Unit- VI

Handling Mode & Ride Mode

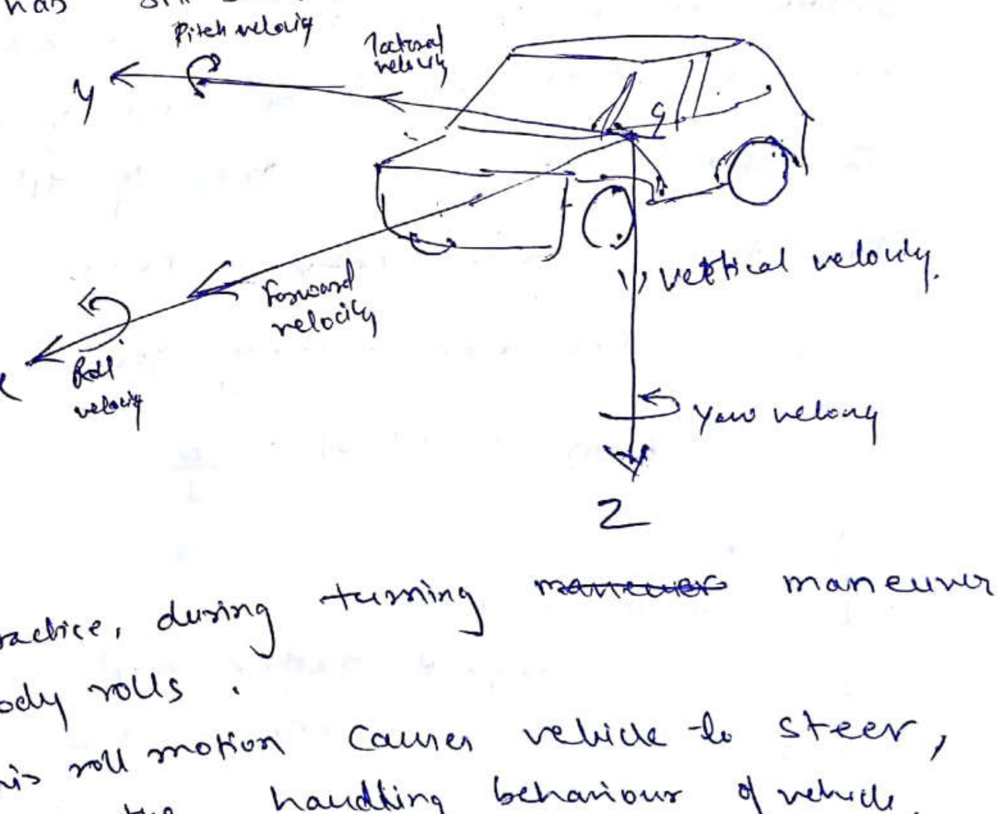
Handling Characteristics of Road Vehicles

Handling characteristics of a road vehicle refers to its response to steering commands & environmental inputs such as wind gust & road disturbances, that affect the direction of motion.

vehicle handling issues.

- control of direction of motion
- ability to stabilize direction of motion against external disturbance

vehicle has six D.O.F.



— In practice, during turning ~~maneuver~~ maneuver vehicle body rolls.

— This roll motion causes vehicle to steer, thus affecting the handling behaviour of vehicle.

— Similarly bounce & pitch motion also affect steering response of vehicle.

The response of the vehicle to steering input and its directional stability associated with a fixed steering wheel, which are usually refered to as fixed control characteristics will ~~be~~ be analysed.

* Steering Geometry

let ~~us~~ us consider a case of cornering of vehicle neglecting centrifugal forces. let us consider front wheel steer vehicle.

at low speed. ~~we know by Ackermann Steering geometry that~~

$$\cancel{\cot \delta_o - \cot \delta_i = \frac{b}{l}}$$

The prime important in designing steering system is minimum tire scrub during turning i.e all tires should be rolling during turning. To statisfy above the axes of all tires should pass through same common point.

So by Ackerman steering geometry, (i.e correct steering condition)

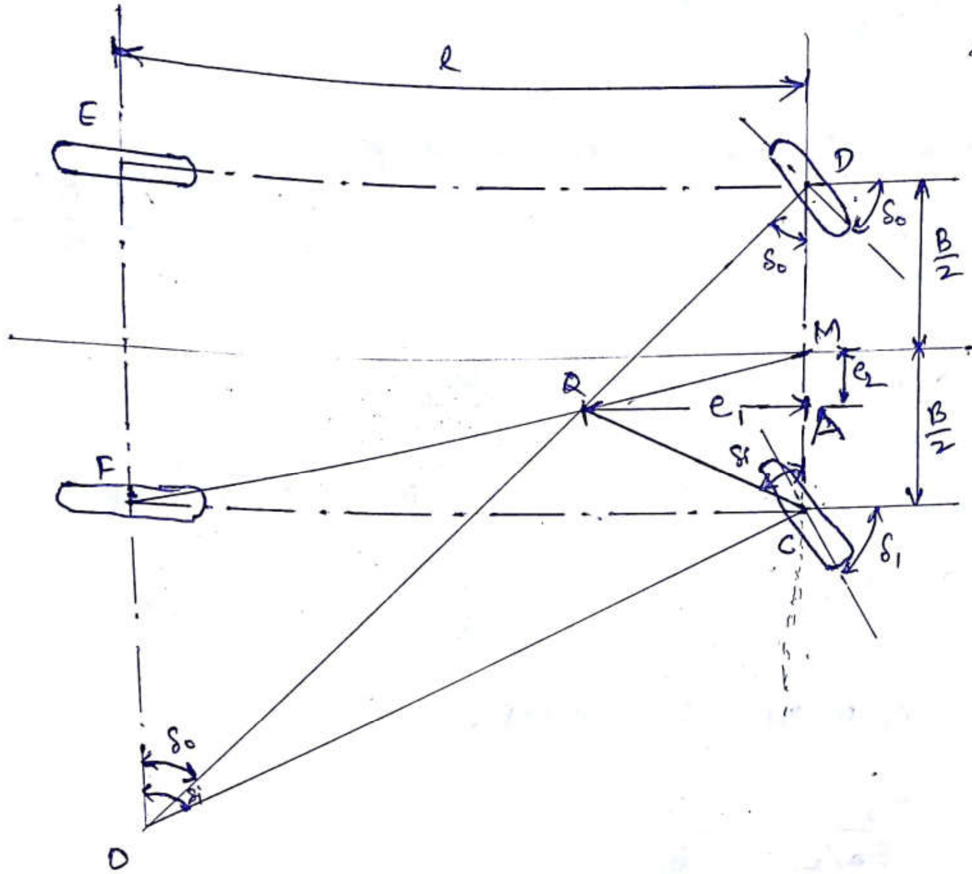
$$\cot \delta_o - \cot \delta_i = \frac{b}{l}$$

b = track

l = wheel base

δ_o = steer angle of outter wheel.

δ_i = — " — " — inner — " —.



Steps

(1) First draw ackerman's steering diagram

(2) draw midline intersecting front axle at M

(3) From M ~~Connect~~ Connect F.

(4) FM will intersect OD at Q..

(5) Connect QC & QD

(6) Then $\angle QCM = \delta_i$ and $\angle QDM = \delta_o$

This can be proved by.

~~In ΔDQA $\cot \delta_o$~~

In ΔDQA,

$$\cot \delta_o = \frac{DA}{QA}$$

$$\cot \delta_o = \frac{\left(\frac{B}{2} + e_2\right)}{e_1}$$

Similarly in ΔQCA.

$$\cot \delta_i = \frac{AC}{QA}$$

$$\cot \delta_i = \frac{\left(\frac{B}{2} - e_2\right)}{e_1}$$

~~and by ackermann relationship~~

$$\cot \delta_o - \cot \delta_i = \frac{1}{e_1}\left[\cancel{\frac{B}{2}} + e_2 - \cancel{\frac{B}{2}} + e_2\right]$$

$$\cot \delta_o - \cot \delta_i = \frac{2e_2}{e_1}$$

~~Comparing with ackermann relationship~~

~~$\cot \delta_o - \cot \delta_i = \frac{b}{l}$~~

$\therefore$ $\Delta MQA \sim \Delta MFC$

So, $\frac{e_2}{B/2} = \frac{e_1}{l}$

$$\frac{e_2}{e_1} = \frac{B/2}{l}$$

So, $\boxed{\frac{2e_2}{e_1} = \frac{B}{l}}$

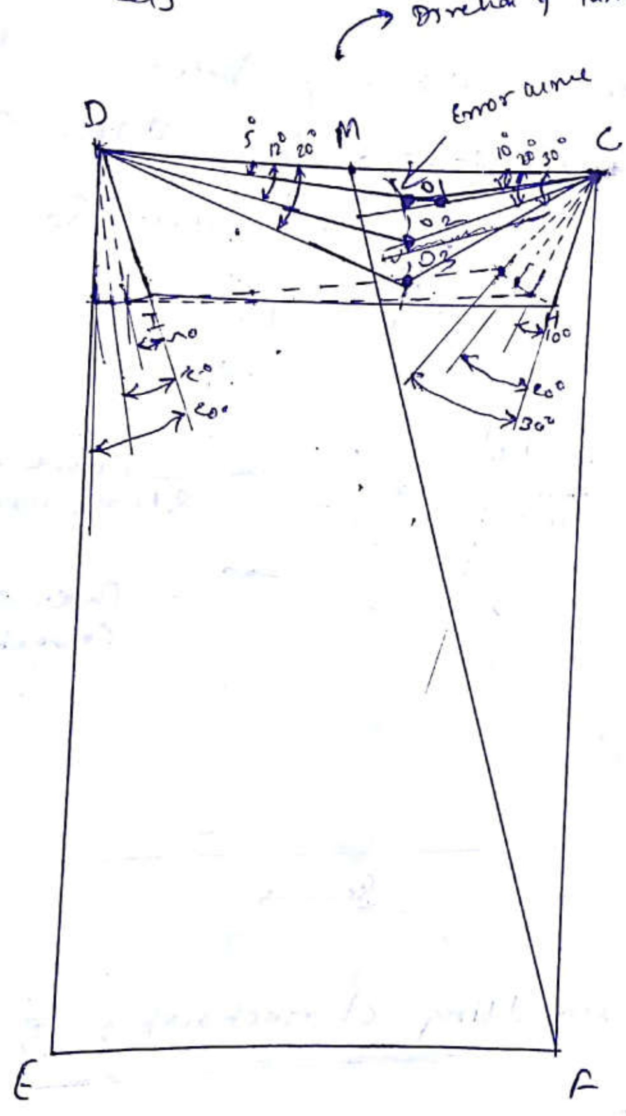
So,

$$\boxed{\cot \delta_o - \cot \delta_i = \frac{B}{l}}$$ — Hence Proved.

This indicate that if the steer angle of the front wheels i.e. δ_i and δ_o satisfy correct steering condition then by laying out the steer angle δ_i and δ_o from front axle, the intersection of the non-common sides of δ_i and δ_o will lie on the straight line connecting the midpoint of front axle and the centre of inside rear wheel

To evaluate the characteristics of a particular steering linkage w.r.t Ackerman steering geometry a graphical method is used. (62)

The steer angle are plotted for ~~input~~ inner & outter wheels

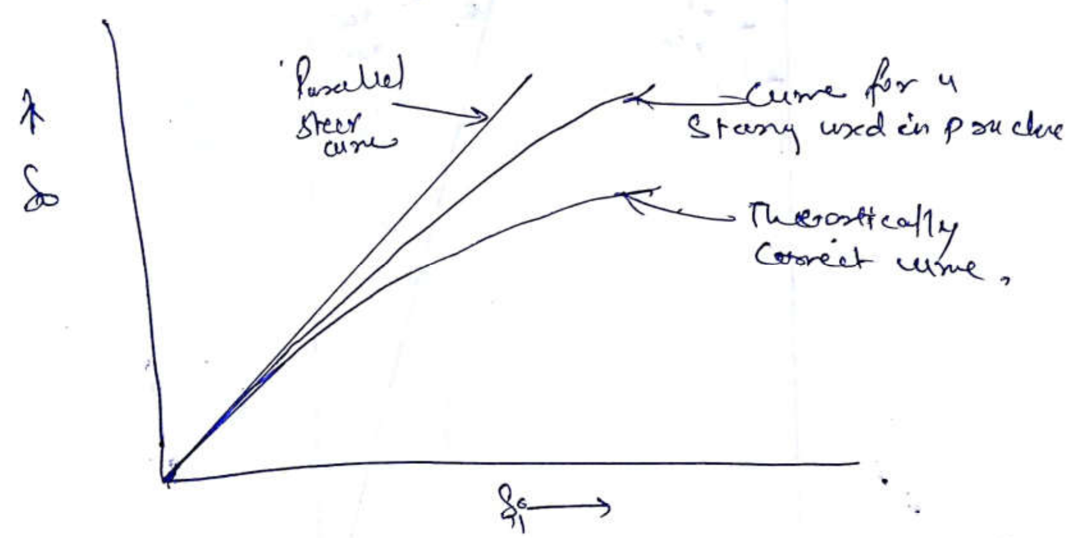


If the steering geometry satisfies correct steering condition the intersections on non-common sides of Si & So will lie on straight line MF.

The deviation curve O1, O2 & O3 from line MF is therefore an indication of error in steering geometry w.r.t ackermann criteria

Steering geometry with an error curve that deviate excessively from line PQF will exhibit considerable tire scrub during cornering. This results in excessive tire wear and increased steering effort.

Fig shows the relationship between δ_o & δ_i for a vehicle with $\frac{b}{l} = 0.56$, as compared to a parallel steer curve ($\delta_o = \delta_i$) and a typical steering geometry



* Steady state Handling characteristics of a Two-axle vehicle :-

Steady - state handling performance is concerned with the directional behaviour of a vehicle during a turn under non-time - varying conditions.

Example :- Steady State turn is a vehicle negotiating a curve with constant radius at a constant forward speed.

