CHAPTER 3
DESIGN OF MOTORCYCLE DISC BRAKE

3.1 BASIC BRAKE SYSTEM DESIGN CONSIDERATIONS

In most cases the brake engineer has the following data available when designing the brakes of a vehicle:

- Empty and loaded vehicle weight
- Static weight distribution lightly and fully laden
- Wheel base
- Center of gravity height lightly and fully laden
- Tyre and rim size

3.2 MEASUREMENT OF CENTRE OF GRAVITY (CG)

The motorcycle’s front wheel load distribution is calculated 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 motorcycle. Figure 3.1 shows the reading which is taken using weighing scale.

The vertical location of CG is measured by lifting the rear wheel to a certain height and the front axle weight is noted down by using weighing scale. After taking required readings, a formula (Derived in Appendix 1) which is given in Equation 3.1 is used to find the vertical location of CG of the motorcycle. Figure 3.2 shows the lifted rear axle for finding the vertical
location of CG. The axle load which is measured using weighing scale is given in Table 3.1 for the different load conditions. The horizontal and vertical location of CG of a typical motor cycle is given in Table 3.2.

Figure 3.1 Weighing scale to measure axle load

Figure 3.2 Lifted rear wheel to find vertical location of CG
Height of CG (h) = \( H_1 + \frac{W_{2f} \cdot L \cdot L_n}{W \cdot H_2} \)  \hspace{1cm} (3.1)

Where

\( W_f \) = Weight on front axle when the motorcycle is level

\( W_r \) = Weight on rear axle when the motorcycle is level

\( W_{lf} \) = Weight on front axle when the motorcycle’s rear wheel is lifted

\( W \) = Total motorcycle weight = \( W_f + W_r \)

\( W_{2f} \) = Weight added on front axle because of rear wheel lift = \( W_{lf} - W_f \)

\( L \) = Length of wheelbase while the motorcycle is level = 1330 mm

\( H_1 \) = Height of front hub off the ground = 300 mm

\( H_2 \) = Height of rear wheel hub above the front wheel hub

(How high the rear-end has been lifted = 355mm)

\( L_n \) = New wheelbase when the motor cycle rear wheel is lifted

\( L_n = \sqrt{L^2 - H_2^2} \)

An iron stem as shown in Figure 3.3 is attached directly above the position of the axle (The line of action of load is passing perpendicular to the axis of axle and through the rear tyre contact patch with the ground). The stem is rigidly fixed with motorcycle frame after removing the seat. Then a dead weight which has a hole in its centre is inserted into the stem. The weight of the each dead weight is 10kg. All dead weights are added in the same manner for taking readings which are substituted in the formulas.
\[
L_e = \frac{W_f \ast L}{W}, \quad L_f = L - L_r, \quad h = H_f + \frac{W_{2f} \ast L_n \ast L}{W \ast H_2}
\]
to find horizontal and vertical CG of the motorcycle.

Figure 3.3 CG Measurement for extra loads on the pillion

3.3 SINGLE VEHICLE BRAKING DYNAMICS

3.3.1 Static Axle Loads

The forces acting on a non-decelerating motorcycle, either stationary or traveling at constant velocity on a level roadway are illustrated in Figure 3.4. Due to the front-to-rear distribution, the front and the rear axle may carry significantly different static axle loads. The static axle load distribution is defined by the ratio of static rear axle load to the total motorcycle weight, designated by the Greek letter \( \Psi \) as

\[
\Psi = \frac{F_{2r}}{W}
\]

