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The Effect of Connection-Plate Thickness on Stress of Truck Chassis with Riveted and Welded Joints under Dynamic Loads

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Abstract: In this study the mechanical behaviour of a semi-heavy truck chassis has been studied using finite element based Ansys Code and the stress distributions in the chassis have been obtained and examined. For this purpose, the FE model of the truck vehicle has been created using 3D shell elements. To validate the FE model of the chassis; firstly, experimental modal analysis has been used. Secondly, the modes of chassis-vibration, natural frequencies and modal shapes have been obtained from the FE analysis and were compared with the results of experimental modal analysis. The dynamic forces due to the unevenness of the road have been calculated using a simple 3D dynamic model of the truck body. Then, stress analysis for the truck chassis have been carried out under static and dynamic loads. Different types of joints and their thickness in the chassis of truck vehicles are one of the important parameters which have significant effect on their strength. To study the effect of the connecting plates on the strength of the chassis, the strength of the welded and also the combined welded-riveted joints has been analyzed with three different plate thicknesses: 5, 8, 12 mm. The results show that the amount of stresses in chassis and connection plates are decreased significantly with increasing the thickness of connection plates. Also, it has been shown that the use of combined welded-riveted joints reduces the stress level of the chassis. The results prove the precision of the FE modeling and they show that the numerical modeling is accurate and therefore, the stresses which have been obtained are reliable and can be used to design the chassis.

Key words: Truck chassis, riveted joints, welded joints, road profile, modal test, finite element method, stress analysis

INTRODUCTION

Every vehicle consists of two parts, chassis and bodywork or superstructure. Vehicle body has to carry both the loads and its own weight. The Conventional chassis frame, which is made of pressed steel members, can be considered structurally as grillage. The chassis frame includes cross- members which locate at critical stress points along the side members. To provide a rigid, box-like structure, the cross- members secure the two main rails in a parallel position. These cross-members are usually attached to the side using welded members or connection plates. The joints are usually riveted or bolted in trucks and welded in trailers. When rivets are used, the holes in the chassis frame are drilled approximately 1/16 (in) larger than the diameter of the rivet (Fitch, 1994). The rivets are then heated to an incandescent red and driven home by hydraulic or air pressure. The hot rivets conform to the shape of the hole and tighten upon cooling. An advantage of this connection is to increase the flexibility of the chassis. Therefore, high stresses are prevented in critical area. The side and cross- members are usually open- sectioned, because they are cheap and easily attached by rivets (Karaoglu and Kuralay, 2002).

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The study of the mechanical behavior of the chassis and their welded or rivet joints are becoming a major concern in recent years. For this purpose, finite element analysis methods have been used extensively. An experimental and numerical analysis of riveted joints was studied by Fung and Smart (1997). In this study to validate the FEM based model of the chassis; an experimental modal analysis has been carried out. The excitation method in experimental tests may be selected to be by shaker or by roving-hammer test, according to the properties of the structure and testing equipment (Ewins, 2000).

Some studies have used extended experimental tests along with mathematical post processing of results (Ewins, 2000; Carneiro *et al.*, 2005; Chester *et al.*, 1997; Ramsey, 1983).

MATERIALS AND METHODS

The chassis of the semi heavy trucks similar to Fig. 1 are usually ladder type and they together with the body and skin panels make the whole structure of the truck. This type of chassis is made by special U shape profiles which are joined by rivets, bolts or weldments. The thickness of these profiles are about 4 or 5 mm and they are made by low carbon steels or steels with average carbon similar to St-44-2 with low sensitivity to the stress construction, high yield stress and strength, good cold forming and high welding ability. The selected chassis is made of average carbon steel with 290 MPa yield stress and 530 MPa ultimate tensile stresses (Yoshitake *et al.*, 1994).

Loads on Chassis Frame

All vehicles are subjected to both static and dynamic loads. Dynamic loads result from driving on uneven roads, while, the static loads are due to the weight of stationary vehicle. Other forces result from braking, acceleration, cornering, torsion and the forces on front and rear axle which are balanced by inertia forces.

Loads acting on the frame cause bending or twisting of the side and the cross members. A simplified free diagram of the major loads which act on the vehicle is shown in Fig. 2. Symmetric loads acting in the vertical direction predominantly cause bending in the side members. Vertical loads additionally arise from lateral forces acting parallel to the frame plane during cornering (Beermann, 1989). Loads imposing on the plane of frame cause bending of the side members and of the cross-members (Beermann, 1989; Kobayashi *et al.*, 1971).

