



STRESS ANALYSIS OF STANDARD TRUCK CHASSIS DURING RAMPING ON BLOCK USING FINITE ELEMENT METHOD

Haval Kamal Asker¹, Thaker Salih Dawood¹ and Arkan Fawzi Said²

¹Mechanical Engineering, University of Duhok, Faculty of Agriculture and Forestry, Iraq

²Mechanical Engineering, University of Duhok, Faculty of Engineering and Applied Sciences, Iraq

E-Mail: Havaldosky78@yahoo.com

ABSTRACT

The frame of the standard dump truck supports all types of complicated loads coming from the road and freight being loaded. So, the intensity and the strength of the frame play a big role in the truck's design. A frame of 6 wheels, standard dump truck has been studied and analyzed using ANSYS package software. The static intensity of the frame has been analyzed when exposed to pure bending and torsion stress, within two cases. First case is when the rear wheels zigzag gets over block (only one side of the chassis steps the block), and the second case is when both wheels gets over the block. Finite element model of a stress analysis of the vehicle chassis has been built using three dimension hyper elastic elements for the modeling. The results show important differences between the two case studies, especially in the torsion and deformations results obtained from the chassis model. Also, vibration modes have been analyzed during the loading conditions. The more damping ratio used, the more stabilizing of the stresses with respect to time.

Keywords: truck frame design, stress analysis, truck chassis, ramping, torsion, numerical.

INTRODUCTION

Now days, the increased demands on trucks have been increased not only on cost and weight, but also on improved complete vehicle features. This result in increasing focus on optimization and modularization which together with the large number of vehicle variants makes it necessary to use efficient analysis methods.

Finite Element-based vehicle analysis has become an important part of the development process for many of vehicle features. A standard dump truck is a truck chassis with a dump body mounted to the frame. The bed is raised by a hydraulic ram mounted under the front of the dumper body between the frames, and the back of the bed is hinged at the back to the truck. The tailgate can be configured to swing on hinges or it can be configured in the "High Lift Tailgate" format wherein pneumatic rams lift the gate open and up above the dump body.

Kim H. *et al.* proposed the hybrid superposition method that combined finite element static and Eigen value analysis with flexible multi body dynamic analysis [1]. Johansson *et al.* presented a method for complete vehicle analysis based on FE-technique used for analysis of complete vehicle features such as vehicle dynamics and durability [2]. C. Karaoglu *et al.* introduced an improved procedure which is based on the modal stresses of FE-MBS hybrid structures [3].

In present work a finite element model has been build up to a 6 wheel standard dump truck chassis in order to simulate the effect of stepping a block in case of (a) when stepping the block zigzag i.e., with one rear wheel side (The right side). (b) When stepping the block with both rear wheels. A dynamic stress analysis with the necessary boundary and loading condition have been applied on the model using ANSYS software and many evaluating points have been distributed along the chassis to evaluate the induced stresses and deformations on the chassis during the two case studies.

The results show important differences between the two case studies especially in the torsion and deformations results obtained from the chassis model. Also, vibration modes have been analyzed during the loading conditions. The effect of changing the stepping conditions on the resulted stresses and deformations on the chassis of the truck to find the best stepping condition have been investigate precisely.

Finite element analysis of chassis

For the FE Analysis, it is necessary to create a solid model of chassis in order to create a FE model. In present work, generally, truck is any of various heavy motor vehicle designed for carrying or pulling loads. Other definition of the truck is an automotive vehicle suitable for hauling. Some other definition are vary depend on the type of truck, such as Dump Truck is a truck whose contents can be emptied without handling; the front end of the platform can be pneumatically raised so that the load is discharged by gravity.

Model of truck chassis

The truck chassis model used is the Scania model. The model is depicted in Figure-1. The model has length of 6.350 m and width of 2.85 m. The material of chassis is Steel with 552 MPa of yield strength and 620 MPa of tensile strength [4]. The other properties of chassis material are listed in Table-1.

Table-1. Properties of truck chassis material.

Modulus elasticity E (Pa)	207 * 10 ⁹
Density ρ (kg/m ³)	7800
Poisson ratio	0.3
Yield strength (MPa)	550
Tensile strength (MPa)	620



In elasto-static problem, each element forms a stiffness matrix, [K], relating forces [F] and displacements [u] at nodes. The size of the stiffness matrix is equal to the number of nodes per element multiplies by the number of freedom per nodes, as in the following

$$[F] = [K] [U]$$

In eigenvalue problem, the characteristic matrix is formed as

$$\{[K] - \omega^2 [M]\} [U] = 0$$

Where M is the mass matrix, ω^2 are eigenvalues, and u is the eigenvectors. In structural dynamics, the values are the natural frequencies and the vectors are mode shapes

