Design and Modeling of Multibody Dynamics of Light Motor Vehicle

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Abstract

This paper presents the design and modelling of multibody dynamics of light motor vehicle with the related search. The study specifies factors influencing the motion of light motor vehicle. These are based on a systematic study of dynamic forces acting on the vehicle body and analysis of stresses acting on it.
Keywords: Tata Sumo, Light Motor Vehicle, Modelling, Dynamic Analysis

I. INTRODUCTION

Multi-body dynamic analysis is the dynamic analysis of mutually interconnected rigid bodies, whose relative motions are constrained by means of joints. The purpose of this analysis is to find out how these bodies move as system and what forces are generated in the process. The multi-body dynamic analysis is typically applied in automobile industry for the modelling and analysis of suspension system. The multi-body suspension models allow precise evaluation of the effect of suspension geometry and the mechanical characteristics of spring and damper on the ride comfort and vehicle handling performance.

In the automotive industry, computer simulations of Light Motor Vehicle durability early on in the design process are becoming more and more important in order to decrease development cost and product time to market. Accurate calculations of force histories are of utmost importance for reliable fatigue life estimates. The forces are often calculated by use of multi-body software and used as input for stress analysis in a FEA package.

As the project is being done in collaboration with TATA MOTORS, the Dynamic Analysis is carried on TATA SUMO.

II. LITERATURE REVIEW

[1] This paper gives information about accurate calculations of force histories for reliable fatigue life estimates. The forces are often calculated by use of multi-body software and used as input for stress analysis in a FE package. The calculated forces are compared with measurements for different road inputs. One type of analysis that is growing in importance is the simulation of forces for durability assessment. However, most (all) methods use multi-body simulation (MBS) codes to perform the calculation of the forces. For truck simulations it is vital to have a flexible representation of the frame.The objective of this study has been to find a more efficient method for the calculation of forces, which allows for fast analysis and easy implementation of flexible bodies.

[2] In this paper, electronic stability control functions are applied on trucks to enhance safety. The challenge of designing such functions is to guarantee robustness for all truck variations, such as lay-out, length, mass, number of axles and number of articulations. Therefore, fundamental understanding of the effect of these variations on the dynamic yaw behaviour of articulated vehicles is required. The results presented in this report pertain to optimal driving conditions. It is recommended to investigate the effect of excitations of the truck other than the steering wheel input on the dynamic performance of truck combinations.

[3] This paper describes a designing and analysis of earth mover – wheel loader vehicle using SolidWorks 2009 and MSC Visual Nastran Desktop 4D. Wheel loader vehicle is a multi body system and to design and analyze such system requires profound knowledge of highly challenging subjects like Computer Aided Design, Finite Element Analysis, machine design,reverse engineering and structural dynamics etc. This project work shows how to implement engineering fundaments into the real world application. In this project, designing has been done by implementing reverse engineering concept. Initially, the dimensions of wheel loader toy have been documented by vernier caliper and full scale ruler.

[5] There are some specific tasks which are to be performed only at the site. For the accomplishment of such tasks heavy machines are employed. When such heavy mechanisms are moved, dynamic forces are induced into the system during

transportation. The magnitude of these forces increases as the speed increases and are transmitted to the vehicle frame which disturbs the smooth movements of the vehicle. The velocity and acceleration of points in the given vehicle volume are determined. These values are then plotted in the given volume which will give clear pattern of the distribution of the acceleration. This distribution will help in identifying the worst and suitable zones. Based on the motion analysis the acceleration values are calculated for the various locations in the vehicle volumes.

[11] Many industries in the last decade successfully implemented Finite Element Analysis as a tool for product development. The success of FEA is now being followed by more widely used motion analysis software. These two numerical tools are often used together to put new designs through their paces, rather than doing almost the same with costly physical prototypes. But the wider use of motion analysis along with FEA clouds the capabilities and limitations of each tool. To clear up the confusion, its use to examine the role of both, describe their range of applications, and explain how these two tools work together.

