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RESEARCH

Comparative Frictional Analysis of Automobile Drum and Disc Brakes

H.P. Khairnar[a], V.M. Phalle[a], S.S. Mantha[a]

[a] *Mechanical Engineering Department, VJTI, Mumbai, India.*

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Corresponding author:

H.P. Khairnar
Mechanical Engineering
Department, VJTI, Mumbai, India
E-mail: hpkhairnar@vjti.org.in

A B S T R A C T

In the present work, a comparative frictional behaviour of drum brakes and disc brakes in automobiles has been investigated. The influential factors; contact force and friction radius were modeled for the estimation of the friction coefficient for drum as well as disc brakes. The effect of contact force and friction radius is studied with varying conditions of parameters; longitudinal force, caliper force and torque on piston side as well as non-piston side. The numerical results obtained have been compared with the similar obtained from virtual Matlab/Simulink models for drum and disc brakes. The results evidenced that friction radius predominantly affects brake pressure and thus the friction coefficient, also the increase in contact force resulted with decrease in friction coefficient both for drum and disc brakes. Further it has been found that disc brakes exhibit gradual decrease of friction coefficient due to the equitable distribution of braking effort while drum brake presents sudden variations in friction coefficient. It can be revealed that frictional behaviour of disc brake is more consistent than drum brake.

1. INTRODUCTION

Todays technological developments in the vehicle technology seek better control on active safety by addressing the need for better and safer braking systems. The braking system is integral and vital part of the active safety control of vehicles. The selection of drum brakes for rear wheels and disc brakes for front wheels has been the trend in the earlier vehicles. Better heat dissipation achieved by disc brakes is responsible for the wide use of disc braking system in the modern vehicles. The available brake torque at the interface of the drum - shoe in drum brakes and on the other hand between pads – rotor disc in the disc brake is responsible for the deceleration of the vehicle. Brake torque is primarily the function of friction coefficient (μ).It is incumbent to compare the performance of the drum and disc brakes considering friction coefficient to optimize the selection of the braking system to achieve the desired active safety control. The functional dependence of the friction coefficient upon a large variety of parameters; including sliding speed, acceleration, critical sliding distance, temperature, normal load, humidity, surface preparation and of course material combination [1]. The variations in the friction coefficient are highly dependent on the frictional materials and braking conditions [2]. The manufacturers of brake systems demand number of requirements concerning the friction materials [3] like i) mean friction coefficient should approach 0.45; (ii)$\Delta\mu$ i.e. the variation of μ during brake stop should approach zero; (iii) braking noise and wheel dust should be avoided, these are difficult to comply with. From the perspective of the emergency braking, factors like drum-shoe/disc-pad friction and usable braking torque are difficult to determine precisely due to variations in the operating conditions. This supports the need for efficient methods of calculating friction coefficient for drum and disc brake systems. During braking in automobiles there is a dynamic transfer of weight onto the front wheels of a vehicle with a corresponding decrease at the rear wheels. Consequently there is a redistribution of usable braking torque at the front and rear brakes in a braking maneuver [4].Hence, the total steering loss can occur when the dynamic friction coefficient is not as per the requirement of vehicle dynamics, in quest of this a significant attention has been paid by the researchers and brake designers. A mathematical model [5] was formulated of the factors influencing the brake force and the results showed that brake force and adhesion coefficients are interdependent. The normal load, sliding speed, ambient conditions and material [6-8] were considered to obtain lower friction coefficient measuring the friction and pull-off forces between a metal pin and plate. However efforts have been made to establish the influence of sliding acceleration to improve the friction coefficient prediction during transient operations. It led to the conclusion that higher the sliding acceleration, higher is the friction coefficient. Most of the materials have exhibited nearly linear dependence of the friction coefficient on the contact pressure in the studied ranges [9]. The braking systems produces a Coulomb friction force which is a result of the contact forces at the contact of the leading, trailing shoes and the brake drum and between disc-pad in the disc brakes This Coulomb friction force is responsible for the brake torque used to decelerate the vehicle and hence it is pertinent to consider its impact. The caliper disc brakes require large actuating force than for drum brakes because they have neither friction moment nor servo action to aid the brake application. Circular pads are generally used in hydraulically actuated caliper brakes whenever the hydraulic pressure is to be increased relatively cheaply since pads themselves are supported entirely by the piston face. Hence circular pads are therefore cheaper to manufacture because no additional supporting structure is required to be created.

Although extensive research has been carried out for brakes using contact analysis [10 and 20], heat dissipation [21], and different material compositions for the frictional surfaces. But the efficient methodology to compute the usable braking torque which is the function of friction coefficient at the interface of the friction surfaces in the drum and disc brakes has not been discussed in the available literature.

The present study intends to estimate and compare the performance of the automobile drum and disc brakes system considering friction coefficient. The multitude of equations was derived for friction coefficient under the equilibrium condition using principles of classical mechanics with friction. Effects of the influential factors such as contact force and friction radius on the friction coefficient were evaluated. The relationship between the above listed factors and friction coefficient has been modeled and the obtained results were compared with virtual Simulink model for drum and disc braking systems

2. ESTIMATION OF FRICTION COEFFICIENT IN DRUM BRAKES

The friction coefficient (μ) at the interface of the drum-shoe was defined as the function of the longitudinal forces existing while the brake is operational.The distribution of the Coulomb friction force is responsible for the variations in the friction coefficient. Hence the computation of the friction coefficient is done for the symmetric and asymmetric shoe length.

2.1 Modeling with normal contact forces

During braking operation the actuation of shoes results due to contact forces F_l and F_t for the leading and trailing shoes respectively as shown in Fig. 1,[11]. The contact forces F_l and F_t are normal to the contact surfaces when contact lining is symmetrical [12]. The prevailing force arrangement leads to maximum friction coefficient due to nonexistence of components of forces. While deriving the equations, F_l and F_t were considered at the distance of the inner drum radius, r. However the Coulomb friction forces were also considered at the inner drum radius, r. The reactions at hinged point, O, of the brake shoes were considered.