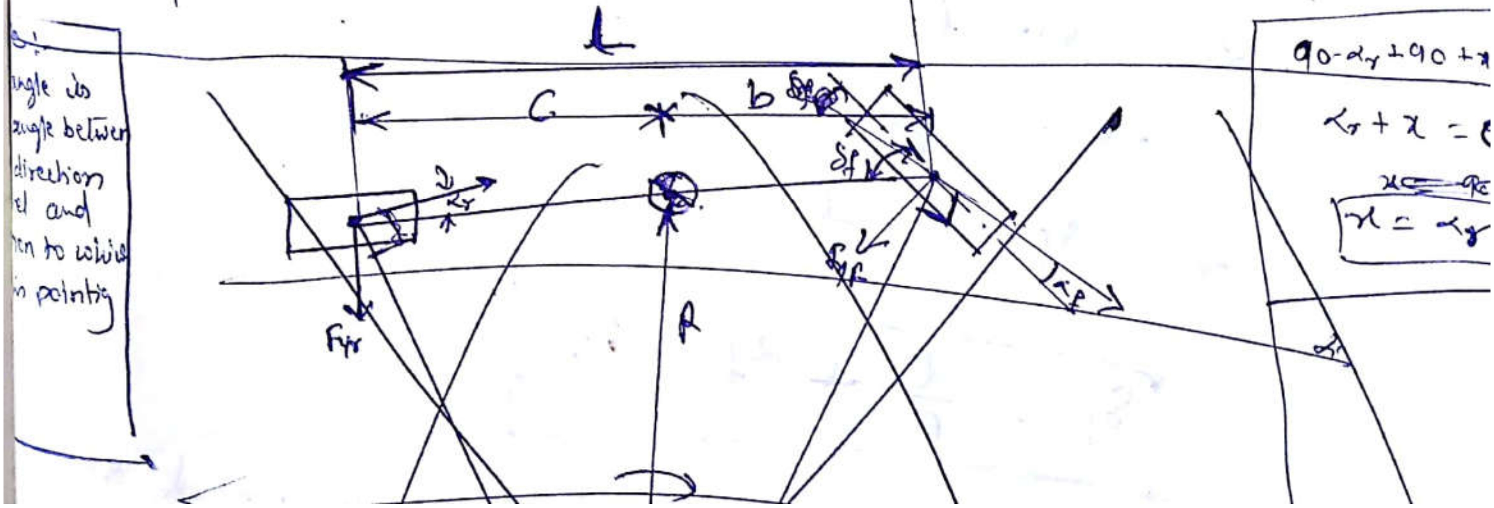
In this analysis inertia properties of vehicle are not considered.

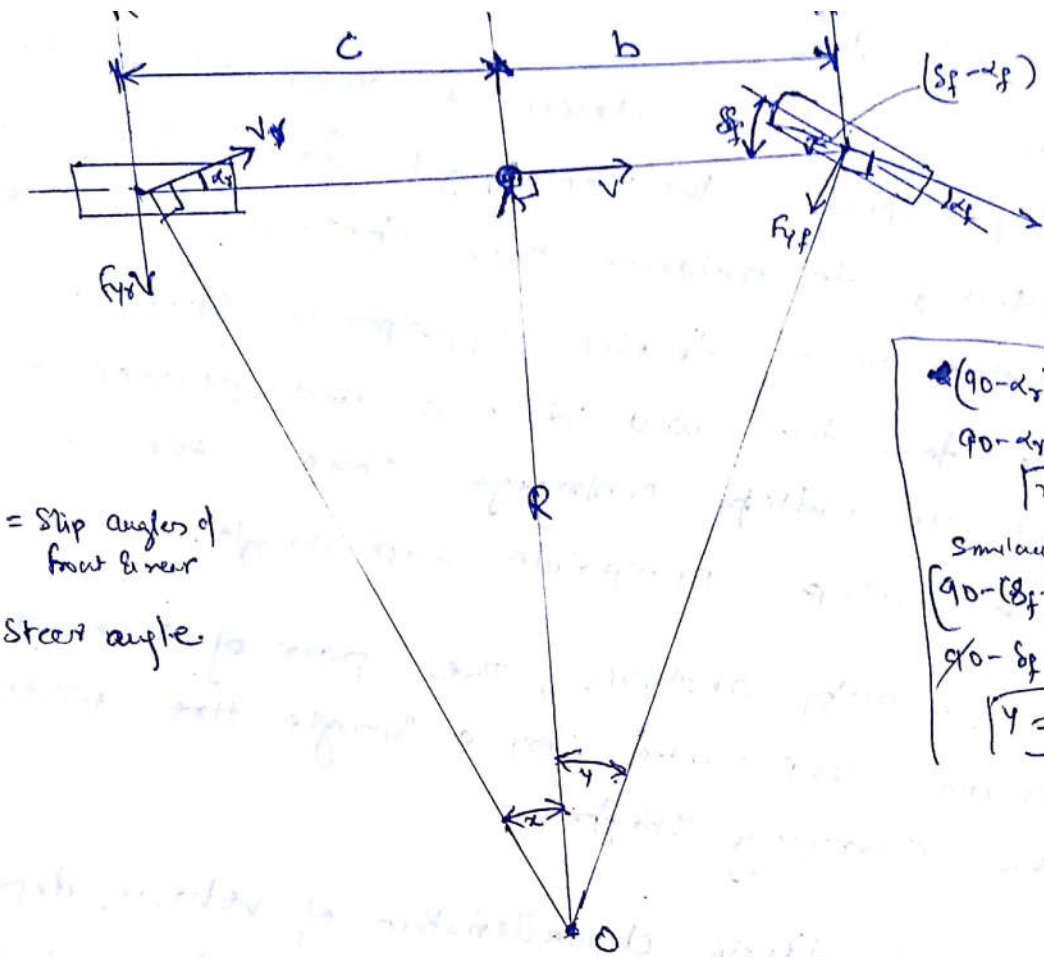
(63)

- When vehicle is taking a turn at moderate or higher speeds, the centrifugal force cannot be neglected, to balance this centrifugal force the tires must develop appropriate cornering force
- The four tires will develop will develop a force to nullify centrifugal force, this will in turn develop appropriate slip angles

- To simplify analysis, the pair of tires on an axles are represented by a single tire with double cornering stiffness

- The handling characteristics of vehicle depend, on relationship between the slip angles of front & rear tires. α_F & α_r





α_r, α_f = Slip angles of front & rear

δ_f = Steer angle

$(90-\alpha_r)+90+x=180$

$90-\alpha_r+90+x=180$

$\boxed{x=\alpha_r}$

Similarly

$[90-(\delta_f-\alpha_f)]+90+y=180$

$\cancel{90}-\delta_f+\alpha_f+\cancel{90}+y=\cancel{180}$

$\boxed{y=(\delta_f-\alpha_f)}$

So, we can say that.

$$R(x+y)=l$$

$$(x+y)=\frac{l}{R}$$

$$\alpha_r+\delta_f-\alpha_f=\frac{l}{R}$$

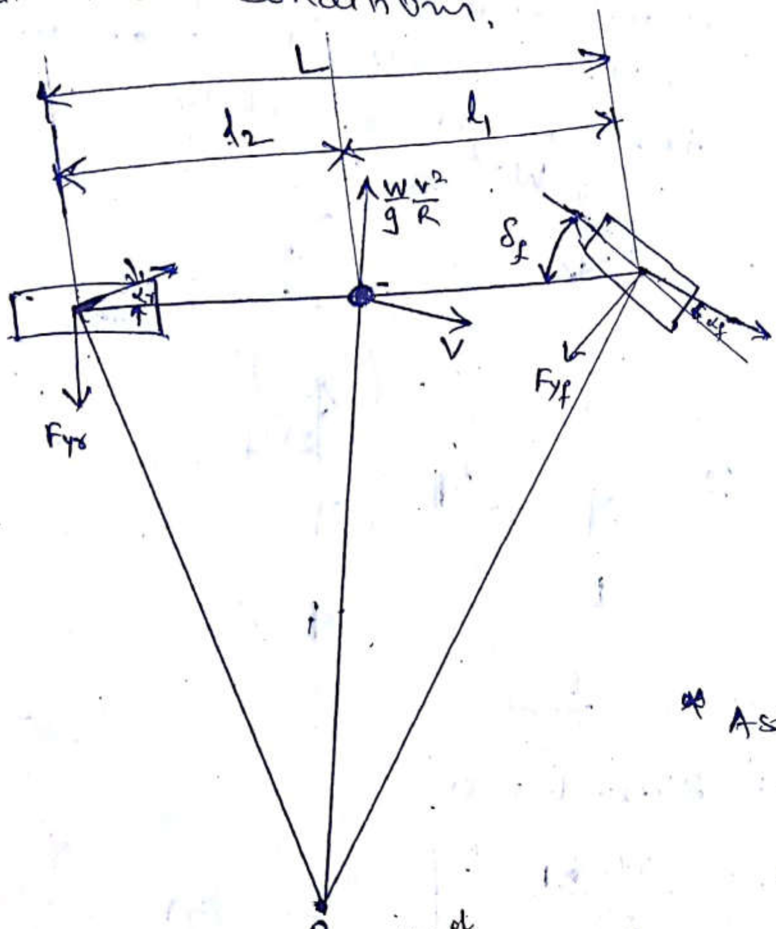
$$\boxed{\delta_f=\frac{l}{R}+\alpha_f-\alpha_r}$$

This is the required steer angle to negotiate the ~~turn.~~ given turn.

The steer angle is the function of slip angles and turning radius.

The ~~steer~~ slip angle are independent on side force acting on tire.

cornering forces can be determined ~~by~~ by equilibrium conditions. (64)



* Assuming δ_f is very Small.

Taking moment at centre of front wheel. (in vertical plane ~~vertical planes~~)

$$\left(\frac{W}{g}\frac{v^2}{R}\right) l_1 - F_{yr} \cdot L = 0$$

$$\boxed{F_{yr} = \frac{W}{g}\frac{v^2}{R}\frac{l_1}{L}} \quad \text{——— (1)}$$

Similarly, Taking moment at centre of rear wheel

$$-\left(\frac{W}{g}\frac{v^2}{R}\right) \cdot l_2 + F_{yf} \cdot L = 0$$

$$\boxed{F_{yf} = \frac{W}{g}\frac{v^2}{R} \cdot \frac{l_2}{L}} \quad \text{——— (2)}$$

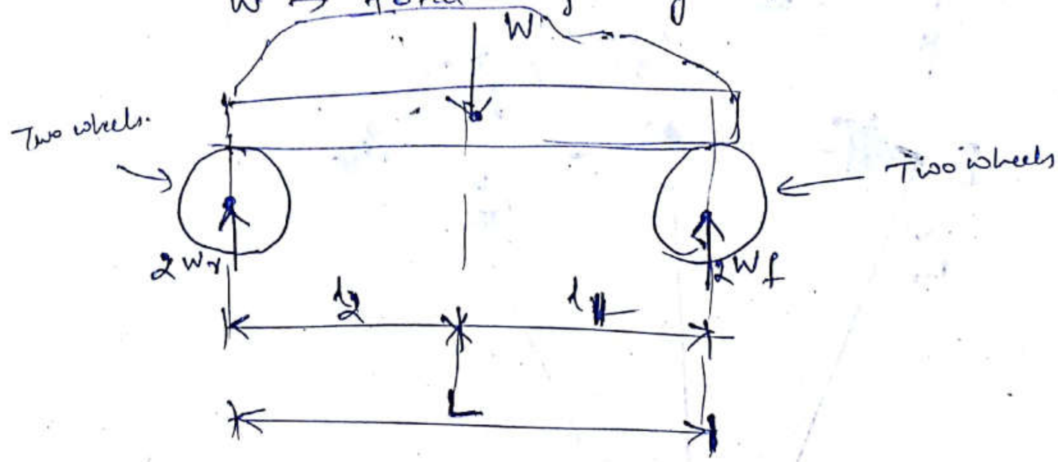
Steering geometry with an error curve

Under static condition.

$W_f \rightarrow$ Normal load on each ~~fron~~ Front tire.

$W_r \rightarrow$ — " — " — " — " — rear — " —

$W \rightarrow$ total weight of vehicle.



Taking moment on front.

$$-W \cdot l_1 + 2W_r \cdot L = 0$$

$$\boxed{W_r = \frac{W l_1}{2L}} \quad \text{——— (3)}$$

Taking moment on rear ~~tube~~ wheel.

$$-2W_f \cdot L + W \cdot l_2 = 0$$

$$\boxed{W_f = \frac{W l_2}{2L}} \quad \text{——— (4)}$$

Substituting values of (3) & (4) in eqution (1) & (2)

$$F_{yr} = \frac{W}{g}\frac{V^2}{R}\frac{l_1}{L}$$

$$= \left(\frac{W_r \cdot 2\cancel{L}}{g\,\cancel{l_1}}\right) \cdot \frac{V^2}{R} \cdot \frac{\cancel{l_1}}{\cancel{L}}$$

$$\boxed{F_{yr} = \frac{2W_r \cdot V^2}{gR}}$$

Similarly,

$$F_{yf} = \frac{W}{g}\frac{v^2}{R}\cdot\frac{l_2}{L}$$

$$= \frac{W_f \cdot (2\cancel{L})}{g \cdot \cancel{l_2}} \cdot \frac{v^2}{R} \cdot \frac{\cancel{l_2}}{\cancel{L}}$$

$$\boxed{F_{yf} = \frac{2W_f v^2}{gR}}$$

within a certain range, (slip angle $\propto$ Cornering force)

Hence,

slip angle of front,

~~$\alpha_f \propto F_{yf}$~~

$$\alpha_f = \frac{F_{yf}}{2C_{\alpha f}}$$

$$\alpha_f = \frac{2W_f v^2}{2C_{\alpha f}\cdot g R}$$

where, $C_{\alpha f}$ = cornering stiffness of each front wheel.

$$\boxed{\alpha_f = \frac{W_f v^2}{C_{\alpha f}\cdot g \cdot R}}$$

Similarly,

$$\alpha_r = \frac{F_{yr}}{2C_{\alpha r}}$$

$$\boxed{\alpha_r = \frac{W_r v^2}{C_{\alpha r}\, g \cdot R}}$$

where, $C_{\alpha r}$ = Cornering stiffness of each rear wheel.

Cornering stiffness depends upon inflatation pressure, normal load, tractive effort, and lateral force. It is constant only for limited range of operation

we known that,

$$\delta_f - \alpha_f + \alpha_r = \frac{L}{R}$$

$$\delta_f = \frac{L}{R} + \alpha_f - \alpha_r$$

$$= \frac{L}{R} + \frac{W_f v^2}{C_{\alpha f} \cdot g \cdot R} - \frac{W_r v^2}{C_{\alpha r}\, g\, R}$$

$$= \frac{L}{R} + \left[\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}\right] \frac{v^2}{gR}$$

$$= \frac{L}{R} + (K_{us}) \frac{v^2}{gR}$$

$$\boxed{\delta_f = \frac{L}{R} + (K_{us}) \frac{a_y}{g}}$$

$= m \cdot \frac{v^2}{r}$

$= (m \cdot a_y)$

*

where,

K_{us} = Understeer coefficient in radians.

a_y = lateral acceleration.

The above equation is the fundamental equation governing the steady-state handling behaviour of road vehicles.

It tells us the steer angle required to negotiate a give curve.