Where, \( F_{2r} \) = Static rear axle load, N

\( W \) = Total motorcycle weight, N
Table 3.1 Measuring of axle load

<table>
<thead>
<tr>
<th>Sl. No</th>
<th>Additional Pillion Load on vehicle</th>
<th>Reaction on front wheel ‘$W_f$’ (kg)</th>
<th>Reaction on rear wheel ‘$W_r$’ (kg)</th>
<th>Total weight $W$ (kg)</th>
<th>Reaction on front wheel in lifted condition ‘$W_{fr}$’ (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Additional 00kg (Without Rider)</td>
<td>61.0</td>
<td>80.0</td>
<td>141</td>
<td>66.0</td>
</tr>
<tr>
<td>2</td>
<td>Additional 00kg (With Rider)</td>
<td>71.0</td>
<td>120.0</td>
<td>191</td>
<td>84.0</td>
</tr>
<tr>
<td>3</td>
<td>Additional 10kg</td>
<td>73.0</td>
<td>128.0</td>
<td>201</td>
<td>87.0</td>
</tr>
<tr>
<td>4</td>
<td>Additional 20kg</td>
<td>74.0</td>
<td>137.0</td>
<td>211</td>
<td>88.0</td>
</tr>
<tr>
<td>5</td>
<td>Additional 30kg</td>
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<td>146.5</td>
<td>221</td>
<td>89.5</td>
</tr>
<tr>
<td>6</td>
<td>Additional 40kg</td>
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<td>156.5</td>
<td>231</td>
<td>91.5</td>
</tr>
<tr>
<td>7</td>
<td>Additional 50kg</td>
<td>74.5</td>
<td>166.5</td>
<td>241</td>
<td>92.5</td>
</tr>
<tr>
<td>8</td>
<td>Additional 60kg</td>
<td>74.5</td>
<td>176.5</td>
<td>251</td>
<td>93.5</td>
</tr>
<tr>
<td>9</td>
<td>Additional 70kg</td>
<td>74.5</td>
<td>186.5</td>
<td>261</td>
<td>94.5</td>
</tr>
</tbody>
</table>
### Table 3.2 Calculation of CG

<table>
<thead>
<tr>
<th>Sl. No</th>
<th>Additional Pillion Load on motor cycle</th>
<th>Distance of CG from Front wheel ‘L_f’ (mm)</th>
<th>Distance of CG from Rear wheel ‘L_r’ (mm)</th>
<th>Height of CG from Ground level ‘h’ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Additional 00kg (Without Rider)</td>
<td>754.60</td>
<td>575.39</td>
<td>470.25</td>
</tr>
<tr>
<td>2</td>
<td>Additional 00kg (With Rider)</td>
<td>835.60</td>
<td>494.39</td>
<td>622.18</td>
</tr>
<tr>
<td>3</td>
<td>Additional 10kg</td>
<td>846.96</td>
<td>483.03</td>
<td>629.59</td>
</tr>
<tr>
<td>4</td>
<td>Additional 20kg</td>
<td>863.55</td>
<td>466.44</td>
<td>614.21</td>
</tr>
<tr>
<td>5</td>
<td>Additional 30kg</td>
<td>881.65</td>
<td>448.34</td>
<td>621.30</td>
</tr>
<tr>
<td>6</td>
<td>Additional 40kg</td>
<td>901.06</td>
<td>428.93</td>
<td>647.96</td>
</tr>
<tr>
<td>7</td>
<td>Additional 50kg</td>
<td>918.85</td>
<td>411.14</td>
<td>653.06</td>
</tr>
<tr>
<td>8</td>
<td>Additional 60kg</td>
<td>935.23</td>
<td>394.76</td>
<td>657.76</td>
</tr>
<tr>
<td>9</td>
<td>Additional 70kg</td>
<td>950.36</td>
<td>379.63</td>
<td>662.10</td>
</tr>
</tbody>
</table>
The relative static front axle load is given by

\[ 1 - \Psi = \frac{F_{sf}}{W} \]  

(3.3)

Where

\( F_{sf} \) = Static front axle load , N

The typical motorcycle has \( \Psi \) values for the empty conditions as high as 0.63, indicating that only 63% of the total weight is carried by the rear axle and that for loaded condition is 0.72.

