Design Analysis and Calculations of an Optimized Braking System.

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Abstract - *This journal focuses on the analysis, design and calculations of the braking system used in All Terrain Vehicles (ATV) and Go-kart vehicles, all over the globe. The following paper focuses on the design theory and analysis of the braking system of a 180 kg vehicle moving with a speed of 60kmph.The primary function of the braking system is to stop the vehicles motion by converting its kinetic energy to heat energy within the minimum stopping distance. The braking system is an important factor which affects the speed and the weight of the vehicle, therefore it should be carefully and thoroughly analyzed. The outcome of this project is an optimized lightweight braking system providing the adequate braking torque for an efficient braking system. Hydraulic disc brake system is used in this project. All the calculation performed, are based on the Industrial Standard Parameters and the design, is verified using NASTRAN Solver and Simulated in Fusion 360.*

Key Words: Braking system, All Terrain Vehicles, Go-kart vehicles, Stopping distance, Braking torque, Hydraulic disc brake, NASTRAN.

1. INTRODUCTION

The braking system of a vehicle is a system which stops the vehicle or reduces the speed of the vehicle by applying frictional force on it. The system absorbs the kinetic energy of the vehicle, converts it into heat energy and allows the vehicle to come to halt. The heat is transferred to the surrounding by the means of conduction, convection and radiation. The system consists of a rotational part connected to wheel (rotor) and a fixed part (caliper) between which the frictional force is produced. During braking, the force is applied on a cylinder, called the master cylinder which contains brake fluid and is connected to the caliper by using brake-lines. The force applied on the master cylinder is transmitted to the caliper and clamps the rotor. The force is transmitted through the brake fluid by Pascal's law. As the required amount of force at caliper to stop the vehicle is greater than that compared to the applied force by the

Driver, so the first step, force reduction is done by using a large diameter of piston at caliper than that of the diameter of master cylinder. Also the Brake pedal is used to increase the applied pedal force on master cylinder 2-4 times using the Lever Rule.

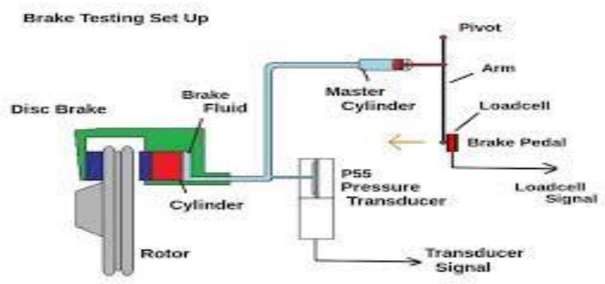


Fig 1 - Diagram of a Typical braking System

1.1 Classification

Brakes are classified into various types depending on their Application, Construction and Actuation. Depending on the Application: i) Service Brakes ii) Parking brakes. Depending on the Construction :i) Drum brakes ii) Disc Brakes. Depending on the Actuation: i) Mechanical Brakes ii) Hydraulic Brakes iii) Vacuum Brakes iv) Electric Brakes.

Most of the commercial vehicles use disc brakes instead of drum brakes depending on certain advantages discussed below. Most of the vehicles use hydraulic braking system while some light vehicles such as bicycles use brake wires. Depending on the position of rotor, the braking system is of two types - Inboard Braking System and Outboard braking system. In an Inboard braking system the rotor (brake disc) is mounted on the brake shaft of the vehicle where as in an Outboard braking system the brake disc is mounted on the wheel hub. Using an inboard brake disc has an advantage of using a larger brake disc with compact packaging of a lower sprung mass. The brake line connection can be classified as X split and FR split, depending on the pressure distribution. To distribute the applied pressure in the brake lines, the balance bars are used.

1.2 Different Parts of a Braking System

Rotors are the crucial part of a braking system. Instead of drum brakes, now most of the vehicles use disc brakes as rotors in their braking system. Disc brakes have advantages over drum brakes as they have fewer moving parts, easy to maintain, low cost and are cheap to manufacture. Certain varieties of brake discs are available in the market such as solid discs. Disc brakes usually have some slots to remove the foreign materials, dusts and helps to dissipate the heat effectively. For heavier vehicles vented brake discs are preferred as they have comparatively larger surface area and an efficient heat dissipation structure. In vehicles such as

ATV's and go-karts solid brake discs are used as they are comparatively lighter and consume low space while providing an effective braking system.