Depending on the values of understeer coefficient (K_{us}) steady state handling may be classified into three categories 66

① Neutral steer ($K_{us} = 0$)

② Understeer ($K_{us} > 0$)

③ Oversteer ($K_{us} < 0$)

① Neutral steer.

when understeer coefficient $K_{us} = 0$,

i.e.

$$\delta_f = \frac{L}{R} + K_{us}\frac{a_y}{g}$$

$$\boxed{\delta_f = \frac{L}{R}}$$

or

$$K_{us} = 0$$

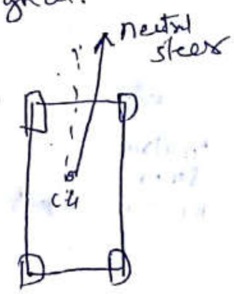
$$\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}} = 0$$

$$\boxed{\frac{W_f}{C_{\alpha f}} = \frac{W_r}{C_{\alpha r}}}$$

or

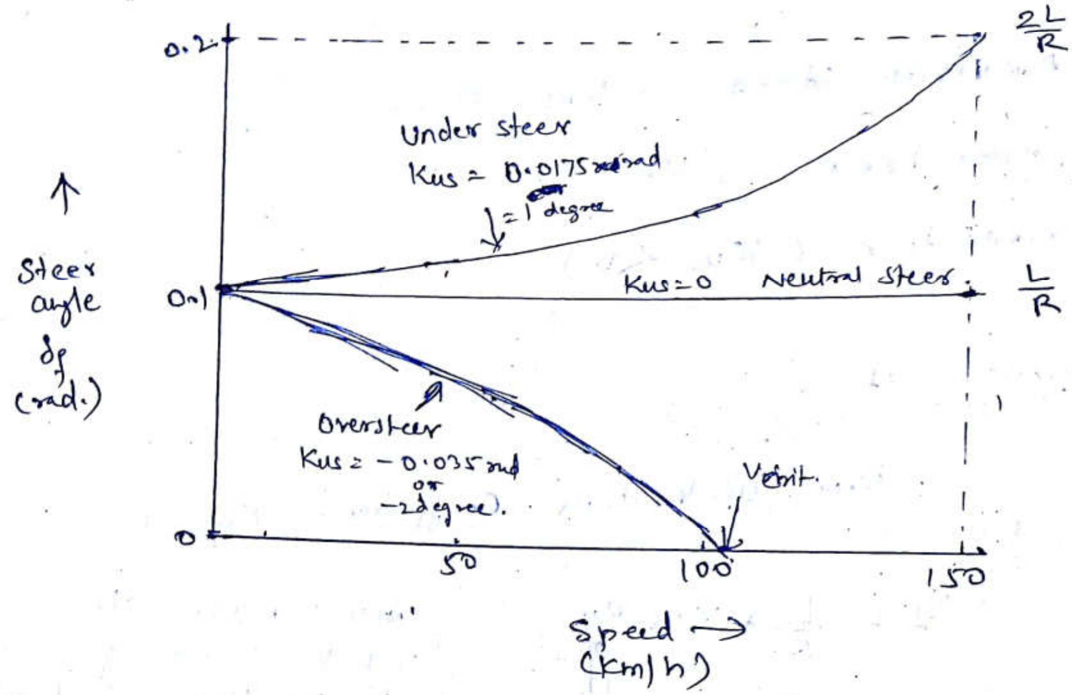
$$\boxed{\alpha_f = \alpha_r.}$$

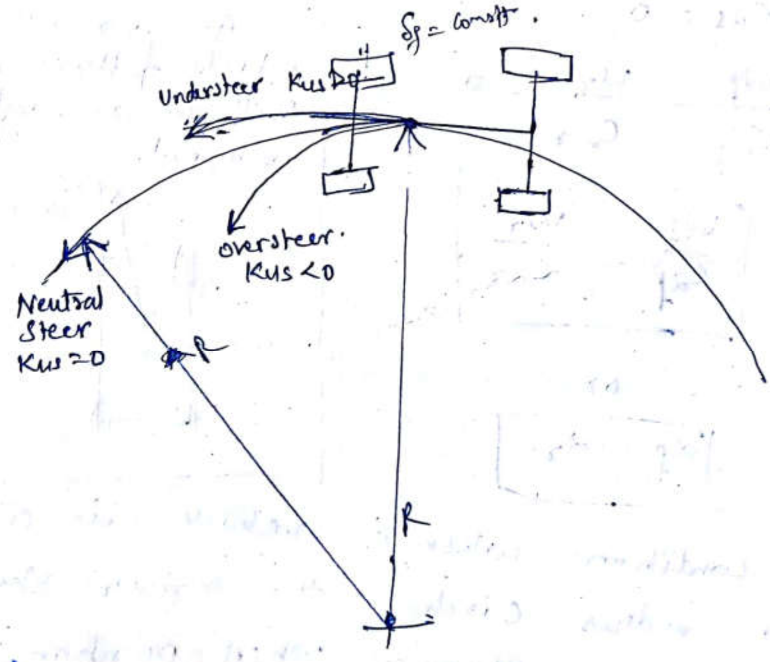
when a neutral steer vehicle is moving on a straight line is subjected to a side force acting at C.G.
so, equal slip angle develop at front & rear tires.
As a result, the vehicle follows a straight path at an angle to the original.



In this condition when a vehicle is accelerated in constant radius circle, the driver should maintain the same steering wheel position.

Condtns:-
Constant Radius $R = 30m$
Wheel base $L = 3m$.





~~Later~~ Charact'c speed is attained when $K_{us} \frac{V_{ch}^2}{gR} = \frac{L}{R}$

$\boxed{V_{ch} = \sqrt{\frac{g}{K}}}$

(2) Under steer

when $K_{us} > 0$.

$$\delta_f = \frac{L}{R} + (K_{us}) \frac{a_y}{g}$$

$$\left[\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}\right] > 0$$

This means.,

$$\boxed{\frac{W_f}{C_{\alpha f}} > \frac{W_r}{C_{\alpha r}}}$$

or

$$\boxed{\alpha_f > \alpha_r}$$

$\boxed{\delta_f \propto a}$

This means δ_f required to negotiate ~~turn~~ a curve increases with "square of forward velocity, or lateral acceleration".

The curve is represented by parabola v/s steer angle vs speed diagram.

we can say that ~~as the~~ when vehicle is accelerated with fixed steering wheel, the turning radius increases.

when travelling in straight path with side force acting on C.G. the front tire will generate ($\alpha_f > \alpha_r$) as a result "yaw motion is initiated" in understeer

For an under steer vehicle, characteristic speed V_{char} is the speed at which the steer angle required to negotiate a turn is equal to $\left(\frac{2L}{R}\right)$

$$\boxed{V_{char} = \sqrt{\frac{gL}{K_{us}}}}$$

(3) Oversteer

$$\therefore \boxed{K_{us} < 0}$$

or

$$\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}} < 0$$

$$\boxed{\frac{W_f}{C_{\alpha f}} < \frac{W_r}{C_{\alpha r}}}$$

or

$$\boxed{\alpha_f < \alpha_r}$$

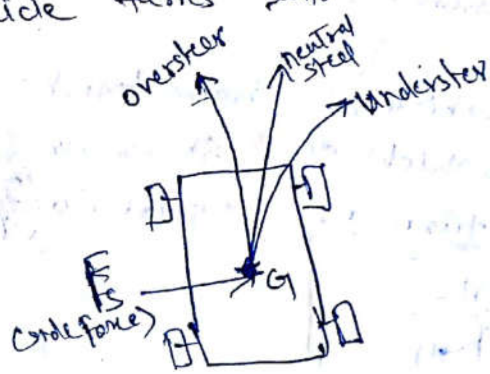
or we can that δ_f will decrease with increase in speed or lateral accelerat

— when vehicle is accelerated in const. radius turn, the driver must decrease the steer angle.

or

— with fixed steering wheel when
* accelerated the turning radius decreases.

— When side force act on vehicle travelling in straight path, the $\alpha_f < \alpha_r$ and as a result yaw motion is initiated and vehicle turns into side force.



For oversteer vehicle, critical speed "V_{crit}" is the speed at which the steer angle required to negotiate any turn is zero.

$$\boxed{V_{crit} = \sqrt{\frac{gL}{-K_{us}}}}$$

Critical speed is attained when

$$-K_{us}\frac{V_{crit}^2}{gR} = \frac{L}{R}$$

$$V_{crit} = \sqrt{\frac{gL}{-K_{us}}}$$

If the vehicle speed goes above critical speed then it will give directional instability.

Factors controlling steady state characteristics →
- weight distribution of vehicle.
- cornering stiffness of tires.

Conditions :

1. Front Engine, front wheel drive (load on front tire.) → understeer
2. Rear Engine, Rear wheel drive (load on rear.) → oversteer.
3. when vehicle accelerates during turn → understeer
   but due to load is on rear wheel, but due to cornering force
4. when vehicle decelerates during turn → oversteer
   load is on front but due to cornering force
5. when four wheel drive equal torque to all wheels, at fix steering wheel position.
   during Acceleration → increased understeer
   — " — deceleration → oversteer.

(6) Mixing radial ply + bias ply tires] → It will effect handling

(7) laterally stiff radial ply tire on front, relatively flexible bias ply tire on rear] → understeer vehicle will change to oversteer

(8) ↓ Inflation pressure in rear tires
↓
cornering stiffness decreases
↓
will change from understeer to oversteer.

(9) lateral load transfer insider to Outside tire during turn will increase slip angle. required to generate a given cornering force

(10) Braking and acceleration during turning will affect the cornering force.

(11) (a) ~~Application~~ For rear wheel drive, application of traction effort during turn reduces cornering properties of rear tire, producing oversteering.

(b) Front wheel drive, application of traction effort during turn, reduces cornering properties on front tire producing understeer.

(12) Roll steer :- Steering motion of front or rear wheel due to relative roll motion of spring mass w.r.t. Unsprung mass.

Roll chamber :- change in camber of wheels due to the relative roll motion of Spring mass w.r.t to Unsprung mass.

Hello.

Compliance steer :- Steering motion of wheels w.r.t sprung mass resulting from compliance in, and force on, the suspension & steering linkages

The effect of Roll steer, Roll camber and compliance steer can be included in a modified form of the understeer coefficient Kus.

(13) — Among the three types of steady state handling behaviour oversteer is not desirable from directional stability point of view

(14) — It is desirable for road vehicle to have small degree of understeer till a lateral acceleration of 0.4g. beyond this understeer can be increased.

(15) Increased understeer at high lateral acceleration would provide greater stability during turns.

A handling diagram is used to determine handling behaviour for changing operating condition

During a turning maneuver, the turning radius R may be difficult to measure directly, however it can be determined from yaw velocity Ω_z reading

yaw velocity $(-\Omega_z)$,

forward velocity, $V = R \cdot \Omega_z$.

So, $\boxed{\Omega_z = \frac{R}{V}}$

steady state handling behaviour

So, our equation will

$$\delta_f = \frac{L}{R} + K_{us}\left(\frac{V^2}{gR}\right)$$

or

$$\delta_f = \frac{L}{R} + K_{us}\left(\frac{a_y}{g}\right)$$

or.

$$\delta_f - \frac{L}{R} = K_{us}\left(\frac{a_y}{g}\right)$$

$$-\left[\frac{L}{R} - \delta_f\right] = K_{us}\left(\frac{a_y}{g}\right)$$

$$\left[\frac{L\Omega_z}{V} - \delta_f\right] = -K_{us}\left(\frac{a_y}{g}\right)$$

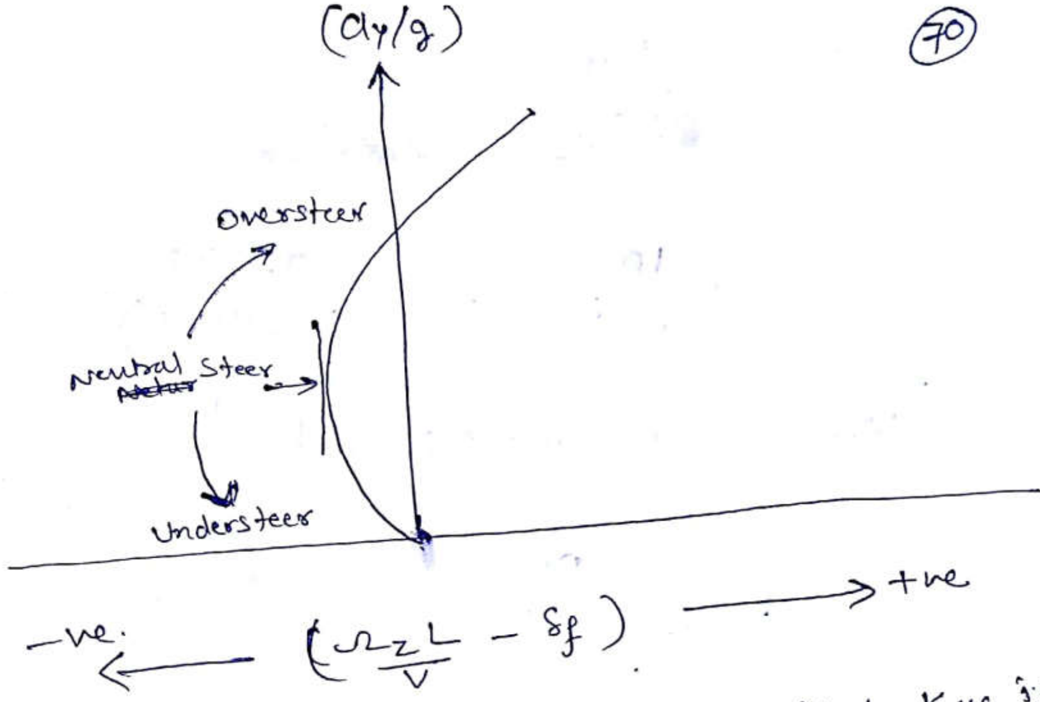
$$\frac{\left(L\Omega_z/V - \delta_f\right)}{(a_y/g)} = -K_{us}$$

$$\boxed{\frac{(a_y/g)}{\left(\Omega_z L/V - \delta_f\right)} = -\frac{1}{K_{us}}}$$

or

$$\boxed{\frac{d(a_y/g)}{d\left(\Omega_z L/V - \delta_f\right)} = -\frac{1}{K_{us}}}$$

So slope of L.H.S term will determine the steer condition



① If slope is -ve, then it implies that Kus is +ve hence understeer

② If Slope is +ve, then it implies that Kus is -ve hence over steer

③ If Slope is infinite, then it imples Kus is zero hence neutral steer.

Ques. $W = 20.105$ KN

$L = 2.8$ m

$W_f = 0.535\, W = 10.756$ KN } static condition

$W_r = 0.465\, W = 9.348$ KN

a) If $C_{\alpha f} = 38.92$ KN — for each tire of front

$C_{\alpha r} = 38.25$ KN — for each tire of rear

determine steady state handling behaviour?

b) If front tire replaced by pair of radial ply

$C_{\alpha f} = 47.82$ KN — for each front tire.