Application of moment balance about the front axle of the stationary motorcycle shown in Figure 3.4 yields

\[ WL_f = F_{R}L \]  

(3.4)

Figure 3.4 Static axle forces
Where

\[ L = \text{Length of wheel base while the motorcycle is level, m} \]
\[ L_f = \text{Horizontal distance from centre of gravity to front axle, m} \]
\[ W = \text{Total motorcycle weight} = mg, N \]

Solved for the horizontal distance \( L_f \) between front axle and CG

\[ L_f = \frac{F_{zR}}{W} L = \Psi L, \text{ m} \] \hspace{1cm} (3.5)

Similarly, for the horizontal distance \( L_r \) between the rear axle and the CG

\[ L_r = (1 - \Psi) L, \text{ m} \] \hspace{1cm} (3.6)

3.3.2 Dynamic Axle Loads

When the brakes are applied, the torque developed by the wheel brake is resisted by the tyre circumference where it comes in contact with the ground. Prior to brake lockup, the magnitude of the braking forces is a direct function of the torque produced by the wheel brake. For hydraulic brakes, Equation (3.7) is used for determining the actual braking forces.

\[ F_x = (P_L - P_0) \ast A \ast \eta \ast BF \left( \frac{r}{R} \right), \text{ N} \] \hspace{1cm} (3.7)

Where

\[ F_x = \text{Braking force, N} \]
\[ P_L = \text{Hydraulic brake line pressure, N/m}^2 \]
\[ P_0 = \text{Pushout pressure, required to bring brake pads in contact with disc, N/m}^2 \]

\[ A = \text{Caliper cylinder area, m}^2 \]

\[ \eta_c = \text{Caliper cylinder efficiency} \]

\[ BF = \text{Brake factor} \]

\[ r = \text{Effective radius of disc, m} \]

\[ R = \text{Effective rolling radius of tyre, m} \]

The forces acting on a motorcycle decelerating on a level road are illustrated in Figure 3.5. Application of moment balance about the rear tyre-to-ground contact patch yields the dynamic normal force \( F_{x, \text{dyn}} \) on the front axle:

\[ F_{x, \text{dyn}} = (1 - \Psi + \chi a)W, \text{ N} \quad (3.8) \]

Where

\[ a = \frac{F_{x, \text{total}}}{W} = \text{Motorcycle deceleration, g-units} \]

\[ F_{x, \text{total}} = \text{total braking force, N} \]

\[ F_{zR} = \text{Static rear axle load, N} \]

\[ W = \text{Total motorcycle weight, N} \]

\[ \chi = \frac{\text{CG height (h) divided by wheel base (L)}}{\text{CG height (h) divided by wheel base (L)}} \]

\[ \Psi = \frac{F_{zR}}{W} \]
Similarly, moment balance about the front tyre-to-ground contact point yields the dynamic rear axle normal force $F_{r\text{R,dyn}}$:

$$F_{r\text{R,dyn}} = (\Psi - \chi a) W, \ \text{N} \quad (3.9)$$

Inspection of Equations (3.8) and (3.9) reveals that the dynamic normal axle forces are linear functions of deceleration ‘a’, i.e., straight-line relationships. The amount of load transfer effect of the rear axle is given by the term $\chi a W$ in the Equations (3.8) and (3.9). The normal rear and front axle loads of a typical motorcycle are given in the Figures 3.6 and 3.7 for unladen, rider-only and additional every 10 kg cases, considering rider weight (50kg) as 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 785 N to only 295 N for a 1g stop at unladen (141 kg) condition, while the front load has increased from 589 N to 1080 N.

3.4 OPTIMUM BRAKING FORCES

3.4.1 Braking Traction Coefficient

The wheel brake torques generate braking or traction forces between the tyre and the ground. The ratio of braking force to dynamic axle load is defined as the traction coefficient $\mu_i$

$$\mu_i = \frac{F_i}{F_{zi,\text{dyn}}}$$  \hspace{1cm} (3.10)

Where

- $F_i$ = Dynamic axle braking force, N
- $F_{zi,\text{dyn}}$ = Dynamic axle normal force, N
- $i$ = designates front (F) or rear axle (R)