[10] Finite Element Analysis (FEA) helps in accelerating the design and development process by minimizing number of physical tests, thereby reducing the cost and time for prototyping and testing. Current study involves static stresses analysis of LCV chassis using FEM to simulate the failure during testing. The commercial finite element package Hyper Mesh and optistruct 0.8 was used for the solution of the problem. To achieve a reduction in magnitude of then stresses at critical stress region, local stiffeners were added. Numerical results showed that stresses on the critical locations where reduced up to 44% by adding the stiffener. Subsequently the chassis was retested and it passed the test.

[9] Truck chassis forms the structural back bone of a commercial vehicle. The main function of the truck chassis is to support the components and payload placed upon it. When the truck travels along the road the chassis is subjected to vibration induced by road roughness and excitation by vibrating components mounted on it. This paper presents the study of the vibration characteristics of the truck chassis that includes the natural frequencies and mode shapes. The response of the truck chassis which includes the stress distribution and displacement under various loading conditions. The method used in the numerical analysis is Finite Element Technique. The result shows that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range.

III. DESIGN CALCULATIONS

A. Motion Analysis:

The velocity and acceleration of points on the sumo chassis are determined for different cases. These values are then plotted in the given volume, which will give clear pattern of the distribution of the acceleration. This distribution will help in identifying the worst and suitable zones. These values of acceleration are determined by the motion analysis. The Motion analysis is done considering following cases.

1) Modes of movements
   - Vehicle in accelerating mode
   - Vehicle in braking mode
   - Vehicle in turning mode
2) Road conditions
   - Vehicle through pit condition
   - Vehicle over bump condition

The acceleration at a point is expressed as $Ap(x,y,z)_{x/y/z}$ ie. Acceleration (location of the point in the volume)$_{direction}$

B. Modes of Movements:

Modes of movements is one of the case considered in motion analysis the vehicle moves under different modes of movements which gives variation in acceleration points on the sumo chassis. The variation of acceleration points gives raise to variations in dynamic forces. The modes of movements like acceleration, braking and turning of vehicle are discussed in a detail.

1) Vehicle in Accelerating Mode:

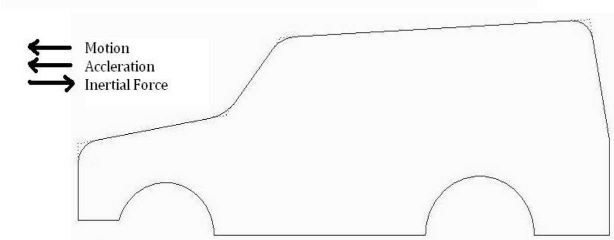


Fig. 1: Vehicle in accelerating mode

When vehicle is moving on a straight path & accelerating in the direction of motion, the inertial forces will be in opposite direction. Suppose that vehicle is moving in one direction the inertial forces will be opposite to the direction of motion.[Fig 1]

Therefore the acceleration of a points on the sumo chassis due to acceleration of the vehicle can be explained as follows.

$$A_A\ p\ (x,y,z)x = \frac{Fa}{m}$$

Where Fa is the accelerating force & can be determined with the help of the torque required, which in turn is found out from the power of the vehicle.

$$P = \frac{2\pi NT}{60}$$

Therefore $T = \frac{60P}{2\pi N}$

and

$$T = Fa\ X\ R_w$$

The acceleration of the point due to acceleration of vehicle will be along the X direction only i.e. along the length of the vehicle.

2) Vehicle in Braking Mode:

When the vehicle is moving on a straight path there is relative acceleration due to change in velocity of the vehicle and the inertial force is acting to the direction of the motion of vehicle. When the brakes are applied, the acceleration due to braking is opposite to the direction of the motion of the vehicle [Fig. 2].