— rear tire unchanged

determine steady state handling behaviour

a) $K_{us} = \frac{W_f}{2C_{\alpha f}} - \frac{W_r}{2C_{\alpha r}}$

$= \frac{10.756}{2(38.92)} - \frac{9.348}{2(38.25)}$

$$\boxed{K_{us} = 0.01598 \text{ rad}}$$

or

$$\boxed{K_{us} = 0.92^\circ}$$

The vehicle is understeer and characteristics speed is

$V_{char} = \sqrt{\frac{gL}{K_{us}}}$

$$\boxed{V_{char} = 41.5 \text{ m/s}}$$

or

$$\boxed{V_{char} = 149 \text{ km/h}}$$

*

(b) $K_{us} = \frac{W_f}{2C_{\alpha f}} - \frac{W_r}{2C_{\alpha r}}$

$= \frac{10.756}{2(47.82)} - \frac{9.348}{2(38.25)}$

$\boxed{K_{us} = -0.00973 \text{ rad}}$ or -0.56°

Scanned by CamScanner

$$V_{crit} = \sqrt{\frac{gL}{-K_{us}}}$$

$$\boxed{V_{crit} = 53.1 \text{ m/s}}$$

or

$$\boxed{V_{crit}.\ 191 \text{ km/h}.}$$

* steady - state Response to steering Input

During turning maneuver steer angle is input. output will be variables like yaw velocity. lateral acceleration and curvature.

The ratio of output to input can be used for comparing response characteristics of different vehicles.

(1) Yaw velocity Response :-

"Yaw velocity gain" is an often used parameter for comparing steering response of road vehicle.

It is defined as the ratio of steady state yaw velocity to steer angle.

Yaw velocity gain (G_{yaw})

$$G_{yaw} = \frac{\Omega_z}{\delta_f}$$

$$G_{yaw} = \frac{V/R}{\frac{L}{R} + (K_{us})\frac{V^2}{gR}}$$

$$\boxed{G_{yaw} = \frac{V}{\left(L + \frac{(K_{us})V^2}{g}\right)}}$$

a) For neutral steer,

$K_{us} = 0$

$$\boxed{G_{yaw} = \frac{V}{L}}$$

yaw velocity gain increases linearly with increase in forward velocity.

b) For understeer,

K_{us} is +ve, the yaw velocity gain increases with increase of forward speed and reaches maximum at particular speed i.e characteristic speed V_{char}.

* c) For oversteer,

K_{us} is −ve, the yaw velocity gain increases with forward speed at an increasing rate at a particular speed, denominator is zero and G_{yaw} becomes infinity, that speed is critical speed V_{crit}.

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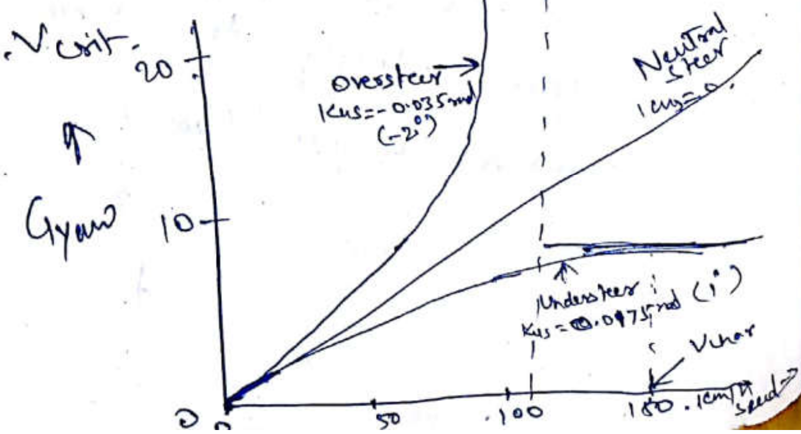


So we can say that, from the point of view of handling response to steering input. the Sensitivity of vehicle is of order. 72

oversteer sensitivity > neutral steer sensitivity > Under steer.

we can also say that,

if $\boxed{G_{yaw} > \frac{V}{L}}$ — oversteer

$\boxed{G_{yaw} < \frac{V}{L}}$ — understeer

2) Lateral Acceleration Response:-

Lateral acceleration gain is defined as the ratio of steady-state acceleration to the steer angle.

$$G_{acc} = \frac{(V^2/gR)}{\delta_f}$$

$$= \frac{(a_y/g)}{\delta_f}$$

$$= \frac{V^2/gR}{\frac{L}{R} + (K_{us})\frac{V^2}{gR}}$$

$$= \frac{V^2/g}{L + (K_{us})\frac{V^2}{g}}$$

lateral acclⁿ →

$$= \frac{\left(v^2/g\right)}{\left(\frac{gL + (K_{us})v^2}{g}\right)}$$

$$\boxed{G_{acc} = \frac{v^2}{gL + K_{us}v^2}}$$

a) For neutral steer,

$K_{us} = 0$

$$\boxed{G_{acc.} = \frac{v^2}{gL}}$$

the Lateral acceleration gain is proportional to square of forward velocity.

(b) For under steer.

K_{us} is +ve

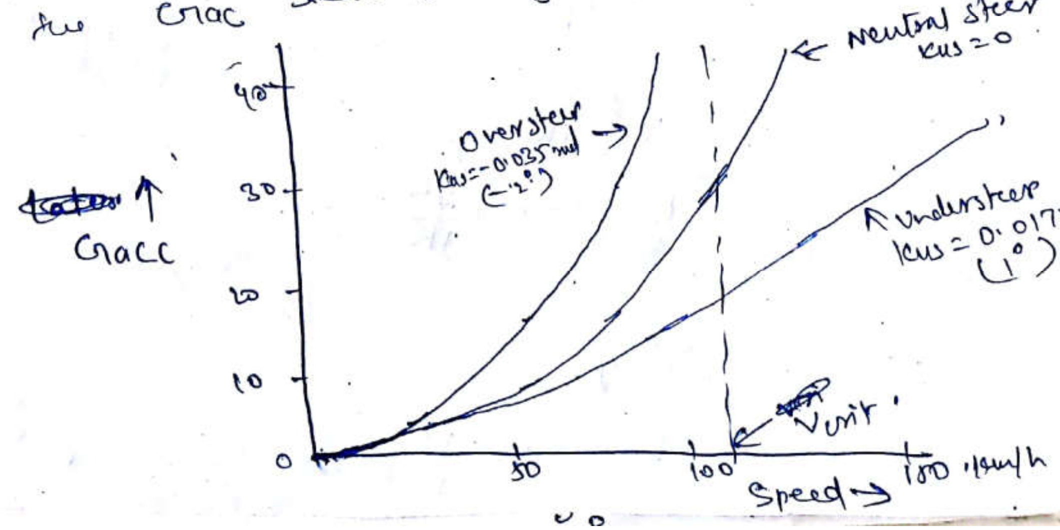
– The G_{acc} increases with increase in speed.

– ~~At~~ At very high speed the term (gL) becomes very small compared to $(K_{us}v^2)$ hence G_{acc} approaches a value of $\left(\frac{1}{K_{us}}\right)$ assympotically.

(C) For oversteer

K_{us} is –ve.

As the speed increases G_{acc} increases with an increasing rate, but at certain value of K_{us} the G_{ac} reaches infinity. ~~to~~ the speed is V_{crit}.



3) Curvature Response :-

The ratio of steady-state curvature ($1/R$) to the steer angle is the parameter commonly used for evaluating the response characteristics of vehicle.

$$\frac{1/R}{\delta_f} = \frac{1/R}{\frac{L}{R} + K_{us}\frac{V^2}{gR}}$$

$$\boxed{\frac{1/R}{\delta_f} = \frac{1}{L + (K_{us})\frac{V^2}{g}}}$$

a) for neutral steer;

$K_{us} = 0$

$$\boxed{\frac{1/R}{\delta_f} = \frac{1}{L}}$$

curvature response is independent of ~~velocity~~ forward speed.

b) for understeer,

K_{us} is positive.

the curvature response decreases with increase in speed.

c) for oversteer,

K_{us} is $-ve$, curvature response increases with forward speed. At particular speed curvature response reaches infinity i.e. This means that the turning radius approaches zero and the vehicle spin out of control", This speed is V_{crit}. of an oversteer vehicle.

Hence we can say that oversteer vehicle has most sensitive handling characteristics while understeer vehicle is least responsive.

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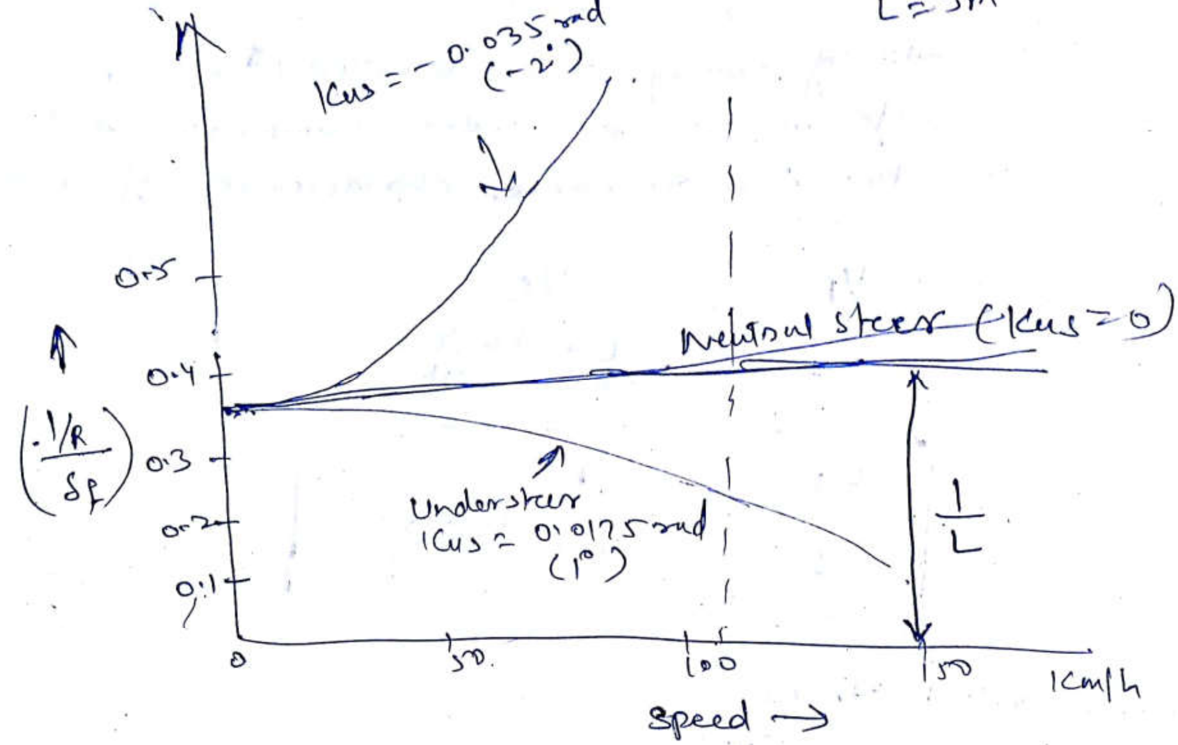
wheelbase

L = 3m

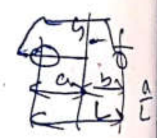


Ans 5.2

$W = 20.105$ KN

$L = 3.2$ m

$$\frac{\text{C.G to Front axle}}{L} = 0.465$$



$C_{\alpha f} = 38.92$ kN | front tire

$C_{\alpha r} = 38.25$ KN | rear tire

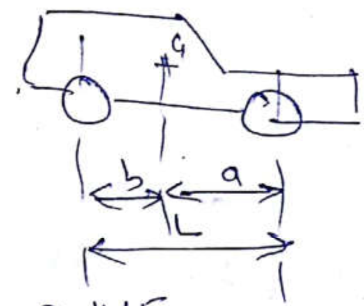
Average steering gear ratio = 25

$G_{yaw} = ?$ } w.r.t δ_f

$G_{acc} = ?$

$\cancel{K_{us} = \frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}}$

=



$$\frac{a}{L} = 0.465$$

$$a = 0.465 \times 3.2$$

$$\boxed{a = 1.488\ m}$$

$$a + b = L$$

$$b = L - a$$

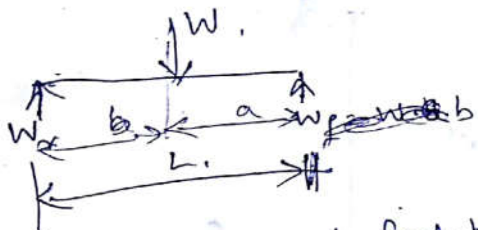
$$\boxed{b = 1.712\ m}$$

or

$$\frac{a}{L} = 0.465$$

and

$$\frac{b}{L} = 0.535$$



Taking momet at front tire

$$= -W.a + W_r.L$$

$$\boxed{W_r = \frac{W.a}{L}} = \underline{9.3481 kN}$$

Taking momet at rear tire

$$= -W_f.L + W.b$$

$$\boxed{W_f = \frac{W.b}{L}} = \underline{10.75\ KN}$$

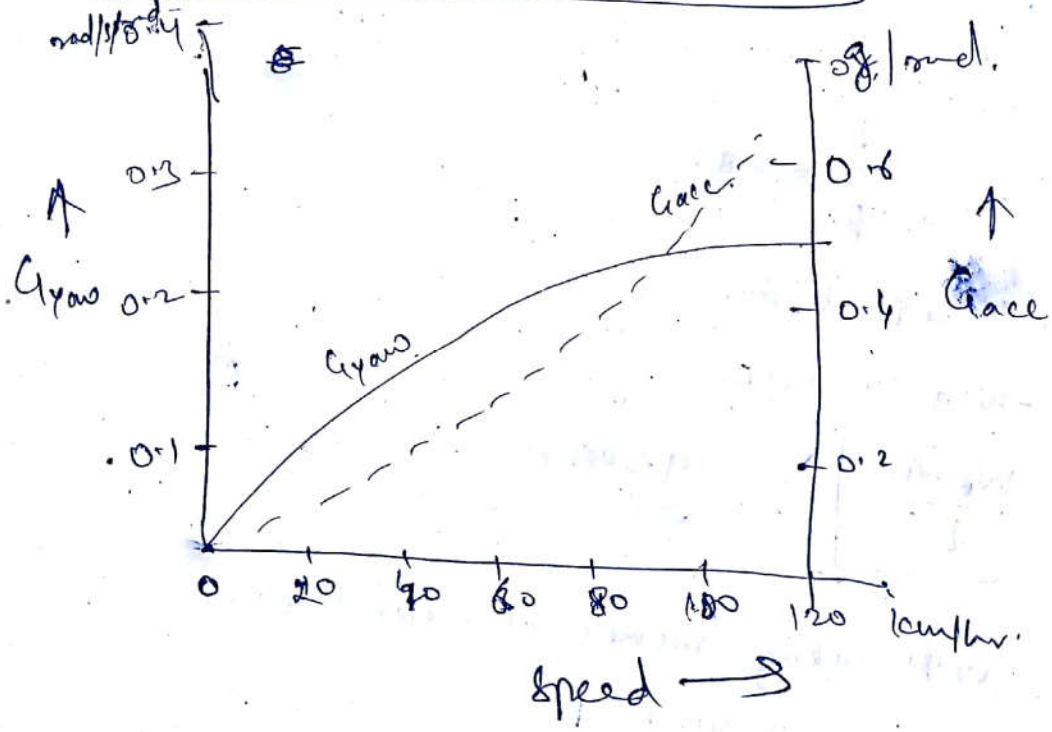
$$K_{us} = \frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}$$

$$= \left[\frac{10.75}{2\times 38.92}\right] - \left[\frac{9.348}{2\times 38.25}\right]$$

$$\boxed{K_{us} = 0.0159 \text{ rad}} \quad \text{or} \quad \underline{0.92^\circ}$$

$$G_{yaw} = \frac{\Omega_z}{\delta_f \cdot \xi_s}$$

$$\boxed{G_{yaw} = \frac{V}{\left[L + \frac{K_{us}V^2}{g}\right]\xi_s}}$$



$$G_{acc} = \frac{a_y/g}{\delta_f \xi_s}$$

$$\boxed{G_{acc} = \frac{V^2}{\left[gL + K_{us}V^2\right]\xi_s}}$$

* Testing of Handling characteristics :- and

(75)

To measure handling behaviour of a road vehicle under steady - state conditions, various types of test can be conducted on a skid pad (which is a large flat paved area).