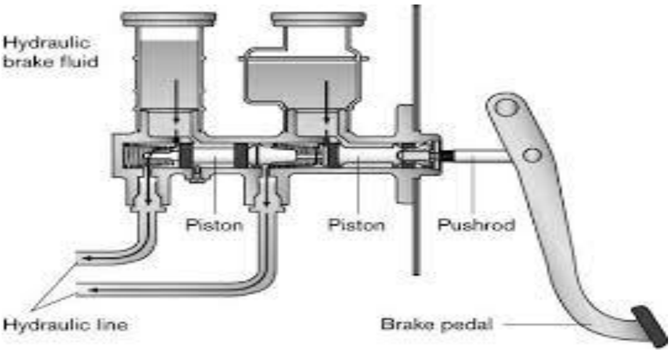


Fig 2 - Diagram of a Tandem Master Cylinder Assembly

Three types of Master Cylinders are used in a braking system – Single master cylinder, Double master cylinder and Tandem master cylinder. They have their advantages in different conditions. Tandem master cylinders are used in an X split system as they require two pistons to perform and can provide two different pressure using a single push-rod.

In a braking system, Calipers are used as the static part which cause the friction to stop the rotor resulting in the stopping of the vehicle. They clamp the rotor between their brake pads and generates required braking forces to stop the rotor. Usually calipers are of two types - Fixed caliper and Floating caliper, depending on the condition whether the pads are fixed or moving. Depending on the no of pistons - single piston caliper and double piston calipers are available in the market.

Brake Fluids: To operate the braking system effectively certain braking fluids (glycols) are used like DOT 3, DOT 4, and DOT 5. These brake fluids are of high boiling point and excellent corrosion resistance to braking systems. The Brake Fluids transfer the kinetic force into pressure, and to efficiently intensify the braking force.

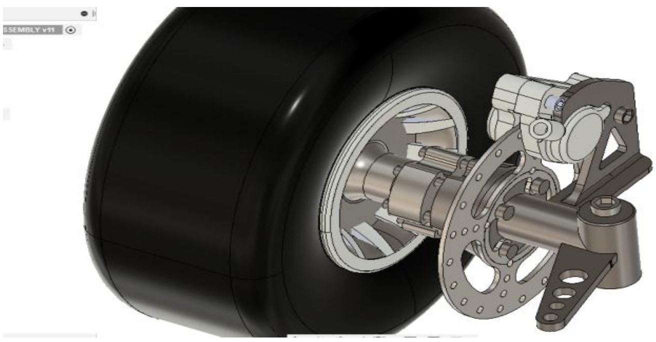


Fig 3 - Diagram of a Front Brake Assembly

2. DESIGN CONSIDERATION

During braking the front load of the vehicle increases and the rear load decreases due to dynamic brake transfer. So it takes more braking force on front than that of the rear to stop the vehicle. We have considered the weight of the vehicle to be 180 kg with the driver and the vehicle is moving with at a maximum speed of 60 kmph. When suddenly the driver applies brakes and all the four wheels are locked simultaneously, which then results in the stopping of the vehicle. The tyre of the vehicle is considered to be a slack type tyre for which the co efficient of friction for tyre road interface was assumed to be 0.85. The dimension of the vehicle for brake calculation are as followed:

Wheel Base	1143 mm
Track Width	969.772 mm
Height of cg	190 mm
Vehicle weight (assumed)	180 kg
Vehicle Speed (assumed)	60kmph

Table 1 – Dimensions of the vehicle for Braking Calculations

2.1 ROLLING RESISTANCE:

Rolling resistance is the resistance provided by the tyre due to deformation. While the vehicle is in a loading condition the loads are concentrated on the front and the rear wheels which are in contact with the road surface. As the wheels of the tyres are composed of rubber, during the rotation of the wheels, hysteresis loss occurs, which results in an opposing force. The ratio of rolling resistance to the load on wheel is called co-efficient of rolling resistance. For our vehicle the coefficient of rolling resistance is calculated as R = C/r =0.1216(r= radius of wheel). As tyre-road surface contact area increases the rolling resistance also increases. In case of suing slack tyres in go-karts, formula cars the rolling resistance is considered but in case of ATV's the value of rolling resistance is negligible.