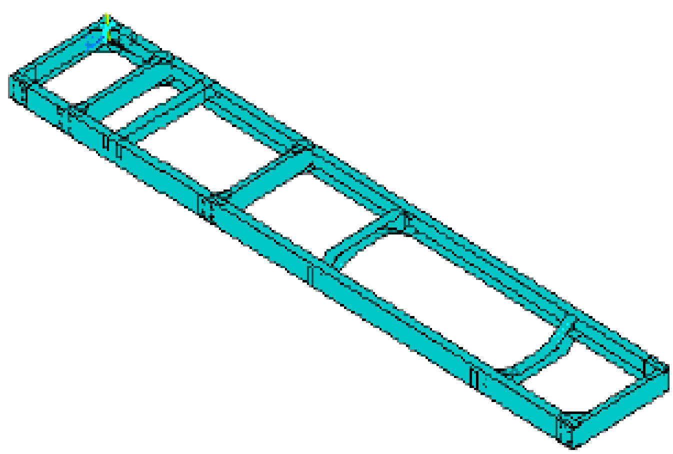

Fig. 1: Ladder type chassis of the truck

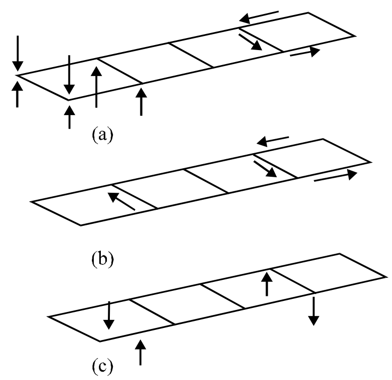

Fig. 2: Loads on the vehicle chassis frame

Modal Analysis

The experimental modal analysis to obtain the dynamic behavior of a system has been extensively discussed in the literature (Hermans and Auweraer, 1999; Yang and Chen, 2002; Coucke *et al*., 2003). The stages of the theoretical analysis of the dynamic behaviour of a system using the modal tests are: firstly a prototype of a system is produced and the actuating forces are imposed on this system; then the forces together with the response of system have been measured. By extracting the frequency functions of the responses and their related parameters, the results of the tests have been analyzed and the vibration modes and their characteristics are obtained. The modal analysis approach is different from the theoretical study of the dynamic behaviour of a system. In the latter, the governing equations are formed and by solving them, the modal model can be obtained and finally the response of system may be calculated. This major difference reveals the advantage and accuracy of the modal test analysis method. The basics of the theoretical methods to analyze the experimental data which are gathered from modal test analysis are explained here briefly.

A real structure is usually assumed to be a set of discontinuous and separate points with different degrees of freedom. Governing equations on the behavior of the system with N degrees of freedom with damping are (Meirovitch, 1967):

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F\} \tag{1}$$

In which M is the mass matrix, C is viscous damping matrix, K stiffness matrix and X is the system-response-vector in specific degrees of freedom. Also {F (t)} is the external applied force vector. The Laplace Transform of Eq. 1 yields:

$$[\underset{\sim}{M}s^2+\underset{\sim}{C}s+\underset{\sim}{K}]\,X(s) = F(s) + (\underset{\sim}{M}s+\underset{\sim}{C})\,x(0) + \underset{\sim}{M}\dot{x}(0) \tag{2}$$

The modal model is fully described by a matrix of frequency response functions $\underset{\sim}{H}_{pq}(\omega)$, where its elements may be obtained by calculating the ratio of the response signal at DOF p, $X_0(\omega)$ to the input signal at DOF q, $F_0(\omega)$ in the frequency domain) (Dimaragonas and Haddad, 1992):

$$H_{pq} = \frac{X_p}{F_q} \tag{3}$$

Assuming zero initial condition, Eq. 2 may be written (in the frequency domain) as:

$$X(\omega) = \underset{\sim}{H}(\omega).F(\omega) \tag{4}$$

Where:

$$\underset{\sim}{H}(\omega) = \left[\underset{\sim}{K} - \omega^2\underset{\sim}{M} + i\omega\underset{\sim}{C}\right]^{-1} \tag{5}$$

The relationship between the natural frequencies, mode shapes and the elements of $\underset{\sim}{H}(\omega)$ matrix can be obtained by Ginsberg (2001) and Maia and Silva (1997):

$$H_{ij}(\omega) = \frac{X_i}{F_j} = \sum_{r=1}^{N} \frac{\phi_{ir}\phi_{jr}}{\omega_r^2 - \omega^2 + i\eta_r\omega_r^2} \tag{6}$$

Frequency response equation is defined as the ratio of output response to excitation force:

$$\alpha(w) = \frac{X}{F} \tag{7}$$

Knowing that the mass and stiffness matrix are orthogonal, the definition of the system with MDOF with damping will be:

$$[\alpha(w)] = [\phi][(\lambda_r^2 - w^2)^+]^{-1}[\phi]^T \tag{8}$$

where, $[\phi]$ is the mass-normalized form of the mode-shapes and Eq. 7 and 8 are the connecting terms between test results and their theoretical application.