Yaw velocity $\Omega_z = \frac{V}{R}$

and lateral accl^{n} $a_y = \frac{V^2}{R}$

so, $\boxed{a_y = \Omega_z \cdot V}$

So if we know a_y and V, then Ω_z can be measured

Three tests :-

1. Constant radius Test
2. — u — Forward speed Test
3. Constant steer angle test.

During tests, steer angle, forward velocity and yaw velocity are usually measured.

(1) Constant Radius Test

In this test, the vehicle is driven along a curve with a constant radius at various speeds. The steer angle (δ_f) at various forward speed together with corresponding lateral acceleration are measured.

we know,

$$\delta_f = \frac{L}{R} + (K_{us}) \frac{a_y}{g}.$$

$\delta_f = \frac{L}{R} + K_{us}\left(\frac{a_y}{g}\right)$

Differentiating w.r.t time

$\frac{d\delta_f}{dt} = \frac{d}{dt}\left(\frac{L}{R}\right) + K_{us}\frac{d}{dt}\left(\frac{a_y}{g}\right)$

$\because$ L and R are constt.

$\frac{d\delta_f}{dt} = K_{us}\frac{d}{dt}\left(\frac{a_y}{g}\right)$

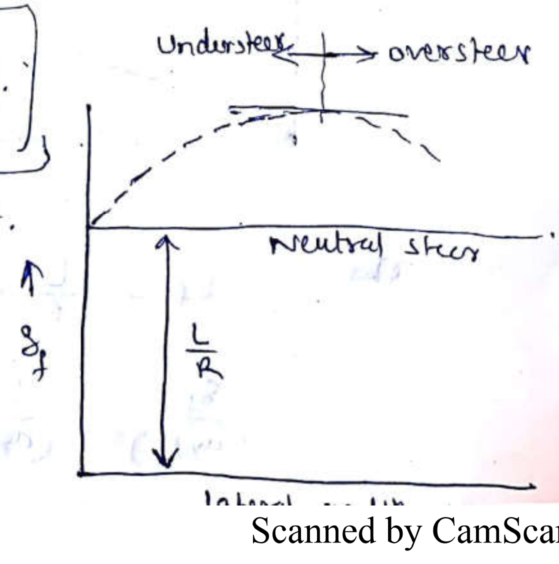
$\boxed{K_{us} = \frac{d\delta_f}{d(a_y/g)}}$

For constant turning radius, "slope of Curve

$$\boxed{\frac{d(\delta_f)}{d\left(\frac{a_y}{g}\right)} = K_{us}.}$$

It is equal to understeer Coeff.

understeer will become oversteer at high lateral acceleration.



- If δ_f required to maintain the vehicle ~~on~~ a constt radius turn is same for all forward speed (i.e. slope of (δ_f) vs (a_y) is zero) than the vehicle is neutral steer.

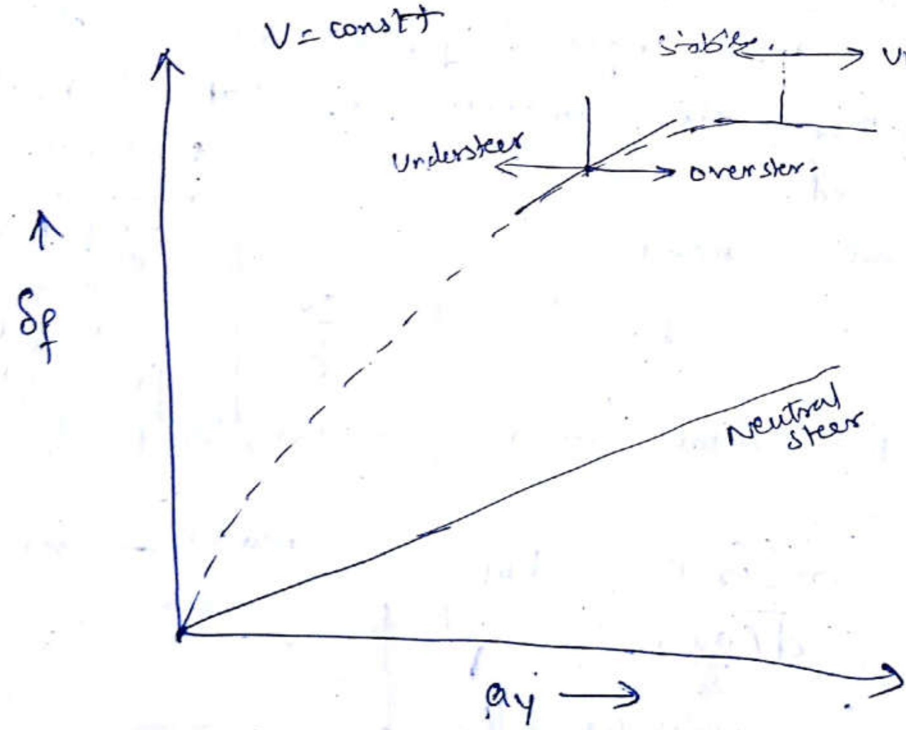
- Slope of (δ_f) vs (a_y) is +ve. — ~~oversteer~~ understeer vehicle

- Slope of (δ_f) vs (a_y) is -ve — oversteer. Vehicle.

(2.) <u>Constant Speed Test</u>

In this test, vehicle is driven at constant forward speed at various turning radii.

The steer angle and the lateral acceleration are measured.



$$\delta_f = \frac{L}{R} + (K_{us})\left(\frac{a_y}{g}\right)$$

$$\frac{\delta_f}{(a_y/g)} = \frac{L}{(a_y/g)} + K_{us}.$$

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and

(76)

$$\frac{\delta_f}{(a_y/g)} = \frac{gL}{V^2} + K_{us}$$

$$\delta_f = \frac{L}{R} + K_{us}\left(\frac{V^2}{gR}\right)$$

$$\delta_f = \left[\frac{L \cdot g}{V^2} + K_{us}\right]\frac{V^2}{gR}$$

~~∴ slope~~ ∴ Differentiate w.r.t time

$$\frac{d\delta_f}{dt} = \left[\frac{gL}{V^2} + K_{us}\right]\frac{d}{dt}\left(\frac{V^2}{gR}\right)$$

∴ term in bracket is const

$$\boxed{\frac{d\delta_f}{d\left(\frac{a_y}{g}\right)} = \frac{gL}{V^2} + K_{us}}$$

So, the slope of curve will be

$$\boxed{\frac{d(\delta_f)}{d(a_y/g)} = \frac{gL}{V^2} + K_{us}}$$

i.e $K_{us} = 0$

- if neutral steer slope $= \frac{gL}{V^2}$.

- For under steer K_{us} is +ve or we can say that slope is greater than the slope of neutral steer

- For oversteer, K_{us} is −ve or we can say that slope is smaller than the slope of neutral steer.

when slope is zero.

$$\frac{gL}{V^2_{crit.}} + K_{us} = 0$$

$$\frac{gL}{V^2_{crit.}} = -K_{us}$$

$$\boxed{V^2_{crit.} = \frac{gL}{(-K_{us})}}$$

This velocity will be the velocity of vehicle operating at critical speed oversteer. the vehicle is at the onset and that directional instability.

(3) Constant Steer Angle Test:-

In this test, the vehicle is driven ~~at~~ with a fixed steering wheel angle at various forward speeds. The lateral accelerations at various speeds are measured.

The results are plotted on $\left(\frac{1}{R}\right)$ vs $\left(\frac{a_y}{g}\right)$ graph.

The radius of curvature is calculated by,

$$a_y = \frac{V^2}{R}$$

$$\boxed{\frac{1}{R} = \frac{a_y}{V^2}}$$

So,

$$\delta_f = \frac{L}{R} + (K_{us})\left(\frac{a_y}{g}\right)$$

$$\therefore \delta_f = 0 \quad \text{— const steer angle.}$$

$$\frac{L}{R} = -(K_{us})\left(\frac{a_y}{g}\right)$$

$$\frac{(1/R)}{(a_y/g)} = \frac{-K_{us}}{L}$$

So, the slope.

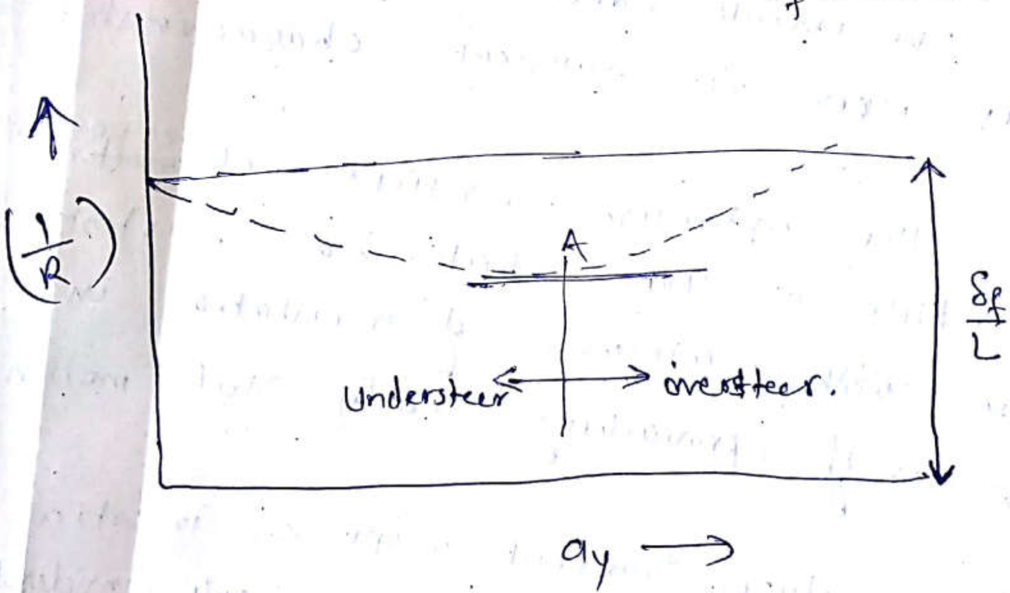
$$\boxed{\frac{d(1/R)}{d(a_y/g)} = \frac{-K_{us}}{L}}$$

- If vehicle is neutral steer $K_{us} = 0$ and the slope of curve will be zero.

- If vehicle is Understeer K_{us} is −ve and the slope will be positive.

- If the vehicle is oversteer K_{us} is +ve and slope will be −ve.

δ_f = Constt.



- The constant speed Test is more representative of the actual road behaviour of a vehicle than the constant Radius Test, as the driver usually maintain more or less constant speed in a turn.

- The constant steer angle is easy to execute

Ride quality → feel of passenger in moving vehicle

Ride problem → vibration of vehicle body

Sources of vibration:

1) Surface irregularities (Road irregularities)
2) aerodynamic forces
3) vibration of Engine and driveline
4) Unbalance of tire/wheel assembly.

The objective is to control the vibration of vehicle so that vibration doesn't exceed the limit of human comfort.

* Methods used to define ride comfort limits

1) Subjective Ride measurement:-

A trained jury rates the ride comfort on relative basis.

However the degree of difference in ride quality, cannot be quantitavely determined by such evaluation.

2) Shake Table test:-

It is intended to identify zones of comfort for human in terms of vibration amplitude, velocity or acceleration. (such as foot to head, side to side or back to chest) over a specified range of frequency.

3) Ride Simulator test

Road input fed to actuator and using a simulator it is possible to establish human tolerance limit

4) Ride measurement in vehicles:

Shake table test and ride Simulator gives laboratory values. So actual Conditions are no considered. Therefore Ride measurement method is formed

This test method attempts to Correlate the response of test in qualitative terms such "unpleasent", or untolerable. with vibration parameters measured at the location where the test subject is Situated under actual driving Condition.

* An guide by International Standard ISO 2631 [7.4, 7.5] says. that the evaluation of vibrational enviroments in transport vehicles as well as in industry, it defines three distinct limits for whole-body vibration in frequency range 1-80 Hz.

1) Exposure limits, which are related to preservation of safety (or health) should not exceed without special justification

2) Fatigue or decreased proficiency boundaries, which are related to preservation of working efficiency and apply to such tasks as driving a road vehicle or tractor

3) Reduced comfort boundaries, which are Concerned with preservation of comfort and in Transport vehicle are related to such functions as reading, writing and eating in vehicles.

* vehicle Ride Models

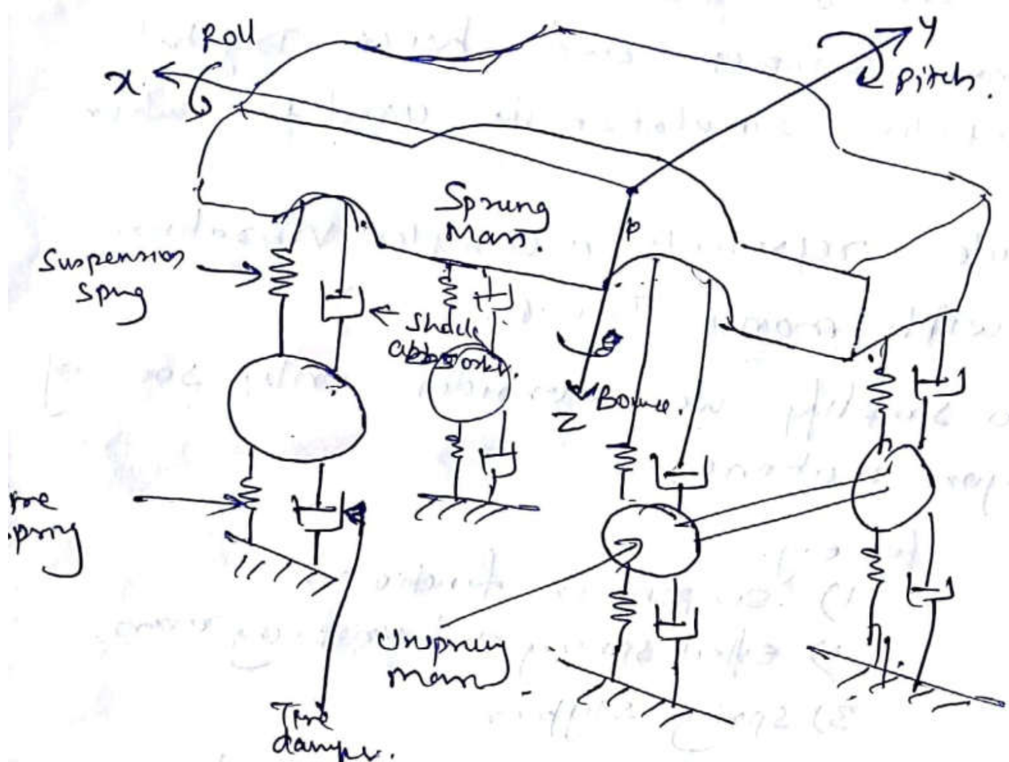


Fig :- A 7-DOF ride model for a passenger car.