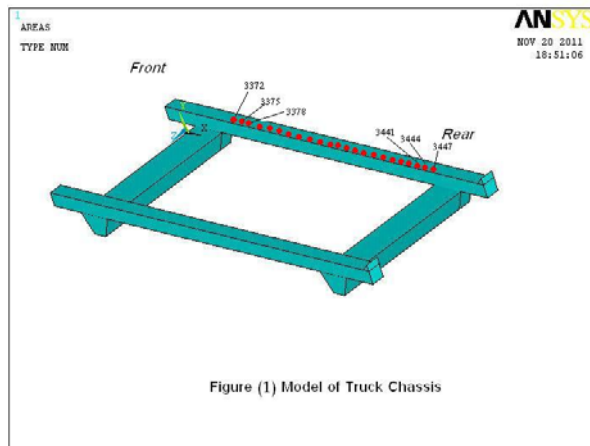


Figure (1) Model of Truck Chassis

Loading

The truck chassis model is loaded by static forces from the truck body and cargo. For this model, the maximum loaded weight of truck plus cargo is 20.000 kg. The load is assumed as a uniform pressure obtained from the maximum loaded weight divided by the total contact area between cargo and upper surface of chassis. Detail loading of model is shown in Figure-2. The magnitude of pressure on the upper side of chassis is determined by [5]:

$$P = \frac{F}{A} = \frac{10000 \text{ Kg} \cdot 9.81 \text{ m/s}^2}{1 \text{ m}^2} = 98100 \text{ N/m}^2 \quad (1) \quad [\text{for each chassis side}].$$

Where:

p = pressure (N/m²), F = force (kg. m/ s²), A = total contact area (m²),

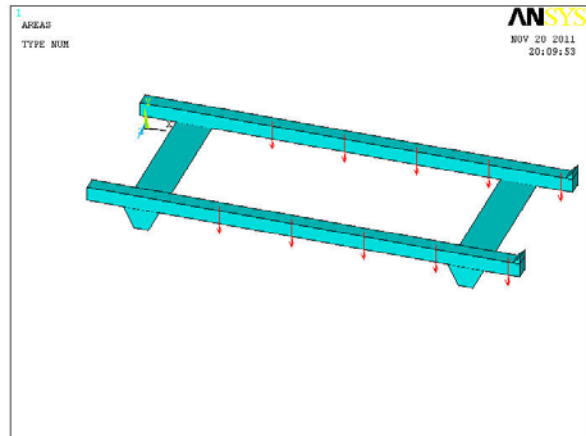


Figure-2. Static load (pressure = 98100 N/m²).

Boundary condition

The boundary conditions applied to this model of chassis can be classified into three general cases: the first boundary condition case applied in the front of the chassis by making the displacement and rotation not allowed in all axes [6].

The second case of boundary condition is applied when the rear wheels simultaneously get over the block, this will cause the suspension on this axle to be displaced, and the compression of the springs causes an upward force on the suspension mounting points. i.e., the rear chassis will be displaced by an amount in the Y direction as a boundary condition. In the second case bending should be observed on the chassis Figure-3.

The third case of boundary condition is applied when only one of the rear wheels ramp the block while the other rear wheel remain on the ground, in this case the chassis will be displaced only from the side of ramped wheel. While the other side of the rear chassis will be fixed (displacement and rotation not allowed in all axes) Figure-4. The third case of boundary condition is very important because it will include the bending and torsion cases which will occur in the chassis during this condition.

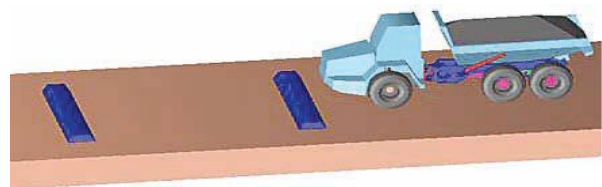


Figure-3. Condition when two wheels simultaneously get over blocks.

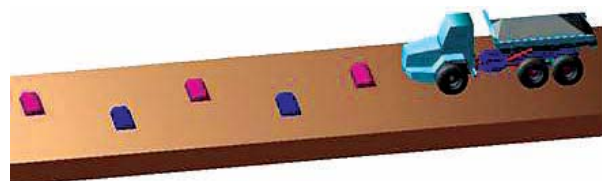


Figure-4. Condition when wheel zigzag gets over block.



The meshed truck chassis model has 6871 elements and 7116 nodes. The element type is Hyper elastic 3Dimension, 8 node Figure-5.

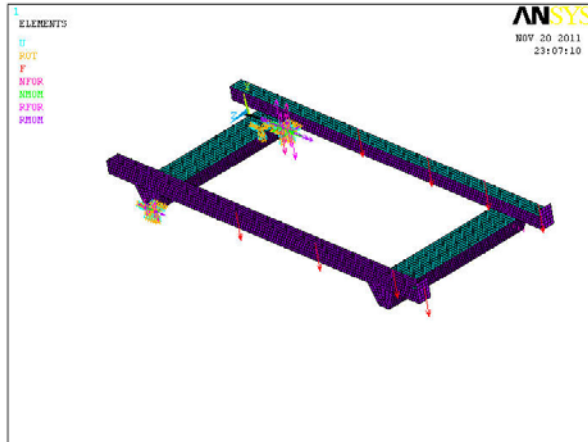


Figure-5. Discretized model of chassis.

Dynamic analysis

The most direct way of theoretically assessing the integrity of structures for dynamic loading, is by dynamic finite element analysis. The objective is to solve for the stresses as time functions, when the model is subjected to time series of loads.

Additional to the difficulty in defining such loads, there are several restrictions with regard to the use of dynamic finite element analyses.