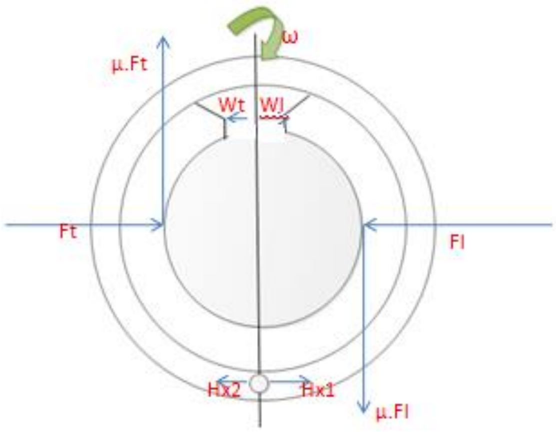


Fig. 1. Equilibrium diagram for the drum brake when contact force is normal.

All the forces resolved considering the equilibrium of both brake shoes. Also the values of actuating forces W_t, W_l and W_t were assumed as equal (W_l = W_t). Hinge reactions (H_{X1}, H_{X2}) for the shoes were considered only for the 'X' direction and not in Y direction due to the direction of the actuating forces W_l and W_t. However the movement of the vehicle was considered in the longitudinal direction only on the ideally horizontal road. The effects of aerodynamic forces are neglected in the present analysis. The Cartesian coordinate system is ox-oy-oz, x axis is along the direction of the vehicle movement, y axis is along the lateral movement of the vehicle and z axis is along the wheel rotation as shown in Fig. 1. The following equations represent the equilibrium of force system as shown in Fig. 1.

$$F_t - F_l + W_l - W_t + H_{X1} - H_{X2} = 0 \quad (1)$$

$$\mu.F_l - \mu.F_t + mg = 0 \quad (2)$$

Solving both the Eq. (1) and (2) leads to Eq.(3) giving the cumulative effect since the Coulomb friction force is acting vertical , the friction coefficient can be estimated using the following equation:

$$\mu = \frac{F_t - F_l + Hx_1 - Hx_2 - mg}{F_l - F_t} \quad (3)$$

The leading shoe is subjected to dynamic weight transfer resulting in reaction from the contact forces; however moments of contact forces considered at the hinge represent the following equation:

$$F_l.h - W_l.r = \mu.F_l.h \quad (4)$$

$$\mu = \frac{F_l.h - W_l.r}{F_l.h} \quad (5)$$

For balancing the forces F_l, W_t and $\mu.F_l$ the moments of trailing shoe at the hinge gives the Eq. (5) establishing the effect of F_t:

$$\mu.F_t.h = W_t.r - F_t.h \quad (6)$$

$$\mu = \frac{W_t.r - F_t.h}{F_t.h} \quad (7)$$

2.2 Modeling with inclined contact forces

The methodology for the estimation of friction coefficient (μ) considering variations in F_l, F_t, at the contact point of the lining-drum is presented below. The friction coefficient is defined through forces involved in the braking process. The Fig. 2 shows the arrangement of longitudinal forces F_l, F_t in inclined position at the lining - drum contact. The resolution of the forces is shown in the Fig. 3, considering components of F_l, F_t, and Coulomb friction force (μF_l, μF_t).

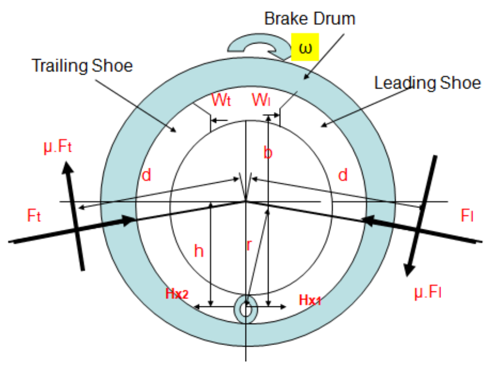


Fig. 2. Equillibrium diagram for the drum brake when contact force is inclined.

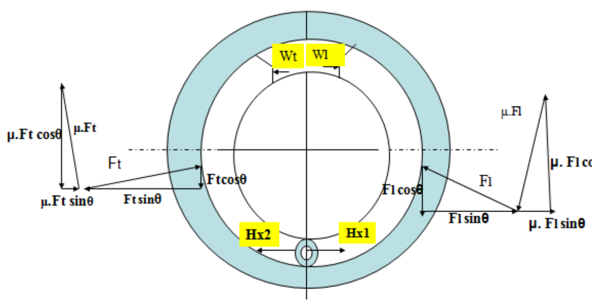


Fig. 3 Force analysis for both the shoes of the drum brake.

The force analysis considering the contact forces (F_l, F_t), actuating forces (W_l, W_t) and hinge reactions (Hx_1, Hx_2) is proposed below. The equilibrium condition of the leading shoe and trailing shoe in the inclined arrangement of the F_l and F_t is described in the following equations:

$$\mu.F_l(d-h.\cos\theta)=F_l.h.\sin\theta-W_l.r \tag{8}$$

$$\begin{aligned}&\mu.F_t.h=W_t.r-F_t.h.\mu.F_l(d-h.\cos\theta)\\&=F_l.h.\sin\theta-W_l.r\end{aligned} \tag{9}$$

$$\begin{aligned}&F_l.\sin\theta-F_t.\sin\theta+W_l-W_t+\mu.F_l.\sin\theta+\\&\mu.F_t.\sin\theta+H_{X1}-H_{X2}=0\end{aligned} \tag{10}$$

$$\begin{aligned}&F_l.\cos\theta+F_t.\cos\theta-\mu.F_l.\cos\theta+\\&\mu.F_t.\cos\theta+mg=0\end{aligned} \tag{11}$$

Eqs. (8-11) are further solved to express the friction coefficient as a function of contact forces (F_l, F_t), actuating forces (W_l, W_t) and distance of Coulomb friction force (d).

$$\mu=\frac{-(F_l-Ft)\sin\theta-Hx_1+Hx_2}{(F_l+F_t)\sin\theta} \tag{12}$$

$$\mu=\frac{-(F_l+F_t)\cos\theta-mg}{(F_t-F_l)\cos\theta} \tag{13}$$

$$\mu=\frac{F_l.h.\sin\theta-W_l.r}{F_l.(d-h.\cos\theta)} \tag{14}$$

$$\mu=\frac{W_t.r-F_t.h.\sin\theta}{F_t.(d-h.\cos\theta)} \tag{15}$$

The magnitude of the friction coefficient can be computed from the above deduced Eqs. (12-15) which is further used in the estimation algorithm. The deduced equations presents database of computed values of the friction coefficient for the asymmetric shoe length.