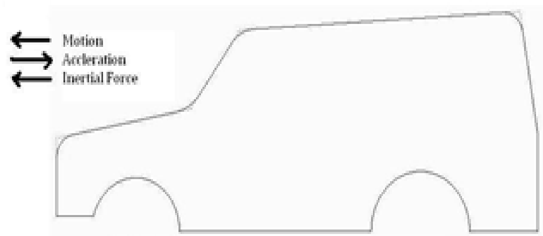


Fig. 2: Vehicle in Braking Mode:

The acceleration of a point on the sumo chassis will be a function of x, y and z direction. Therefore the acceleration of a point due to the braking is in the x direction i.e. direction along the length of the vehicle and can be determined as follows.

$$A_B\ p(x,y,z)x = \frac{Fb}{m}$$

The acceleration in y and z direction will be zero. The acceleration due to braking will be same along the length of the vehicle i.e. in the x direction. So there is no variation in acceleration along the x axis.

3) Vehicle in Turning Mode:

There can be a situation when the vehicle is moving on a curved track. Suppose that the vehicle is turning in right direction.The steering gear mechanism is used for changing the direction of the wheel axles with reference to the chassis, so as to move the vehicle in any desired path. Usually the back wheels have a common axis, which is fixed in direction with reference to the chassis of the vehicle and the steering is done by means of the front wheels. The front wheels are placed over the front axle. The back wheels are placed over the back axle. When the vehicle takes a turn, the front wheels along with the respective axle turns about the respective pivoted points. The back wheels remain straight and do not turn therefore the steering is done by means of front wheels only.

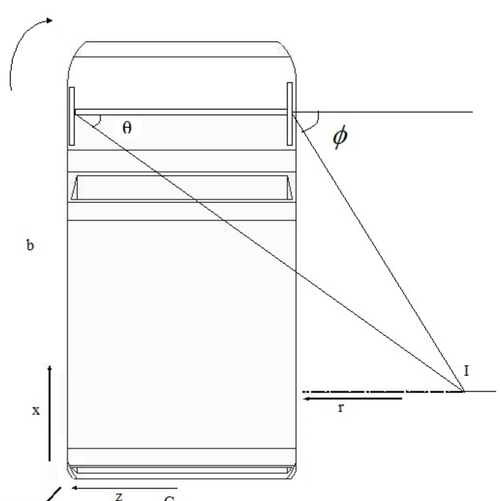


Fig 3.3 Carbody in Turning mode

Fig. 3: Vehicle in Turning Mode

In order to avoid skidding (i.e. slipping of the wheels side ways), the front wheels must turn about the same instantaneous center 'I' which lies on the axis of the back wheels. If the instantaneous center of the front wheel do not coincide with the instantaneous center of the back. wheels, the skidding on the front or back wheels will definitely take place which will cause more wear and tear of the tyres.

Thus the condition for correct steering is that all wheels must turn about the same instantaneous centre. The axis of the inner wheels makes a large turning angle θ than the angle ϕ subtended by the axis of outer wheel.

The fundamental equation for correct steering is Cot ϕ – Cot θ = C/b. If this condition is satisfied, there will be no skidding of the wheels, when the vehicle takes turn.

When the vehicle is moving on a curved track of Radius Rc, the vehicle will experience an angular speed with respect to the radius of curve about vertical y axis.

$$\omega_c = Vv / Rc$$

Every body in equilibrium in a rotating system (ie in an accelerating frame, or non-inertial frame) experience a force directed away from the center of the circular path described by it.

The magnitude of this out ward force is equal to that of the centripetal force. The out ward force is called centrifugal force. This centrifugal force is called a Pseudo force. It is simply represents the effect of the acceleration of the rotating frame. Every point on the sumo chassis is influenced by the centrifugal force due to different acceleration of the points on the sumo chassis.