A further differentiation may be made in terms of the solving method. One group of methods uses the direct integration method (called 'dynamic transient analysis' by Bishop), solving for the displacements after each small time increment by direct integration. A complete analysis is therefore performed at each time step, except for the compilation of the mass, damping and stiffness matrices.

RESULTS AND DISCUSSIONS

It is worth mentioning that the indicated numbers (1, 2, 3, 4...16) from Figures 6 to 13 are denote to the nodes (3372, 3375, 3378...3447) in Figure-1.

Figure-6 shows a comparison between the torsion stresses among the two case studies (one rear wheel get on a ramp and both rear wheels gets over the ramp). This Figure clearly shows that the torsion stress in the case of one rear wheel get over the ramp is greater than the torsion stress of the case when both rear wheels gets over the ramp.

Also, it can be seen that the maximum torsion stress of the two case studies occur near node (12) [max. torsion stress for one wheel get over the ramp = 485 Mpa and max. torsion stress for both wheel get over the ramp = 51 Mpa].

The comparison between the bending stresses is shown in Figure-7. It has been found that the maximum bending stress in the case of both rear wheels gets over the

ramp is occur near node (1) [max. bending stress for both wheels get over the ramp = 1760 Mpa], while in the case of one rear wheel get over the ramp the maximum bending stress occur near node (16) [max. bending stress for one wheel get over the ramp = 895 Mpa].

Figure-8 shows the displacements in the X - direction for the two case studies. A similar behavior can be obtained for the U_x displacement in the both two cases [one rear wheel and both rear wheels gets over the ramp] and the maximum displacement occur near node (9) [U_x max. = -9.5 mm], here the minus sign indicates to the direction of the displacement.

The displacements in Y - direction for both two case studies are presented in Figure-9. Also the same behavior is obtained for both two cases and the maximum U_y occur near node (16), [max. U_y for one rear wheel get over the ramp = 176.4 mm and U_y for both rear wheels get over the ramp = 192.3 mm].

In Figure-10, it can be concluded that there is a difference in the displacements in the Z - direction between the two case studies. It seems that the U_z displacements are approximately stable in the case of both rear wheels gets over the ramp. While the U_z displacements show a different behavior and the maximum U_z displacement occur near node (16) [U_z max. = 38.5 mm].

The rotational displacement about X - axis has been cleared in Figure-11 for the both two cases of study, and it indicate to higher rotational displacement in X - direction for the case of one rear wheel get over the ramp than the other case of study. While the behavior of the rotational displacement is approximately is the same for the both two cases of study. The maximum R_x occur between node (14) and node (15) [R_x max. for both rear wheels over the ramp = 0.12 and R_x max. for one rear wheel on the ramp = 0.827].

Figure-12 shows the difference in the rotational displacements about the Y - axis for the two case studies. Higher rotational displacement is obtained when of both rear wheels gets over the ramp. The maximum R_y is occur near node (1) for the case of both rear wheels over the ramp, while the maximum R_y is occur near node (9) for the case of one rear wheel get over the ramp [R_y max. for both wheels on the ramp = 0.02 and R_y max. for one wheel on the ramp = -0.1149].

The results of the rotational displacement about the Z - axis were expressed in Figure-13. The behavior of the rotational displacement in Z - direction of the both two case studies is similar. The maximum R_z is occur between node (8) and node (9) [R_z max. for both wheels on the ramp = 0.59 and R_z max. for one wheel on the ramp = 0.554].

The results of the dynamic simulation are expressed from Figure-14 to Figure-21. Figure-14 and Figure-15 shows the dynamic behavior for the torsion stress in the case of both rear wheels gets on the ramp using damping ratio equal to 0.05 and 0.08 respectively. While Figure-16 and Figure-17 shows the torsion stress for the case of one wheel get over the ramp using two



damping ratios (0.05 and 0.08). Also Figure-18 and Figure-19 indicates the bending stresses for the case of both rear wheels gets over the ramp for the two damping ratios (0.05 and 0.08). Figure-20 and Figure-21 represents the bending stress for the case of one rear wheel get over the ramp using damping ratio = 0.05 and 0.08 respectively. These Figures indicate clearly the more damping ratio used, the more stabilizing the stresses with respect to time. The time of the dynamic simulation was 5 seconds and the substep was equal to 10 sub steps, each substep = 0.5 second.

Figures 22 to 31 show the model under different deformations and stresses mode.

In Figure-32 the behavior of the torsion stress in reference can be compared with the behavior of the torsion stress of the present work Figure-6. The behavior shows a good compatibility with the behavior of reference [7].

CONCLUSIONS

The model analysis for vehicle that considers the elastic characteristic of frame was applied to the rear frame of articulated dump truck, and the result express the behavior of the dump truck chassis during ramping block in the two case studies (both wheels ramp the block and zigzag wheels ramp the block). As a result, it was confirmed that this analysis can be used to predict the bending and torsion stresses of frames when a vehicle ramp a block.

Numerical simulation result shows that the critical point of stress occurred when the truck zigzag ramp the block. The big effect was given to the case of zigzag wheels of the dump truck ramp the block because there was great difference in the torsion stress values in both two case studies.