3. ESTIMATION OF FRICTION COEFFICIENT IN DISC BRAKES

The rotational deceleration of the wheel and rotor are based on the torque caused by the force of friction between the brake pad and the disc rotor [19]. The friction force responsible for deceleration is a function of the force applied by the driver to the pedal. The combined effect of forces acting on the wheel shows that the vehicle speed decreases as the force of the road on the tire overcomes the force of the axle on the wheel.

The conventional design [13] equations for brakes had predicted on one of two assumptions as uniform wear or uniform pressure. If an annular disc brake is replaced with full- faced rigid discs on both input and output shafts and with a lining material that covers the entire face of one of the discs so that $r_i=0$, the torque capability of the brake may be given initially by equation:

$$T=2/3\pi\mu\rho(r_0^3-r_i^3) \tag{16}$$

An analysis of the force lines [14] along a caliper shows that the suitable position for the measurement of the clamping and tangential forces can be found in the friction surface of the brake pads, since all forces required for the braking process must be supported by the brake pads with the inclusion of these forces in the back plate of the brake pad, a correct prediction becomes quite difficult due to the multiple contact points with the caliper. The consideration of the friction radius on the clamping force due to a stationary brake enables radial shifting to the outer edge of the brake disc. During braking process the application of tangential force causes a shifting in the tangential direction. Clamping force is exerted due to the combined effect of hydraulic pressure and the piston surface. The tangential force at the contact of the disc pad and rotor disc is:

$F_{tP}= F_p.\ \mu$ (friction area of the piston side) (17)

$F_{tnp}= F_{np}.\ \mu$ (friction area of the non piston side) (18)

The tangential force causes, together with mean friction radius R, the braking torque:

$$B.T. = (F_{tp}+F_{tnp}).R. \tag{19}$$

Ultimately braking torque is the quantity which results with the help of the dynamic tire radius in the contact patch of the tire, the desired braking force and the deceleration of the vehicle. Due to the support of tangential force on the stator reaction F_1 to the piston force results, considering this force F_1, the clamping force does not correspond to the piston force during a braking process.

It was concluded that[14] one cannot start with a uniform surface pressing and that the friction radius shifts while braking .The caliper expands itself under the high clamping force due to which shifting of the friction radius occurs and a bigger friction radius can be assumed. There are no identical shiftings on piston side and non piston side, since the non piston side has more elastic qualities than piston side with regard to the construction. The variations in the actual and average values of the braking torque are caused due to the deviations between the actual and the nominal geometry of the contact between the brake pads and the disc [18].

$$\text{Braking Force(B.F.)} = \int_{-\theta}^{\theta}\int_{R}^{r} Fy.\rho v..Rd.Rd.\phi \qquad (20)$$

Braking force is transmitted hydraulically through the fluid, for cylinders of the same size force transmitted from one is the similar value as the force applied to other by using cylinders of different sizes forces can be increased or reduced allowing users to achieve desired braking force for each wheel:

$$\text{Braking Force (B.F.)} = \int_{-\theta}^{\theta}\int_{R}^{r} Fy.\rho v.R^2.dR.d\theta \qquad (21)$$

The uniform wear out has been assumed [15] in the brake pads and disc, the braking torque can be calculated as a function of brake inner disc and outer diameter ,the pressure between brake pad and disc and friction coefficient (μ) between the pad and the disc. To prevent the numerical instabilities, the transition between the positive and negative braking torque should be smooth.
The clamping force F_c acting normal to the contact surface of the disc and the caliper is balanced by the resulting force F_2 to attain the equilibrium condition, tangential force F_{tp} is acting the downward as shown in Figs. 4 and 5. Similarly forces at the non piston side of the disc are acting to maintain the equilibrium in the braking process and to avoid the skidding of the vehicle. But due to the actuation mechanism of the piston at the brake disc, the forces at the piston and non-piston side are not equal in magnitude. F_{np} is the contact force at the contact of the disc and caliper, F_2 is the reaction force and F_{tnp} is the tangential force. It is pertinent to note that due to the forces F_1 and F_2 the piston force does not correspond to the clamping force. Due to this phenomenon, variation in the friction radius occurs. Equilibrium equations at the rotor disc in the longitudinal direction results:

$$F_p - F_2 = 0 \text{ (piston side)} \qquad (22)$$

$$F_1 - F_{np} = 0 \text{ (non piston side)} \qquad (23)$$

Weight component, mg, in the downward direction takes into account the accumulated mass of the disc and caliper. Due to the inequality of the forces on the piston and non piston side friction coefficients respectively μ_p and μ_{np} causes the friction forces in the lateral direction:

$$mg + \mu_p.F_p = 0 \text{ (piston side)} \qquad (24)$$

$$mg + \mu_{np}.F_{np} = 0 \text{ (non piston side)} \qquad (25)$$

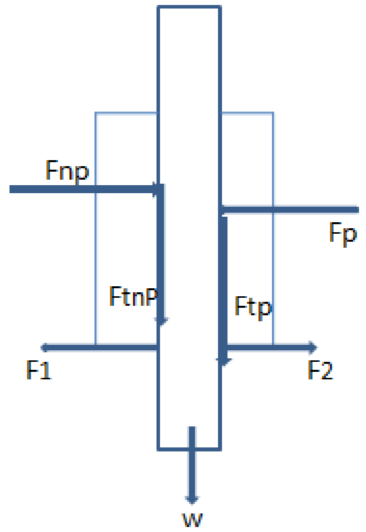


Fig. 4. Forces with variations in active friction radius.