In the first step of a modal test analysis, the elements of at least one full line or one full column of the $[\alpha(\omega)]$ matrix should be measured. In the next stage, for calculating the modal parameters, the maximum points in every diagram, which refers to the natural frequency of the system, are selected and the natural frequency and damping ratio are calculated using the frequency response at these points. The ratio between amplitudes of different frequency response diagrams in a certain natural frequency shows the shape and mode of vibration at this frequency.

Truck and Chassis Modeling

In this study, the strength of the chassis is analyzed against the imposed forces and therefore, attached parts to the cassis such as doors, windows etc have not been considered. However, due to the effect of different types of loads, such as torques which unstable the vehicle in cornering, those parts which play a role in cornering have been taken into account. The parts that have been considered in this truck- model are: truck chassis, truck, cabin and shock absorber rubbers which are installed between truck chassis and truck, suspension system.

For Finite Element modeling of the truck, first two dimensional sketches of the chassis and its main parts have been produced and with using them, a 3D model has been developed in Ansys user manual. For this purpose, shell 63 elements have been used. Cabin and truck also have been included in this model. Mass and other mechanical properties of the model have been introduced based on the accurate measurements of the real parts. Since, there is damping absorbers between truck and chassis, these elements have been added in to the model using elastic solid elements and their stiffness have been taken into account. The final 3D FE model is shown in Fig. 3.

To validate the FEM model of the chassis, experimental modal analysis has been carried out (Fig. 4). A free- free condition was chosen for modal analysis. For this purpose, the chassis has been

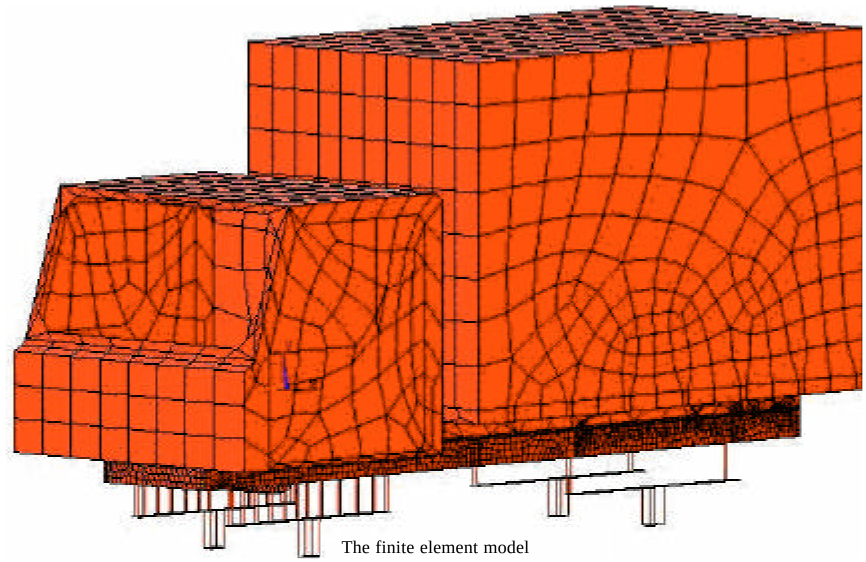

Fig. 3: FE model of the truck body to carry out stress analysis

Fig. 4: Experimental modal test

suspended along its symmetric axes from a frame by using a hook attached to chassis. Hammer test was used to provide the required exciting forces on the structure. In this test, a hammer (Type 8202, B and K Inc., Denmark) and four transducers with a sensitivity of 0.94 pc/N have been used. The mass of hammer is 21 g and the force has been applied to 7 different points. Direction of forces is along the Z-axis (Fig. 6) and measurements including the frequency response have been made by transducers and accelerometer type 4504, B and K Inc., Denmark, with mass of 14 g. In these tests, seven points have been used for measurements. Points with maximum amplitude give the natural frequency of the part. The results have been obtained and presented as frequency response (FRF) curves for different points of chassis which are shown in Fig. 5. The measured- frequency-data range from 0 to 100 Hz, twelve natural frequencies have been obtained for chassis which are shown in Table 1. Also

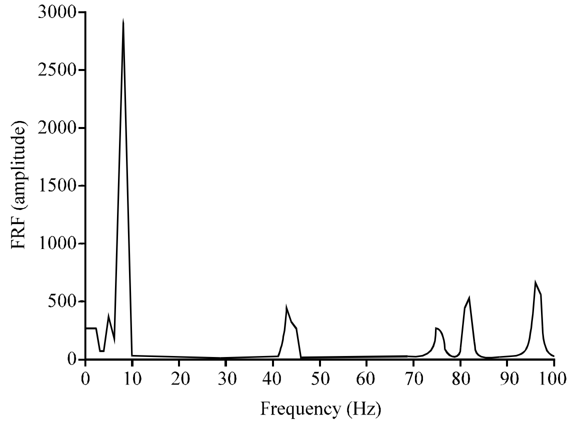

Fig. 5: Typical frequency response curves from the hammer test (FRF)