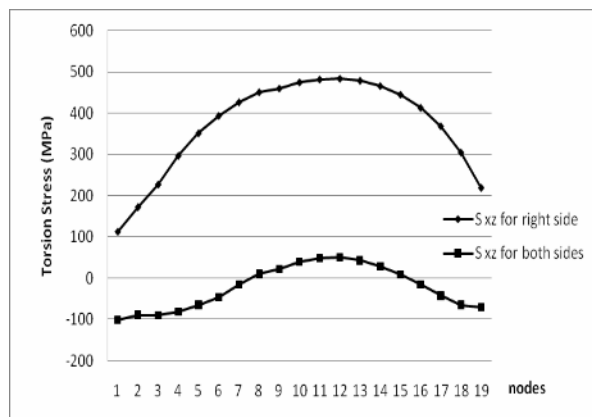


Figure-6. Torsion stresses for the two cases (right rear side and both rear sides on the block).

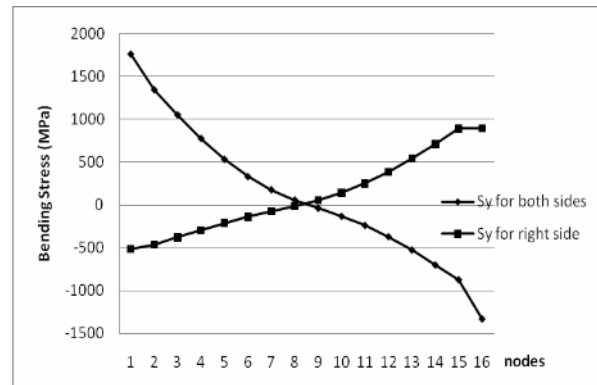


Figure-7. Bending stresses for the two cases (right rear side and both rear sides on the block).

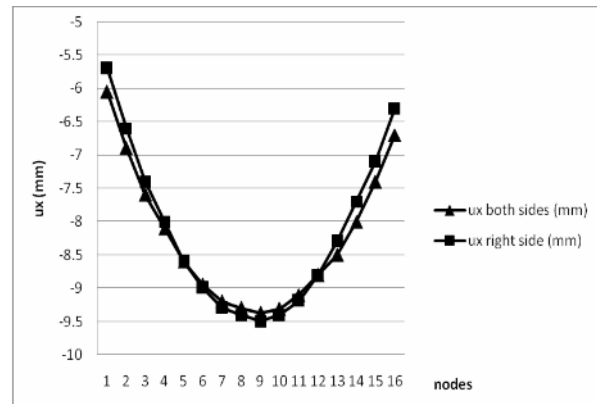


Figure-8. Displacement in X direction for the two cases (both rear wheels and one rear wheel getting on the block).

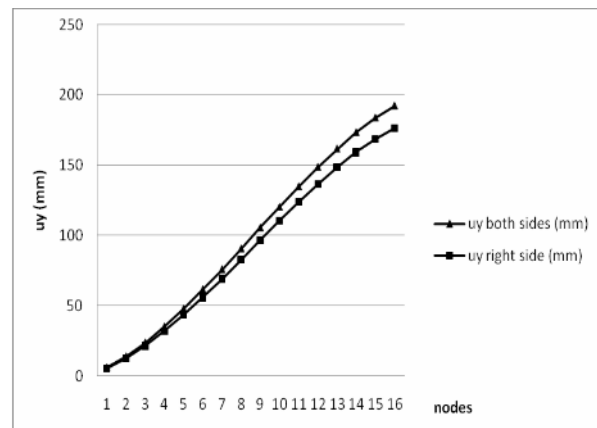


Figure-9. Displacement in Y direction for the two cases (both rear wheels and one rear wheel getting on the block).

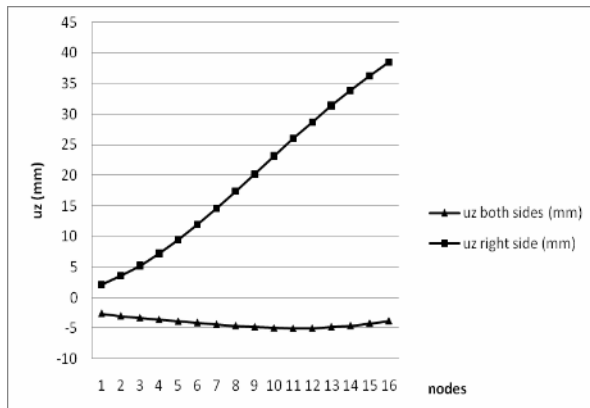


Figure-10. Displacement in Z direction for the two cases (both rear wheels and one rear wheel getting on the block).

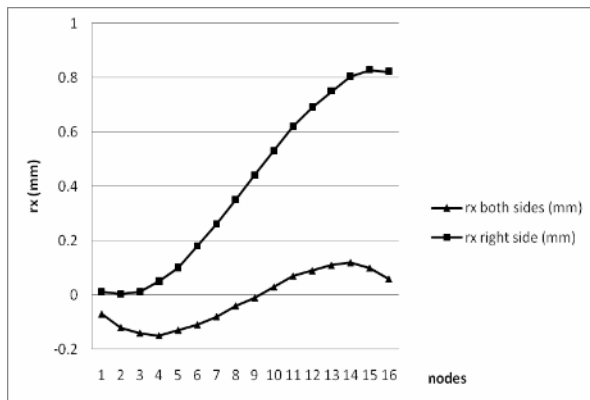


Figure-11. Rotation in X direction for the two cases (both rear wheels and one rear wheel getting on the block).

