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Control of Braking Force under Loaded and Empty Conditions on Two Wheeler

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Abstract**—**The Automobile Braking System has a crucial role for safety of the passenger and riding quality of the vehicle. The braking force mainly depends on normal reaction on the wheel and the co-efficient of friction between the tire and the road surface. Whenever a vehicle is loaded, the normal reaction on the rear wheel is increased. Thus the amount of braking force required to halt the vehicle with minimum stopping distance, is based on the pillion load on the vehicle. In this work, in order to vary the braking force in two wheelers, the mechanical leverage which operates the master cylinder is varied based on the pillion load. Thus the amount of braking force developed between ground and tire is varied. This optimum braking force on the disc brake helps in attaining the minimum vehicle stopping distance. In addition to that, it also helps in preventing sliding. Thus the system results in reducing the stopping distance of the two wheelers and providing a better braking efficiency than the conventional braking system.

Keywords**—**Braking force, Master cylinder, Mechanical leverage, Minimum stopping distance.

I. INTRODUCTION

A large number of studies have shown that accidents on vehicles are the root cause of millions of death each year. Especially, two wheeler accidents are increasing rapidly when compared with four wheeler accidents. Two wheeler accidents are mainly due to sliding and increased stopping distance [3], as two wheelers are less stable when compared to four wheelers.

The stopping distance of a two wheeler is the distance between a point where the brake is applied and to the point where linear velocity of the two-wheeler becomes zero. In order to reduce accidents, minimum stopping distance should be achieved. It is accomplished with the help of maximum deceleration of the vehicle. This maximum deceleration depends on maximum braking force [8]. If the braking force exceeds the maximum friction limit between tire and ground, sliding will occur due to the locking of wheels. Hence an optimum braking force [4], [5] which helps in attaining minimum stopping distance without sliding [10] should be attained. Sliding can be avoided if the locking of wheels is prevented.

In the conventional or present braking system, there is no additional system to achieve minimum stopping distance for different pillion load. This may be one of the reasons for the occurrence of two-wheeler accidents.

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II. HOW MINIMUM STOPPING DISTANCE CAN BE ACHIEVED

Normal reaction on the tire is one of the key factors to be considered for varying the braking force. Because the braking force mainly depends on co-efficient of friction between the tire-ground and normal reaction on the tire [3]. Here, in order to vary the braking force in accordance with the normal reaction of the two-wheeler, and is related as follows:

$$F_r = \mu F_{ZRdyn} \quad (1)$$

$$F_r = (P_1 - P_0) * A_{wc} * \eta_c * BF(r/R) \quad (2)$$

where

F_r = Rear axle Braking force, N

μ = Coefficient of friction between tire and road surface

F_{ZRdyn} = Rear axle dynamic load

P_1 = Hydraulic brake line pressure, N/m^2

P_0 = Push out pressure, required to bring brake pads in contact with disc in N/m^2

A_{wc} = Caliper cylinder area in m^2

η_c = Wheel cylinder efficiency

BF = Brake factor

R = Effective radius of disc in m

R = Effective rolling radius of tire in m

The factors that vary the braking force are:

A. Effective Radius of the Caliper

As the effective radius of the disc increases the braking force increases in accordance with the pillion load on the two-wheeler. The optimum braking force which helps in attaining minimum stopping distance without sliding is obtained. But this has a disadvantage on modifying the whole caliper system. (The effective radius of the disc is varied only with the help of caliper system).

B. Master Cylinder Cross Section

The modification of the master cylinder cross section area with respect to the pillion load is not possible.

C. Number of Piston

Varying the number of pistons in the caliper is a tedious job during manufacturing process and it is not economical.

D. Leverage Distance Ratio

The mechanical leverage which operates the master cylinder is varied based on the pillion load. The variation of the leverage distance ratio will be simple and efficient when compared with all the above factors.

This is achieved from *The Law of Lever* concept.

$$F * a = F_1 * b \tag{3}$$

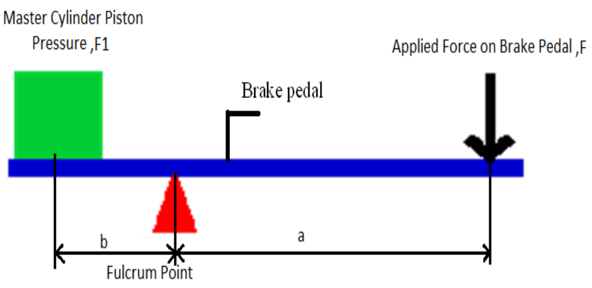

Fig. 1 Layout of rear brake pedal with master cylinder

Taking moment balance with respect to the fulcrum point,

$$F_1 = F * (a/b) \tag{4}$$

where,

'*F*' is the Input force (force applied on the rear brake pedal)

'F_1' is the output force (Hydraulic force developed in master cylinder)

'*a*' is the distance between the input force(brake pedal) and the fulcrum centre.