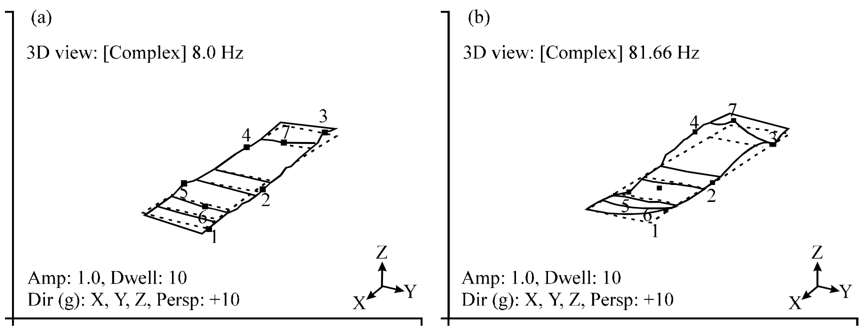

Fig. 6: Typical modal shapes obtained from tests, (a) bending (8 Hz) and (b) bending (81.66 Hz)

Table 1: Comparison of the natural frequencies and mode shapes from hammer tests and FEM solutions

Case	FEM (Hz)	Experimental (Hammer test) (Hz)	Mode shape	Error (%)
1	0	-	Rigid	-
2	0	-	Rigid	-
3	0	-	Rigid	-
4	0	-	Rigid	-
5	0	-	Rigid	-
6	0	-	Rigid	-
7	8.66	8.00	1st bending mode	5
8	29.60	26.71	2nd bending mode	10
9	37.90	34.54	3rd bending mode	10
10	52.09	43.21	Complex undefined mode shape	-
11	86.85	81.66	4th bending mode	6

for each test point, modal shape has been produced. Figure 6 shows two typical modal shapes. The results in Table 1 show that the first 6 modes are zero which are the rigid modes. The maximum error between computational results and average hammer test results is 10%. This error-level is acceptable and is due to the limitations of the test-rig.

The difference/error between frequencies which have been obtained using FE analysis and those from modal analysis are between 5 to 10%. The errors can be attributed to the simplifying assumptions, limitations in FEM modeling and other parameters related to the experiment itself and

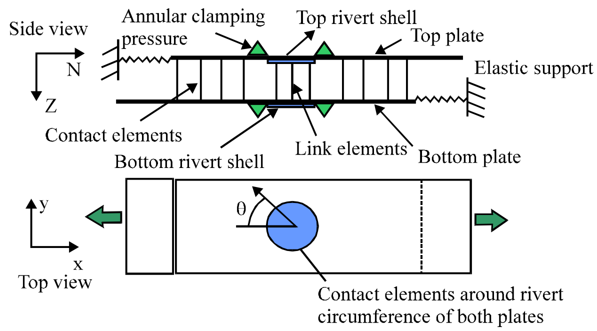

Fig. 7: FEM modeling of the riveted joints

the accuracy of the measurements. The simplified modeling of chassis, assumption of unitized body and neglecting the contact areas in joints can also be sources of this error. Common sources of error, including data acquisition, leakage, windowing, filtering and other computational errors exist and may affect the results (Ewins, 2000). Other experiment limitations such as variation of the applied transient force in hammer test also exist (Larbi and Lardies, 2000).

The results, in spite of 5 to 10% error level and also six zero fist modes prove that the numerical FEM based solution is accurate and reliable.

Modeling of Weld and Riveted Joints

After modeling the truck and chassis, the joints have been modeled. The results of tensile and fatigue tests show that in ladder chassis, the yielding or fatigue failure never start from the welded joints because they strengthen using supporting plates. On the other hand, due to the required accuracy, the joints have been assumed to be perfect-joint and Partition Method has been used for their modeling (Harish *et al*., 1998). For modeling the rivets and joint plate's shell 63, contact 52, truss and link10 elements have been used (Liao *et al*., 2001). The diameter of rivets is about 11 mm and the schematic diagram of rivet-modeling has been shown in Fig. 7.

Suspension Modeling

To carry out the FEM analysis of the model, it is necessary to be able to impose required displacement and forces. Therefore, it is necessary to model the suspension system of the truck. This has been done using spring, mass and dampers. The stiffness of the springs and their strength has been defined and modeled using Combine 14 element. Also the lower springs to simulate/model the flexibility and damping of the tires have been introduced. Also, plates with defined mass have been placed between springs to act as unsprung mass. The model which has been developed to carry out the dynamic analysis is shown in Fig. 8. This model is a combination of sprung (M_s), unsprung masses (m_i), tire damping (C_t), suspension springs (K_i) and suspension dampers (C_i) (Marco and Rogeria, 1995; Wong, 2001). Table 2 shows the values of sprung, unsprung, stiffness and damping of tires, suspension systems (Shakori, 2006).