Assuming that the vehicle is having a uniform speed while turning. This acceleration of the points on the sumo chassis will be in xz plane ie in the z direction. The centrifugal acceleration will be

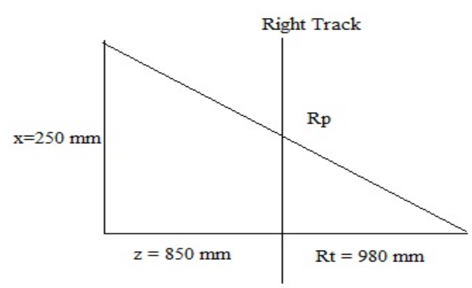
$$A_t\, p(x,y,z) = \omega^2 Rp$$

Where, A_t is the acceleration of a point expressed in the coordinate system and Rp is the radial distance of the point from the center of turn 'I' of the y axis [Fig 3.4]

The R_t is expressed in terms of the radius of turn and the x & z are the coordinates in the xz plane as the vehicle is turning about y axis

$$\text{ie } Rp = \sqrt{(R_t + z)^2 + x^2}$$

$$\text{ie }; A_t p(x,y,z) = \omega^2 \sqrt{(R_t + z)^2 + x^2}$$



Therefore acceleration expressed in terms of the velocity in xz plane will be

$$A_t\, p(x,y,z) = [Vv/Rc]^2 * \sqrt{(R_t + z)^2 + x^2}$$

From the above equation it is clear that the acceleration increase as the x & z value increases. Thus the acceleration is higher at the front and the rear over hangs as well as it is higher at the left and right over hangs.

So, the overhangs regions are the worst zones on the sumo chassis. The safest zone of minimum acceleration zone is the inner wheel position and the central zone of the sumo chassis.

C. Road Condition:

Road condition is the second case considered in motion analysis. The different road conditions are pits and bumps. When vehicle is passing through pit the vehicle experience vertical motion in downward direction and when vehicle is moving over bumps the vehicle will experience vertical motion in upward direction. Further the vehicle through pit condition and over bump condition are discussed in detail.

1) Vehicle through Pit Condition:

Considering that only one wheel of the sumo chassis is going through the pit. When the wheel is passing through the pit, it will be having a downward motion, ie it will have downward velocity. This downward velocity will be causing a downward acceleration.

Let the front right wheel of sumo chassis is passing through the pit [Fig 3.4].

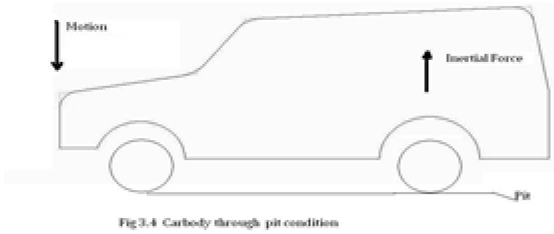


Fig. 4: Vehicle through Pit Condition

Therefore the front right corner of the sumo chassis will move in downward direction. Also the trailer will move in forward direction ie the sumo chassis will move along +ve x axis and -ve y direction. The sumo chassis of the vehicle will experience rotation about the rarer track and rotation about the left track. This rotation results in the rotation about diagonally opposite corner.

The chassis of the vehicle will experience the motion of the point about diagonally opposite corner. The front right corner of sumo chassis will perform circular motion. This point will have radial acceleration as well as tangential acceleration. So the resultant acceleration is to be calculated using radial component of acceleration and tangential component of acceleration.

The angular velocity will be

$$\omega_{p}\, p(x,y,z)_x = \frac{V_p\, p(x,y,z)_{xy}}{Z}$$

For rotation about left track and

$$\omega_{p}\, p(x,y,z)_z = \frac{V_p\, p(x,y,z)_{xy}}{X}$$

For rotation about rear axle of the sumo chassis

The resultant of the above two rotations will be

$$\omega_{p}\, p(x,y,z)_{xz} = \frac{V_p\, p(x,y,z)_{xy}}{XZ}$$

For rotation about diagonally opposite corner

The acceleration due to these angular velocities will be

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_x.z \text{ about left track}$$

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_z.x \text{ about rear axle.}$$

The result of these two acceleration will be

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_{xz}.xz$$

Considering the different speed of the vehicle ie (30 kmph (8.33 m/s), 40 kmph (11.11 m/s), 50 kmph (13.88 m/s), 80 kmph (22.22 m/s), 100 kmph (27.77 m/s), 120 kmph (33.33 m/s)), the velocity in downward direction is calculated using the following relation.

$$V_p\, p(x,y,z)_{xy} = \frac{V_v}{2\cos^2(\phi/2)}$$

The values of acceleration at these points for different speeds (30 kmph (8.33 m/s), 40 kmph (11.11 m/s), 50 kmph (13.88 m/s), 80 kmph (22.22 m/s), 100 kmph (27.77 m/s), 120 kmph (33.33 m/s)) so that zone with higher acceleration can be identified on the sumo chassis. The tangential acceleration is calculated for different point on the sumo chassis. It is seen that the front right axle region is having max. acceleration and it goes on decreasing towards rear axle.

