume (m^3) = $\pi(D^2-d^2)$*thickness of disc*(1/4) = 0.0000214812 m^3

So, ΔT = 10929.04096*0.988766374 / (7700*500*0.0000214812) = 130.664126 ˚C

2.7 Parking on an Incline

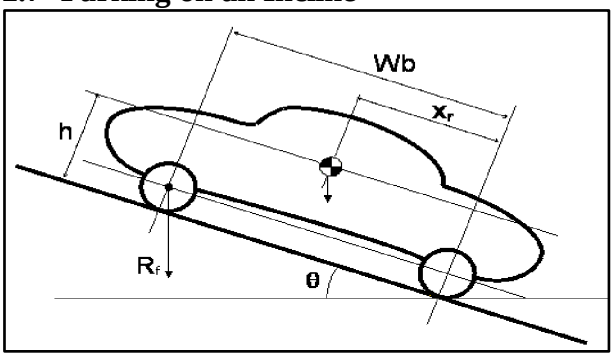

Fig 4: Vehicle Parked on an Incline(ɵ=45˚)

2.7.1 Axle Loads

$R_f =M(x – h.S)/W_b$

where,

R_f = Front axle load (Kg)

x = Horizontal distance from C.G. to rear axle on the level (m)

$R_f =240*(0.8682 – 0.434*1)/0.9652 = 104.0743287$ Kg

2.7.2 Traction Force

$T_{fr} = M.g.S/ (1+S^2)^{1/2}$

where,

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T_{fr} = Traction force required (N)

$T_{fr} = 240*9.81*1/ (1+1^2)^{1/2} = 1664.812206$ N

3. FEM USING ANSYS 14.0

ANSYS is one of the useful software for design analysis in mechanical engineering. This software utilizes the FEM-Finite Element Method to simulate the working conditions of your designs to predict its behavior.

ANSYS makes it possible to optimize a design effectively.

Following steps were taken to analyze the disc brake:

1) CAD Modeling of Disc Brake (Building of Prototype).
2) Thermal analysis of Disc Brake and coupling it with Structural Module (Analyzing the Prototype).
3) Interpreting the results obtained and using them to optimize the prototype.

3.1 CAD Modeling of Disc Brake

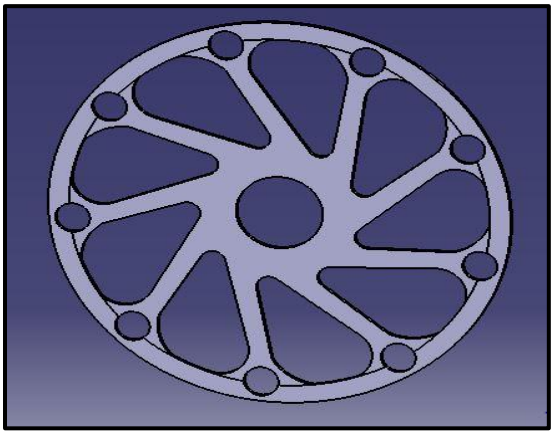

Fig 5: CAD Model of Disc Brake (Final Prototype)

3.2 Thermal Analysis of Disc Brake in ANSYS

Table 1. Inputs Required for Thermal Analysis

Material Used	Stainless Steel
Ambient air Temperature	29˚C
Input Heat Flux	33918 W/m^2
Process of flow	Convection
Type of Support	Fixed(at mounting)

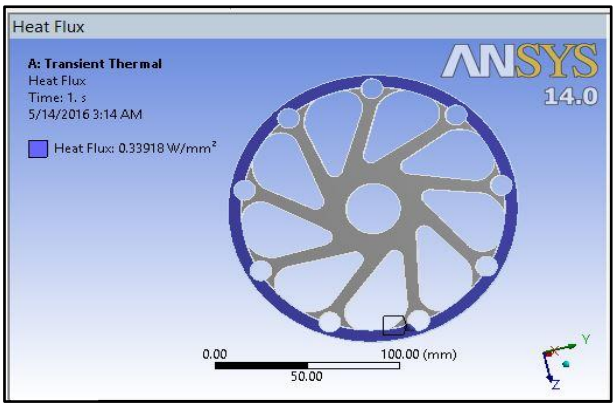

Fig 6: Total Input Heat Flux (perdisc)