Simulating the Boundary Condition

The boundary condition has been imposed on the supports, where springs attached to chassis or to panels. They are also given in the tire-road contact surfaces. The sprigs act in one direction, therefore the motion of truck in longitudinal and lateral directions has been restricted using the boundary conditions. Also the rotation of the truck around the vertical axis has been restricted.

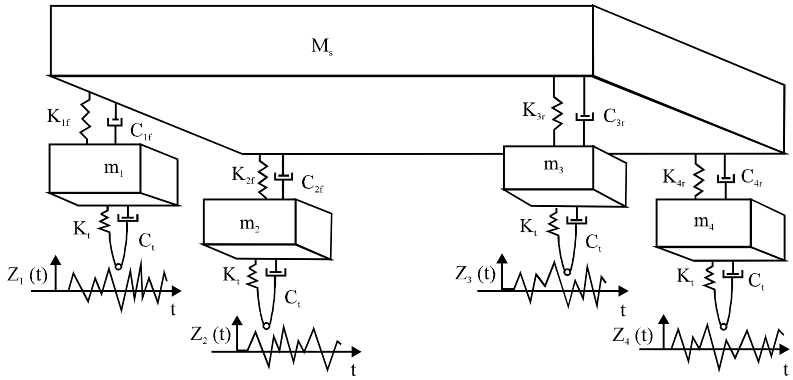

Fig. 8: Modeling of the suspension system for exerted dynamic loads

Table 2: Parameters of tires and suspension model

Case	Unit	Value
M_s : Sprung mass	(kN)	50
m_1, m_2, m_3, m_4: Unsprung masses	(kN)	5
K_t : Tire stiffness	(kN m^{-1})	177
C_t : Tire damping	(N sec m^{-1})	2860
K_{1f}, K_2f: Front suspension spring	(kN m^{-1})	85
K_{3r}, K_{4r}: Rear suspension spring	(kN m^{-1})	250
C_{1f}, C_{2f}: Front suspension damper	(Nsec m^{-1})	58
C_{3r}, C_{4r}: Rear suspension damper	(Nsec m^{-1})	68

Table 3: Classification of roughness types according to ISO; S_o is rated roughness parameters based on ISO. Ramji *et al.* (2004)

Classification and road roughness	Roughness parameters $S(K_0)/10^{-6}\frac{m^3}{cycle}$
Very good	2-4
Good	8-32
Average	32-128
Poor	128-512
Very poor	512-2048

The Simulation of the Truck Passing over Uneven Roads

Kawamura *et al.* (1998) have used numerical methods to produce the profile of the uneven roads and have studied the fatigue life of the vehicle parts by moving them along these road profiles. Kawamura *et al.* (1998) and Davis and Thompson (2001) have also defined the road roughness using Power Spectral Density (PSD). Using these results, they have calculated the loads due to the road profile and applied them on the vehicle body to obtain the stress distribution. Ramji *et al.* (2004) have used ISO standards to classify the road roughness based on PSD method. These classifications are presented in Table 3.

To simulate the movement of the truck on an uneven road, a road profile shown in Fig. 9 has been used.

The Fig. 9 shows that along the first 80 cm of the road, there is a 5 cm bump and along the next 80 cm the road is even and then along the last 80 cm the road comes to its first level.

Since the speed of the truck passing along the uneven road has an important role, it has been considered as a time function based on the IRI (International Reference Index) (Sayers, 1995) and is shown in Fig. 10. The IRI index is the ratio of the sum of the vertical distances covered by the one