The traction coefficient is the level of tyre-road friction needed by the braked tyre so that it will just not lock up. The traction coefficient varies as either braking force or dynamic axle normal force change and, consequently, is a vehicle and deceleration-dependent parameter. In general, the front and rear axle traction coefficient will be different. Only when the numerical values of traction and tyre road friction coefficients are equal does the tire lock up.
Figure 3.6 Dynamic rear axle loads for unladen and various pillion loads
Figure 3.7 Dynamic front axle loads for unladen and various pillon loads
Figures 3.6 and 3.7 are combined and plotted as shown in Figure 3.8 for unladen and laden conditions for determining the condition of optimum motorcycle deceleration in ‘g’ units. It is determined that optimum deceleration in ‘g’ units for unladen condition is 0.28 and that for laden condition is 0.52.
3.4.2 Dynamic Braking Forces

Multiplication of the dynamic axle loads by the traction coefficient yields the dynamic braking forces $F_{df}$ for the front axle:

$$F_{df} = (1 - \Psi + \gamma a)W_{\mu_{\text{TF}}} \text{, N} \tag{3.11}$$

Similarly the dynamic braking force $F_{dr}$ for the rear axle is given by

$$F_{dr} = (\Psi - \gamma a)W_{\mu_{\text{TR}}} \text{, N} \tag{3.12}$$

Where,

$\mu_{\text{TF}} = $ Front traction coefficient.

$\mu_{\text{TR}} = $ Rear traction coefficient.

The tyre-road friction coefficient $\mu_f$ or $\mu_r$, existing at either the front or the rear tyre, is indicative of the ability of a road surface to allow traction to be produced for a given tyre and, as such, is a fixed number. A braked tyre will continue to rotate as long as the traction coefficient is less than the tyre-road friction coefficient, otherwise it will lock up. At the moment of incipient tyre lock up, the traction coefficient equals the tyre-road friction coefficient. When both the axles of the motorcycle are braked at sufficient levels so that the front and the rear wheels are operating at incipient or peak friction conditions, then the maximum traction capacity between the tyre road system is utilized. Under these conditions the motorcycle deceleration will be a maximum, since the traction coefficients of front and rear are equal, and are also equal to the motorcycle deceleration measured in g-units.
3.4.3 Optimum Straight-Line Braking

For straight-line braking on a level surface in the absence of any aerodynamic effects and rolling resistance of tyre, optimum braking in terms of maximizing motorcycle deceleration is defined by

\[ \mu_F = \mu_R = a \]  (3.13)

Where,

- \( a \) = Motorcycle deceleration, g-units.
- \( \mu_F \) = Front tyre-road friction coefficient.
- \( \mu_R \) = Rear tyre-road friction coefficient.

The optimum braking forces may be determined by setting the traction coefficient equal to motorcycle deceleration in Equations (3.11) and (3.12), resulting in the optimum braking force \( F_{x_F,\text{opt}} \) on the front axle

\[ F_{x_F,\text{opt}} = (1 - \Psi + \chi a)W_a, \text{ N} \]  (3.14)

And the optimum braking force \( F_{x_R,\text{opt}} \) on the rear axle

\[ F_{x_R,\text{opt}} = (\Psi - \chi a)W_a, \text{ N} \]  (3.15)

Inspection of Equations (3.14) and (3.15) reveals a quadratic relationship relative to deceleration ‘a’. The graphical representation is a parabola as illustrated in Figure 3.9 for different pillion loads on the motor cycle. Any point on the curve represents an optimum point identified by deceleration equal to friction coefficient. Inspection of the empty case reveals that for decelerations greater than 0.8 to 1g, optimum rear braking begins to decrease due to the significant load transfer onto the front axle. For the loaded case, the higher static rear axle load yields an increasing optimum rear brake force with higher deceleration.
The deceleration scales for the unladen and laden vehicle are different as shown in Figure 3.10. A simplification can be obtained by expressing the optimum braking forces relative to vehicle weight, or per one Newton of weight, by dividing the optimum braking forces by vehicle weight. Consequently, the optimum braking forces $F_{x\text{F, opt}}$ and $F_{x\text{R, opt}}$ are

Front: $\frac{F_{x\text{F, opt}}}{W} = (1 - \Psi + \gamma a)a$ \hspace{1cm} (3.16)

