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A Review on Design of Hydraulic Disc Brakes and Calculations

Engineering

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ABSTRACT

The calculation and verification of braking force is a crucial step in the design process of an automobile as the braking system directly factors as a good control and safety feature in the product. While designing, the main objective is to generate more braking force than ideally required to account for inefficiencies in mechanical linkages and hydraulic systems. This paper reviews the basic principles and considerations in the braking system design process and further explains the general procedure for the same. The paper also explains a validation method for any designed braking system using a brake performance triangle (BPT).

INTRODUCTION
The main objective of any braking system is to attain a reduction in speed of a vehicle to facilitate speed control and maneuverability keeping in mind various constraints like cost, stopping distance, etc. Hence accurate calculation of braking force becomes a crucial design concern. Like every design, braking force calculations also start with certain assumed parameters which are later corroborated or modified according to the results of these calculations.

PROBLEM STATEMENT
As the most critical application of braking is during sudden stopping of a vehicle, the condition of decelerating from a reference speed to zero speed is used to calculate braking force. Consider a vehicle moving at an initial speed of u m/s and time in which the vehicle should be stopped is t seconds. Let the braking acceleration or deceleration is a_b m/s^2 and mass of the vehicle is m kg. For calculating a_b, kinematic equations are used.

From the first kinematic equation,

$$0 = u - a_b \times t \qquad (1)$$

$$\therefore a_b = \frac{u}{t} \qquad (2)$$

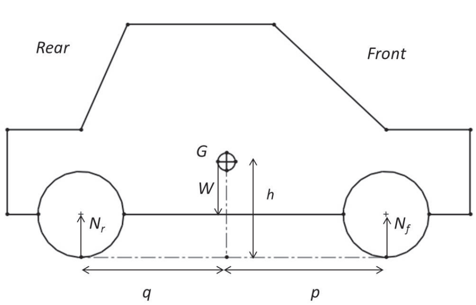
In equation (2), the time t is exclusive of driver's reaction time as well as system response time[1]. Also, from these calculations and second kinematic equation we can find out the stopping distance s m (3), (4).

$$u^2 = 2 \cdot a_b \cdot s \qquad (3)$$

$$\therefore s = \frac{u^2}{2 \cdot a_b} \qquad (4)$$

REQUIRED BRAKING FORCE

Figure 1: Force Body Diagram (FBD) of a vehicle in static condition



In figure 1, the force body diagram of a stationary vehicle is shown. The normal reactions by the ground on the vehicle at the tires at the front and rear are *Nf* and *Nr* respectively. The position of center of gravity is at a height of *h* denoted by *G*. Also, *p* and *q* denote the horizontal distance of *G* from the front and rear tire contact points respectively. The weight of the car is *W* N.

From the Newton's third law of motion,

$$N_f = \frac{W \cdot q}{p+q} \qquad (5)$$

$$N_r = \frac{W \cdot p}{p+q} \qquad (6)$$

$$x_1 = \frac{N_f}{N_f + N_r} \times 100 \qquad (7)$$

The static normal reaction distribution on the car on front two wheels with respect to rear two wheels is x_1, is expressed as a percentage in equation (7).

Now, consider that the vehicle is decelerating by a_b m/s^2. In this case the car is acted upon by frictional force. The force body diagram is shown in the Figure 2.

The normal reactions at the tires are different from the initial static condition. The normal reactions in this case are N_f' and N_r' calculating using equations (8) and (9). The frictional forces at the front and rear tires are *Bf* and *Br* respectively. These values are given by equations (10), (11).

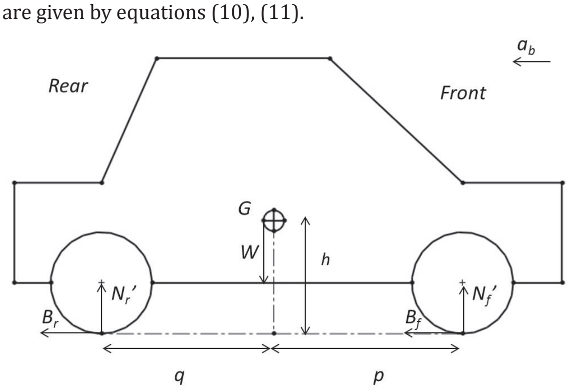


Figure 2 Force Body Diagram (FBD) of a vehicle in dynamic condition

$$N_f{}' = \frac{W \cdot q}{p+q} + \frac{m \cdot a_b \cdot h}{p+q} \quad (8)$$

$$N_r{}' = \frac{W \cdot p}{p+q} - \frac{m \cdot a_b \cdot h}{p+q} \quad (9)$$

$$B_f = \mu_t \cdot N_f{}' \quad (10)$$

$$B_r = \mu_t \cdot N_r{}' \quad (11)$$

In equations (8) to (11), μ_t is the co-efficient of friction between tires and the road surface. Thus, B_f and B_r are the required braking forces at the front and rear tires respectively. The generated braking force must be more than the required force to account for inefficiencies in mechanical linkages and hydraulic systems.

BRAKING ASSEMBLY AND APPLIED BRAKING FORCE
In the dynamic state, the required braking forces on front and rear tires may or may not be equal. Hence some method for biasing braking forces needs to be introduced in the design. The most preferred methods for the same are:

1) Pressure Distribution Valve
2) Different Brake Assemblies at the front and rear wheels.