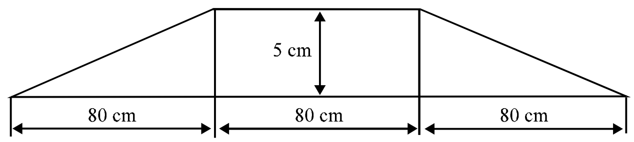

Fig. 9: Road profile

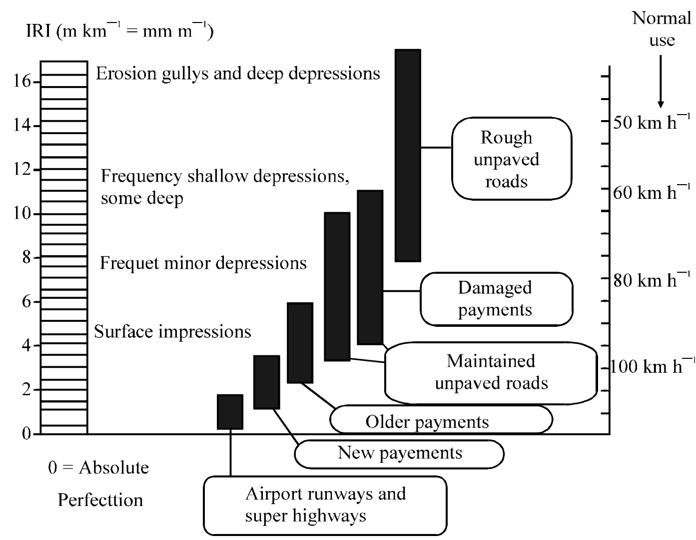

Fig. 10: Diagram of the international reference index

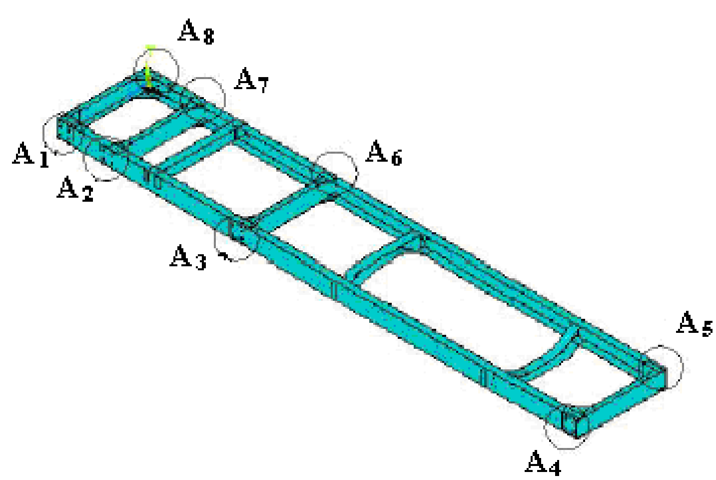

Fig. 11: Numbering of the rivet joints

fourth-model of the vehicle to the horizontal distance passed by it with a constant average speed of 80 km h^{-1} (Sayers and Karamihas, 1998).

Figure 11 is showing the numbering of the rivet joints. For the stress analysis of the chassis in passing over the uneven road three steps have been considered:

- Wheels in one side of the truck are passing over the hump where the other side-wheels are passing over a reverse dimple or pit at the same time
- In the second type of excitation, one front wheel and at the same time, an opposite rear wheel are passing over a hump
- In the third type of excitation, only one front wheel is passing over the hump

RESULTS AND DISCUSSION

In this study, the finite element analysis of semi truck vehicle has been carried out. To insure and validate the modeling of the chassis, experimental modal analysis has been used. The effect of three different thicknesses 5, 8 and 12 mm of joint plates has been studied over the strength of the chassis while passing over uneven roads. The results show that the one of the (first type) excitations is the most critical case which produces the maximum stresses. The critical region of the chassis which has been identified with * mark is shown in Fig. 12.

Also the strength of welded joints in chassis and the combined welded and riveted joints have been analyzed. For combined welded and riveted joints, the thickness of the reinforcing plate is 8 mm. Maximum stress level for this case is 460 MPa. Also, results for some other case studies are given in Fig. 13-16. The results given in Table 4 and 5 show that the maximum level of the stress for chassis with combine welded and riveted joints is less than the stress level for chassis with welded joints.

Table 4: Results for chassis with welded joints

	Maximum stress on total chassis (MPa)			Maximum stress on joints (MPa)		
	Joints thickness (mm)			Joints thickness (mm)		
Type of loads	5	8	12	5	8	12
Road roughness (type 1)	553	547	520	553	129	112.0
Road roughness (type 2)	159	147	101	159	107	33.7
Road roughness (type 3)	160	147	134	160	66	52.0

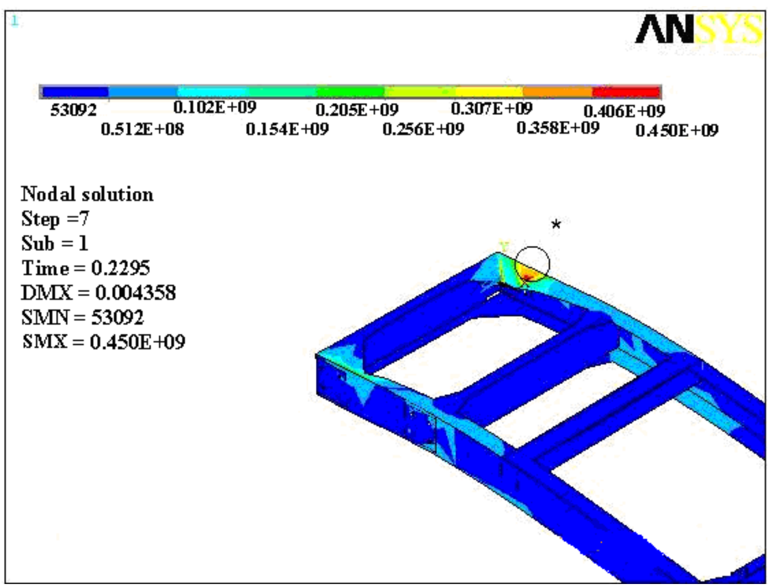

Fig. 12: Stress distribution for the type 1 road profile and with combined welded-riveted joints chassis (joint plate thickness is 8 mm)

