APPENDIX 1

Derivation of Equation 3.1 (Determination of the center of gravity of motorcycle)

Taking moment balance with respect to a point at the contact patch between the rear tyre and the road surface as shown in Figure A1.1 yields:

\[
W_f = \frac{W \times L_f}{L} \tag{A1.1}
\]

\[
L_f = \frac{W_f \times L}{W} \tag{A1.2}
\]

Figure A1.1 Weight distribution of motorcycle

Figure A1.2 Weight distribution of motorcycle when rear axle is lifted up
It is assumed clockwise moment is positive and counter clockwise moment is negative. Taking moment balance with respect to point ‘O’ as shown in Figure A1.2 yields:

\[ W_{lf} \cdot L_n - W \cdot \cos \alpha \cdot L_e - W \cdot \sin \alpha \cdot H = 0 \]  \hspace{1cm} (A1.3)

\[ \cos \alpha = \frac{L_n}{L} \]  \hspace{1cm} (A1.4)

\[ \sin \alpha = \frac{H_2}{L} \]  \hspace{1cm} (A1.5)

\[ W_{lf} \cdot L_n - W \cdot \frac{L_n}{L} \cdot L_e - W \cdot \frac{H_2}{L} \cdot H = 0 \]  \hspace{1cm} (A1.6)

\[ H = \frac{W_{lf} \cdot L_n \cdot L}{W \cdot H_2} - \frac{L_n \cdot L_e}{H_2} \]  \hspace{1cm} (A1.7)

Substitute Equation (A1.2) in Equation (A1.7)

\[ H = \frac{W_{lf} \cdot L_n \cdot L}{W \cdot H_2} - \frac{L_n \cdot \frac{W_{lf} \cdot L}{W}}{H_2} \]  \hspace{1cm} (A1.8)

\[ = \frac{W_{lf} \cdot L_n \cdot L - L_n \cdot W_{lf} \cdot L}{W \cdot H_2} \]  \hspace{1cm} (A1.9)

\[ = \frac{(W_{lf} - W_{lf}) \cdot L_n \cdot L}{W \cdot H_2} \]  \hspace{1cm} (A1.10)

Let take, \( W_{lf} - W_{lf} = W_{2f} \)

\[ H = \frac{W_{2f} \cdot L_n \cdot L}{W \cdot H_2} \]  \hspace{1cm} (A1.11)

\[ h = H_1 + \frac{W_{2f} \cdot L_n \cdot L}{W \cdot H_2} \]  \hspace{1cm} (A1.12)
APPENDIX 2

Sample calculation for finding normal reaction on tyres, optimum braking force, braking force, effective radius of disc, deceleration of motorcycle when rear brake is disconnected and front brake is disconnected for unladen and laden conditions.

Table A2.1 Motorcycle geometry and braking system hardware data

<table>
<thead>
<tr>
<th>S1.No</th>
<th>Motorcycle geometry and braking system hardware data</th>
<th>Unladen</th>
<th>Loaded</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Total motorcycle Weight ( (\bar{W}) )</td>
<td>191 kg</td>
<td>261 kg</td>
</tr>
<tr>
<td>02</td>
<td>( F_{R} )</td>
<td>120 kg</td>
<td>186.5 kg</td>
</tr>
<tr>
<td>03</td>
<td>( h )</td>
<td>622.18 mm</td>
<td>662.10 mm</td>
</tr>
<tr>
<td>04</td>
<td>( L )</td>
<td>1330 mm</td>
<td>1330 mm</td>
</tr>
<tr>
<td>05</td>
<td>( \Psi )</td>
<td>0.6283</td>
<td>0.7146</td>
</tr>
<tr>
<td>06</td>
<td>( \chi )</td>
<td>0.4678</td>
<td>0.4978</td>
</tr>
<tr>
<td>07</td>
<td>( a ) in ‘g’ units</td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td>08</td>
<td>( \mu_{1R} )</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>09</td>
<td>( \mu_{1F} )</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>10</td>
<td>( P_{L} )</td>
<td>6.20 N/mm²</td>
<td>6.20 N/mm²</td>
</tr>
<tr>
<td>11</td>
<td>( P_{0} )</td>
<td>0.05 N/mm²</td>
<td>0.05 N/mm²</td>
</tr>
<tr>
<td>12</td>
<td>( A )</td>
<td>803.84 mm²</td>
<td>803.84 mm²</td>
</tr>
<tr>
<td>13</td>
<td>( \mu_{F} )</td>
<td>0.98</td>
<td>0.98</td>
</tr>
<tr>
<td>14</td>
<td>BF</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>15</td>
<td>( R )</td>
<td>300 mm</td>
<td>300 mm</td>
</tr>
</tbody>
</table>

Sample calculation for Unladen Condition (191kg)

\[
\Psi = \frac{F_{R}}{\bar{W}} = \frac{120}{191} = 0.6283
\]
\[ \chi = \frac{h}{L} \]
\[ = \frac{622.18}{1330} \]
\[ = 0.4678 \]

\[ F_{x, \text{Dyn}} = (1 - \Psi + \chi)aW \]
\[ = (1 - 0.6283 + 0.4678 \times 0.4)191 \times 9.81 \]
\[ = 1047.06 \text{N} \]

\[ F_{x, \text{Dyn}} = (1 - \Psi + \chi)aW \]
\[ = (0.6283 - 0.4678 \times 0.4)191 \times 9.81 \]
\[ = 826.64, \text{N} \]