3. CALCULATIONS

All the calculations are performed considering initial velocity of 60 kmph and final velocity of 0 kmph. During braking, there is a weight transfer due to which load on the front axle increases. By taking the moments about the Rear and Front wheels, we get the following equations of dynamic load transfer

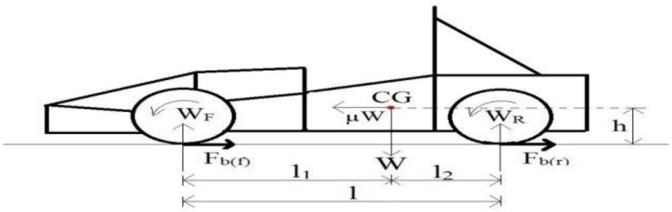


Fig 4 – Diagram of a go-kart vehicle

For static condition ,$W_{front} = W (L_2/L)$ $W_{rear} = W (L_1/L)$

For dynamic condition, $W_{front} = W (L_2/L) + (h/L) F_b$ $W_{rear} = W (L_1/L) - (h/L) F_b$ Where transferred load = $(h/L) F_b$;

Maximum braking force provided between the road and tyre is given by $F_b \max = \mu W = W (d/g)$

$F_b(f) = W_f(d/g) = (d/g)[W(L_2/L)+(h/L)\mu W]$ $F_b(r) = W_r (d/g) = (d/g)[W(L_2/L)+(h/L)\mu W]$

Where, W= Weight of the vehicle F_b = Braking Force (total); The following equations are for an ideal condition where the braking efficiency is 100%.The Coefficient of rolling resistance is $(f_r) = 0.1216$

Considering the rear wheel drive, the deceleration during braking is found to be: Maximum deceleration, d=7.848 m/s^2 . Considering the values as per the design of our team's go-kart vehicle: l= 1.143m; l_1= 0.6286 m; l_2= 0.5143 m; h= 0.19 m; W= 180*9.81= 1764 N; μ=0.85

From these values, the dynamic load transfer is found to be = 246.92 N. Hence braking forces at axles are:

$F_b (f) = W_f (d/g) = (d/g) [W (L_2/L) + (h/L) \mu W] = 832.5$ N

$F_b(r) = W_r (d/g) = (d/g) [W (L_2/L) + (h/L) \mu W] = 578.7$ N

$F_b \max = \mu W = W (d/g) = 1411.2$ N Percentage biases on the front and the rear wheel respectively are: $K_b (f) = F_b (f)/F_b = 0.58$; (ratio of the braking force at front to the total braking force) $K_b(r) = F_b(r)/F_b = 0.41$; (ratio of the braking force at rear to the total braking force) Braking torques at the front and the rear axles are respectively are: $T_f = (F_b(f) * R_{wheel})/2 = 52.86$ Nm $T_r = (F_b(r) * R_{wheel})/2 = 40.509$ Nm Where, R_{wheel} = 0.127m (front) =0.14m (rear)