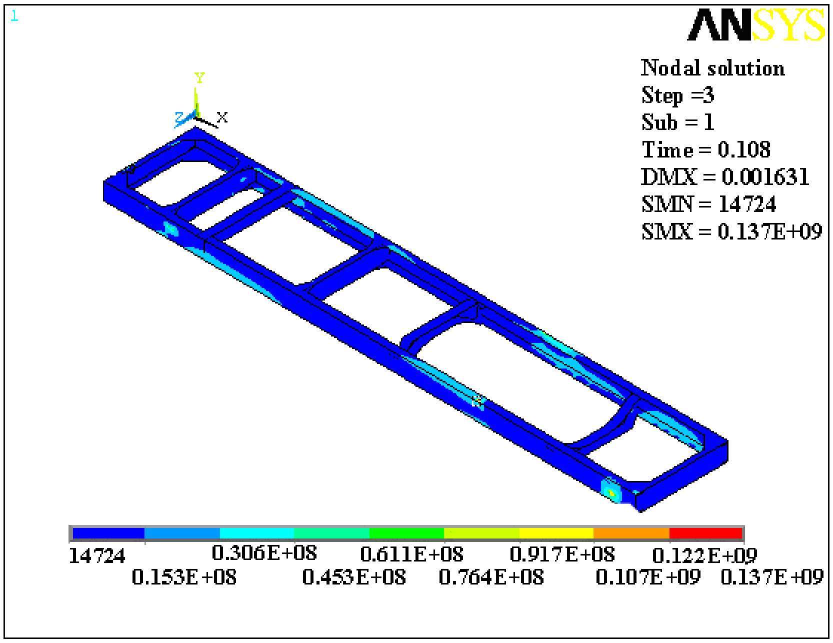

Fig. 13: Stress distribution for the type 2 road profile and with combined welded-riveted joints chassis (joint plate thickness is 8 mm)

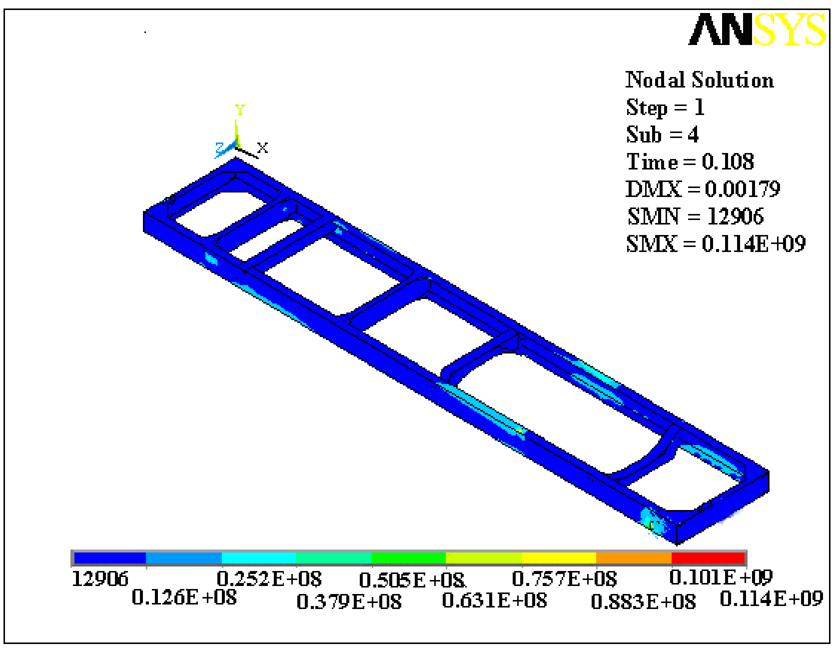

Fig. 14: Stress distribution for the type 3 road profile and with combined welded-riveted joints chassis (joint plate thickness is 12 mm)