To study ride quality of ground vehicles, the above 7-D.O.F model is used.

motions considered

1) Roll of vehicle body
2) Pitch — " — " — " —
3) Bounce — " — " — " —
4) Bounce of two front wheel
5) — " — " — Rear axle.

mass of vehicle body → sprung mass.

— " — " — running gears → unsprung mass.

Newton's 2nd law method is used for formulation of equation and determine the principal modes of vibration of vehicle body.

- if excitation is known than response can be determined by equation of motion.

- However with increase in D.O.F system becomes complex and hence Digital Computer simulation is used for solution.

A vehicle represents a complex vibration system with many D.O.F.

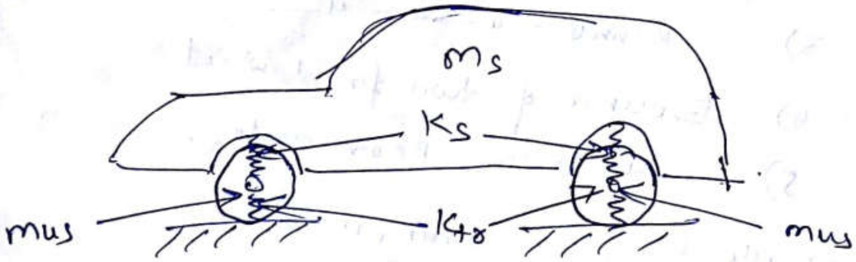
To simplify we consider only some of its major motions:

for e.g.

1) suspension functions.
2) effect spring and unsprung mass.
3) spring stiffness.
4) damping of vehicle. vibration

A linear model of 2-D.O.F may be used.

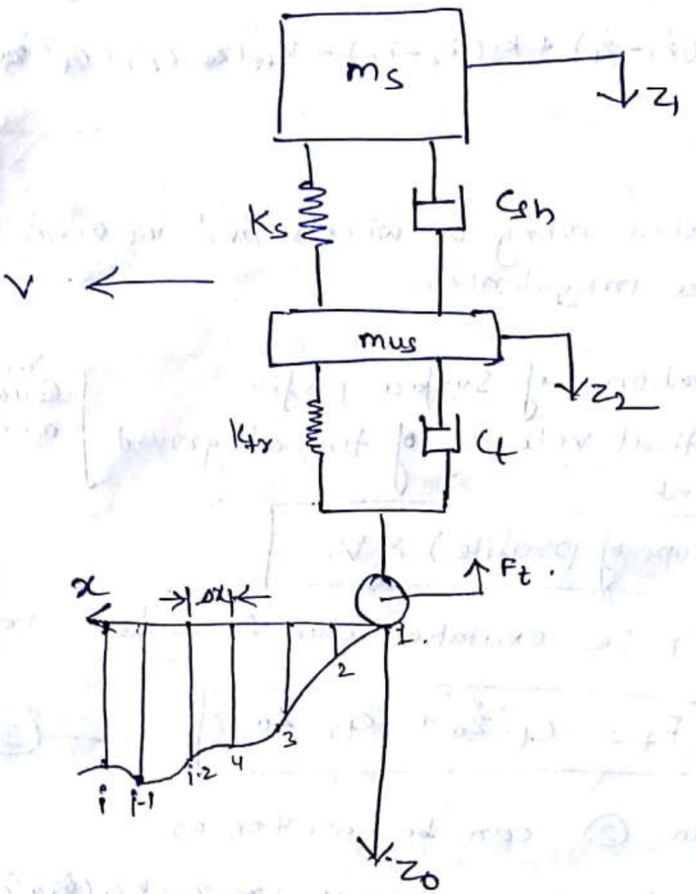
* **2-D.O.F vehicle model of spring and Unsprung mass :-**



The D.O.F includes motion of

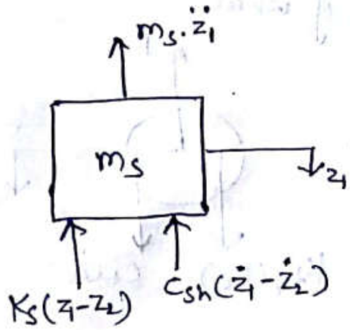
- Spring mass (Z_1) (vehicle body)
- Unsprung mass (Z_2) (wheels and associated components)

Quarter car model

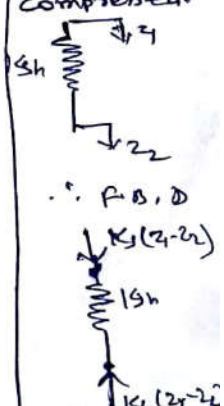


Assume, $z_1 > z_2$

F.B.D of ~~unsp~~ spring mass



As $z_1 > z_2$ so spring is compressed.

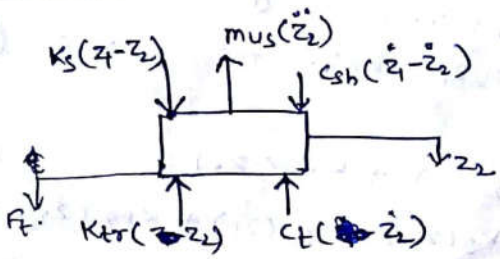


In compressed spring load applied is -

So by newton's law:

$$m_s \cdot \ddot{z}_1 + K_s(z_1 - z_2) + C_{sh}(\dot{z}_1 - \dot{z}_2) = 0 \quad \text{--- (1)}$$

Similarly F.B.D of Unspring mass



By newton's 2nd law.

$$m_{us}\ddot{z}_2 - C_{sh}(\dot{z}_1-\dot{z}_2) - K_s(z_1-z_2) + K_{t}(\cancel{z_0 -} z_2) + C_t(\cancel{\dot{z}_0 -}\dot{z}_2) = F_t$$

$$m_{us}\cdot\ddot{z}_2 + C_{sh}(\dot{z}_2-\dot{z}_1) + K_s(z_2-z_1) + K_{tr}(\cancel{z_0 -} z_2) + C_t(\cancel{\dot{z}_0 -}\dot{z}_2) = F_t \quad \text{--- (2)}$$

where,

F_t : excitation acting on wheels and induced by surface irregularities.

and,

z_0 = elevation of surface profile
$\dot{z}_0$ = vertical velocity of tire at ground point
] In fig of Quarter car model

$$\boxed{\dot{z}_0 = (\text{slope of profile}) \times V.}$$

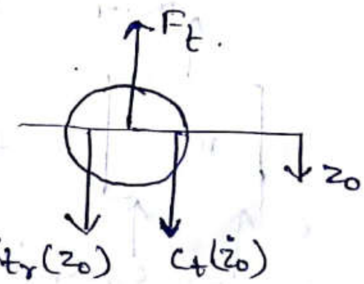
so, F_t i.e excitation due to surface irregularities

$$\boxed{F_t = C_t\cdot\dot{z}_0 + K_{tr}\cdot z_0} \quad \text{--- (3)}$$

So that equn (2) can be written as

$$m_{us}\cdot\ddot{z}_2 + C_{sh}(\dot{z}_2-\dot{z}_1) + K_s(z_2-z_1) + K_{tr}(\cancel{z_0 -} z_2) + C_{tr}(\cancel{\dot{z}_0 -}\dot{z}_2) = C_t\cdot\dot{z}_0 + K_{tr}z_0 \quad \text{--- (4)}$$

As F.B.D of wheel.



Hence Now to determine natural frequency of 2-D.O.F the equation of motion of (1) and (2) (without considering damping)

equn (1),
$$m_s\cdot\ddot{z}_1 + K_s(z_1-z_2) = 0$$
$$m_s\cdot\ddot{z}_1 + K_s z_1 - K_s z_2 = 0 \quad \text{--- (5)}$$

and equn (2),
$$m_{us}(\ddot{z}_2) + K_s(z_2-z_1) + K_{tr}(z_2) = 0$$
$$m_{us}(\ddot{z}_2) + K_s(z_2) - K_s(z_1) + K_{tr}(z_2) = 0 \quad \text{--- (6)}$$

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Solution of equtn (5) and (6) is

$$z_1 = Z_1 \cos \omega_n t \quad \text{------ (7)}$$
$$z_2 = Z_2 \cos \omega_n t \quad \text{------ (8)}$$

where,

ω_n = Undamped angular natural frequency

Z_1, Z_2 = Amplitudes of spring and unspring mass.

Substituting the assumed solution into equtn (5) and (6).

i.e. equn (5),

$$m_s \ddot{z}_1 + K_s z_1 - K_s z_2 = 0$$

$\therefore$

$z_1 = Z_1 \cos \omega_n t$ and $z_2 = Z_2 \cos \omega_n t$

$\dot{z}_1 = -Z_1 \omega_n \sin \omega_n t$, $\dot{z}_2 = -Z_2 \omega_n \sin \omega_n t$

$\ddot{z}_1 = -Z_1 \omega_n^2 \cos \omega_n t$, $\ddot{z}_2 = -Z_2 \omega_n^2 \cos \omega_n t$

So,

$$m_s(-Z_1 \omega_n^2 \cos \omega_n t) + K_s(Z_1 \cos \omega_n t) - K_s(Z_2 \cos \omega_n t) = 0$$

$$\left[(-m_s \omega_n^2 + K_s) Z_1 - K_s Z_2\right] \cos \omega_n t = 0.$$

or

$$(-m_s \omega_n^2 + K_s) Z_1 - K_s Z_2 = 0. \quad \text{------ (9)}$$

Similarly for equtn (6).

$$m_{us}(\ddot{z}_2) + K_s(z_2) - K_s(z_1) + K_{tr}(z_2) = 0$$

$$m_{us}(-Z_2 \omega_n^2 \cos \omega_n t) + K_s(Z_2 \cos \omega_n t) - K_s(Z_1 \cos \omega_n t) + K_{tr}(Z_2 \cos \omega_n t) = 0$$

$$-m_{us} Z_2 \omega_n^2 + K_s \cdot Z_2 - K_s Z_1 + K_{tr} \cdot Z_2 = 0$$

$$-K_s Z_1 + (-m_{us} \cdot \omega_n^2 + K_s + K_{tr}) \cdot Z_2 = 0. \quad \text{------ (10)}$$

Now solving equtn (9) and (10) by matrix determinant method.

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So equtn (9) and (10) are.

$$(-m_s \omega_n^2 + K_s) Z_1 - K_s Z_2 = 0$$

$$-K_s Z_1 + (-m_{us} \omega_n^2 + K_s + K_{tr}) Z_2 = 0$$

By matrix, ~~det~~

$$\begin{vmatrix} (-m_s \omega_n^2 + K_s) & -K_s \\ -K_s & (-m_{us} \cdot \omega_n^2 + K_s + K_{tr}) \end{vmatrix} = 0.$$

Expanding the determinant leads to characteristic equtn.

$$\left[(-m_s \omega_n^2 + K_s)(-m_{us} \omega_n^2 + K_s + K_{tr})\right] - \left[K_s^2\right] = 0$$

$$\left[\omega_n^4 (m_s \cdot m_{us}) - K_s (m_{us}) \omega_n^2 - m_s K_s \omega_n^2 + K_s^2 - m_s K_{tr} \omega_n^2 + K_{tr} \cdot K_s\right] - \left[K_s^2\right] = 0$$

$$\omega_n^4 (m_s m_{us}) + \omega_n^2 (-m_s K_s - m_s K_{tr} - m_{us} K_s) + K_s K_{tr} + \cancel{K_s^2} - \cancel{K_s^2} = 0$$

$$\therefore \omega_n^4 (m_s m_{us}) + \omega_n^2 (-m_s K_s - m_s K_{tr} - m_{us} K_s) + K_s K_{tr} = 0$$

So, solution of above equtn is

$$\omega_{n_{1,2}}^2 = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A}$$

where,

$$B = -(m_s K_s + m_s K_{tr} + m_{us} K_s)$$

$$A = m_s \cdot m_{us}$$

$$C = K_s \cdot K_{tr}$$

Although each of these leads to frequencies $\pm \omega_{n_1}$ and $\pm \omega_{n_2}$, the negative values are discarded as being of no physical significance.

So, the corresponding natural frequency is Hz.

$$f_{n_1} = \frac{\omega_{n_1}}{2\pi}$$

$$f_{n_2} = \frac{\omega_{n_2}}{2\pi}$$

For Passenger cars,

$$m_s > m_{us}$$

and $K_s < K_{tr}$

Because of this two condition an approximate method is used to determine the two natural frequency of two system

The approximate values of undamped natural frequency in Hz

Spring mass frequency,

$$\boxed{f_{n-s} = \frac{1}{2\pi}\sqrt{\frac{\left[\frac{K_s \cdot K_{tr}}{(K_s + K_{tr})}\right]}{m_s}}}$$

Unsprung mass frequency,

$$\boxed{f_{n-us} = \frac{1}{2\pi}\sqrt{\frac{K_s + K_{tr}}{m_{us}}}}$$

* **Note**-

① $f_{n-us} > f_{n-s}$.

② In passenger car & for shock absorber is 0.2 to 0.4 and the damping of tire is relatively Insignificant. Hence there is little difference between Undamped and damped natural frequency and hence frequency of ~~damper~~ undamped natural frequency is used to characterise the system.