\[ F_{x, \text{opt}} = (1 - \Psi + \chi)aW \]
\[ = (1 - 0.6283 + 0.4678 \times 0.4)191 \times 9.81 \times 0.4 \]
\[ = 418.82, \text{N} \]

\[ F_{x, \text{opt}} = (\Psi - \chi)aW \]
\[ = (0.6283 - 0.4678 \times 0.4)191 \times 9.81 \times 0.4 \]
\[ = 330.68, \text{N} \]

\[ a_F = \frac{(1 - \Psi_{1F})}{1 - \chi_{1F}} \]
\[ = \frac{(1 - 0.6283)0.7}{1 - 0.4678 \times 0.7} \]
\[ = 0.3869 \]

\[ a_R = \frac{\Psi_{1R}}{1 + \chi_{1R}} \]
\[ = \frac{0.6283 \times 0.7}{1 + 0.4678 \times 0.7} \]
\[ = 0.3313 \]

\[ F_{x, R} = (P_L - P_0) \times A \times \eta_k \times BF \left( \frac{r}{R} \right) \]
\[ = 0.3313 \times 191 \times 9.81 = (6.2 - 0.05) \times \frac{3.14}{4} \times 32 \times 32 \times 0.98 \times 0.7 \times \left( \frac{r}{300} \right) \]
\[ r = 54.91 \text{mm} \]
Sample calculation for loaded Condition (261 kg)

\[ \Psi = \frac{F_R}{W} \]  
\[ = \frac{186.5}{261} \]  
\[ = 0.7146 \]  

\[ \chi = \frac{h}{L} \]  
\[ = \frac{662.10}{1330} \]  
\[ = 0.4978 \]  

\[ F_{F, \text{dyn}} = (1 - \Psi + \chi a) W \]  
\[ = (1 - 0.7146 + 0.4978 * 0.4) 261 * 9.81 \]  
\[ = 1240.57, N \]  

\[ F_{F, \text{opt}} = (\Psi - \chi a) W * a \]  
\[ = (0.7146 - 0.4978 * 0.4) 261 * 9.81 \]  
\[ = 496.23, N \]  

\[ F_{R, \text{opt}} = (\Psi - \chi a) W * a \]  
\[ = (0.7146 - 0.4978 * 0.4) 261 * 9.81 * 0.4 \]  
\[ = 527.94, N \]  

\[ a_F = \frac{(1 - \Psi)_{\text{IF}}}{1 - \chi_{\text{IF}}} \]  
\[ = \frac{(1 - 0.7146) 0.7}{1 - 0.4978 * 0.7} \]  
\[ = 0.3066 \]  

\[ a_R = \frac{\Psi_{\text{IF}}}{1 + \chi_{\text{IF}}} \]  
\[ = \]
\[
\frac{0.7146 \times 0.7}{1 + 0.4978 \times 0.7} = 0.3709
\]

\[F_R = (P_L - P_g) \times A \times \eta_p \times BF \left( \frac{r}{R} \right)\]  \hspace{1cm} (A2.18)

\[
0.3709 \times 261 \times 9.81 = (6.2 - 0.05) \times \frac{3.14}{4} \times 32 \times 32 \times 0.98 \times 0.7 \times \left( \frac{r}{300} \right)
\]

\[
r = 84.02 \text{ mm}
\]
APPENDIX 3

Figure A3.1 Parabola of normalized dynamic braking and driving forces for load (201 kg) condition
APPENDIX 4

Figure A4.1 Parabola of normalized dynamic braking and driving forces for load (211 kg) condition
Figure A5.1  Parabola of normalized dynamic braking and driving forces for load (221 kg) condition
Figure A6.1 Parabola of normalized dynamic braking and driving forces for load (231 kg) condition
Figure A7.1  Parabola of normalized dynamic braking and driving forces for load (241 kg) condition
Figure A8.1 Parabola of normalized dynamic braking and driving forces for load (251 kg) condition
APPENDIX 9

Motorcycle deceleration, braking efficiency and stopping distance calculation for unladen and laden condition.

Table A9.1 Braking system hardware data.

<table>
<thead>
<tr>
<th>Sl. No</th>
<th>Motorcycle geometry and braking system hardware data</th>
<th>Unladen</th>
<th>Loaded</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Total motorcycle Weight (W)</td>
<td>191 kg</td>
<td>261 kg</td>
</tr>
<tr>
<td>02</td>
<td>$F_{sr}$</td>
<td>120 kg</td>
<td>186.5 kg</td>
</tr>
<tr>
<td>03</td>
<td>h</td>
<td>622.18 mm</td>
<td>662.10 mm</td>
</tr>
<tr>
<td>04</td>
<td>L</td>
<td>1330 mm</td>
<td>1330 mm</td>
</tr>
<tr>
<td>05</td>
<td>$\Psi$</td>
<td>0.6283</td>
<td>0.7146</td>
</tr>
<tr>
<td>06</td>
<td>$\chi$</td>
<td>0.4678</td>
<td>0.4978</td>
</tr>
<tr>
<td>07</td>
<td>a in 'g' units</td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td>08</td>
<td>$\mu^v$</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>09</td>
<td>$\mu^r$</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>10</td>
<td>$P_L$</td>
<td>6.20 N/mm$^2$</td>
<td>6.20 N/mm$^2$</td>
</tr>
<tr>
<td>11</td>
<td>$P_0$</td>
<td>0.05 N/mm$^2$</td>
<td>0.05 N/mm$^2$</td>
</tr>
<tr>
<td>12</td>
<td>A</td>
<td>803.84 mm$^2$</td>
<td>803.84 mm$^2$</td>
</tr>
<tr>
<td>13</td>
<td>$\eta_f$</td>
<td>0.98</td>
<td>0.98</td>
</tr>
<tr>
<td>14</td>
<td>BF</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>15</td>
<td>R</td>
<td>300 mm</td>
<td>300 mm</td>
</tr>
</tbody>
</table>

When the brakes are applied, the kinetic energy of the motorcycle is converted into heat energy between the tire and the ground. Hence the total kinetic energy of the motorcycle is equal to the frictional energy developed between the tire and the ground.