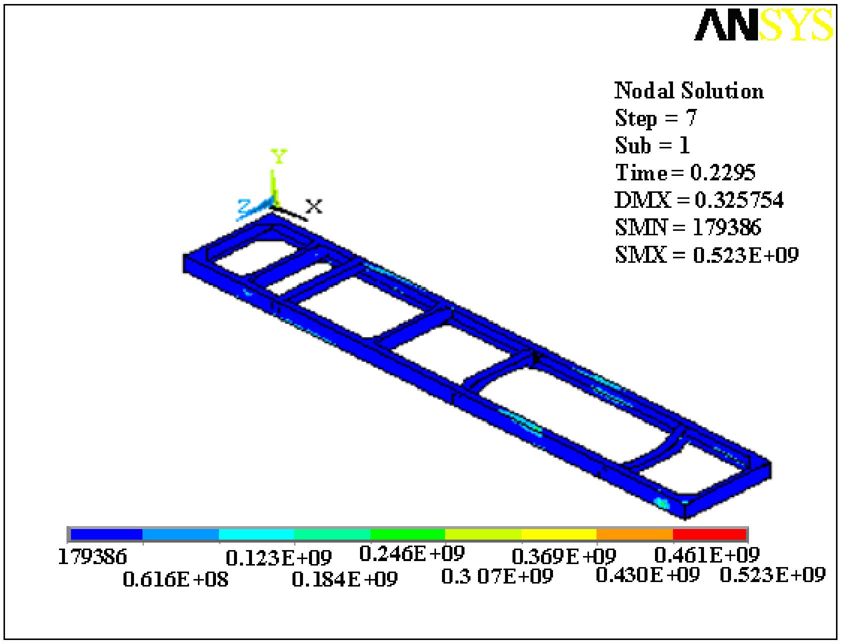

Fig. 15: Stress distribution for the type 1 road profile and with combined welded-riveted joints chassis (joint plate thickness is 5 mm)

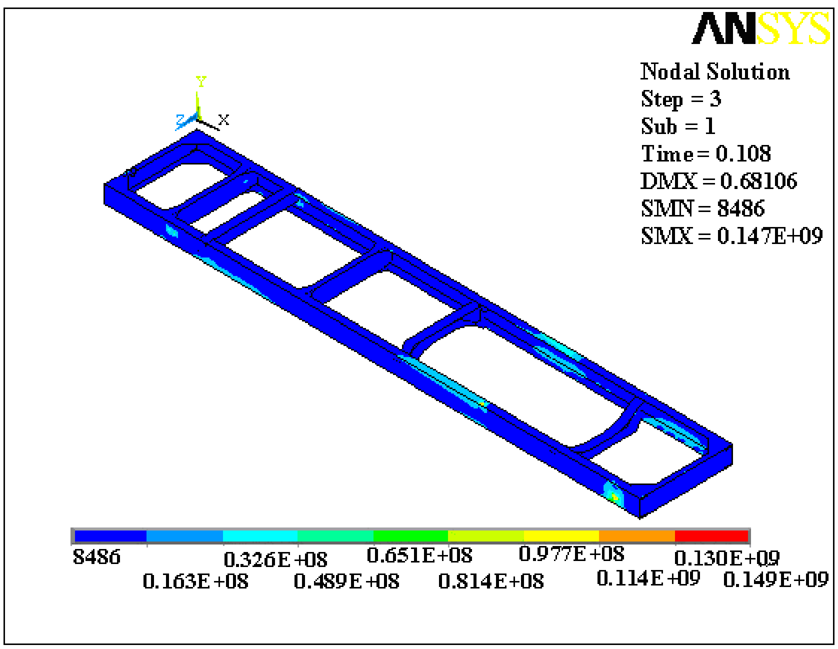

Fig. 16: Stress distribution for the type 2 road profile and with welded joints chassis (joint plate thickness is 8 mm)

Table 5: Results for chassis with combined welded and riveted joints

	Maximum stress on total chassis (MPa)			Maximum stress on joints (MPa)		
	Joints thickness (mm)			Joints thickness (mm)		
Type of loads	5	8	12	5	8	12
Road roughness (type 1)	513	453	376.0	500	288	134
Road roughness (type 2)	137	134	87.4	118	115	31
Road roughness (type 3)	137	115	114.0	118	115	41

CONCLUSION

In this study the stresses have been obtained and examined for a semi heavy Truck Chassis due to the static and dynamic loads. The strength of welded joints in chassis and also the joints with welds and rivets have been analyzed. The effect of three different thicknesses 5, 8, 12 mm for the connection plates have been studied over the strength of the chassis. To validate the FEM based model of the chassis; an experimental model analysis has been carried out. The results obtained can be summarized as follows:

- The results of modal analysis and tests can be used to validate the 3D FE model.

It has been shown that the maximum levels of the stress for chassis with combine welded and riveted joints is less than the stress level for chassis with welded joints.

By increasing the thickness of the reinforcing plates, the stress level reduces significantly. However, by increasing the thickness of the reinforcing plates, the weight of the chassis will increase.

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