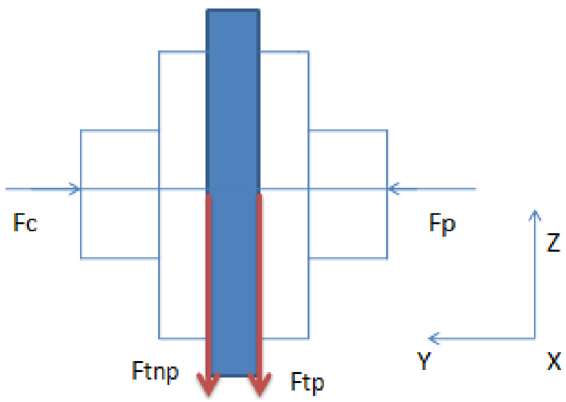


Fig. 5. Forces with constant active friction radius.

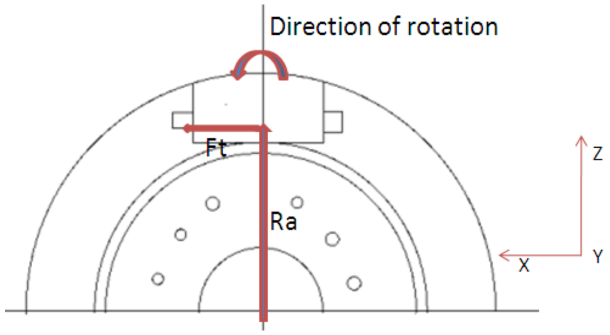


Fig. 6. Tangential force at the calliper.

The clamping force F_c shown in Fig. 5 is the result of the pressure exerted by the fluid and the piston surface. The clamping force is responsible for the tangential force F_t at the caliper and the disc interface shown in Fig. 6, tangential force F_t on piston as well as on the non piston side is expressed below:

$$F_t = F_C.(\mu_{np} + \mu_p) \tag{26}$$

The tangential force together with the friction radius 'R_a' causes the braking torque:

$$T_b = F_t.R_a \tag{27}$$

The braking torque causes the deceleration of the vehicle, to balance the braking effort required at the front and rear axle torque at the piston and non piston side as calculated below:

$$T_p = \mu_p.F_p.R \text{ (piston side)} \tag{28}$$

$$T_{np} = \mu_{np}.F_{np}.R \text{ (non piston side)} \tag{29}$$

$$T = 2/3.F.\mu.(r_o^3 - r_i^3).\theta \tag{30}$$

Torque increases with the actuation effort, velocity and temperature [16]. The total braking torque for the front and rear axles takes into account the angle (θ) for contact patch of the caliper with disc and inner as well as the outer radius.

4. OPERATING AND GEOMETRIC PARAMETERS

The input parameters used for the estimation of the friction coefficient (μ) are listed below. The parameters for the drum braking were referred from the dual braking system set up shown in Fig. 8.

For Drum Brakes

Table 1. Operating and geometric parameters.

Sr. No.	Parameter	Value (cm)
1	r	24.3
2	d	16
3	h	12
4	W_l, W_t	24
5	m (shoe)	0.4

For Disc Brakes

Following Table 2 shows the calculated and referred parameters from the reference vehicle system [17].

Table 2. Input parameters.

Sr No	Parameters	Value
1	Input drive force (lb)	90
2	Pedal ratio	4:1
3	Brake pedal force (lb)	360
4	Force from master cylinder (psi)	935.44
5	Outer radius (in)	8.2
6	Inner radius (in)	4.2
7	Torque at the rotor (In-lb)	(0-437289.8)
8	Force at the piston and non piston side (lb)	2644.9
9	Density of the disc (Kg/m^3)	7850
10	Density of the friction plate (Kg/m^3)	6000

5. DRUM BRAKING SYSTEM SET UP

Fig. 7 shows the set up of simple dual braking system and Fig. 8 indicate the layout of the set up. The purpose of a dual air brake system is to accommodate a mechanically secured parking brake that can be used during a service brake failure and to accommodate the pipes connecting service reservoir and brake chamber are installed with the two pressure gauges as shown in Fig. 7. The brake actuator is connected to the compressor through hand brake and sensing reservoir. Brake chamber is accessing the compressed air through another service reservoir. Service reservoir serves the purpose of safety during the failure. The air compressor is driven by the engine through V-belt.

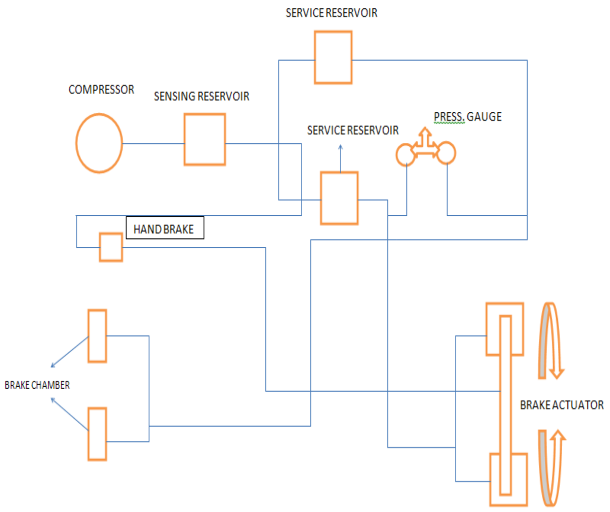


Fig. 7. The layout of Braking System set up.

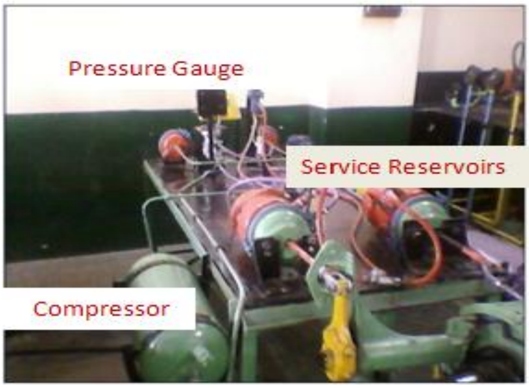


Fig. 8. Braking System set up.

The test was executed as follows:

- The prestart test ensured enough compressed air in the system to actuate the brake application.
- Air compressor supplied the air pressure of 8 bar for actuating the brakes.
- The set up was actuated to demonstrate working of brake linkage mechanism and to acquire the data regarding operating and geometrical parameters.

From the above steps; a set of data shown in Table 1 was measured under a given operating condition. The measured data was processed with the earlier derived equations to compute the friction coefficient (μ).