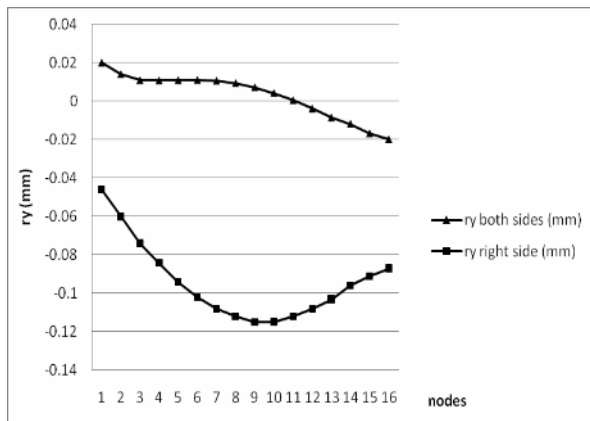


Figure-12. Rotation in Y direction for the two cases (both rear wheels and one rear wheel getting on the block).

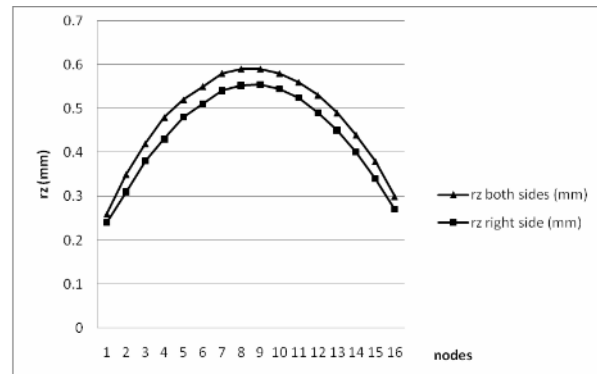


Figure-13. Rotation in Z direction for the two cases (both rear wheels and one rear wheel getting on the block).

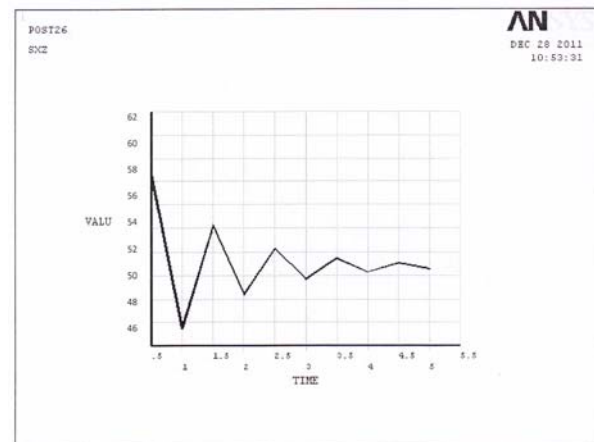


Figure-14. Dynamic torsion stress for both wheels when damping coefficient = 0.05.

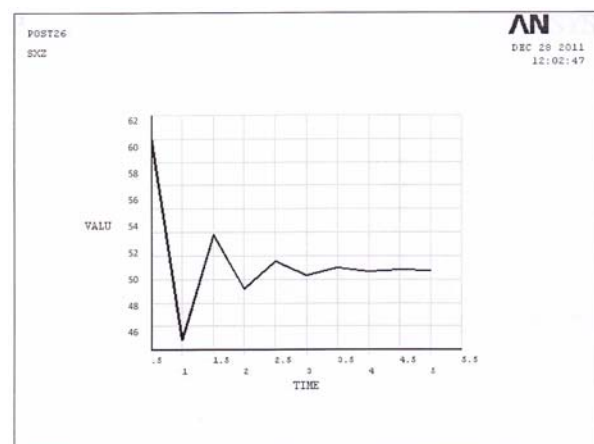


Figure-15. Dynamic torsion stress for both wheels when damping coefficient = 0.08.

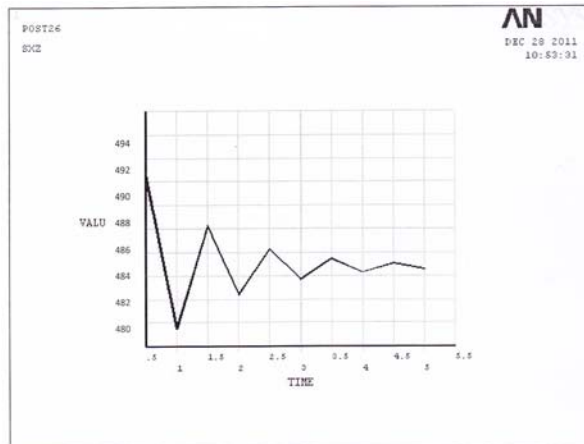


Figure-16. Dynamic torsion stress for one wheel when damping coefficient = 0.05.