2) Vehicle Over Bump Condition:

It is considered that only one wheel is passing over the bump. [Fig 3.5]

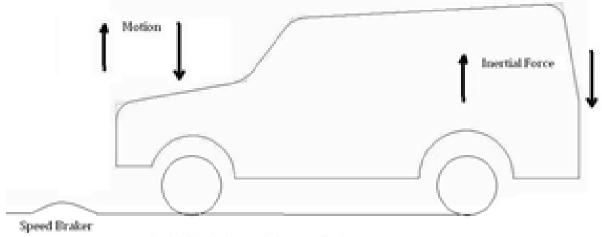


Fig. 5: Vehicle Over Bump Condition:

The vehicle will experience a velocity in y direction thus the sumo chassis of the vehicle will experience angular velocity about two axes.

The angular speed about the respective axes will depend on the vehicle speed, height and width of the bump.

Considering the different speed of the vehicle the velocity in y direction is calculated using the following relation :

$$V_{BP}\,P(x.y.z) = \frac{0.5wV_v}{\sqrt{(0.5w)^2 + h^2}}$$

D. Combination of Modes of Movements And Road Conditions :

This case includes the combination of modes of movements and road condition. Initially individual cases were consider and examine. But now this case will be having the effect of two individual cases the vehicle while turning through pit, while turning over bumps can be studied. Here only combination that is vehicle through turning pit is considered.

E. Vehicle In Turning Mode Through Pit Condition :

Supposed that the vehicle is turning right side and the back left wheel of the sumo chassis is passing through pit. Due to turning motion of sumo chassis, centrifugal forces will act on different point on sumo chassis. The maximum centrifugal forces will be act back left corner on sumo chassis. As the back left wheel is passing through pit at the same time it will have the maximum vertical velocity as well as acceleration in down word direction.

F. Dynamic Forces:

The criteria for deciding whether to treat the loads as static or dynamic, is ratio of the time required to apply the load (i.e. to increase it from zero to its full value) to the natural period of vibration. If the time of loading is greater than three times the natural period, it can be assumed as static load. On the other hand if the loading time is less than half the natural period, it is definitely an impact or dynamic load. In between in grey area an important difference between static and impact loading is that statically loaded parts must be designed to carry loads, where as parts subjected to impact must be designed to absorb energy.

For determining of dynamic forces a relation between vehicle velocity, acceleration and clearance between linkages is established. The effect of acceleration of mass in the vehicle volume, have relative motion due to the available clearance.

When both M (mass of the vehicle) and m (mass of the link) are moving in the same direction with same velocity there will be no impact. When there is relative velocity ie M and m have different velocities, the link will hit due to the kinetic energy. At the place of the contact the kinetic energy of the link will be transferred to the sumo chassis and get converted to potential energy. This potential energy is stored as strain energy. A relation can be established between stress, force, acceleration and clearance which will give the values of dynamic forces.

The relation is

$$F = \sqrt{\frac{2EmaSA}{l}}$$

Where, E = Elasticity ($2X10^5 N/mm^2$)
m = Mass of link (50 Kg)
a = Acceleration
S = Clearance
A = Area (Table5.1(a))
l = Length (Table5.1(a))

G. Modes of Movements

1) Vehicle In Accelerating Mode :

The acceleration of a points on the sumo chassis due to acceleration of the vehicle can be explained as follows.