Rear: $\frac{F_{x\text{R, opt}}}{W} = (\Psi - \gamma a)a$ \hspace{1cm} (3.17)

The graphical representation of Equations (3.16) and (3.17) is illustrated in Figure 3.11 for the unladen and loaded cases. Any point on the optimum braking force curve represents optimum braking, i.e., a condition under which the tyre-road friction coefficient for the front and the rear equals motor cycle deceleration. For example, for the unladen vehicle and for a deceleration of 0.7 g, the relative optimum front braking force is 0.5 and the relative rear optimum braking force is 0.2. The corresponding approximate values for the laden condition are 0.45 and 0.25, respectively. If the actual brake torque balance front to rear were to be distributed according to the optimum ratios indicated, simultaneous front and rear brake lockup would occur, yielding minimum stopping distance.

Only one deceleration scale is used in Figure 3.11 for both loading conditions. The lines of constant deceleration run under an angle of 45 deg, assuming equal scales are used for the front and the rear braking forces. The reason for the 45-deg angle follows from Newton’s Second Law, expressed by the relative optimum braking forces as

$$\frac{F_{x\text{R, opt}}}{W} + \frac{F_{x\text{F, opt}}}{W} = a$$ \hspace{1cm} (3.18)
Figure 3.9 Dynamic braking forces for various loads
Figure 3.10 Dynamic braking forces unladen and laden
Inspection of Figure 3.11 reveals, for example, for the 0.8 g-deceleration line substituted into Equation (3.18), that for the unladen case: 0.58 + 0.22 = 0.8; and for the laden case: 0.55 + 0.25 = 0.8. The numerical values indicate that the front braking force decreased by 0.58 - 0.55 = 0.03 on the vertical axis, which is the increase experienced by the rear braking force on the horizontal axis. The result is a right-angled triangle with two sides of equal distance, namely the vertical and horizontal components, thus yielding a 45-degree slope for the deceleration lines.

Inspection of Equations (3.16) and (3.17) reveals that the optimum braking forces are only a function of the particular motor cycle geometry and weight data, i.e., $\Psi$ and $\gamma$, and vehicle deceleration $'a'$. They are not a function of the brake system hardware that has been installed.

To better match actual with optimum braking forces, it becomes convenient to eliminate motor cycle deceleration by solving Equation (3.16) for deceleration $'a'$, and substituting into Equation (3.18). The result is the general optimum braking forces equation.

![Figure 3.11 Normalized dynamic brake forces](image-url)
\[
\left( \frac{F_{SR,\text{opt}}}{W} \right) = \sqrt{\frac{(1-\Psi)^2}{4\chi^2} + \left( \frac{1}{\chi} \right)^2 \left( \frac{F_{SF,\text{opt}}}{W} \right) - \left( \frac{1-\Psi}{2\chi} \right) - \left( \frac{F_{SF,\text{opt}}}{W} \right)}
\]

Equation (3.19) allows computation of the appropriate optimum rear axle braking force associated with an arbitrarily specified (optimum) front braking force.

The graphical representation of Equation (3.19) is that of a parabola illustrated in Figure 3.12. The optimum curve located in the upper right quadrant represents braking, the lower left acceleration. Only the braking quadrant and then only the section exhibiting deceleration ranges of interest are of direct importance to brake engineers. The optimum curves shown in Figure 3.12 represent the section of interest relative to deceleration ranges encountered frequently.

The entire optimum braking/acceleration forces diagram, however, is used to develop useful insight and design methods for matching optimum and actual braking forces for brake design purposes. In addition, the methods will also be used in the reconstruction of actual motorcycle accidents involving braking and loss of directional stability due to premature rear brake lock up.

### 3.5 LINES OF CONSTANT FRICTION COEFFICIENT

For increasing deceleration, assuming that the tyre-road friction is high enough, the optimum braking of the rear axle begins to decrease and reaches zero where it intercepts the front braking axis. At this point the deceleration of the motorcycle is sufficiently high that the rear axle begins to lift off the ground due to excessive load transfer effect.
Similarly, in the case of increasing acceleration, the front axle begins to lift off the ground when the optimum acceleration curve intercepts the rear braking force axis.