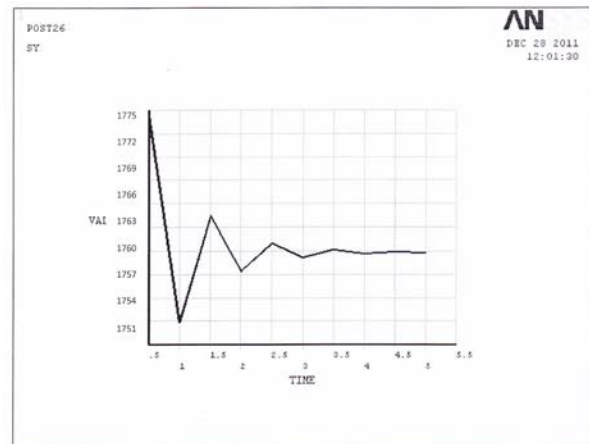


Figure-19. Dynamic bending stress for both wheels when damping coefficient = 0.08.

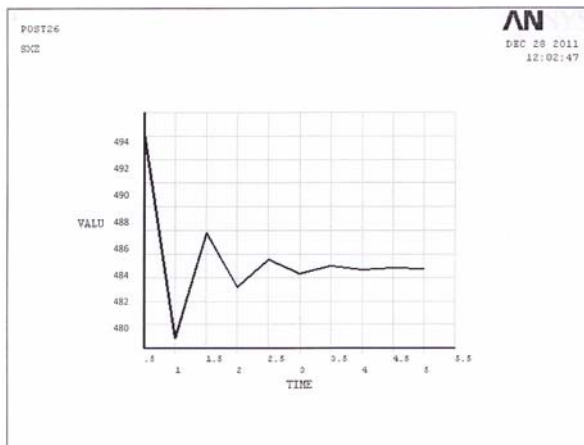


Figure-17. Dynamic torsion stress for one wheel when damping coefficient = 0.08.

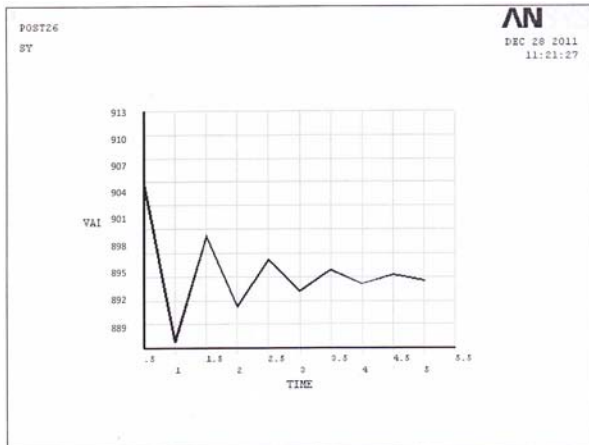


Figure-20. Dynamic bending stress for one wheel when damping coefficient = 0.05.

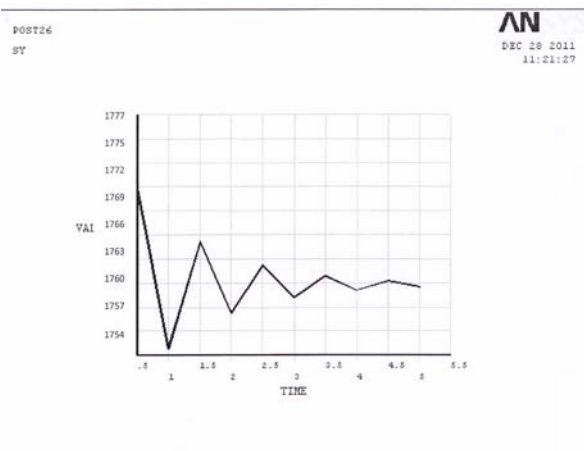


Figure-18. Dynamic bending stress for both wheels when damping coefficient = 0.05.

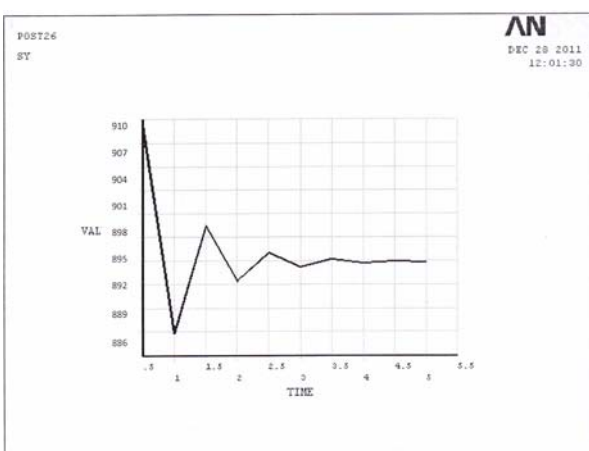


Figure-21. Dynamic bending stress for one wheel when damping coefficient = 0.08.

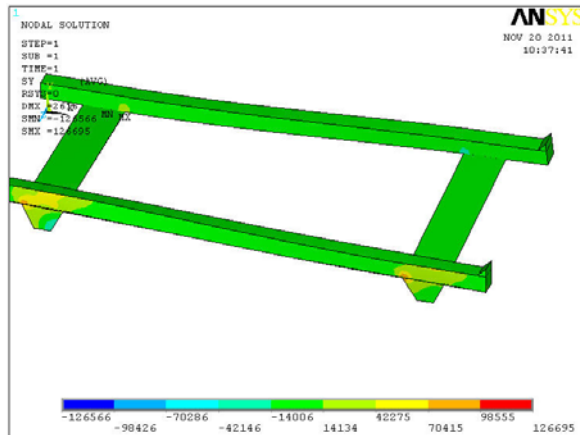


Figure-22. Bending in case of one rear side on the block.