'b' is the distance between the output force(master cylinder piston) and the fulcrum centre.

The ratio of output to input force is given by the ratio of the perpendicular distances between the lines of action of forces from the fulcrum. This is known as *The Law of lever*.

$$F_1/F = a/b \text{ (LAW OF LEVER)} \tag{5}$$

$$b = a * F/F_1 \tag{6}$$

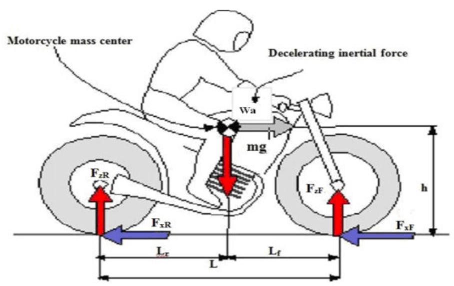

Fig. 2 Load transfer effect during braking

During the deceleration phase, the load on the front wheel increases while the load on the rear wheel decreases, due to load transfer effect [6]-[9]

Static front axle load,

$$F_{ZF} = mg(L_r/L) \tag{7}$$

Static rear axle load,

$$F_{ZR} = mg(L_f/L) \tag{8}$$

The dynamic load [1] on the front wheel is equal to the sum of the static load and of the load transfer effect.

$$F_{ZFdyn} = mg\left(\frac{L_r}{L}\right) + F\left(\frac{h}{L}\right) \tag{9}$$

While the dynamic load on the rear wheel is equal to the difference between the static load and the load transfer effect:

$$F_{ZRdyn} = mg\left(\frac{L_f}{L}\right) - F(h/L) \tag{10}$$

where,

$F(h/L)$ is the load transfer effect on the road wheels.

It can be concluded that the dynamic load on front wheel is increased due to the addition of load transfer effect to the front wheel while the dynamic load on the rear wheel is decreased due to the load transfer effect from the rear to the front.

III. Measurement of Centre of Gravity (CG)

The motorcycle front wheel load distribution is determined by placing the front wheel on a weighing scale. After finding the front wheel load distribution, an application of balance moment about the rear axle is taken to find the horizontal location of centre of gravity (CG) of the motor cycle.

Vertical location of CG is measured by lifting the rear wheel to certain height and noted down the front axle weight by using weighing scale.

After obtaining required readings, (11) is used to find the vertical location of CG of the motorcycle. The rear axle is lifted for finding vertical location of CG. The axle load which is measured using weighing scale is determined for the different load conditions.

TABLE I
Measuring of Axle Loads

Weight (kg)	Reaction on front wheel W_f(kg)	Reaction on rear wheel W_r(kg)	Reaction on front wheel in lifted condition W_{lf} (kg)
191 Rider(Unladen)	71	120	84
261 Rider+70kg	74.5	186.5	94.5

kg – kilogram

TABLE II
Calculation of CG

Weight (kg)	Distance of CG from Front wheel L_f(mm)	Distance of CG from Rear wheel L_r(mm)	Height of CG from Ground level h(mm)
191Rider(Unladen)	835.60	494.39	622.18
261 Rider+70kg	950.36	379.63	662.10

kg – kilogram, mm- millimetre

$$\text{Height of CG (h)} = H_1 + \frac{(W_{2f} * L * L_n)}{(W_t * H_2)} \tag{11}$$

where

W_f =Weight on front axle when the motorcycle is level

W_r =Weight on rear axle when the motorcycle is level

W_{1f} =Weight on front axle when the motorcycle rear wheel is lifted

W_t =Total weight of the motor cycle = $W_f + W_r$

W_t =Weight added on front axle because of rear wheel lift $= W_{lf} - W_f$

W_{2f} =Length of wheelbase while the motor cycle is level = 1330 mm

L =Height of front hub off the ground = 300 mm

H_1 =Height of rear wheel hub above the front wheel hub (how high the rear-axle has been lifted) = 355mm

H_2 =New wheel base when the motor cycle rear wheel is lifted.