6. ESTIMATION ALGORITHM FOR COMPUTATION OF THE FRICTION COEFFICIENT

For Drum Brakes

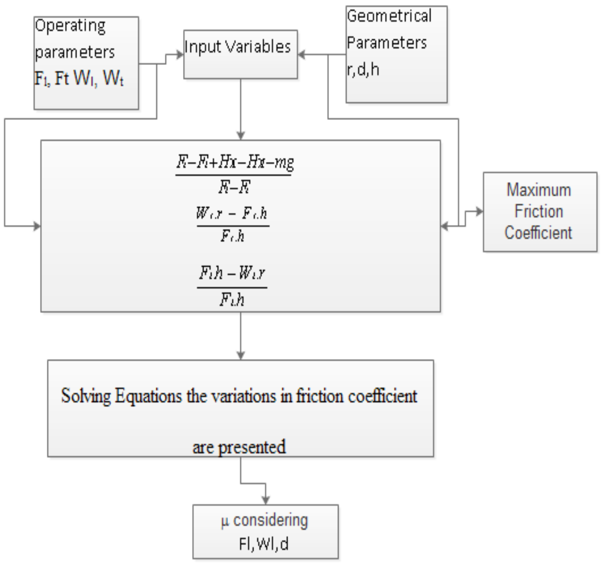


Fig. 9. The flowchart for friction coefficient estimation.

For Disc Brakes

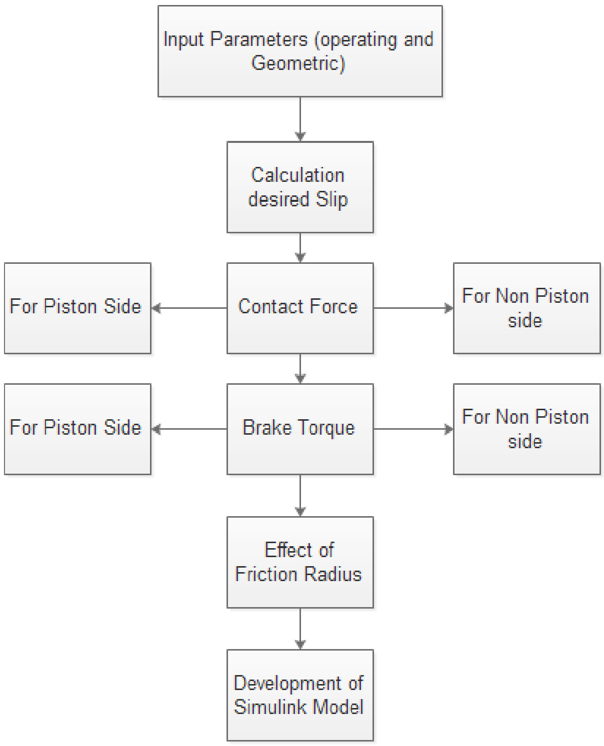


Fig. 10. Algorithm for friction coefficient estimation.

7. DEVELOPMENT OF SIMULINK MODEL

The model presented in the Fig. 11 and Fig. 12 simulates the dynamics at the brake drum-shoe and disc rotor-pad interface during the braking process. The models present a single wheel for drum and disc brake which can be replicated a number of times to represent a model for multi wheel vehicle. The brake drum and rotor disc are at initial velocity corresponding to the vehicle speed before the brakes are applied. From the available data slip between the vehicle speed and drum as well as disc speed is fixed at 0.8 represented by the constant block parameter. In the present work bang-bang controller is used based upon the actual slip and the desired slip. The control of the brake pressure is considered through a first order lag that represents the delay associated with the hydraulic lines of the brake system. The models then integrate the filtered rate to yield the actual brake pressure. The gain factor for the contact force is fixed at the '1'. The calculated brake torque from the contact force and the distance at the shoe – drum and disc rotor-pad interface is set as the block parameter. Further the friction coefficient is obtained by ratio of the forces.

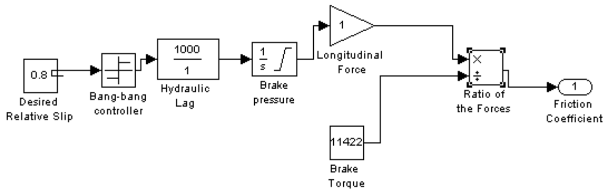


Fig. 11. Simulink model for single wheel drum brake.

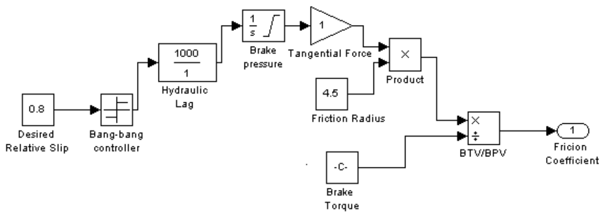


Fig.12. Simulink model for single wheel disc brake.

8. Validation

For Drum Brakes

The validation of the proposed methods is based on comparison between estimated Friction coefficient (μ) with the help of output from the deduced equations solved by using the acquired data from set up and friction coefficient resulted from Simulink model of the braking system shown in Fig. 13. The friction coefficient between 0.3-0.7 as selected from the database of the numerically simulated results are pertaining in the range proposed by (Mortimer,R.G., Campbell, J.D. ,1970)[4] in the earlier work for man- vehicle interface while present study concentrates on brake shoes-drum interface. The operating parameters were obtained from the test runs of the braking trainer set up.

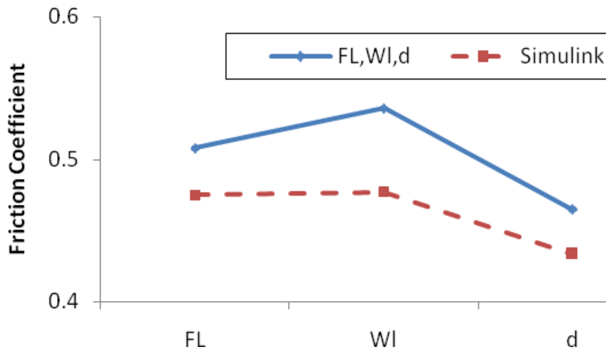


Fig. 13. Comparison of friction coefficient from equations and simulink brake model.