Kinetic energy = Frictional energy

$$\frac{1}{2} \cdot m \cdot V^2 = F_{sr} \cdot SD \quad (A9.1)$$

Where, $m$ = Total mass of motorcycle
\( V \) = Motorcycle velocity

\( F_{xR} \) = Dynamic braking force for rear axle, N

\( SD \) = Stopping distance

Since only the rear wheel is braked, \( F_{xR} \) alone is considered for frictional force between the tyre and ground.

\( F_{xR} \) is derived from Newton’s second law as follows

\[
F = m \times a_1 \quad \text{(A9.2)}
\]

Where, \( F \) = Braking force between tyre and road surface (N)

\( m \) = Total motorcycle mass (kg)

\( a_1 \) = Motorcycle deceleration (m/s\(^2\))

Since only the rear brake is applied and in the absence of aerodynamic force and rolling resistance of tyre,

\[
F = F_{xR} \quad \text{(A9.3)}
\]

Where, \( F_{xR} \) is dynamic braking force for rear axle

Substituting equation (A9.3) in (A9.2)

\[
F_{xR} = m \times a_1 \quad \text{(A9.4)}
\]

\[
F_{xR} \leq F_{zR,dyn} \times \mu_R \quad \text{Without wheel sliding} \quad \text{(A9.5)}
\]

Where \( F_{zR,dyn} \) = Dynamic rear axle normal force

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

\[
F_{zR,dyn} = (\Psi - \chi_a R)W \quad \text{(A9.6)}
\]
Where, 

\( W \) = Total motorcycle weight, N  
\( \chi \) = CG height (h) divided by wheel base (L)  
\( \Psi \) = The ratio of static rear axle load to the total vehicle weight  
\( a_r \) = Deceleration in ‘g’ units with front brake disconnected  

Substituting Equation (A9.6) in (A9.5), 

\[ F_{sr} \leq (\Psi - \chi a_r)W \]  

(A9.7)  

The magnitude of the braking force developed between the tyre and the ground by braking system hardware must be less or equal to 

\( (\Psi - \chi a_r)W \) for without wheel sliding. If the magnitude is less, the deceleration of vehicle is increased, that leads to increase the stopping distance. If the magnitude is more, the wheel is about to lock up and it leads to increase the stopping distance. If it is equal, the maximum traction capacity between the tyre and the ground is utilized, that leads to increase vehicle deceleration and reduces the stopping distance of the vehicle. 

\[ m = \frac{W}{g} \]  

(A9.8)  

Substituting Equations (A9.7) and (A9.8) in (A9.4), 

\[ (\Psi - \chi a_r)W = \frac{W}{g}a_i \]  

(A9.9)  

\[ a_r = \frac{\Psi W}{\chi W + a_i} \]  

(A9.10)
Where, \( \frac{a_i}{g} = a_g \)

Substituting Equations (A9.10) in Equation (A9.6)

\[
F_{zR,dyn} = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W
\]

(A9.11)

Substituting Equation (A9.11) in Equation (A9.5)

\[
F_{sR} = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W \cdot \mu_R
\]

(A9.12)

Substituting Equation (A9.12) and (A9.8) in Equation (A9.1)

\[
\frac{1}{2} \cdot \frac{W}{g} \cdot V^2 = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W \cdot \mu_R \cdot SD
\]

(A9.13)

\[
SD = \frac{V^2 (1 + \gamma_{LR})}{2 \cdot g \cdot \Psi \cdot \mu_R}
\]

(A9.14)

From the Equation (A9.14) we get the stopping distance (SD) for unladen and laden condition

\[ V = 40 \text{ kmph} = 11.11 \text{ m/s} \]

Unladen (\( m = 191\text{kg} \))

\[
SD = \frac{(11.11)^2 \cdot (1 + 0.4678 \cdot 0.7)}{2 \cdot 9.81 \cdot 0.6283 \cdot 0.7}
= 18.988 \text{m}
\]

Laden (\( m = 261 \text{ kg} \))
\[ \text{SD} = \frac{(11.11)^2 \times (1 + 0.4978 \times 0.7)}{2 \times 9.81 \times 0.7146 \times 0.7} \]
\[ = 16.959 \text{m} \]

\[ \text{V} = 50 \text{ kmph} = 13.89 \text{ m/s} \]

Unladen (\( m = 191 \text{ kg} \))

\[ d = \frac{(13.89)^2 \times (1 + 0.4678 \times 0.7)}{2 \times 9.81 \times 0.6283 \times 0.7} \]
\[ = 29.68 \text{m} \]

Laden (\( m = 261 \text{ kg} \))

\[ \text{SD} = \frac{(13.89)^2 \times (1 + 0.4978 \times 0.7)}{2 \times 9.81 \times 0.7146 \times 0.7} \]
\[ = 26.508 \text{m} \]

\[ \text{V} = 60 \text{ kmph} = 13.89 \text{ m/s} \]

Unladen (\( m = 191 \text{ kg} \))

\[ \text{SD} = \frac{(16.66)^2 \times (1 + 0.4678 \times 0.7)}{2 \times 9.81 \times 0.6283 \times 0.7} \]
\[ = 42.698 \text{m} \]