L_n $L_n = \sqrt{L^2 - {H_2}^2}$

IV. Dynamic Axle Loads

Application of moment balance about the rear tire to ground contact patch yields dynamic normal force [1] on the front axle

$$F_{ZFdyn} = (1 - \Psi + \aleph a)W, \text{ N} \qquad (12)$$

where,

a $= F_{x,total}/W$ Motorcycle deceleration, g-units

$F_{x,\ total}$ = total braking force, N

F_{ZR} = Static rear axle load, N

W = motorcycle weight, N

$\aleph$ = CG height (h) divided by wheel base (L)

Ψ $= F_{ZR}/W$

Similarly, moment balance about the front tire-to-ground contact point yields the dynamic rear axle normal force

$$F_{ZRdyn} = (\Psi - \aleph a)\text{W, N} \qquad (13)$$

Inspection of equations reveals that the dynamic normal axle forces are linear functions of deceleration [8] ‘a’, i.e., straight-line relationships. The amount of load transfer effect of the rear axle is given by the term $\aleph aW$ in the equations. The normal rear and front axle loads of a typical motorcycle are for the unladen, rider -only and additional every 10kg cases, considering rider weight (50kg) are constant. Inspection of the axle loads reveals that the rear axle load is significantly less at higher decelerations than that associated with the front axle. For example, the rear axle load has decreased from a static load of 785N to only 295N for a 1g stop at unladen (141 kg) condition, while the front load has increased from 589 N to 1080 N.

The relative centre of gravity height $\aleph$ of a typical motorcycle does not change significantly, if at all, from the rider-only to the fully laden condition.

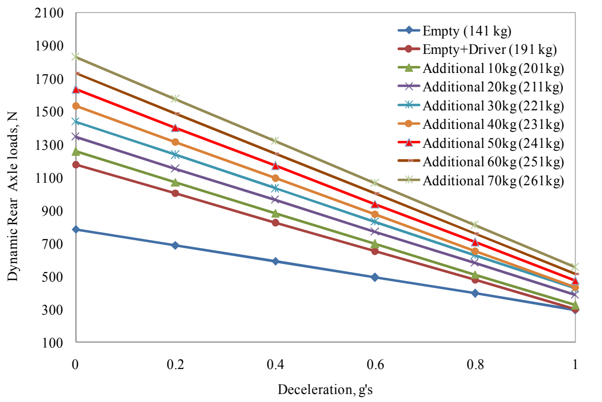

Fig. 3 Dynamic rear axle loads for unladen and various pillion loads

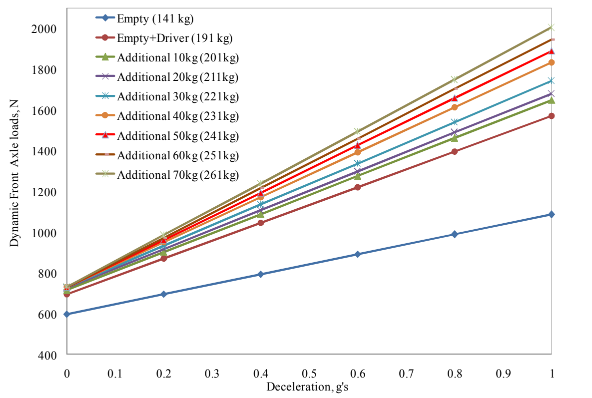

Fig. 4 Dynamic front axle loads for unladen and various pillion loads

V. Optimum Braking Forces

The wheel brake torque [12] generates braking forces between the tire and ground. The ratio of braking force to dynamic axle load is defined as the traction co-efficient μ.

$$\mu_{Ti} = (F_{xi}/F_{Zi,dyn}) \qquad (14)$$

where,

F_{xi} = axle braking force, N

$F_{Zi,dyn}$ = dynamic axle normal force, N

i = designates front or rear axle

The braking coefficient is the level of tire-road friction needed by the braked tire so that it will just not lock up. The braking coefficient varies when either braking force [4], [11] or dynamic axle normal force changes and, consequently, it is a vehicle geometry and deceleration-dependent parameter. In general, the front and rear axle braking coefficient will be different. Only when the numerical values of braking and tire road friction coefficient are equal does the tire lock up.