(3) so, as $f_{nus} > f_{ns}$.

so if a tire hits bumps the impulse will set wheel to oscillate. with f_{n-us} while ~~the~~ for the Spring mass the excitation will be the vibration of Unspring mass.

therefore ~~for~~ for spring mass the ratio of excitation ~~fr~~ frequency to natural frequency i.e (ω/ω_n) will be.

$$\left(\frac{f_{n-us}}{f_{n-s}}\right).$$

but since $f_{nus} > f_{n-s}$

" the amplitude of spring mass will be very small." ∴

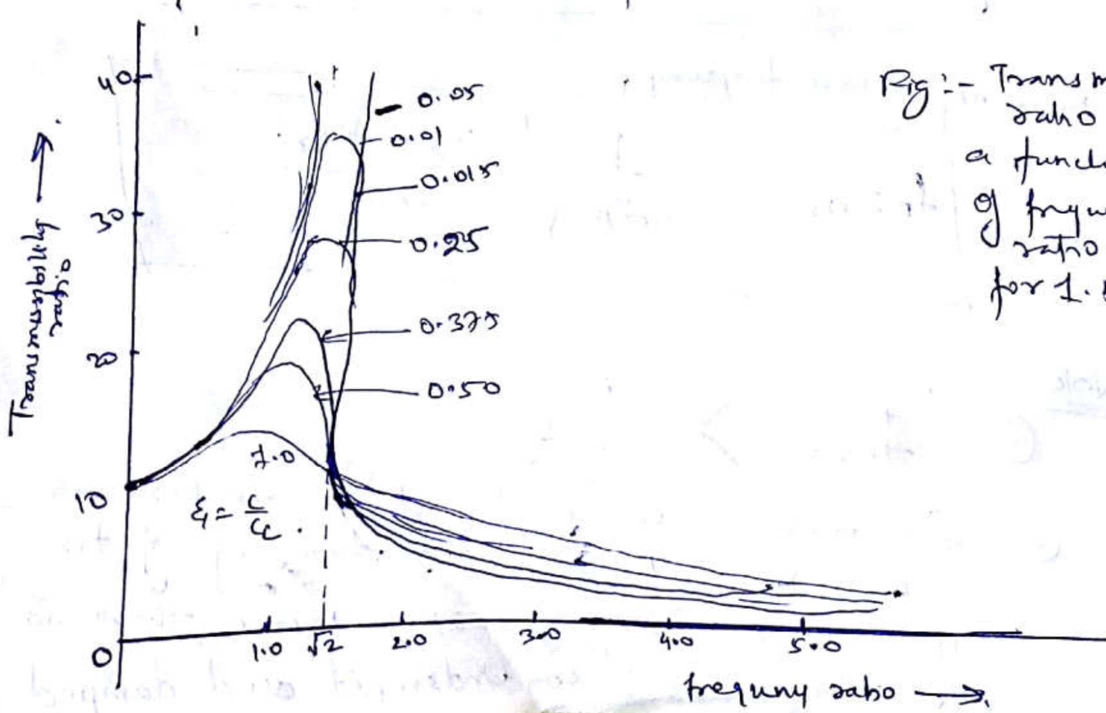


Fig :- Transmissibility ratio as a function of frequency ratio for 1.0.0.12

when the vehicle travels over an irregular surface, the excitation will normally consist of wide range of frequencies. As seen in the fig of M.F Vs frequency ratio.

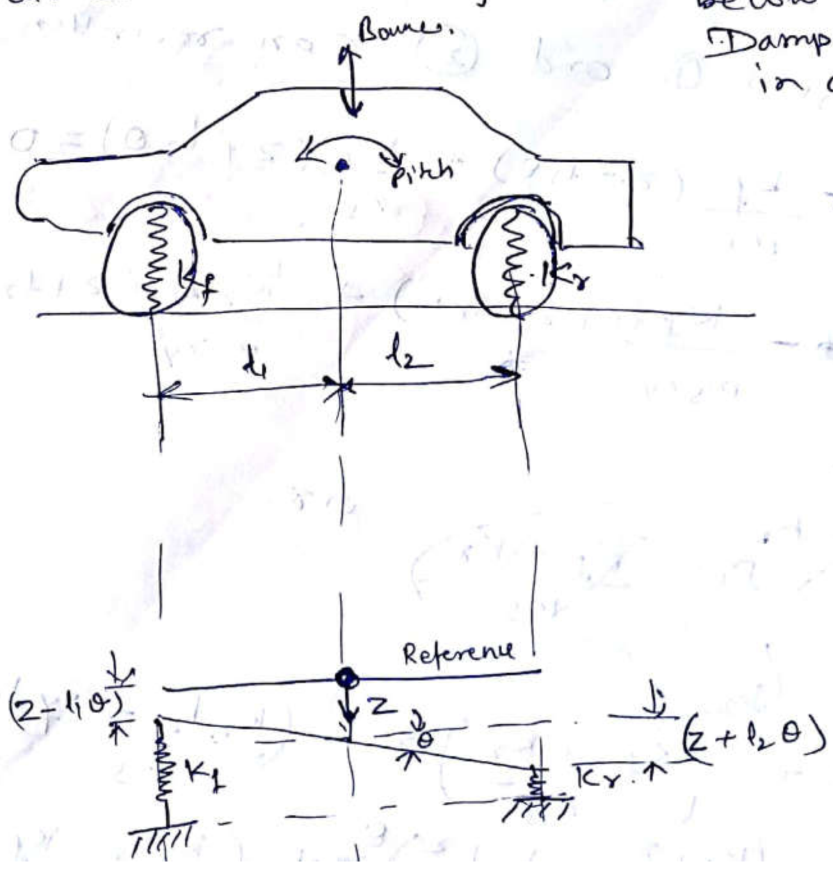
At high frequency ratio $\left(\frac{f_{n-us}}{f_{n-s}}\right)$ i.e at high frequency input can be effectively ~~[illegible]~~ isolated through the Suspension.

But low frequency excitation can however be transmitted to the vehicle body unimpeded, or even amplified as transmissibility ratio is high when excitation is close to natural frequency of spring mass

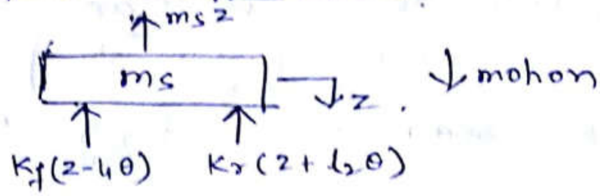
2-D.O.F vehicle Model for Pitch and Bounce :-

Because of different natural frequency of sprung and unsprung mass bounce and pitch of the vehicle body and the motion of wheels may be considered to exist almost independently. It can be studied by below model.

Damping is neglected in above model.



(i) ~~Eqn~~ F.B.D for Bounce motion

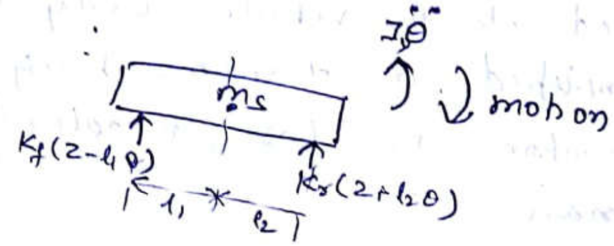


so By newton's 2nd law eqn becom

$$m_s(\ddot{z}) + K_f(z-l_1\theta) + K_r(z+l_2\theta) = 0 \quad \text{--- (1)}$$

This is the equation of motion for bounce.

(ii) F.B.D for Pitch motion



~~Iy~~ By Newton's 2nd law of motion

$$I_y\ddot{\theta} - K_f(z-l_1\theta)\cdot l_1 + K_r(z+l_2\theta)\cdot l_2 = 0$$

$$(m_s r_y^2)\ddot{\theta} - K_f l_1(z-l_1\theta) + K_r\cdot l_2(z+l_2\theta) = 0 \quad \text{--- (2)}$$

The eqns (1) and (2) can rewritten as

$$\ddot{z} + \frac{K_f}{m_s}(z-l_1\theta) + \frac{K_r}{m_s}(z+l_2\theta) = 0$$

and $$\ddot{\theta} - \frac{K_f l_1}{m_s r_y^2}(z-l_1\theta) + \frac{K_r l_2}{m_s r_y^2}(z+l_2\theta) = 0$$

or.

~~So let $D_1 = \frac{K_f + K_r}{m_s}$, D_2~~

$$\ddot{z} + \left(\frac{K_f + K_r}{m_s}\right)z + \left(\frac{K_r l_2 - K_f l_1}{m_s}\right)\theta = 0.$$

$$\ddot{\theta} + \left(\frac{K_f l_1^2 + K_r l_2^2}{m_s r_y^2}\right)\theta + \left(\frac{K_r l_2 - K_f l_1}{m_s r_y^2}\right)\cdot z = 0$$

So,

let.

$$D_1 = \frac{K_f + K_r}{m_s}$$

$$D_2 = \frac{(K_r l_2 - K_f l_1)}{m_s}$$

$$D_3 = \frac{1}{m_s r_y^2}(K_f l_1^2 + K_r l_2^2)$$

So equtn can further be written as.

$$\ddot{z} + D_1 z + D_2 \theta = 0 \quad \text{--- (3)}$$

$$\ddot{\theta} + D_3 \theta + \frac{D_2}{r_y^2} \cdot z = 0 \quad \text{--- (4)}.$$

It is evident from both equtn that D_2 is coupling coeff. for bounce & pitch motion. and these motion uncouples when $D_2 = 0$ i.e $K_r l_2 = K_f l_1$

when this occurs,

(1) A force applied on C.G. will induces only bounce motion

(2) while a moment applied will produce pitch motion.

In such case natural frequency is given by:

$$\omega_{nz} = \sqrt{D_1}$$

$$\omega_{n\theta} = \sqrt{D_3}$$

It is found this results in poor ride.

But in general

Pitch and Bounce are coupled and an force on front ~~and~~ or rear will excite both motions.

So it become 2-D.O.F problem and solution for the equtn comes as

$$z = Z \cos \omega_n t$$

$$\theta = \Theta \cos \omega_n t$$

$$\ddot{z} = -Z\omega_n^2 \cos \omega_n t$$

$$\ddot{\theta} = -\Theta \omega_n^2 \cos \omega_n t$$

So substituting in equn. (3) ~~and (4)~~

$$-Z\omega_n^2 \cos\omega_n t + D_1 Z \cos\omega_n t + D_2 \Theta \cos\omega_n t = 0.$$

$$-Z\omega_n^2 + D_1 Z + D_2 \Theta = 0$$

$$(D_1 - \omega_n^2) Z + D_2 \Theta = 0. \qquad \text{—— (5)}$$

Similarly in equtn (4)

$$-\Theta \omega_n^2 \cos\omega_n t + D_3 \Theta \cos \omega_n t + \frac{D_2}{r_y^2} \cdot Z \cos\omega_n t = 0$$

$$-\Theta \omega_n^2 + D_3 \Theta + \frac{D_2}{r_y^2} \cdot Z = 0$$

$$\left(\frac{D_2}{r_y^2}\right) Z + (D_3 - \omega_n^2)\Theta = 0 \qquad \text{—— (6)}$$

Solving equtn (5) and (6) by matrix Determinant method.

$$(D_1 - \omega_n^2) Z + D_2 \theta = 0$$

$$\left(\frac{D_2}{r_y^2}\right) Z + (D_3 - \omega_n^2)\theta = 0$$

$$\begin{vmatrix} (D_1 - \omega_n^2) & D_2 \\ \left(\frac{D_2}{r_y^2}\right) & (D_3 - \omega_n^2) \end{vmatrix} = 0$$

$$(D_1 - \omega_n^2)(D_3 - \omega_n^2) - \frac{D_2^2}{r_y^2} = 0.$$

$$D_1 \cdot D_3 - D_1\omega_n^2 - D_3\omega_n^2 + \omega_n^4 - \frac{D_2^2}{r_y^2} = 0$$

$$\omega_n^4 - (D_1 + D_3)\omega_n^2 + \left(D_1 D_3 - \frac{D_2^2}{r_y^2}\right) = 0$$

Solving the

$$\omega_{n_{1,2}}^2 = \frac{(D_1 + D_3) \pm \sqrt{(D_1 + D_3)^2 - 4\left(D_1 D_3 - \frac{D_2^2}{r_y^2}\right)}}{2}$$

$$\omega_{n_{1,2}} = \frac{(D_1 + D_3) \pm \sqrt{D_1^2 + D_3^2 + 2D_1D_3 - 4D_1D_3 - \frac{4D_2^2}{r_y^2}}}{2}$$

$$\omega_{n_{1,2}} = \frac{1}{2}(D_1 + D_3) \pm \frac{1}{2}\sqrt{D_1^2 + D_3^2 - 2D_1D_3 - \frac{4D_2^2}{r_y^2}}$$

$$\omega_{n_{1,2}} = \frac{1}{2}(D_1 + D_3) \pm \frac{1}{2}\sqrt{(D_1 - D_3)^2 - \frac{4D_2^2}{r_y^2}}$$

so the solution becoms.

$$\omega_{n_1} = \frac{1}{2}(D_1 + D_3) - \sqrt{\frac{1}{4}(D_1 - D_3)^2 - \frac{D_2^2}{r_y^2}} \quad \text{—— (7)}$$

$$\omega_{n_2} = \frac{1}{2}(D_1 + D_3) + \sqrt{\frac{1}{4}(D_1 - D_3)^2 - \frac{D_2^2}{r_y^2}} \quad \text{—— (8)}$$

from equtn (5) and (8) the amplitude ratio of the bounce and pitch oscillations for two natural frequency can be determined.

for ω_{n_1}, Amplitude ratio i.e. $\frac{\text{Bounce}}{\text{Pitch}}$ from equtn (

~~Amp~~

$$\left.\frac{Z}{\theta}\right|_{\omega_{n_1}} = \frac{D_2}{-(D_1 - \omega_{n_1}^2)}$$

$$\left.\frac{Z}{\theta}\right|_{\omega_{n_1}} = \frac{D_2}{\omega_{n_1}^2 - D_1}$$

Similalarly

for ω_{n_2}, Amplitude ratios i.e $\frac{\text{Bounce}}{\text{Pitch}}$ from equtn

$$\left.\frac{Z}{\theta}\right|_{\omega_{n_2}} = \frac{D_2}{\omega_{n_2}^2 - D_1}$$

— To farther illustrate the characteristics of bounce and pitch modes of oscillation, the concept of "oscillation centre" is introduced.

— It measured from C.G. of vehicle. and ~~it~~ is determined by Amplitude ratios and denoted by l_o.