Laden (\( m = 261 \text{ kg} \))

\[ \text{SD} = \frac{(16.66)^2 \times (1 + 0.4978 \times 0.7)}{2 \times 9.81 \times 0.7146 \times 0.7} \]
\[ = 38.135 \text{m} \]

Motorcycle deceleration, braking efficiency, and stopping distance for various effective disc radius:
The maximum braking force developed between the tire and ground

is equal to

$$F_{sr} = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W \cdot \mu_{LR}$$

For unladen condition

$$F_{sr} = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W \cdot \mu_{LR}$$

$$= \left[ \frac{0.6283}{1 + 0.4678 \times 0.7} \right] \cdot 191 \times 9.81 \times 0.7$$

$$= 620.79 \text{ N}$$

For loaded condition

$$F_{sr} = \left[ \frac{\Psi}{1 + \gamma_{LR}} \right] \cdot W \cdot \mu_{LR}$$

$$= \left[ \frac{0.7146}{1 + 0.4978 \times 0.7} \right] \cdot 261 \times 9.81 \times 0.7$$

$$= 949.80 \text{ N}$$

But the magnitude of $F_{sr}$ depends on braking system hardware that is equal to

$$F_{sr} = (P_L - P_0) \cdot A \cdot \eta_c \cdot BF \left( \frac{r}{R} \right) \quad (A9.15)$$

Where

$F_{sr}$ = Rear axle Braking force

$P_L$ = Hydraulic brake line pressure (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).

For an effective disc radius of 10 mm \(r = 10\) mm

Substituting \(r = 10\) mm in Equation (A9.15)

\[
F_{sr} = (P - P_o) \times A \times \eta_k \times BF \left(\frac{r}{R}\right)
\]

\[
= (6.2 - 0.05) \times \frac{3.14}{4} \times 32 \times 32 \times 0.98 \times 0.7 \times \left(\frac{r}{300}\right)
\]

\[
= 113.04, \text{N}
\]

For unladen condition \((m = 191 \text{ kg})\)

\[
F_{sr} = m \times a_i \quad (A9.16)
\]

\[
a_i = \frac{F_{sr}}{m}
\]

\[
a_i = 0.592 \text{ m/s}^2
\]

Braking efficiency \(\frac{a_i}{g}\) \(= 0.06\)

\(= 6\%\)

Stopping distance \(= \frac{V^2}{2 \times a_i}\) \(\quad (A9.18)\)

\(V = 40 \text{ kmph}\)
Stopping distance \(= \frac{V^2}{2a_i} \)
\[= \frac{(11.11)^2}{2 \times 0.592} \]
\[= 104.25 \text{ m} \]

\(V = 50 \text{ kmph}\)

Stopping distance \(= \frac{V^2}{2a_i} \)
\[= \frac{(13.88)^2}{2 \times 0.592} \]
\[= 162.92 \text{ m} \]

\(V = 60 \text{ kmph}\)

Stopping distance \(= \frac{V^2}{2a_i} \)
\[= \frac{(16.66)^2}{2 \times 0.592} \]
\[= 234.61 \text{ m} \]

For laden condition \((m = 261 \text{ kg})\)

\(F_{sr} = m \times a_i \)
\[a_i = \frac{F_{sr}}{m} \]
\[a_i = 0.433 \text{ m/s}^2 \]

Braking efficiency \(= \frac{a_i}{g} \)
\[= 0.044 \]
\[= 4.4 \% \]
Stopping distance = \( \frac{V^2}{2a_i} \)

\( V = 40 \text{ kmph} \)

\[
\text{Stopping distance} = \frac{V^2}{2a_i} = \frac{(11.11)^2}{2 \times 0.433} = 142.53 \text{ m}
\]

\( V = 50 \text{ kmph} \)

\[
\text{Stopping distance} = \frac{V^2}{2a_i} = \frac{(13.88)^2}{2 \times 0.433} = 222.46 \text{ m}
\]

\( V = 60 \text{ kmph} \)

\[
\text{Stopping distance} = \frac{V^2}{2a_i} = \frac{(16.66)^2}{2 \times 0.433} = 320.50 \text{ m}
\]

For an effective disc radius of 10 mm the braking force (\( F_{\text{sr}} = 113.04 \text{ N} \)) developed by the braking system hardwares, between the tyre and the ground is less than maximum traction capacity (\( F_{\text{sr, max}} = 620.79 \text{ N} \)) between the tyre and the ground. So the motorcycle deceleration decreases leading to an increase in the motorcycle stopping distance.
For an effective disc radius of 55 mm ($r = 55\,\text{mm}$)

\[
F_{\text{fr}} = (P_L - P_0) \ast A \ast \eta_k \ast BF \left(\frac{r}{R}\right)
\]

\[
= (6.2 - 0.05) \ast \frac{3.14}{4} \ast 32 \ast 32 \ast 0.98 \ast 0.7 \ast \left(\frac{r}{300}\right)
\]

\[
= 621.74, \text{N}
\]

For unladen condition ($m = 191\,\text{kg}$)

\[
F_{\text{fr}} = m \ast a_1
\]

\[
a_1 = \frac{F_{\text{fr}}}{m}
\]

\[
a_1 = 3.255\,\text{m/s}^2
\]

Braking efficiency $= \frac{a_1}{g}$

\[
= 0.33
\]

\[
= 33\%
\]