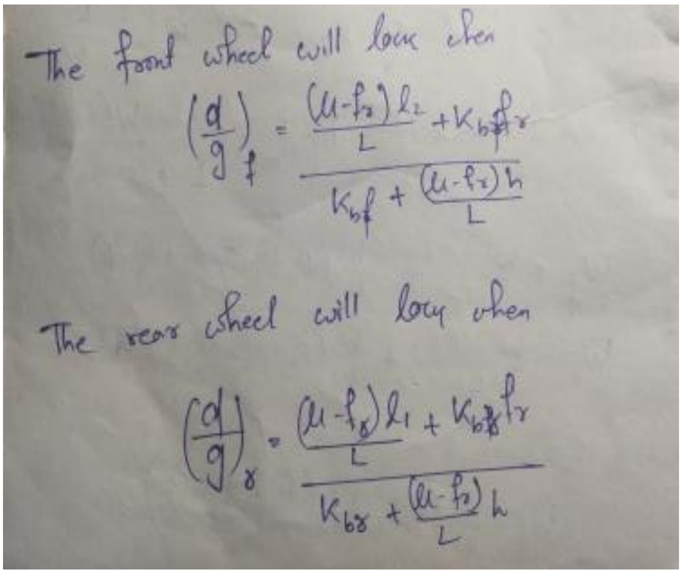


Fig 5 - The d/g ratio for front wheel and rear wheel

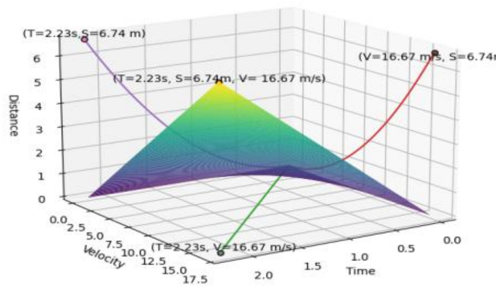
Friction Coefficient	0.85	K_b	Front – 0.58 Rear – 0.42
Rolling Resistance	0.1216	Deceleration	7.84m/s^2
Braking Force	Front – 832.5N Rear – 578.7N	Stopping Distance	6.723 m
Applied Torque	Front -34.35 N Rear – 49.07N	Stopping Time	2.23 s

Table 2 – Parameters and Values of the Calculations

3.1. GRAPHS:

1. Velocity~Time~Distance Graph :

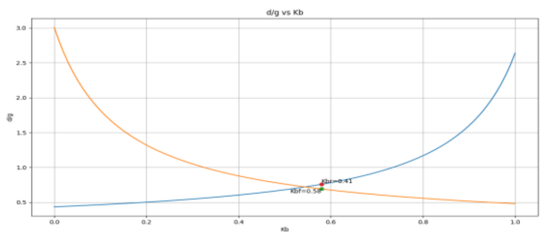
This graph shows the variation of the maximum Braking Distance and Time with respect to the Velocity at which brakes are applied. For the maximum velocity of the vehicle 16.67 m/s, the maximum stopping distance and maximum braking time is found out to be 6.74 m and 2.23 sec respectively.



Graph 1 – Velocity ~ Time ~ Distance

2. D/g vs. Kb Graph :

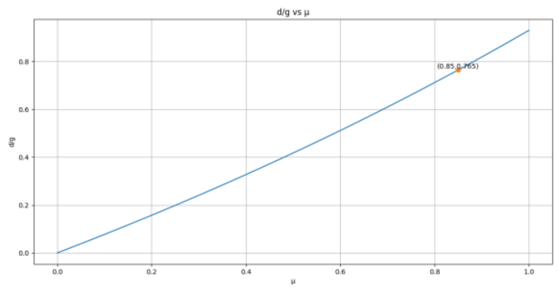
This graph shows the relation between the deceleration of the vehicle and the brake force distribution on the front and rear wheels. Here Kb_f is the ratio of brake force at front to the total braking force for effective braking. And Kb_r shows the ratio of the rear braking force to the total braking force.



Graph 2 - D/g vs. Kb

3. D/g vs. μp Graph :

This graph shows the variation of deceleration of the vehicle with respect to the effective coefficient of friction (μ).The optimum ratio of D/g for our vehicle is found to be 0.765 considering the value of the effective coefficient of friction to be 0.85.



Graph 3 - D/g vs. μp

3.2. PART SELECTION PROCEDURE:

Taking into account, that the Driver applies a pedal force FP = 200N, on the brake pedal having a leverage 2.3,

Force on the push rod of the master cylinder = 2.3*200 = 460N As per market availability and requirement, a Maruti 800 Master Cylinder has been chosen having the master cylinder diameter = 19.05mm. Pressure in master cylinder =force on MC/area of MC piston =1.614 MPa Due to the dynamic load transfer, required braking force increases on the front axle so based on market availability we have decided to use 2 solid brake discs on the front wheel and 1 rear inboard solid brake disc on the rear axle.

So required braking Torque on Front wheel =Torque on front axle/2 = 52.86/2 = 26.43 Nm Braking Torque on Rear wheel = Torque on front axle/1= 40.5 Nm

A dual piston fixed caliper of diameter 25.4 mm is chosen due to its light weight, simple design and better maintainability. So Force at caliper = Pressure in master cylinder * Area of piston Caliper *No. of pistons = 1.614MPa* 0.000506 * 1 = 817.78 N Clamping Force = 2 * force on caliper = 1635.56N

Let the coefficient of friction between the disc and the pad (μ) in the brake caliper be 0.3. Choosing Stainless Steel 410 brake disc of 160mm diameter for front and 220 mm diameter for rear results in Effective radius, $R_{eff} = (r_1+r_2)/2$; where R_1= Radius of circle formed by inner edge of friction pad R_2= Radius of circle formed by outer edge of friction pad.