For Disc Brakes

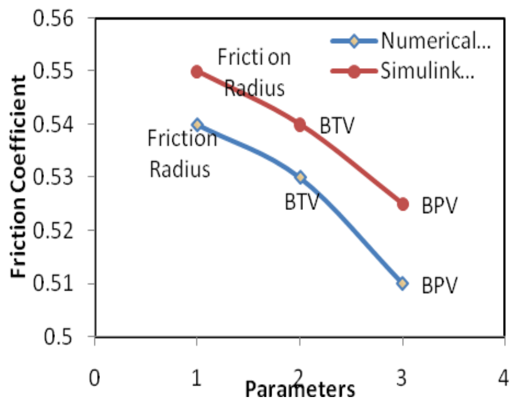


Fig. 14. Comparison of output for numerical and simulink based estimation.

The validation of the proposed methods is based on comparison between estimated friction coefficient (μ) obtained from the deduced equations solved by using the acquired data from reference system and friction coefficient resulted from Simulink model of the braking system as shown in Fig. 14. Also the friction coefficient between 0.3-0.7 as selected from the database of the simulated results are pertaining in the range proposed by Degestein et al.(2006)[14] in the earlier work for man-vehicle interface while present study concentrates on brake disc rotor-pad interface.

9. RESULTS AND DISCUSSION

The frictional behaviour of drum brake and disc brake depicting variations in the friction coefficient by use of the deduced equations has been discussed. The friction coefficient for drum and disc brake for contact forces and friction radius considering the effects of tangential and longitudinal forces for drum brake and tangential force, caliper force, torques(piston, non-piston)for disc brake are evaluated.

9.1 Contact force (F_l)

Drum Brake

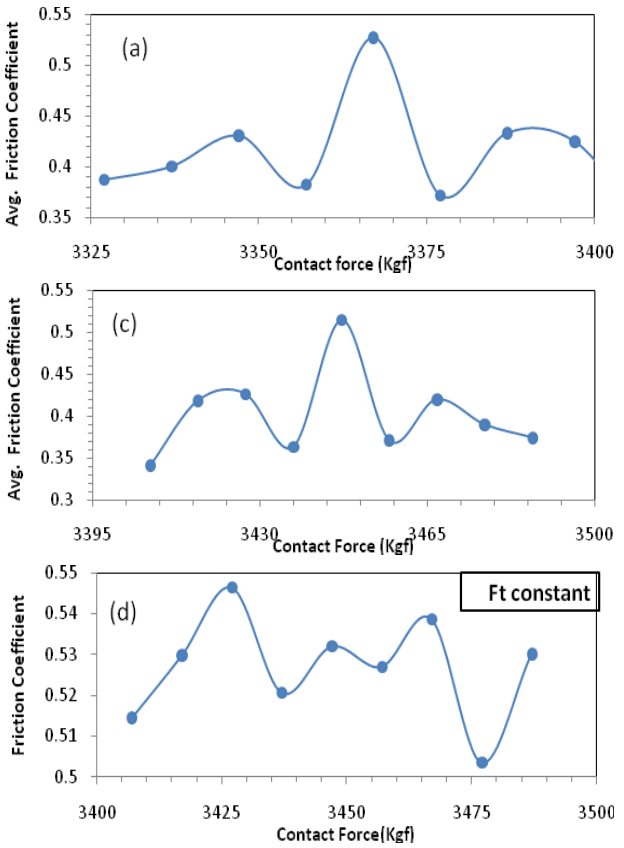
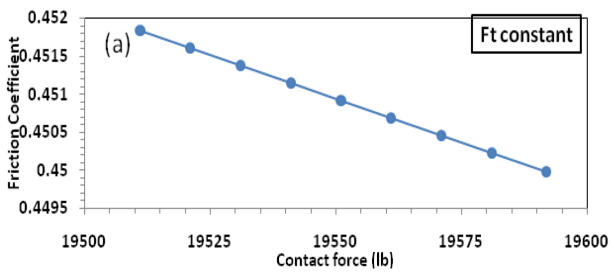


Fig. 15(a-d). Friction Coefficient profiles for variations in contact force for drum brake.

Disc Brake



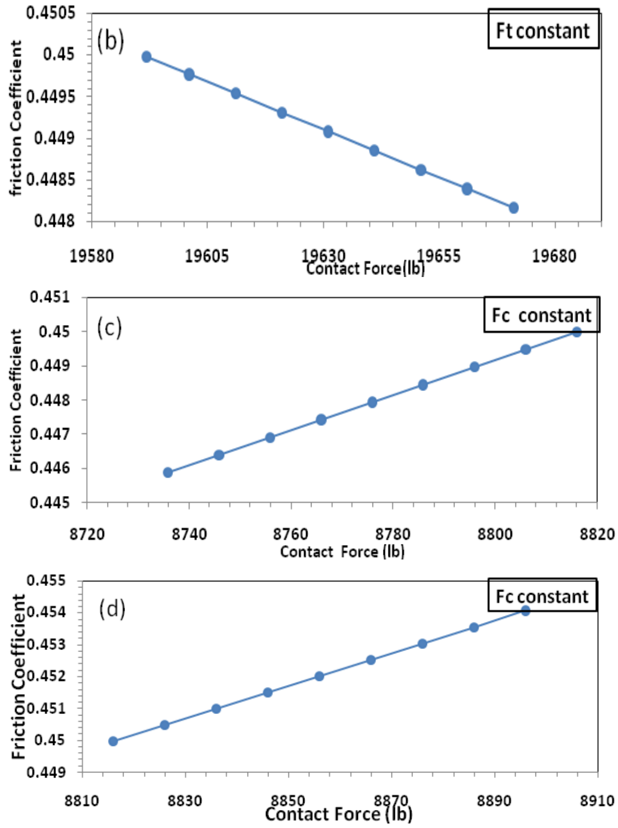


Fig. 16 (a-d). Friction Coefficient profiles for variations in contact force for disc brake.