— There will be two centres
- one centre associated with ω_{n_1}
- second — " — " — " — ω_{n_2}

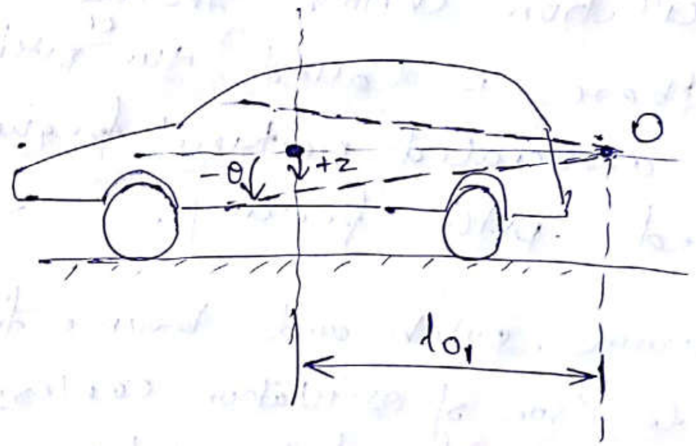
For ω_{n_1} it is

$$l_{o_1} = \frac{D_2}{\omega_{n_1}^2 - D_1}$$

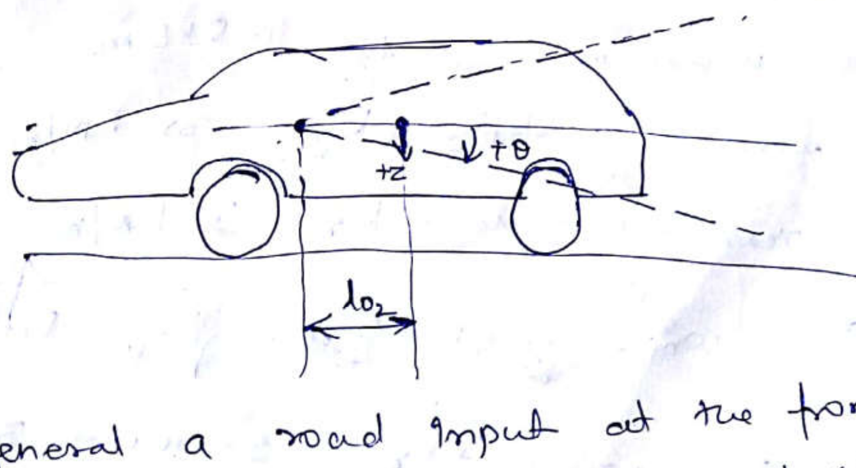
and for ω_{n_2}

$$l_{o_2} = \frac{D_2}{\omega_{n_2}^2 - D_1}$$

- when Amplitude ratio is -ve, oscillation center will be located to right of C.G. of vehicle. as shown in fig.



- when Amplitude ratio is +ve, the oscillation centre is located left to C.G of vehicle. as shown in fig.



In general a road input at the front or rear wheel will cause a moment about each oscillation center, and therefore will excite both bounce and pitch oscillations.

or we can say,

"the body motion will be the sum of oscillations about the two centers".

- Oscillation center ~~lieing~~ ~~lyi~~ outside the wheelbase is called "bounce center" and associated frequency is called bounce frequency.

- Oscillation center inside the wheelbase is called the "pitch center" and associated natural frequency is called pitch frequency.

<u>Ques</u> Determine pitch and bounce frequency and location of oscillation centers of an vehicle with following data

Spring mass, $m_s = 2120\,kg$

radius of gyration, $r_y = 1.33\,m$.

Dist. from front axle to C.G $d_1 = 1.267\,m$

— " — " rear — " — " — " — $d_2 = 1.548\,m$.

front spring stiffness $K_f = 35\,kN/m$.

rear — " — " — $K_r = 38\,kN/m$

<u>Ans</u>

$$\omega_{n_1}^2 = \frac{1}{2}(D_1 + D_3) - \sqrt{\frac{1}{4}(D_1 - D_3)^2 + \left(\frac{D_2}{r_y}\right)^2}$$

$$\therefore D_1 = \frac{K_f + K_r}{m_s} = \frac{35000 + 38000}{2120}$$

$$\boxed{D_1 = 34.43\ s^{-2}}$$

$$D_2 = \frac{K_r l_2 - K_f l_1}{m_s}$$

$$= \frac{(38000 \times 1.548) - (35{,}000 \times 1.267)}{2120}$$

$$\boxed{D_2 = 6.83 \ m/s^2}$$

$$D_3 = \frac{K_f l_1^2 + K_r l_2^2}{m_s r_y^2}$$

$$= \frac{[35000 \times (1.267)^2] + [38000 \times (1.548)^2]}{2120 \times (1.33)^2}$$

$$\boxed{D_3 = 39.26 \ s^{-2}}$$

$$\boxed{\left(\frac{D_2}{r_y}\right)^2 = 26.37 \ s^{-4}}$$

$$D_3 + D_1 = 73.69 \ s^{-2}$$

$$D_3 - D_1 = 4.83 \ s^{-2}$$

so,

$$\boxed{\omega_{n_1}^2 = 31.17 \ s^{-2}}$$

$$\boxed{\omega_{n_1} = 5.58 \ s^{-1}}$$

$$\boxed{f_{n_1} = 0.89 \ Hz}$$

Similarly

$$\omega_{n_2} = \frac{1}{2}(D_1 + D_3) + \sqrt{\frac{1}{4}(D_1 - D_3)^2 + \left(\frac{D_2}{\gamma_4}\right)^2}$$

$$\boxed{\omega_{n_2}^2 = 42.52\, s^{-2}}$$

$$\omega_{n_2} = 6.52\, s^{-1}$$

$$\boxed{f_{n_2} = 1.04\, Hz}$$

location of oscillation center.

$$l_{o_1} = \left.\frac{z}{\theta}\right|_{\omega_{n_1}} = \frac{D_2}{\omega_{n_1}^2 - D_1}$$

$$= \frac{6.83}{31.17 - 34.43}$$

$$= -2.09\, m.$$

$$l_{o_2} = \left.\frac{z}{\theta}\right|_{\omega_{n_2}} = \frac{D_2}{\omega_{n_2} - D_1}$$

$$= \frac{6.83}{42.52 - 34.43}$$

$$= +0.84\, m.$$

Active and Semi-active Suspension

- As discussed in vehicle ride model, to acheive good vibration isolation for the spring mass over a wide range of frequency a soft suspension spring is required.

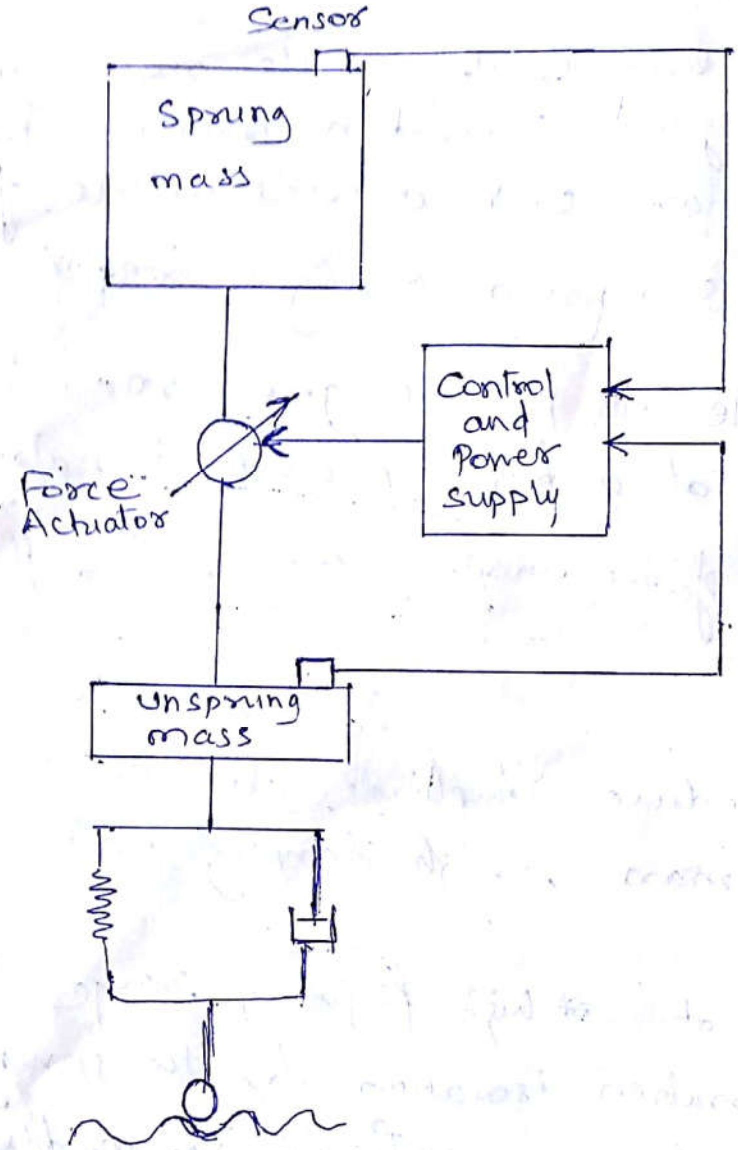
- While to provide good road holding capability at a frequency close to natural ~~frequency~~ frequency of the unsprung mass, a stiff spring is required.

- To reduce amplitude close to ω_n of ~~Spring~~ Spring mass, high damping ratio is required.

- While at ~~of~~ high frequency range to provide good vibration isolation for the spring mass a low damping ratio (ξ) is preffered

- To provide good roadholding capability in high frequency range, a high damping ratio is required.

These requirements can't be met by conventional (Passive) Suspension system, because the characteristics of spring and shockabsorber are fixed and cannot be changed in accordance to operating conditions.



Active suspension system provides:

1) Better ride quality
2) —"— handling
3) —"— performance under various operating conditions.

Spring and shock absorber (conventional or Passive Suspension system) $\xrightarrow[\text{by}]{\text{Replaced}}$ Actuator (Active Suspension system)

*Note :- The actuator can also be Installed parallel with Conventional System.

- In active suspension system, sensor continously monitor the operating conditions of vehicle.

- Based on the signal obtained by the sensors, the control system continously changes the force in the actuator to achieve improved ride, handing and performance.

~~Generally the~~ Generally the optimum control strategy is defined as the one that minimizes the following:-

1) The root mean square (rms) value of the spring mass acceleration
2) The rms of suspension travel.
3) The rms value of the dynamic tire deflection.

Usually these quantities are multiplied by weighting ~~factors~~ factors, then combined to form an evaluation function.

An active suspension can also be used to ~~to~~ control the height, roll, dive (forward pitching) and squat (rearward pitching). of the vehicle body.

Following advantages

1) height of vehicle can be kept constant despite. changes in load
2) Adequate suspension can be provided for negotiating bumps.
3) Reduce Aerodynamic resistance and lift.

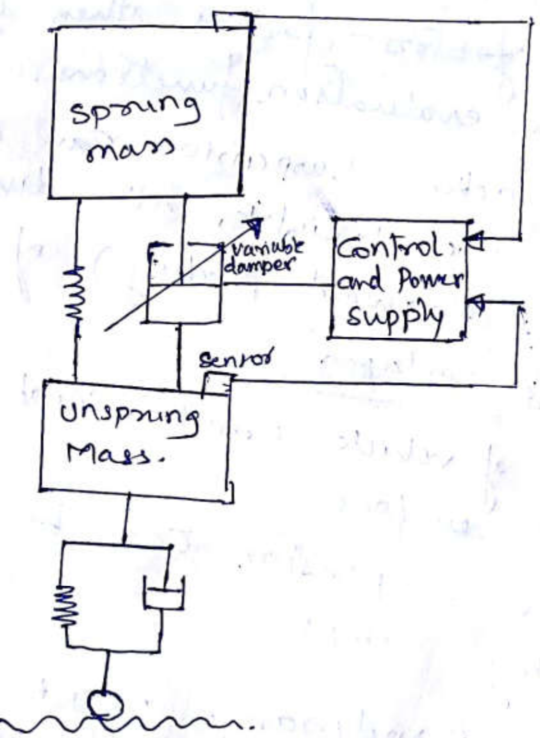
3) ground clearance can be adjusted easily
4) During ~~roll~~ cornering, Anti-roling forces Can be produced

Disadvantages

1) Requires significant external power to operate.
2) It makes the ~~complex~~ system complex.
3) less reliable.
4) more cost and weight.

Semi-Active Suspension System :-

To reduce complexity and cost, the concept of semi-active suspension has emerged.
In this system the ~~damper~~ damping force of damper can be modulated in accordance of operating condition. The spring remains same as conventional system.



The variation in damping force can be achieved by following means :-

1) Adjusting orifice area in the shock absorber (i.e making holes in piston of damper.) hence changing resistance to fluid flow.
2) using fluids like electro-rheological and magneto-rheological fluids in the damper.

Electro-rheological fluid :-

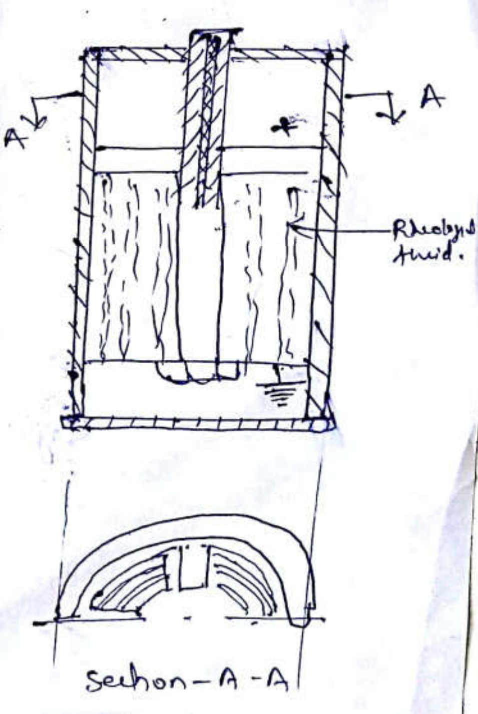
It is mixture of dielectric base oil and fine semi-conducting particles.

In electric field, this fluid thickens, and hence apparent viscosity can be controled and hence resistance to flow can be achieved.

The electric field can be varied by ~~app~~ regulating voltage across the fluid.

Hence, by regulating the voltage the damping force can be varied.

major disadvantage of the system is to develop a rheological fluid which can produce adequate shear strength fluid that could operating in a temperature $-40^\circ C$ to $+120^\circ C$.



Section - A - A

Magneto-rheological fluid :-

A magneto-rheological fluid is mixture of micron-sized magnetizable particles suspended in carrier fluid like silicon oil.

By changing magnetic field, the apparent viscosity of damper oil can be changed.

Hence resistance to flow can be changed by changing the magnetic field around oil.

Advantages

1) As comparision to electro-rheological fluid, the magneto-rheological fluid properties have greater stabality over wide range of temperature.

2) Magneto-rheological fluid have high shear strength.

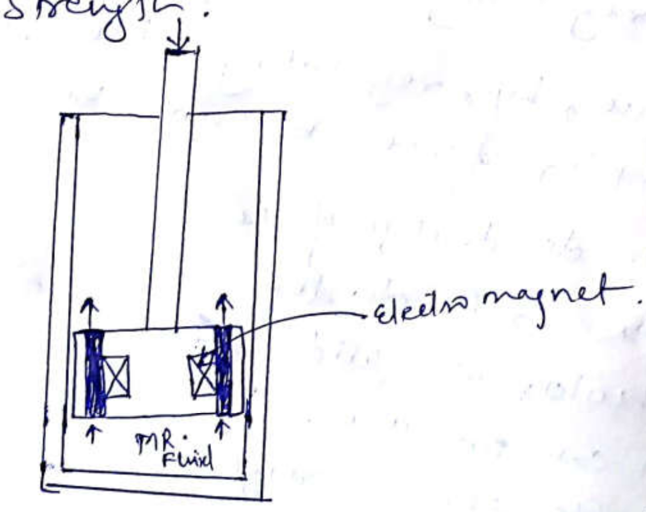


Fig:- ~~Electro~~ Magneto-rheological damper.