So R_{eff} (front) = 0.07m; R_{eff} (rear) = 0.10m Applied Braking force, F (friction) = 2 * μ *$F_{calliper}$ = 2 * 0.3 * 817.78 N= 490.668 N Applied Braking Torque,T(front)=F(friction)*R_{eff}(front)= 490.668*0.07m = 34.35 Nm (>26.43 Nm i.e. the required front wheel torque) T (rear) =F(friction)*R_{eff}(rear)=490.668*0.10m = 49.07 Nm (>40.51 Nm i.e. the required rear wheel torque)

From the above calculation, we see the applied braking torque is more than the required braking torque for both front axle and rear axle. Hence a pedal force of 200 N would provide sufficient braking torque to lock the wheel.

Master Cylinder	BOSCH 19.05 mm
Brake Caliper	Apache RTR 200
Brake Fluid	DOT 3
Brake Line	Solid Brake Line
Disc Material	Stainless Steel 410
Front Disc Diameter	150 mm
Rear Disc Diameter	210 mm
Brake Pedal Area	2.66 in^2

Table 3 – Part Specification Table

3.3. BRAKING PARAMETERS:

For the front wheel to be locked, the deceleration value must be greater than or equal to 6.82 m/s^2. For the rear wheel to be locked the value of d must be 7.42 m/s^2 .But the maximum deceleration is 7.84 m/s^2 .So during braking the front wheel locks first. Braking Efficiency: η = min (d/g front, d/g rear) / (μ +f_r) Braking efficiency is = 87% From the calculation, Maximum Deceleration of the Vehicle, d_{max} = 7.848 m/s^2 .Stopping Distance, S = 6.724 m; Maximum braking time, t =2.23 Sec

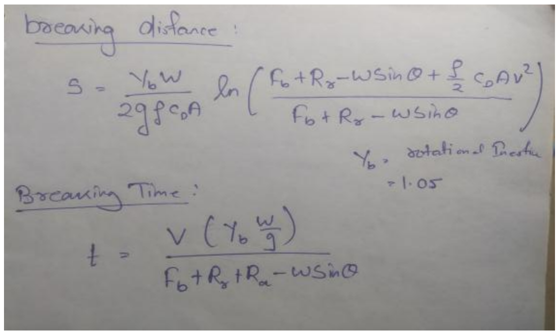


Fig 6 – equations for calculating braking distance & braking time

3.4. THERMAL CALCULATION:

During braking the total kinetic energy is converted into thermal energy. So the Transmissional Kinetic Energy is, For Front axle, KE = 0.5*(Wfront /g)*v*v = 14829.64 J. For Rear axle, KE = 0.5*(Wrear /g)*v*v = 10308.6 J.

As per some industrial standards, The Rotational KE is found nearly equal to 3% of Transmissional KE. So Rotational Kinetic Energy is,

For Front axle, KE = 448.88 J, For Rear axle, KE = 309.25 J.

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Total KE for Front Axle =15274.52 J Total KE for the front wheel, KE_{front} = 7367.26 J

Total KE for Rear Axle =10617.85 J Total KE for rear wheel, KE_{rear} = 5308.925 J

Braking Power of Tyre, P_{front} = KE/t = 3303.7 W P_{rear} = 2380.68 W

Heat Flux in to one side of the disc, Q = P / area = (4*P) / 3.141*(D+d)*(D-d); where D=outer diameter of the disc; d =minimum diameter of disc in contact. For Front Q_f = 519406.81 W/m^2 For Rear, Qr = 259049.57 W/m^2

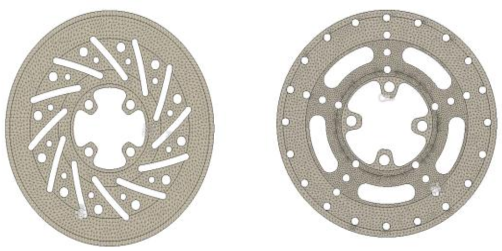
So Single Stop Temperature rise of the disc, ΔT =0.527*Q*{t / (ρ*C*K)}$^{1/2}$ For Stainless Steel 410, ρ = density of disc material = 7.8 g/cc C = Specific heat capacity = 460 J/kg-K

K = Thermal conductivity = 24.9 W/m-K ΔT_{front} = 43.245 K; ΔT_{rear} = 21.57 K

Hence after a single stop, the final temperature of the brake disc for front and rear is 70.245 degree and 48.57 Celsius respectively.