Stopping distance $= \frac{V^2}{2 \ast a_1}$

$V = 40\,\text{kmph}$

\[
\text{Stopping distance} = \frac{V^2}{2 \ast a_1}
\]

\[
= \frac{(11.11)^2}{2 \ast 3.255}
\]

\[
= 18.96\,\text{m}
\]
V = 50 kmph

Stopping distance = \( \frac{V^2}{2a_1} \)

\( = \frac{(13.88)^2}{2 \times 3.255} \)
\( = 29.59 \text{ m} \)

V = 60 kmph

Stopping distance = \( \frac{V^2}{2a_1} \)

\( = \frac{(16.66)^2}{2 \times 3.255} \)
\( = 42.64 \text{ m} \)

For laden condition (m = 261 kg)

\( F_{sr} = m \times a_1 \)
\( a_1 = \frac{F_{sr}}{m} \)
\( a_1 = 2.383 \text{ m/s}^2 \)

Braking efficiency = \( \frac{a_1}{g} \)
\( = 0.2428 \)
\( = 24.28 \% \)

Stopping distance = \( \frac{V^2}{2a_1} \)

V = 40 kmph
Stopping distance = $\frac{V^2}{2a_i}$

$= \frac{(11.11)^2}{2 \times 2.383}$

$= 25.89 \, \text{m}$

$V = 50 \, \text{kmph}$

Stopping distance = $\frac{V^2}{2a_i}$

$= \frac{(13.88)^2}{2 \times 2.383}$

$= 40.43 \, \text{m}$

$V = 60 \, \text{kmph}$

Stopping distance = $\frac{V^2}{2a_i}$

$= \frac{(16.66)^2}{2 \times 2.383}$

$= 58.25 \, \text{m}$

For an effective disc radius of 55 mm the braking force ($F_{\text{br}} = 621.74 \, \text{N}$) developed by braking system hardware, between the tyre and the ground is equal to traction capacity ($F_{\text{tr}, \text{max}} = 621.72 \, \text{N}$) between the tyre and the ground. Also, the full traction capacity available between the tyre and the ground is utilized. So the motorcycle deceleration decreases leading to an increase in the motorcycle’s stopping distance.

From Table A9.2, braking efficiency is improved by 37% for unladen condition compared with laden condition at an effective disc radius of 55mm because the maximum traction capacity between the tyre and the
The ground is utilized. Also, if the effective disc radius is increased at the unladen condition, it leads to wheel lock up. The stopping distance of the motorcycle for locked wheel cannot be predicted easily since it depends on various parameters of the tyre and the wheel slip. Hence the theoretical stopping distance for an effective radius more than 55 mm is not given in table A9.3.

Similarly the braking efficiency for laden condition is high (37.08%) at an effective disc radius of 84 mm compared with the different effective disc radius because the maximum traction capacity between the tyre and the ground is utilized at the disc radius of 84 mm. Also, if the effective disc radius is increased further at the laden condition, it leads to wheel lock up. The stopping distance of the motorcycle with locked wheel cannot be predicted easily since it depends on various parameters of the tyre and the wheel slip. Hence the theoretical stopping distance for an effective radius more than 85 mm is not given in Table A9.3.

If the effective disc radius is more than 55 mm for unladen condition, the wheel locks during the brake test. The rider observed that wheel locked at an effective radius of 70mm, 84mm and 110mm. But, the wheel prematurely locked at an effective radius of 110 mm compared with the wheel locking at an effective disc radius of 84 and 70mm. So the stopping distance is increased slightly for an effective disc radius of 70mm, 84mm, and 110.

The braking force (621.72N) developed for the effective radius of 55mm, 65mm, and 84mm are same at unladen condition. But it is important that how fast (time duration) the braking force (621.72N) developed which influence the stopping distance of the motorcycle i.e. for an effective radius of 84mm, the braking force (621.72N) developed very fast (Braking system takes less time to develop 621.74 N of force) which leads earlier wheel lock-up, compared with 65mm wheel lock-up.
Table A9.2 Motorcycle deceleration and braking efficiency

<table>
<thead>
<tr>
<th>Sl No</th>
<th>Loading condition</th>
<th>Effective disc radius (mm)</th>
<th>Braking force Developed (N)</th>
<th>Maximum traction capacity (N)</th>
<th>Motorcycle Deceleration (m/s²)</th>
<th>Braking efficiency</th>
<th>Braking efficiency %</th>
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<tr>
<td>01</td>
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<td>113.04</td>
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<td>621.72</td>
<td>----</td>
<td>---</td>
<td>---</td>
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<td>949.53</td>
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<td>0.3708</td>
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Table A9.3 Stopping distance for various effective disc radius

<table>
<thead>
<tr>
<th>Sl No</th>
<th>Loading condition</th>
<th>Effective disc radius (mm)</th>
<th>Braking force Developed (N)</th>
<th>Maximum traction capacity (N)</th>
<th>SD V=40kmph (m)</th>
<th>SD V=50kmph (m)</th>
<th>SD V= 60kmph (m)</th>
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<td>---</td>
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APPENDIX 10

Relation between tractive (or) braking effort and longitudinal slip of tyres

When a driving torque is applied to a pneumatic tyre, a tractive force is developed at the tyre-ground contact patch as shown in Figure A10.1. At the same time, the tyre treads in front of and within the contact patch is subjected to compression. A corresponding shear deformation of the side wall of the tyre is also developed.