The zero point on the front braking force axis is determined by setting the relative rear axle braking force equal to zero in Equation (3.19) and solving for the relative front braking force, resulting in

\[
\frac{F_{df, opt}}{W} = 0 \Rightarrow \frac{F_{dr}}{W} = \frac{-(1 - \Psi)}{\chi}
\]

Similarly, setting the relative front axle braking force equal to zero in Equation (3.19) yields

\[
\frac{F_{dr, opt}}{W} = 0 \Rightarrow \frac{F_{df}}{W} = \frac{\Psi}{\chi}
\]

Any point on the optimum braking forces curve represents the condition under which the front and the rear tyre-road friction coefficients are equal to each other as well as to the deceleration of the motor cycle. Under these conditions all available tyre-road friction is utilized for vehicle deceleration. For example, at the 0.7 g point the front and the rear tyre-road friction coefficients are also equal to 0.7.

At the respective zero points, the tyre traction forces, either braking or accelerating, are zero regardless of the level of friction coefficient existing between the tyre and the ground due to the normal forces between the tyre and the ground being zero.

A straight line connecting the zero point \(\frac{-(1 - \Psi)}{\chi}\) and a point of the optimum force curve represents a condition of constant coefficient of
friction between the front tyre and the ground. For example, connecting the zero point with the 0.7 g optimum point establishes a line of front tyre friction coefficient of 0.7 constant along the entire line.

Similarly, by connecting the rear zero point with points on the optimum curve, lines of constant rear tyre friction coefficient are obtained.

Inspection of Figure 3.12 reveals that the constant front friction line of 0.7 intercepts the front braking force axis with the rear braking force equal to zero at a deceleration of approximately 0.3859 g. In other words, when braking on a road surface having a tyre-road friction coefficient of 0.7 with the rear brake failed or disconnected, the front brakes are at the moment of lock up while the motor cycle decelerates at 0.3859 g.

On the other hand, when the front brake is disconnected, the rear brake locks up at a deceleration of approximately 0.33135 g while braking on a road surface with a 0.7 coefficient of friction, as indicated by the interception of the 0.7 constant rear friction line with the rear braking force axis.

The deceleration \( a_r \) achievable with the rear brake disconnected is derived from Newton's Second Law and Equation (3.11), however, with the traction coefficient being equal to the front tyre-road friction coefficient since the front brake is about to lock up:

\[
F_{af} = (1 - \Psi + \chi_{rb})_{rb}W = a_rW
\]

\[
(1 - \Psi)_{rb} = a_r(1 - \chi_{rb})
\]

Solving for deceleration \( a_r \) with the rear brakes disconnected, yields
\[ a_F = \frac{(1-\Psi)\mu_F}{1-\chi \mu_F}, \text{ g-units} \quad (3.21) \]

A similar derivation yields the deceleration \('a_R'\) with the front brakes disconnected:

\[ a_R = \frac{\Psi \mu_F}{1+\chi \mu_F}, \quad (3.22) \]

where \( \mu_F \) = Front tyre-road friction coefficient

\( a_F \) = Deceleration with the rear brake disconnected in ‘g’ units

\( \mu_F \) = Rear tyre-road friction coefficient

\( a_R \) = Deceleration with the front brake disconnected in ‘g’ units

It becomes convenient to use Equations (3.21) and (3.22) along with the optimum points to draw the lines of constant friction coefficient rather than using the zero points.

Figure 3.13 shows the lines of constant friction coefficient for the motor cycle under loaded condition.