A combination of both the methods may also be used successfully to achieve the most accurate results. We shall limit our discussion to the second method owing to its popularity and ease of design. In this paper, the design will involve biasing by distinct set of assemblies for front and rear parts. The following notations will be used in the further part of the paper.

d_{mc} =Master Cylinder Diameter (m)

r =Pedal Ratio

d_{cp} =Caliper Piston Diameter (m)

d_{rf} =Front Rotor Diameter (m)

d_{rr} =Rear Rotor Diameter (m)

μ_{br} =Friction Coefficient between Brake Pad and Rotor

d_{tf} =Front Tire Diameter (m)

d_{tr} =Rear Tire Diameter (m)

F =Applied Force (N)

B_{fa} =Applied Braking Force on Front Two Tire (N)

B_{ra} =Applied Braking Force on Rear Two Tires (N)

The mechanical force applied by the driver is amplified by the pedal arrangement. The final force which acts on the master cylinder is given by equation (12).

$$F_{mc} = F \times r \quad (12)$$

This force causes a fluid pressure to be exerted on the piston cylinder arrangement, given by

$$p_{mc} = \frac{F_{mc}}{\frac{\pi}{4} \times d_{mc}^{\,2}} \quad (13)$$

Unless a pressure distribution valve is used for biasing, the pressure remains constant throughout the hydraulic circuit according to Pascal's law of constant pressure.

This pressurized fluid applies a force on the caliper piston which causes it to move and transmit the force on the rotor. The force applied by caliper piston and the other side of caliper on two opposite sides of rotor are in exact opposite directions, both forces will cancel out each other and but torques on the rotor will add[2]. This can be understood clearly from Figure 3.

So the force by caliper piston (assuming the caliper has only one piston) on the rotor can be expressed as

$$F_p = p_m \times \left(\frac{\pi}{4} d_p^{\,2} \right) \quad (14)$$

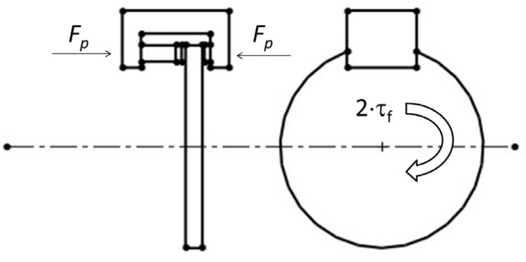


Figure 3: FBD of and torque applied on the rotor

The torques applied by the caliper (two pistons) on the front and rear rotor will be given by equations (15) and (16) respectively[3].

$$\tau_f = 2\mu_b \cdot F_p \cdot \left(\frac{d_f}{2} \right) \quad (15)$$

$$\tau_r = 2\mu_b \cdot F_p \cdot \left(\frac{d_r}{2} \right) \quad (16)$$

So the final braking forces acting on the front two and rear two tires is given by equations (17), (18).

$$B_{fa} = 2 \cdot \tau_f \cdot \left(\frac{d_{tf}}{2} \right) \quad (17)$$

$$B_{ra} = 2 \cdot \tau_r \cdot \left(\frac{d_{tr}}{2} \right) \quad (18)$$

BRAKE PERFORMANCE TRIANGLE

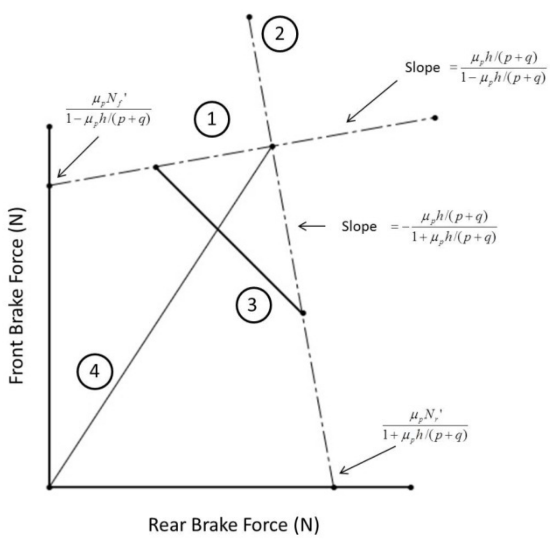


Figure 4: Brake Performance Triangle[4] Graph

This is a test to check the performance for any braking system after its design. In figure 4, the four lines represent

1. Front Lockup Line
2. Rear Lockup Line
3. Constant Deceleration Line
4. Proportionating Line

The area inside the two lockup lines i.e. front/rear lockup lines individually signifies the efficient working of front/rear brakes.

The triangular area enclosed by the two lockup lines and constant deceleration line represents the range where both of the brakes will work for the deceleration below the deceleration line limit.

The proportionating line (line 4) represents ideal braking. If the point representing the designed braking system falls inside the triangle as well as on the proportionating line, the design can be considered to be optimum.

The validity of every braking system can be checked using this triangle and system performance for various values of coefficient of friction, deceleration rate, etc. can be corroborated.

RESULT
The braking force acting on the vehicle depends on specifications of rotor, caliper piston, master cylinder, pedal ratio and co-efficient of friction.

CONCLUSION
For the brakes to work effectively, the applied braking force should be more than required braking force. There are various methods to achieve this, each having distinct advantages.

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