Figure A10.1 Behavior of a tyre under the action of a driving torque

As tread elements are compressed before entering the contact region, the distance that the tyre travels when subject to a driving torque will
be less than that in free rolling. This phenomenon is usually referred to as deformation slip. The slip of the vehicle running gear, when a driving torque is applied, is usually defined by

\[ i = \left( 1 - \frac{V}{r_\omega} \right) \times 100\% = \left( 1 - \frac{R}{r_T} \right) \times 100\% \]  \hspace{1cm} (A10.1)

Where V is the translatory speed of the tyre center (Motorcycle velocity), \( \omega \) is angular velocity of the wheel, R is the effective rolling radius of the tyre, and \( r_T \) is the rolling radius of the tyre, which is the ratio of the translatory speed of the tyre center to the angular speed of the tyre. When a driving torque is applied, the tyre rotates without the equivalent translatory progression; therefore, \( R \omega > V \).

As the tractive force (or braking force) developed by a tyre is proportional to the applied wheel torque under steady-state conditions, slip is a function of tractive effort. Generally speaking, at first the wheel torque and tractive force increase linearly with slip, because initially slip is mainly due to elastic deformation of the tire tread. This corresponds to section OA of the curve shown in Figure 2. Further increase of wheel torque and tractive force results in part of the tyre tread sliding on the ground. Under these circumstances, the relationship between the tractive force and the slip is nonlinear. This corresponds to section AB of the curve shown in Figure A10.2. Based on an available experimental data, the maximum tractive force of a pneumatic tyre on hard surface is usually reached somewhere between 15 and 20% of slip. Any further increase of slip beyond that results in an unstable condition, with the tractive effort to the vertical load of a tyre falling rapidly from the peak value \( \mu_p \) to the pure sliding value \( \mu_s \) as shown in Figure A10.2.
A general theory that can accurately predict the relationship between the tractive effort and the longitudinal slip of pneumatic tyres on hard surfaces has yet to be evolved. However, several theories have been proposed that could provide a basic understanding of the physical nature of the processes involved. One of the earliest and the simplest theoretical treatments on the relationship between the tractive effort and the longitudinal slip of pneumatic tyres was made by Julian et al.

In Julian’s theory, it is assumed that the tyre tread can be regarded as an elastic band, and that the contact patch is rectangular and the normal pressure is uniformly distributed. Consider that a driving torque is applied to a tyre. In the region in front of the contact patch, the driving torque produces a longitudinal strain $\varepsilon$ (in compression) in the tread. It remains constant in the adhesion region of the contact patch, where no sliding between the tyre tread...
and the ground takes place. Let $e_0$ be the longitudinal deformations of the tyre tread in front of the contact patch, and let ‘e’ be the longitudinal deformation of the tread at a distance of $x_t$ behind the front contact edge

$$e = e_0 + x_t \varepsilon$$

(A10.2)

Assume that $e_0$ is proportional to $\varepsilon$, and $e_0 = \lambda \varepsilon$. Then

$$e = (\lambda + x_t)\varepsilon$$

(A10.3)

It is further assumed that within the adhesion region, where no sliding between the tyre tread and the ground occurs, the tractive force per unit contact length is proportional to the deformation of the tread. Thus

$$\frac{dF_t}{dx} = k_i e = k_i (\lambda + x_t)\varepsilon$$

(A10.4)

Where $k_i$ is the tangential stiffness of the tyre tread and $F_t$ is the tractive force.

The tractive force developed between o and x is then given by

$$F_t = \int_{0}^{X} k_i (\lambda + x_t)dx = k_i \lambda X \varepsilon \left(1 + \frac{X}{2\lambda}\right)$$

(A10.5)

Let $p_i$ be the normal pressure, $b$ be the width of the contact patch, and $\mu_p$ be the peak value of the coefficient of road adhesion. Then the condition for adhesion, that is no sliding occurs between the tread and the ground, is

$$\frac{dF_t}{dx} = k_i (\lambda + x_t)\varepsilon < p_i b \mu_p$$

(A10.6)
This implies that if a point at a distance of \( x_t \) behind the front edge is in the adhesion region, then \( x_t \) must be less than a characteristic length \( l_c \), tread and the ground takes places, that is

\[
x_t \leq l_c = \frac{p}{k} \frac{\mu_p}{\varepsilon} - \lambda = \frac{\mu_p W_t}{l_c k \varepsilon} - \lambda
\]  

(A10.7)

Where \( W_t \) is the normal load on the tire and \( l_c \) is the contact length of the tyre.

If \( l_t < l_c \) then the entire contact area is in the adhesion region. Putting \( X = l_t \) in Equation 10.5, the tractive force becomes

\[
F_t = k \lambda l_c \left( 1 + \frac{l_t}{2l_c} \right) = K_t \varepsilon
\]  

(A10.8)

Where, \( K_t = k \lambda l_c \left[ 1 + \left( \frac{l_t}{2l_c} \right) \right] \)

Since the longitudinal strain \( \varepsilon \) is a measure of the deformation slip of the tyre, it is concluded that if the entire contact patch is in the adhesion region, the relationship between tractive force \( F_t \) and slip \( i \) is linear. This corresponds to the region between points O and A on the tractive effort coefficient-slip curve shown in Figure (A10.2).