A. Dynamic Braking Forces

Multiplication of the dynamic axle loads by the traction coefficient yields the dynamic braking forces [1] for the front axle and the rear axle.

$$F_{XF} = (1-\Psi+\aleph a)\, W\mu_{TF}, \text{ N} \quad (15)$$
$$F_{XR} = (\Psi-\aleph a)\, W\mu_{TR}, \text{ N} \quad (16)$$

For straight-line braking on a level surface in the absence of any aerodynamic effects and rolling resistance of tire, optimum braking in terms of maximizing motorcycle deceleration is equal to coefficient of friction.

$$\mu_R = \mu_F = \mu_{TF} = \mu_{TR} = a$$

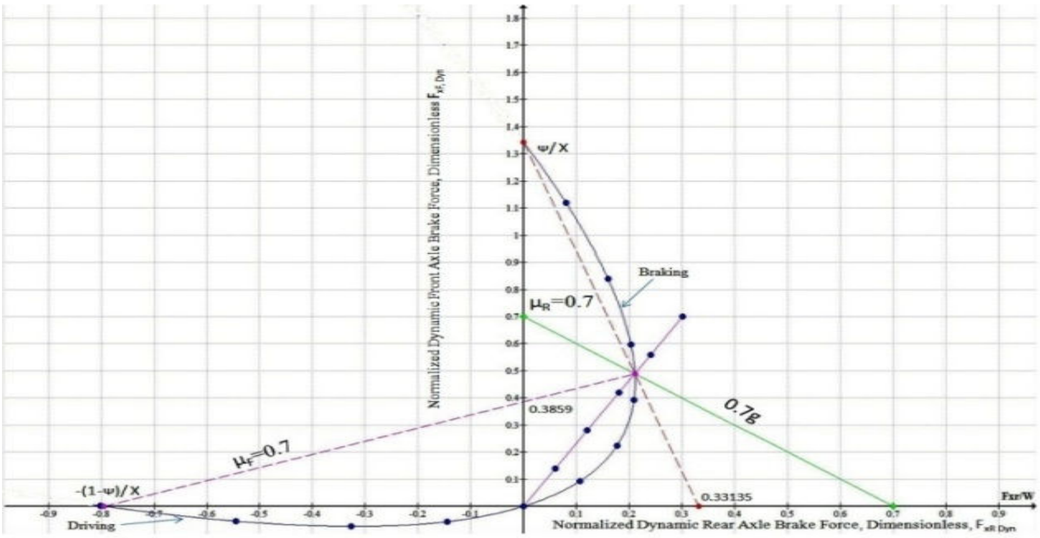

Fig. 5 Parabola of normalized dynamic braking and driving forces for unladen (191 kg) condition

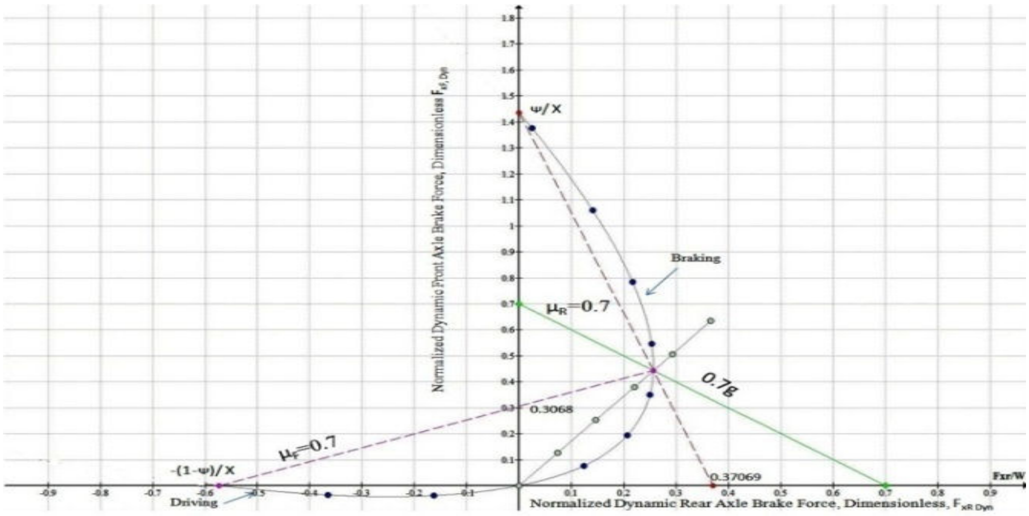

Fig. 6 Parabola of normalized dynamic braking and driving forces for loaded (261 kg) condition

TABLE III
CALCULATION OF F_r/w

Weight (kg)	Loading Condition	F_r/w
191	Rider(Unladen)	0.33135
261	Rider+70kg	0.37069

kg – kilogram

VI. Effect of B Value with Respect to the Vehicle Weight

$F_1/F = a/b$ (Law of Lever)

Hence if b value is varied with respect to the vehicle weight, optimum braking force corresponding to the vehicle weight is applied which helps in attaining minimum stopping distance without sliding.