The effect of contact forces for drum and disc brake have been studied by varying the tangential and longitudinal forces for drum brake and tangential force, caliper force, torques (piston, non-piston) for disc brake. The results are indicated in the Fig. 15 (a-d) and Fig. 16 (a-d) for drum and disc brake respectively. The longitudinal forces F_l and F_t (3407Kgf) are kept constant to study the variations in the friction coefficient for drum brake. Similarly tangential force (8815.95 lb), caliper force (19591.83 lb), torques (218644 in-lb, 204169.7 in-lb) have been kept constant to obtain the profiles of friction coefficient for disc brake. The friction coefficient variations are more consistent in drum brake than in disc brake.

- When tangential force is constant; friction coefficient is decreasing in disc brake with increasing contact force while its variations are between 0.4-0.6 in drum brake with slight increase. Drum brake demonstrated more abrupt changes than disc brake.
- The friction coefficient profile is increasing when caliper force is constant in disc brake.

- For drum brake sudden increase in the profile of friction coefficient can be observed at the contact force between 3350-3450 kgf due to the increase in the contact pressure at the interfacial surface.

9.2 Friction Radius (d, R_a)

Drum Brake

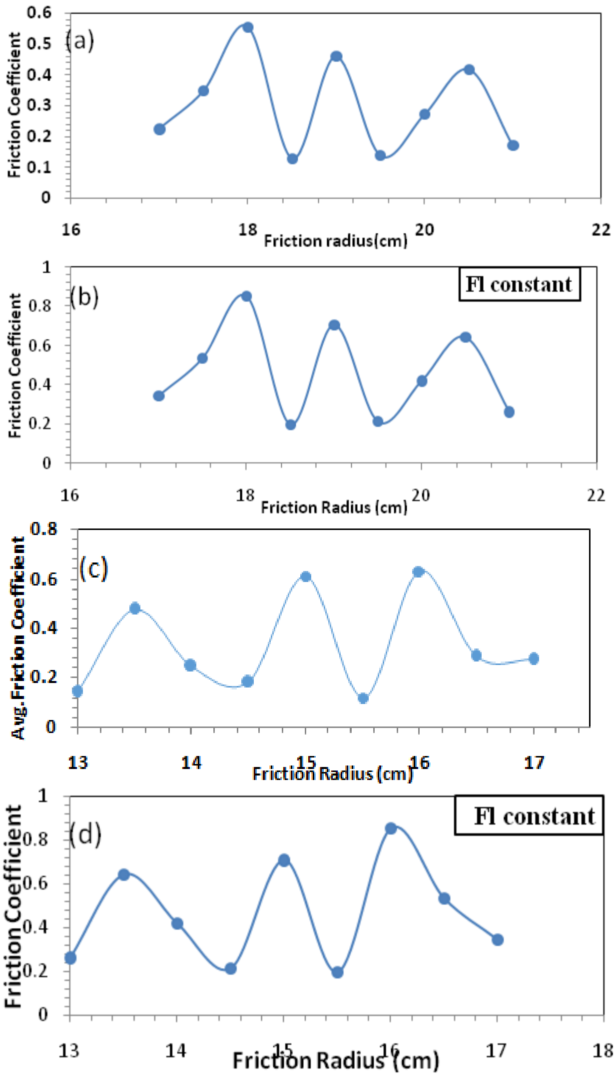
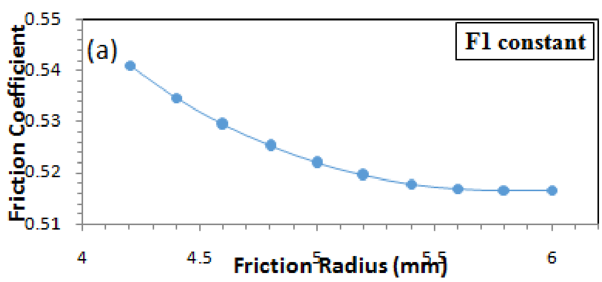


Fig. 17 (a-d) Friction coefficient profiles for variations in friction radius for drum brake.

Disc Brake



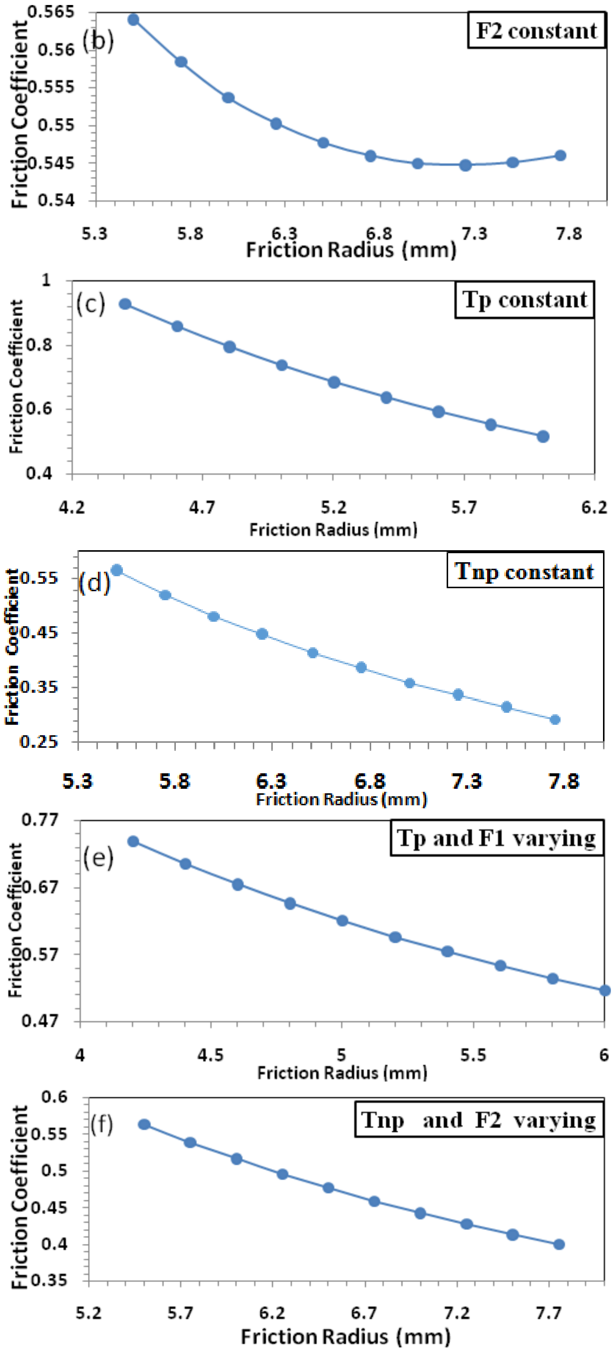
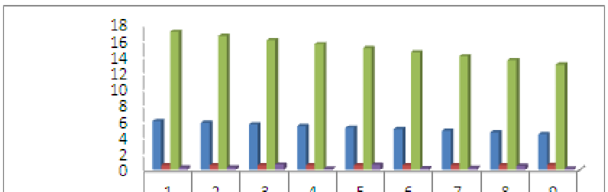


Fig 18 (a-f). Friction coefficient profiles for variations in friction radius for disc brake.