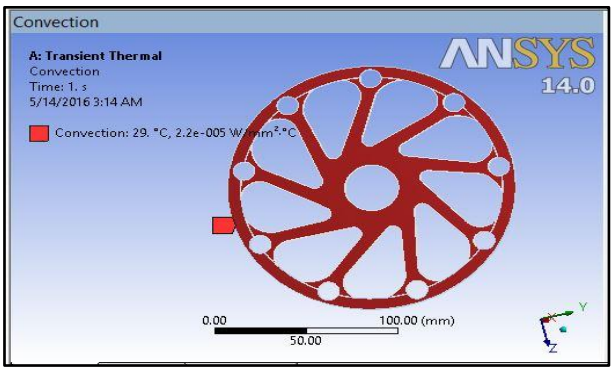

Fig 7: Convection process taking place at disc

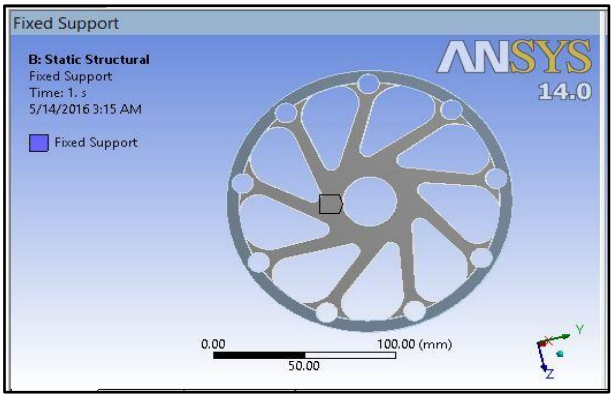

Fig 8: Fixed Support provided at mounting

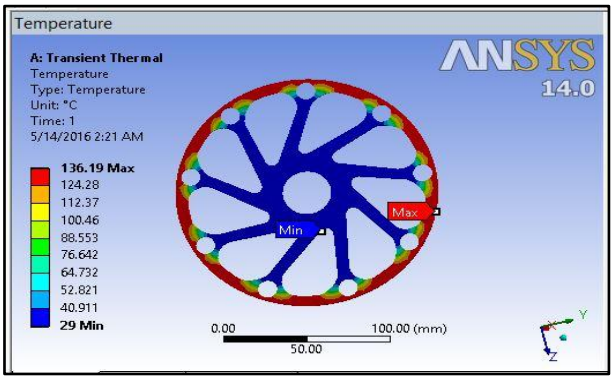

Fig 9: Temperature Distribution

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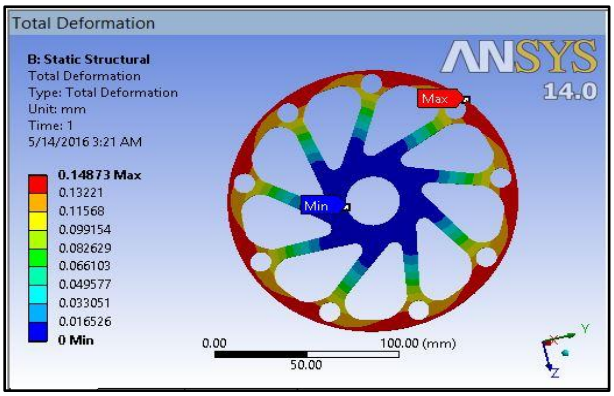

Fig 10: Total Deformation of Disc Brake

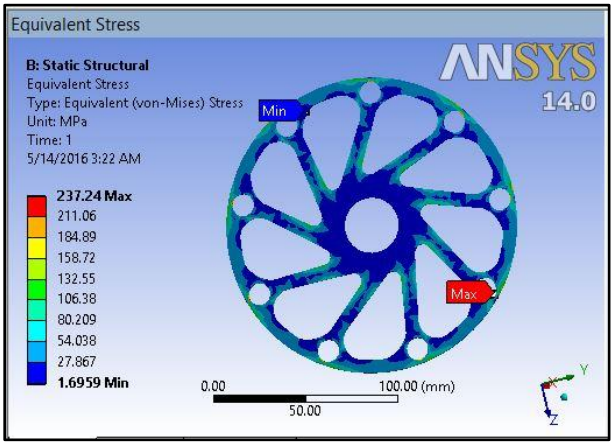

Fig 11: Von-Mises Stress Distribution of Disc Brake

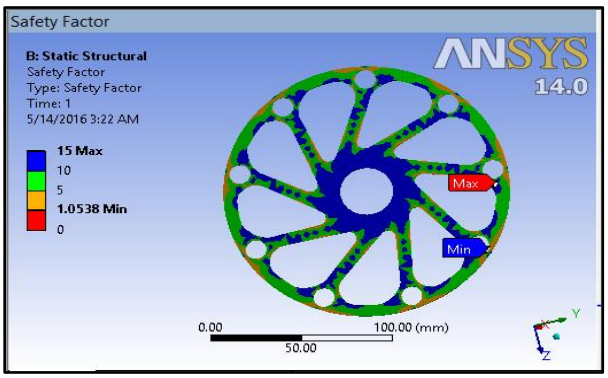

Fig 12: Factor of safety of Disc Brake

4. CONCLUSIONS

After thorough study and market survey Stainless Steel was preferred for disc material. Using this material required brake forces and stopping distance were calculated. The standard disc brake model for 3-wheeled tadpole hybrid was analyzed in ANSYS and thermal-structural (coupling) analysis was done to calculate temperature distribution, total deformation and Equivalent (von-Mises) Stress. It was determined that minimum FOS determined was 1.0538 and in majority of disc FOS ranged from 5-15 as seen in diagram.

5. ACKOWLEDGEMENT

We sincerely thank Mr. Sandeep Sharma, Technical Head, Aerosphere who has always motivated us to carry this research work. His proper guidance led to the completion of this research. We would also like to thanks our friend for his continuous advice and being our motivation. We would also like to thank our parents for their never-ending support.

6. REFERENCES

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