3.5. STRUCTURAL AND THERMAL ANALYSIS:

After the calculation phase, different designs for brake disc and pedals are designed and analyzed using CAD and CAE softwares like SOLIDWORKS and AUTODESK FUSION 360. To be used as a rotor, the brake must have high strength, high corrosion resistance and easily machinable. To achieve this purpose 2 different material was chosen and analyzed with 2 different designs. After analysis, the design was chose for the manufacturing and further prepared to be used.



Model 1 Model 2

These two models are designed in SOLIDWORKS and for the analysis of these designs, they are meshed with meshing size of 2mm and tetrahedron element type. The material for brake disc were chosen as Stainless Steel 410 and gray cast iron as the disc material and different values were analyzed for choosing the proper manufacturing material.

For the thermal analysis the portion of brake disc which was used to be clamped by the caliper was given a maximum temperature of 140.5 °C (which is 2 times of the calculated maximum single stop temperature) and the mounting point of brake disc was given the temperature of 25 °C. Radiation and convection was applied to the other faces of brake disc with an emissivity of 0.43 and convection coefficient of 24.9 W/ (m^{2}K). For the structural analysis of the brake disc, the mounting part of the disc was kept fixed and a moment of 40 Nm was applied to the clamped faces. Then maximum stress and the Factor of Safety were compared based on analysis results.

Material	Stainless Steel 410	Gray Cast Iron
Max Temp	140.5 °C	140.5 °C
Heat Flux	8.3 W/cm^2	7.4 W/cm^2
Max Stress	309 MPa	225.7 MPa
FOS	3.96	3.35

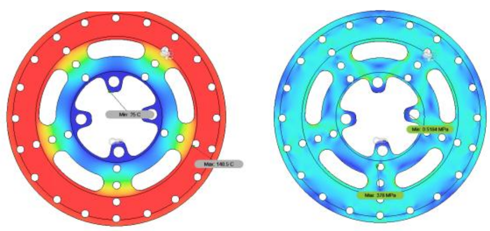
Table 4 - For Model 1 Design

Temperature Stress

Fig 7 – Equivalent Temperature & Stress for Model 1

Material	Stainless Steel 410	Gray Cast Iron
Max Temp	140.5 °C	140.5 °C
Heat Flux	5.9 W/cm^2	4.5 W/cm^2
Max Stress	378 MPa	272 MPa
FOS	3.21	2.73

Table 5 - For Model 2 Design



Temperature Stress

Fig 8 – Equivalent Temperature & Stress for Model 2

From the above analysis results we have to choose a design which must have high factor of safety and a high heat flux. Comparing the values Model 1 is selected and to be used as our brake disc which is Stainless Steel 410.

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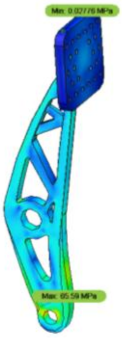

Fig 9 – Brake Pedal Stress

Considering the driver can apply a pedal force of 200 N on a brake pedal, brake pedal is designed of having a pedal leverage of 2.3 and material of Aluminum 7065. The designed model was analyzed under 200 N force and the resulted FOS is 3.57 with a max stress of 65 MPa.

4. CONCLUSIONS

The design calculations are done to find the force being applied on the brake discs, the maximum stress that can be endured during the braking system. We have used Solidworks and Autodesk Fusion 360 for designing the various parts of the braking system. We have used NASTRAN solver for determining the thermal stress and the temperature. From our above calculations, we have selected Model 1, for our vehicle. The model is of Stainless Steel 410 and it can endure a maximum stress of 309 Mpa with a fos of 3.96.

REFERENCES

[1] Rudolf Limpert, Brake Design and Safety , Society of Automotive Engineers,2011

[2] Thomas D. Gillespie, Fundamentals of Vehicle Dynamics

[3] Milliken and Milliken , Race Car Vehicle Dynamics

[4] J. Wong ,Theory of Ground Vehicles, Wiley, 1993

[5] Alternate Material in Automobile Brake Disc, y Telang A.K.

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