$$A_A\,p\,(x,y,z)x = \frac{Fa}{m}$$

Where, Fa is the accelerating force
M is the mass of link

a) Power (P) :

$$P = \frac{2\pi NT}{60}$$

Get specified value of torque and engine speed is 223 Nm and 1600 RPM

$$P = \frac{2*\pi*1600*223}{60}$$

P = 37364 w

P = 37.364 Kw

Therefore,

$$T = Fa * R_w$$

$$F_a = \frac{T}{Rw}$$

Wheel radius (R_w) = 27 cm

$$F_a = \frac{223}{0.27}$$

$$F_a = 825.92 \text{ N}$$

The acceleration of the point due to acceleration of vehicle will be along the X direction only i.e. along the length of the vehicle.

b) Vehicle In Braking Mode :

The acceleration of a point due to the braking is in the x direction i.e. direction along the length of the vehicle and can be determined as follows.

$$A_B\ p(x,y,z)x = \frac{Fb}{m}$$

The acceleration in y and z direction will be zero. The acceleration due to braking will be same along the length of the vehicle i.e. in the x direction. So there is no variation in acceleration along the x axis.

$$F_b = F_a = 825.92 \text{ N}$$

c) Vehicle In Turning Mode :

The vehicle is moving on a curved track, the vehicle will experience an angular speed with respect to the radius of curve about vertical y axis.

$$\omega_c = \frac{Vv}{Rc}$$

where, V_v = Linear velocity of vehicle = 30 km/hr = 8.33 m/s
R_c = Radius of curve of turn = 1500 mm

$$\omega_c = \frac{8.33}{1.5}$$

$$\omega_c = 5.553 \text{ rad/s}$$

Assuming that the vehicle is having a uniform speed while turning. This acceleration of the points on the sumo chassis will be in xz plane i.e, in the z direction. The centrifugal acceleration will be

$$A_t\ p(x,y,z) = \omega_c^2 * Rp$$

Where, A_t is the acceleration of a point expressed in the coordinate system and
Rp is the radial distance of point from the center of turn 'I' of the y axis in fig.1
R_t is expressed in terms of the radius of turn

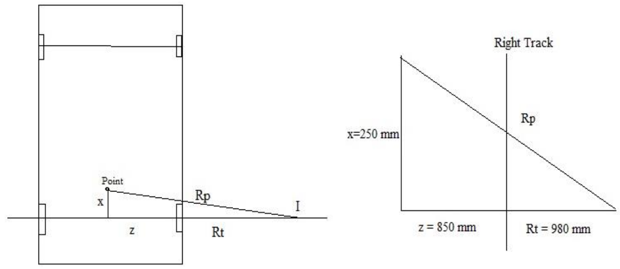


Fig. 6: Radius of Turn

i.e, $Rp = \sqrt{(R_t + z)^2 + x^2}$

$$Rp = \sqrt{(980+850)^2 + 250^2}$$

$$Rp = 1847 \text{ mm}$$

$$A_t\ p(x,y,z) = \omega_c^2 * Rp$$

$$= (5.553)^2 * 1847$$

$$A_t\ p(x,y,z) = 56.95 \text{ m/s}^2$$

H. Road Condition

1) Vehicle through Pit Condition:

The angular velocity for rotation about left track:

$$\omega_p\, p(x,y,z)_x = \frac{V_p\, p(x,y,z)_{xy}}{Z}$$

$$\omega_p\, p(x,y,z)_x = \frac{5}{0.85}$$

$$\omega_p\, p(x,y,z)_x = 5.88 \text{ rad/s}$$

The angular velocity for rotation about rear axle of the sumo chassis:

$$\omega_p\, p(x,y,z)_z = \frac{V_p\, p(x,y,z)_{xy}}{X}$$

$$\omega_p\, p(x,y,z)_z = \frac{5}{0.25}$$

$$\omega_p\, p(x,y,z)_z = 20 \text{ rad/s}$$

The resultant of the above two rotations for rotation about diagonally opposite corner:

$$\omega_p\, p(x,y,z)_{xz} = \frac{V_p\, p(x,y,z)_{xy}}{XZ}$$

$$\omega_p\, p(x,y,z)_{xz} = \frac{5}{0.25*0.85}$$

$$\omega_p\, p(x,y,z)_{xz} = 23.52 \text{ rad/s}$$

The acceleration due to these angular velocities about left track :