When the brakes are applied, the torque developed by the wheel brake is resisted by the tire circumference were it comes in contact with the ground. Prior to break lockup, the magnitude of the braking forces is a direct function of the torque produced by the wheel brake. For hydraulic brakes, is used for determining the actual braking forces.

Equation (2) is

$$F_r = (P_l - P_0) * A_{wc} * \eta_c * BF\left(\frac{r}{R}\right), \text{ N}$$
$$(P_l - P_0) = F_1/A_{mc} \quad (17)$$

$$(P_l - P_0) = F * (a/b)/A_{mc}$$
$$(P_l - P_0) = F * Y/A_{mc} \quad (18)$$

where
F_r = Rear axle Braking force, N
P_l = Hydraulic brake line pressure, N/m^2
P_0 = Pushout pressure, required to bring brake pads in contact with disc (0.05*10^-6 N/ m^2)
A_{wc} = Caliper cylinder area (804.2477*10^-6 m^2)
η_c = Wheel cylinder efficiency is 0.98
BF =Brake factor is 0.7
r = Effective radius of disc (0.115m)
R = Effective rolling radius of tire (0.3m)
$\frac{a}{b}$ = Y (Leverage ratio)
A_{mc} = Master cylinder cross section area (226.98*10^-6 m2)

TABLE IV
EFFECT OF B VALUE WITH RESPECT TO THE VEHICLE LOADING CONDITIONS

Weight (kg)	Loading Condition	b value in mm
191	Rider(Unladen)	20.57
261	Rider+70kg	13.53

kg – kilogram, mm - millimetre

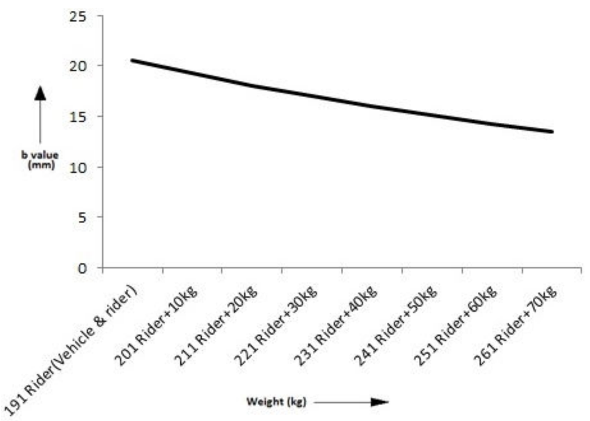

Fig. 7 comparing b with respect to vehicle loading conditions

VII. THEORETICAL STUDY

The theoretical braking distance can be found by determining the work required to dissipate the vehicle's kinetic energy.

The kinetic energy E is given by the formula:

$$E = (1/2)mv^2, \quad (19)$$

where,
'm' is the vehicle's mass in kg.
'v' is the vehicle speed in m/s.

The work (W) done by braking is given by:

$$W = \mu mgd \quad (20)$$

where,
'μ' is the coefficient of friction between the road surface and the tires.
'g' is the acceleration due to gravity in m/s^2
'd' is the stopping distance in m

The braking distance (which is commonly measured as the slide length) given at initial driving speed (v) is then found by replacing work done (W) by kinetic energy (E) of the two-wheeler which gives:

$$d = v^2/2a \quad (21)$$
$$\mu = 0.7$$

We know that

$$\mu R_R = Wa \quad (22)$$
$$\mu(\Psi - \aleph a)W = Wa$$

$$a = \left(\frac{\mu\Psi}{1+\mu\aleph}\right) * 9.81\ m/s^2 \quad (23)$$

Refer to "(21)" It is found that
a=3.239 m/s^2 for 191 kg
a=3.638 m/s^2 for 261 kg.

VIII. PRACTICAL PROCEDURE

In this modified braking system, there are two slots in the brake lever, one for laden condition and the other for unladen condition.