The condition for sliding at the rear edge of the contact area is given by

\[
l_t = l_c = \frac{\mu_p W_t}{l_c k \varepsilon} - \lambda
\]  

(A10.9)

This means that if the slip and tractive force reach the critical values \( i_c \) and \( F_c \) given below, sliding in the trailing part of the contact patch begins.

\[
i_c = \frac{\mu_p W_t}{l_c k (l_t + \lambda)}
\]  

(A10.10)
A further increase of slip or tractive force above the respective critical values results in the spread of the sliding region from the trailing edge toward the leading part of the contact patch. The tractive force developed at the sliding region \( F_{xs} \) is given by

\[
F_{xs} = \mu_p W_t \left( 1 + \frac{1}{2\lambda_c} \right)
\]  
(A10.12)

And the tractive force at the adhesion region \( F_{xa} \) is given by

\[
F_{xa} = k_{\lambda_c} l_c \left( 1 + \frac{1}{2\lambda} \right)
\]  
(A10.13)

Where \( l_c \) is determined by Equation (A10.7)

Hence the relationship between the total tractive force and slip when part of the tyre tread sliding on the ground is expressed by

\[
F_t = F_{xs} + F_{xa} = \mu_p W_t - \frac{\lambda \left( \mu_p W_t - K' \right)}{2l_c K'}
\]  
(A10.14)

Where \( K' = k_{\lambda_c} l_c \)

Equation (A10.14) clearly indicates the nonlinear behavior of the tractive effort coefficient –longitudinal slip relationship when sliding occurs in part of the contact area. This corresponds to the region between A and B of the curve shown in Figure A10.2.
When sliding extends over the entire contact patch, the tractive force $F_t$ is equal to $\mu_p W_t$. Under this condition the slip $i$ is obtained by setting $l_c$ equal to zero in 8. The value of the maximum deformation slip $i_m$ is equal to $\frac{\mu_p W_t}{l_t k_c \varepsilon}$ and corresponds to point B shown in Figure A10.2. A further increase of tyre slip results in an unstable situation with the value of the coefficient of road adhesion falling rapidly from the peak value $\mu_p$ to the pure sliding value $\mu_s$.

In practice the normal pressure distribution over the tyre ground contact patch is not uniform. There is a gradual drop of pressure near the edges. It is expected, therefore, that a small sliding region will be developed in the trailing part of the contact area even at extremely low tractive effort.

When a braking torque is applied to the tyre, the stretching of the tread elements occurs prior to entering the contact area as shown in figure 3, in contrast with the compression effect for a driven tyre. The distance that the tyre travels when a braking torque is applied, therefore, will be greater than that in free rolling. The severity of braking is often measured by the skid of the tyre $i_s$ which is defined as

$$i_s = \left(1 - \frac{r\omega}{V}\right) \times 100\% = \left(1 - \frac{r}{R}\right) \times 100\%$$ (A10.15)

For a locked wheel, the angular speed $\omega$ of the tyre is zero. Whereas the transtatory speed of the tyre center is not zero. Under this condition, the skid is denoted 100%.
Figure A10.3 Behavior of a tyre under the action of a braking torque

Usually the relationship between the braking effort coefficient, which is defined as the ratio of the braking effort to the normal load of the tyre, and skid exhibits similar characteristics to that between the tractive effort coefficient and slip discussed previously. Figure A10.3 shows the variation of the braking effort coefficient with skid for a bias-ply passenger car tyre over various surfaces.

Surface conditions as well as tyre design and construction are the most important factors affecting the coefficient of road adhesion. Among the operational parameters speed and vertical load have noticeable effects on the tractive (braking) effort-slip (skid) characteristics. Figure A10.4 shows the influence of speed on the braking effort coefficient- skid characteristics of a
truck tyre on asphalt. The effect of vertical load on the braking effort-skid relationship is shown in Figure A10.5.

Figure A10.4 Effect of speed on the braking performance of a truck
10.00x20/F on asphalt
Figure A10.5 Effect of normal load on the braking performance of a truck 10.00x20/F on asphalt

From the above discussion it is concluded that the braking (or tractive) effort is mainly depends on tractive effort coefficient, normal reaction on tyre, vehicle speed, wheel skid, tangential stiffness of the tyre, contact length of the tyre, cornering stiffness of tyre. Hence, the theoretical calculation of the stopping distance for locked wheel may not be so easy to determine.
APPENDIX 11

Design of ‘C’ clamp

Point ‘EGBCHF’ denotes the ‘C’ clamp in Figure A11.1.

Point ‘AEFD’ denotes the caliper which is fixed at the end of ‘C’ clamp using bolts.

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

Diameter of caliper cylinder \( (d_{cc}) = 32 \text{ mm} \)

Caliper cylinder area \( (A) = 804 \text{ mm}^2 \)

\[
P = P_L \times A \tag{A11.1}
\]

Where, \( P = \) Force acting on each halve of the caliper

\( P = 4986 \text{ N} \)

\( L_3 = L_4 = 30 \text{ mm} \)

\( I_3 = I_4 = 90 \times 35^3/12 \text{ mm}^4 \)
Consider the member ‘BC’ which is subject to axial load and bending moment as shown in Figure A11.2. FEA (Finite element analysis) model of member ‘BC’ is given in Figure A11.3.

**Figure A11.2 Free body diagram of member ‘BC’**

The member ‘BC’ should withstand the axial load developed by brake fluid pressure. So the axial deflection should be very minimal for
effective braking. Elongation in the member ‘BC’ on both sides is assumed as 0.000208 mm because the member ‘BC’ is considered as rigid member.

\[
\text{Elongation in the member ‘BC’} = \frac{P \cdot L_{BC}}{2 \cdot A_{BC} \cdot E} \quad (A11.2)
\]

Where, \( E \) = Young’s modulus.