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_x * z$$

$$A_p\, p(x,y,z)_{xy} = 5.88^2 * 0.85$$

$$A_p\, p(x,y,z)_{xy} = 29.38 \text{ m/s}^2$$

The acceleration due to these angular velocities about rear axle :

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_z * x$$

$$A_p\, p(x,y,z)_{xy} = 20^2 * 0.25$$

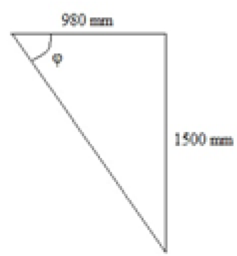
$$A_p\, p(x,y,z)_{xy} = 100 \text{ m/s}^2$$

The result of these two acceleration:

$$A_p\, p(x,y,z)_{xy} = \omega_p^2\, p(x,y,z)_{xz}.xz$$

$$A_p\, p(x,y,z)_{xy} = 23.52^2 * 0.25*0.85$$

$$A_p\, p(x,y,z)_{xy} = 117.55 \text{ m/s}^2$$



$$\text{Sin } \varphi = \frac{opposite}{hypotenous} = \frac{1500}{1792}$$

$$\Phi = \sin^{-1} \frac{1500}{1792} \quad ,$$

$\Phi = 56.83^0$

I. Considering the Different Speed of the Vehicle:

1) For 30km/hr (8.33m/s);

The velocity in downward direction is calculated using the following relation :

$$V_p\, p(x,y,z)_{xy} = \frac{V_v}{2\cos^2(\phi/2)}$$

$$V_p\, p(x,y,z)_{xy} = \frac{8.33}{2\cos^2(\frac{56.83}{2})}$$

$$V_p\, p(x,y,z)_{xy} = 5.38 \text{ m/s}$$

2) For 40km/hr (11.11m/s);

$$V_p\, p(x,y,z)_{xy} = 7.18 \text{ m/s}$$

3) *For 50km/hr (13.88m/s);*

$$V_p\ p(x,y,z)_{xy} = 8.97 \text{ m/s}$$

4) *For 80 km/hr (22.22 m/s);*

$$V_p\ p(x,y,z)_{xy} = 14.36 \text{ m/s}$$

5) *For 100 km/hr (27.77 m/s);*

$$V_p\ p(x,y,z)_{xy} = 17.94 \text{ m/s}$$

6) *For 120 km/hr (33.33 m/s);*

$$V_p\ p(x,y,z)_{xy} = 21.54 \text{ m/s}$$

J. Vehicle Over Bump Condition:

Considering the different speed of the vehicle the velocity in y direction is calculated using the following relation :

1) *For 30 km/hr (8.33m/s);*

$$V_{BP}\ P(x.y.z) = \frac{0.5wV_v}{\sqrt{(0.5w)^2 + h^2}}$$

$$V_{BP}\ P(x.y.z) = \frac{0.5*1.7*8.33}{\sqrt{(0.5*1.7)^2 + 1.925^2}}$$

$$V_{BP}\ P(x.y.z) = 3.36 \text{ m/s}$$

2) *For 40 km/hr (11.11m/s);*

$$V_{BP}\ P(x.y.z) = 4.48 \text{ m/s}$$

3) *For 50 km/hr (13.88 m/s) ;*

$$V_{BP}\ P(x.y.z) = 5.60 \text{ m/s}$$

4) *For 80 km/hr (22.22 m/s) ;*

$$V_{BP}\ P(x.y.z) = 8.97 \text{ m/s}$$

5) *For 100 km/hr (27.77 m/s) ;*

$$V_{BP}\ P(x.y.z) = 11.21 \text{ m/s}$$

6) *For 120 km/hr (33.33 m/s) ;*

$$V_{BP}\ P(x.y.z) = 13.46 \text{ m/s}$$

K. Dynamic Forces:

1) *For 30 km/hr (8.33 m/s);*

$$F = \sqrt{\frac{2EmaSA}{l}}$$

Where E = Elasticity = $2X10^5$ N/mm^2
m = Mass of link = 50 Kg
a = Acceleration = Velocity / Time;
t = 4 sec(assume)
S = Clearance = 190 mm
A = Area (From Table No. 5.1(a))
l = Length (From Table No. 5.1(a))