Initially, an experiment is conducted without pillion load. Then the brake lever is adjusted such that the b value (the distance between the output force and the fulcrum centre) is 20.57mm and the vehicle is accelerated to various speeds. When the brake is applied, the corresponding stopping distances are measured.

Then the experiment is conducted with an additional pillion load of 70kg. The brake lever is adjusted such that the b value is 13.53mm and the vehicle is again accelerated to various speeds. When the brake is applied the corresponding stopping distances are measured.

The stopping distance of the vehicle is reduced for the modified system when it is compared with the conventional system.

In conventional system, the leverage ratio (*Y =4.14*) remains the same for all loading conditions.

In modified system, the leverage ratio (*Y)* for the loading condition (191 kg) is 7.046 and for the loading condition (261 kg) it is 10.712.

Fig. 8 Fabrication of the Modified system

TABLE V
COMPARISON OF STOPPING DISTANCE BETWEEN CONVENTIONAL, THEORETICAL AND MODIFIED SYSTEM FOR 191 KG

Vehicle Speed (Km/h)	Conventional System (m)	Theoretical Values (m)	Modified System (m)
40	22.10	19.05	19.70
50	33.65	29.78	30.23
60	49.60	42.89	44.50

km/h – kilometre per hour, m- metre

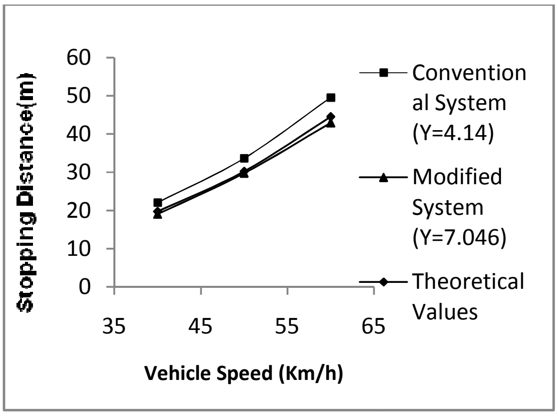

Fig. 9 Variation of Speed and Stopping distance for 191kg

TABLE VI
COMPARISON OF STOPPING DISTANCE BETWEEN CONVENTIONAL, THEORETICAL AND MODIFIED SYSTEM FOR 261 KG

Vehicle Speed (Km/h)	Conventional System (m)	Theoretical Values (m)	Modified System (m)
40	24.16	16.96	17.85
50	40.21	26.51	30.41
60	54.60	38.19	42.15

km/h – kilometre per hour, m- metre

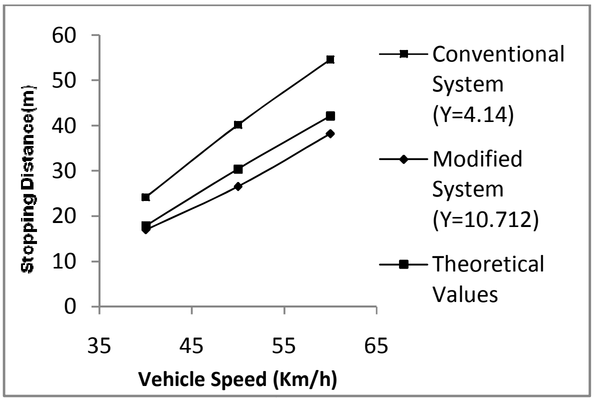

Fig. 10 Variation of Speed and stopping distance for 261kg

IX. COMBINED EFFECT OF MODIFIED BRAKING SYSTEM WITH ANTILOCK BRAKING SYSTEM

Anti-lock braking system [2], [13] helps in preventing the vehicle sliding but it does not help in reducing stopping distance for varying load [14] at varying speeds. But this braking system helps in achieving minimum stopping distance without sliding. Hence ABS supported by this new braking system will be very effective in preventing the accidents.

X. CONCLUSION

In this work, the application of leverage mechanism is introduced by varying the position of master cylinder on the brake pedal based on the pillion load. Thus the braking force is also varied according to the pillion load and the following conclusions are drawn.

- The ‘b’ value varies inversely in accordance with the pillion load.
- The reduction of stopping distance at 191 kg is 11% and for 261 kg, it is about 24%. This is due to the presence of optimum braking force between the tire and the road surface.
- Sliding of vehicle is reduced.

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