\( A_{BC} = \text{Cross sectional area of member ‘BC’} \)

\[
0.000208 = \frac{4986 \cdot 45}{2 \cdot A_{BC} \cdot 2 \times 10^5}
\]

\( A_{BC} = 2696 \text{ mm}^2 \)

So, the cross section is selected as 30 X 90 mm for the member ‘BC’

The member is also subjected to a bending moment

\[
[M_{BC} = P \cdot (L_4 + L_5 + L_6) = 398880 \text{ N-mm}]
\]

The maximum vertical deflection at both ends of member is given as

\[
y_{BC} = \text{Bending moment at } C^1 \text{ for the conjugate beam.}
\]

\[
y_{BC} = \text{Load } B^* B^1 * C^{*} C^1 * X \text{ distance of its CG from } C^1
\]

\[
y_{BC} = \frac{M_{BC} \cdot 11.25}{E I_{BC}} \quad (A11.3)
\]

\[
= \frac{398880 \cdot 12 \cdot 22.5 \cdot 11.25}{2 \cdot 10^5 \cdot 90 \cdot 30^3}
\]

\[
y_{BC} = 2.493 \times 10^{-3} \text{ mm}
\]

The axial deflection and the vertical deflection of member ‘BC’ are very less. Hence the member may be considered as a rigid one. The axial deflection and the vertical deflection of member ‘BC’ are determined using ANSYS. The results are given in Figure 11.4 and 11.5.
Figure A11.3 Free body diagram of member ‘BC’- ANSYS

Figure A11.4 Axial deflection of member ‘BC’- ANSYS
The right hand side member AEGB may be considered as cantilever beam with tip load at ‘A’ and fixed at point ‘B’ as shown in Figure A11.7.
The caliper volume loss coefficient for fixed caliper designs for one caliper can be approximated by

\[ V_{ci} = K_d \cdot P_L \]  \hspace{1cm} (A11.4)

The value for \( K_d \) is a function of the caliper cylinder size.

\[ K_d = 482 \times 10^{-6} d_c - 1.632 \times 10^{-6}, \text{cm}^3 / \text{N/cm}^2 \]  \hspace{1cm} (A11.5)

Where, \( K_d = \) fluid volume loss coefficient
\[ d_{wc} = \text{Diameter of caliper cylinder, cm} \]

\[ K_d = 482 \times 10^{-6} \times 3.2 - 1.632 \times 10^{-6}, \text{cm}^3/\text{N/cm}^2 \]

\[ k_d = 1.5407 \times 10^{-3}, \text{cm}^3/\text{N/cm}^2. \]

Substitute \( k_d \) in Equation (A11.4)

\[ V_d = 1.5407 \times 10^{-3} \times 620 \]

\[ V_{cl} = 0.9553 \text{ cm}^3 \]

Brake fluid stroke loss in caliper = \( \frac{V_d}{A} \)

= 1.18 mm.

Brake fluid stroke loss in caliper is equal to deflection of caliper body due to brake fluid pressure. So the deflection of caliper body may be restricted to 1.18 mm. If the deflection at point ‘A’ is more than 0.29 mm, brake may not be efficient.

From Figure (A11.7)

\[ E^*E_2^* = \frac{PL_1}{EI_1} \]

= \( 5.193 \times 10^{-4} \)

\[ E^*E_3^* = \frac{PL_1}{EI_2} \]

= \( 1.2465 \times 10^{-5} \)

\[ G^*G_2^* = \frac{P(L_1 + L_2)}{I_2E} \]

= \( 2.077 \times 10^{-5} \)
\[ G'G_3^* = \frac{P(L_1 + L_2)}{I_3E} \]
\[ = 3.876 \times 10^{-6} \quad \text{(A11.10)} \]

\[ B'B_1^* = \frac{P(L_1 + L_2 + L_3)}{I_3E} \]
\[ = 6.202 \times 10^{-6} \quad \text{(A11.11)} \]

\( y_A = \text{Bending moment at A}^* \text{ for the conjugate beam} \)

\[ = \text{Load A}^*E'E_2^* \times \text{distance of its CG from A}^* + \text{Load E'}E_3^*G'G_2^* \times \text{distance of its CG from A}^* + \text{Load G'}G_3^*B'B_1^* \times \text{distance of its CG from A}^* \]

Where, \( y_A \) – Deflection of ‘C’ clamp at point A

\[ = \frac{1}{2} \times 5.193 \times 10^{-4} \times 30 \times 2 \times 30 \times 3 \]
\[ + \frac{1}{2} \left[ 1.2465 \times 10^{-5} + 2.077 \times 10^{-5} \right] \times 20 \]
\[ \times \left[ 30 + 20 \times \frac{2 \times 1.2462 \times 10^{-5} + 2.077 \times 10^{-5}}{1.2465 \times 10^{-5} + 2.077 \times 10^{-5}} \right] \]
\[ + \frac{1}{2} \left[ 3.876 \times 10^{-6} + 6.202 \times 10^{-6} \right] \times 30 \]
\[ \times \left[ 50 + 30 \times \frac{2 \times 3.876 \times 10^{-6} + 6.202 \times 10^{-6}}{3.876 \times 10^{-6} + 6.202 \times 10^{-6}} \right] \]

\( y_A = 0.155 + 0.0135 + 0.01 \)

\( y_A = 0.1785 \text{mm} \)

The deflection at point ‘A’ is less than the brake fluid stroke loss in caliper. Hence the design is safe. An FEA model of ‘C’ clamp and caliper halves are shown in Figure A11.8. The horizontal deflection and bending moment diagram of ‘C’ clamp and caliper halves determined using ANSYS, are shown in Figure A11.9 and A11.10.
Figure A11.8 Free body diagram of ‘C’ clamp and caliper halves-ANSYS

Figure A11.9 Horizontal deflection of ‘C’ clamp and caliper halves-ANSYS

Figure A11.10 Bending moment diagram of ‘C’ clamp and caliper halves-ANSYS