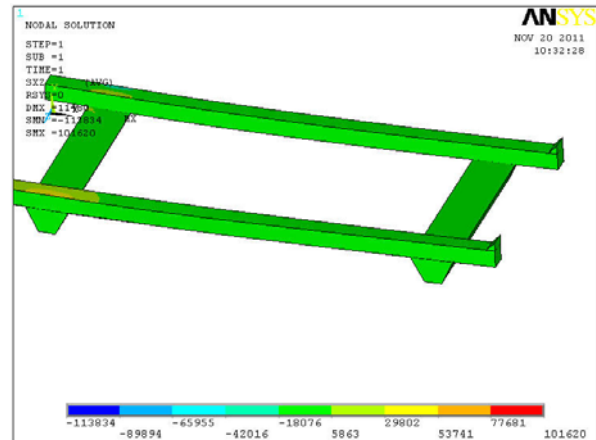


Figure-25. Torsion in case of both rear sides on the block.

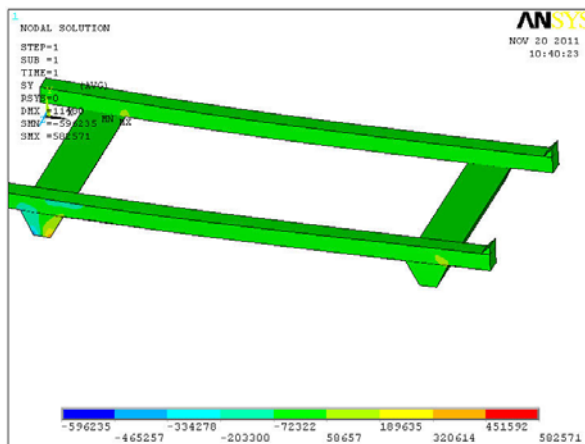


Figure-23. Bending in case of both rear sides on the block.

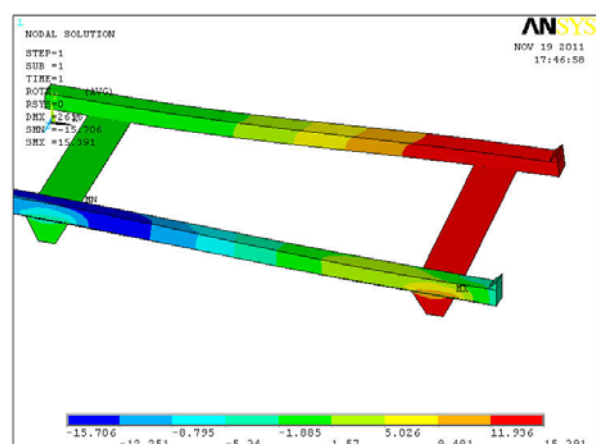


Figure-26. Deformation in x direction in case on one rear side on the block.

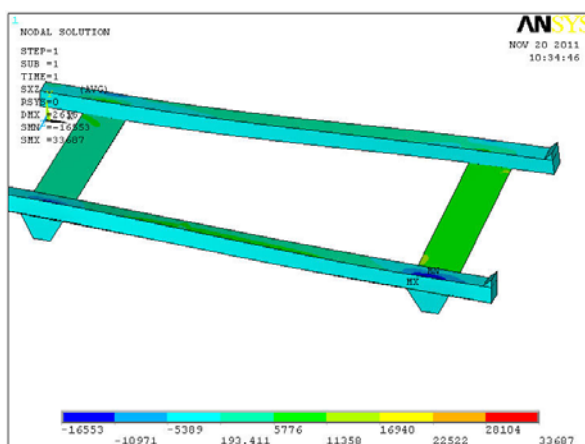


Figure-24. Torsion in case of one rear side on the block.

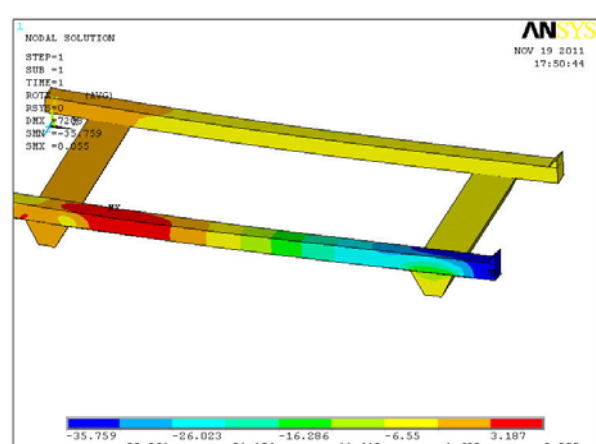


Figure-27. Deformation in x direction in case on both rear sides on the block.