$$F = \sqrt{\frac{2*2E5*490.5*7.5*0.19*4.682}{2.580}}$$

$$F = 22524.89 \text{ N}$$

2) *For 40 km/hr (11.11 m/s);*

$$F = 26009.5 \text{ N}$$

3) *For 50 km/hr (13.88 m/s);*

$$F = 29079.51 \text{ N}$$

4) *For 80 km/hr (22.22 m/s);*

$$F = 36783.03 \text{ N}$$

5) *For 100 km/hr (27.77 m/s);*

$$F = 41124.64 \text{ N}$$

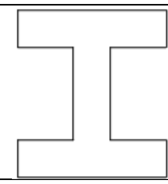
6) *For 120 km/hr (33.33 m/s);*

$$F = 45049.79 \text{ N}$$

The table shows the details of each component of sumo chassis.

Table – 1
Details of Each Component Of Sumo Chassis

Name of Component	*Front & rear track*

Cross Section (I – Section) *Depth 300mm, flanges 127mm X 10mm & web 10mm thick*	
Length	*2580mm*
Area	*4682mm^2*
Volume	*12.08 X 10^6 mm^3*
Weight	*95kg*
No. of Component	*02*

IV. Results and discussion

The dynamic forces obtained from the calculations are tabulated below

Table – 1
Dynamic Forces Obtained From the Calculations

SPEED (kmph)	***DYNAMIC FORCES (N)***
30	*22524.89*
40	*26009.50*
50	*29079.51*
80	*36783.03*
100	*41124.64*
120	*45049.79*

V. Conclusion

The dynamic force generated is less for 30 kmph as compared to other values. As the speed of the vehicle increases, the dynamic force generated also increases.

References

[1] F.Oijer, "FE-based Vehicle Analysis of Heavy Trucks, Part II: Prediction of Force Histories for Fatigue Life Analysis", Volvo Truck Corporation, S-405 08 Goteborg, Sweden.
[2] M.F.J. Luijten, "Lateral Dynamic Behaviour of Articulated Commercial Vehicles", Eindhoven University of Technology, August 2010.
[3] N.M.Shah, "Design and Analysis of Earth Mover – Wheel Loader Vehicle",California State University, fall 2010
[4] Matthew C. Snare,"Dynamics model for predicting maximum and typical acceleration rates of passenger vehicles", Virginia Polytechnic Institute and State University, August 26, 2002.
[5] R. Chakravaty, M.Tech. Project, Dynamic Forces Due to Heavy Linkages Mounted on Vehicle in Motion, VNIT Nagpur.
[6] www.technetalliance.com/uploads/tx_caeworld/DesCommVehChassBodyPart1990
[7] www.dot.state.mn.us/design/rdm/english/3e.pdf Road design manual Dec-2004.
[8] R.S. Khurmi & J.K. Gupta "Theory of Machine",S chand & company LTD, New Delhi,1976, p.p.226-229.
[9] Teo Han Fui, Roslan Abd. Rehman, Static & Dynamic Structural Analysis of a 4.5 Ton Truck Chassis Dec-2007, No.24,56-57.
[10] S.S.Sane, Ghanashyam Jadhav And Anandaraj.H, Stress Analysis of Light Commercial Vehicle Chassis By Fem.
[11] Paul Dvorak, Paul Kurowski, a closer look at motion analysis Jul-7-2005, London Ontario, Canada.
[12] Johansson and N.Gustavsson, FE-Based vehicle analysis of heavy trucks part- I: methods and models, department of vehicle dynamics and CAE, Volvo truck corporation, S-40508 Goteborg, Sweden.