	1	2	3	4	5	6	7	8	9
R (disc)	6	5.8	5.6	5.4	5.2	5	4.8	4.6	4.4
µ disc	0.540	0.537	0.535	0.535	0.536	0.539	0.544	0.551	0.559
d(drum)	17	16.5	16	15.5	15	14.5	14	13.5	13
µ drum	0.273	0.285	0.633	0.118	0.611	0.181	0.245	0.476	0.141

Fig. 19. Comparison of friction coefficient (µ) with friction radius.

The effect of friction radius on behaviour of friction coefficient for drum and disc brake have been studied by varying the longitudinal forces for drum brake and tangential force, torques(piston, non-piston)for disc brake. The results are indicated in the Fig.17 (a-d) and Fig. 18(a-f) for drum and disc brake respectively. The longitudinal forces F_l and F_t (3407Kgf) are kept constant to study the variations in the friction coefficient for drum brake. Similarly tangential force (8815.95 lb), caliper force (19591.83 lb), torques (218644 in-lb, 204169.7 in-lb) has been kept constant to obtain the profiles of friction coefficient (μ) for disc brake. Abrupt variations in the profiles of the friction coefficient (μ) were observed in the drum brake with increase in the friction radius while variations have been gradual in the disc brake.

- With constant longitudinal force at the leading shoe and increasing friction radius sudden increase in the friction coefficient was observed at the friction radius 18cm due to increase in the pressure intensity at 18 cm for drum brake as shown in Fig. 18 (a) and (b) for disc brake friction coefficient is decreasing with the increase in friction radius as indicated in Fig. 18 (a) and (b), since the contact pressure distribution is also reducing.
- With constant longitudinal force at the leading shoe and decreasing friction radius sudden increase in the friction coefficient upto 0.8 was observed at the friction radius 16 cm can be attributed the increase in pressure intensity at 16 cm for drum brake as shown in Fig. 17 (b) and (d). Average friction coefficient obtained after simulating all the deduced equations have exhibited quite abrupt changes due to the fact that optimum performance can only be observed for friction radius of 15cm for drum brake as reported in Fig.18 (a) and (c).
- For constant torques (piston, non-piston) and tangential force at the disc brake the dependence of friction coefficient on friction can be observed from the profiles; as friction coefficient is decreasing with the increase in friction radius as indicated in Fig. 18 (e) and (f).
- For disc brake having constant tangential force there is gradual decrease in the friction coefficient with decreasing friction radius due to the reduction in the braking effort as shown in Fig.18 (a) and (b).
- Fig. 19 shows comparative behaviour of drum and disc brakes. With increase in friction radius increase in friction coefficient for drum and disc brake was observed but for lower friction radius regimes drum brake could not present optimum results.

10. CONCLUSION

The study presents an estimation methodology of the friction coefficient for drum-shoe and rotor disc-pad interface in the automobiles. The output from deduced equations was compared with the similar obtained from virtual Simulink models for drum and disc brakes. The relationship of influential factors, contact forces (F_l, F_t) and friction radius with friction coefficient were used in the estimation algorithm to compute the friction coefficient.

- The increase in contact force resulted with decrease in friction coefficient both for drum as well as disc braking system. But the disc brakes exhibit gradual decrease of friction coefficient due to the equitable distribution of braking effort.
- It can be judged from the results that friction radius predominantly affects brake pressure and thus the friction coefficient as shown in Fig. 16.The disc brakes presented least of the variations in the friction radius as compared to drum brakes due to the better control on the actuating mechanism on the piston side hence friction coefficient was in the range 0.535-0.559 which is useful to attain suitable braking torque.
- We can conclude that the parameter identification and state estimation method presented can be useful in designing a braking system and proper combination of braking system for a vehicle.

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NOMENCLATURE

μ: Friction Coefficient

F_l: contact Force for leading shoe (Kgf)

F_t: contact Force for trailing shoe (Kgf)

W_l: actuating forces for leading and trailing shoes (Kgf)

W_t: actuating forces for leading and trailing shoes (Kgf)

Hx_1,Hx_2: Hinge Reaction in x direction (Kgf)

mg: Weight component of the brake shoes

θ: Angle of the longitudinal force with vertical (deg)

h: Distance between the centre of the pivot and centre of the drum (cm)

r: Distance between the centre of the pivot and inclined force (cm)

d: Distance between the centre of the pivot and Coulomb force (cm)

μF_l: Coulomb friction force for leading shoe

μF_t: Coulomb friction force for trailing shoe

D_f: Input drive force (lb)

P_R: Pedal ratio

B_p: Brake pedal force (lb)

F_m: Force from master cylinder (lb)

r_o: Outer radius (in)

r_i: Inner radius (in)

T_b: Torque at the rotor (in-lb)

F_1, F_2: Force at the piston and non piston side (lb)

ρ: Density of the disc (kg/m3)

ρ_f: Density of the friction plate (kg/m3)

F_P: Force on Piston side (lb)

F_{np}: Force on non-Piston side (lb)

F_c: Caliper Force (lb)

Ft: Tangential Force (lb)

R_a: Friction Radius (cm)

μ_p: Friction coefficient on piston side

μ_{np}: Friction coefficient on non piston side

T_p: Torque on piston side (in-lb)

T_{np}: Torque on non piston side (in-lb)