It is concluded that normalized dynamic rear axle load is increased from 0.33135 to 0.37069 based on the motorcycle geometry. Hence the effective disc radius may be varied for controlling rear braking force based on pillion load in the motorcycle. The required rear wheel braking force for various pillion load is found out using Figures which are shown in Appendices (A3.1) to (A8.1).
3.6 CALCULATION OF EFFECTIVE DISC RADIUS

\[ F_{SR} = (P_L - P_0) \times A \times \eta_c \times BF \left( \frac{r}{R} \right) \text{, N} \tag{3.23} \]

Where \( F_{SR} \) = Dynamic braking force for rear axle (calculated from Figure 3.12 and Figure 3.13 for unladen and laden condition respectively).

\( P_L \) = Hydraulic brake line pressure (Approximately 6.20 N/mm\(^2\)).

\( P_0 \) = Pushout pressure, required to bring brake pads in contact with the disc (0.05 N/mm\(^2\)).

\( A \) = Caliper cylinder area (Diameter 32mm).

\( \eta_c \) = Caliper cylinder efficiency (Assumed 0.98).

\( BF \) = Brake factor (Assumed 0.7).

\( r \) = Effective radius of disc.

\( R \) = Effective rolling radius of tire (300 mm).

When the brake is applied, the braking force is linearly varied with time. But, in this research work the maximum braking force between tyre and ground (the product of normal reaction on wheel and friction coefficient between tyre and ground) is controlled based on pillion load on the two-wheeler by changing the effective radius of disc. For brakes without a booster, the brake system should be designed so that for a maximum pedal force of 425 to 489 N, a theoretical deceleration of 1g is achieved when the vehicle is loaded at GVW (Gross vehicle weight).

\[ P_L = \frac{F_p \times l_p \times 1_p}{A_{mc}} \tag{3.24} \]
Where,

\[
P_L = \text{Hydraulic brake line pressure (Measured 6.2 N/mm}^2\text{).}
\]

\[
F_p = \text{Pedal force, N}
\]

\[
I_p = \text{Pedal lever ratio = } \frac{145}{35} = 4.142.
\]

\[
\eta_p = \text{Pedal lever efficiency = 0.8 (approximately)}.
\]

\[
A_{mc} = \text{Master cylinder bore area = } 3.14 \times 17 \times 17 / 4 = 226.98 \text{ mm}^2.
\]

Pedal lever efficiency is defined as the ratio of the force developed in the pushrod of the master cylinder to the force applied in the brake pedal. In general there would be some losses due to the friction between master cylinder piston and cylinder. Moreover the return spring connected to the brake pedal develops a restoring force (opposite to applied pedal force) that must be balanced. Hence the net force acting on the master cylinder pushrod is less than the applied force.

Brake line fluid pressure is measured by using pressure gauge fitted in the brake fluid line, as shown in the Figure 3.14. Brake pedal force is calculated from that measured fluid pressure value.

Substituting all the values in the Equation (3.24), we get \( F_p = 425 \text{ N}. \)

![Figure 3.14 Determination of brake line pressure](image)
Using Equation (3.23), the effective radius of disc for various pillion loads on the motorcycle is determined and given in the Table 3.3. The radial movement of pad is to be adjusted from 54.84 mm (191kg -unladen condition) to 83.83 mm (261kg-Laden condition). Hence the total radial movement of pad is 29mm (say 30mm). Sample calculations are given in Appendix 2.

Table 3.3 Calculation of effective disc radius

<table>
<thead>
<tr>
<th>Weight (kg)</th>
<th>Loading Condition</th>
<th>F_{sr} / W (Deceleration in ‘g’ units)</th>
<th>Effective disc Radius (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>191</td>
<td>Rider (unladen)</td>
<td>0.33135</td>
<td>54.84</td>
</tr>
<tr>
<td>201</td>
<td>Rider +10kg</td>
<td>0.33697</td>
<td>58.69</td>
</tr>
<tr>
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<td>62.64</td>
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<td>66.68</td>
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<td>75.06</td>
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<td>83.83</td>
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</table>

3.7 SUMMARY

This chapter dealt with the measurement of the centre of gravity of the motorcycle for the different load conditions and analyzed the response of the two-wheeler due to the forces produced by the braking system. The optimum braking forces for the straight-line, level-surface braking process are presented. The effective disc radius is calculated based on pillion load on the two-wheeler.