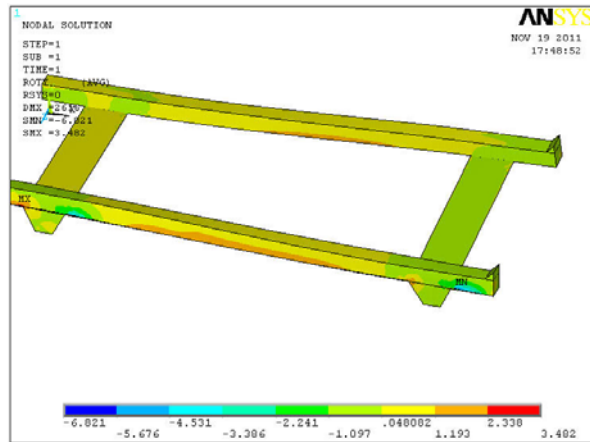


Figure-28. Deformation in y direction in case on one rear side on the block.

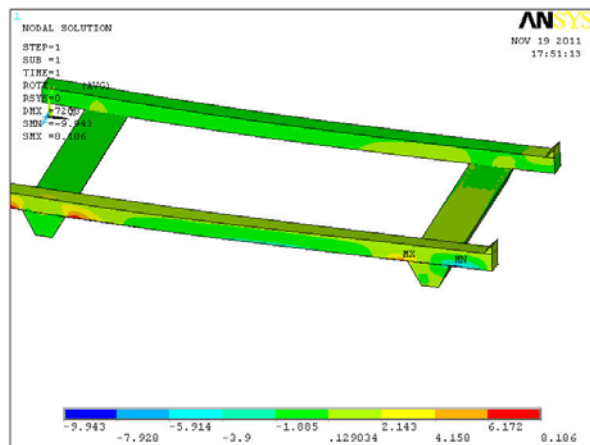


Figure-29. Deformation in y direction in case on both rear sides on the block.

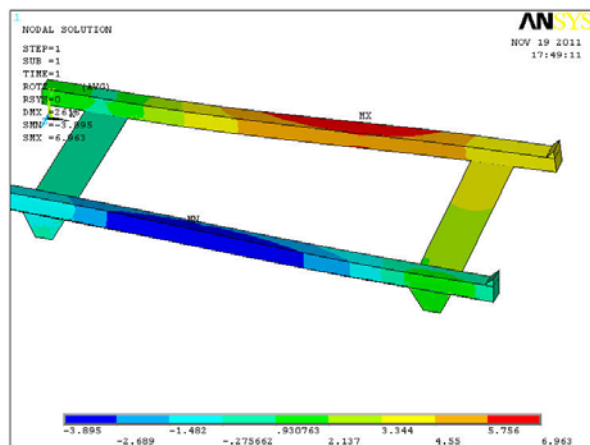


Figure-30. Deformation in z direction in case on one rear side on the block.

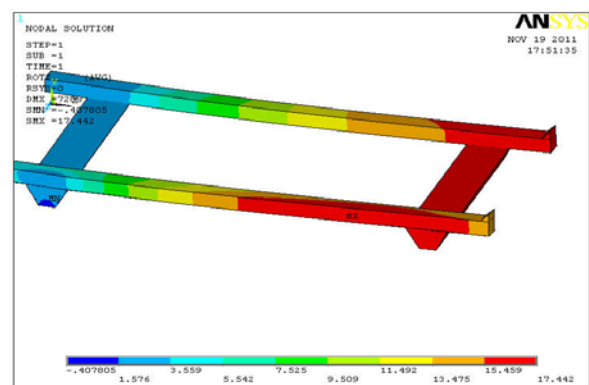
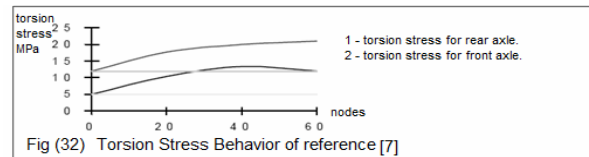


Figure-31. Deformation in z direction in case on both rear sides on the block.



REFERENCES

- [1] H. S. Kim *et al.* 2010. Dynamic Stress Analysis of a Bus Systems, Commercial Vehicle Engineering and Research Center, Hyundai Motor Company.
- [2] Johansson and S Eslund. 1993. Optimization of Vehicle Dynamics in Truck by use of Full Vehicle FE Models. I. Mech. E. - C466/016/93. pp. 181-193.
- [3] C. Karaoglu and N. S. Kuralay. 2000. Stress Analysis of a Truck Chassis with Riveted Joints. Elsevier Science Publishers B. V. Amsterdam, the Netherlands. 38: 1115-1130.
- [4] R. C. Juvinall and K. M. Marshek. 2006. Fundamental Machine Component Design. John Wiley and Son, Inc., USA.
- [5] N.K. Ingole, D.V. Bhope. 2011. Stress Analysis of Tractor Trailer Chassis for Self Weight Reduction. International Journal of Engineering Science and Technology (IJEST).
- [6] Sujatha C and V Ramamurti. 1990. Bus Vibration Study-Experimental Response to Road Undulation. Int. J. Vehicle Design. 11(4/5): 390-400.
- [7] A. Dubey, V. Dwivedi. 2003. Vehicle Chassis Analysis: Load Cases and Boundary Conditions for Stress Analysis. 11th National Conference on Machines and Mechanisms held at the Indian Institute of Technology Delhi, New Delhi on December 18-19, 2003. (NaCoMM 2003), <http://www.nacomm03.ammindia.org/Articles/Nav009.pdf>.