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STATICS AND DYNAMICS STRUCTURAL ANALYSIS OF A 4.5 TON TRUCK CHASSIS

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ABSTRACT

Truck chassis forms the structural backbone 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 include the natural frequencies and mode shapes. The responses of the truck chassis which include the stress distribution and displacement under various loading condition are also observed. The method used in the numerical analysis is finite element technique. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range. The mode shape results determine the suitable mounting locations of components like engine and suspension system. Some modifications are also suggested to reduce the vibration and to improve the strength of the truck chassis.

Keyword: *Truck chassis, vibration, torsion stiffness, stress distribution.*

1.0 INTRODUCTION

Malaysia is getting involved in the latest technology to become a well-developed country by 2020 and has made development in various sections of industries. The automotive industry is one of the industries which Malaysia has made large investment and development. The industry is still in the development phase compared to the sophisticated technology used in some foreign countries like US and Japan. Further research and development in this industry is very important.

Truck chassis used in off-road vehicles have almost the same appearance since the models were developed more than 30 years ago. This indicates that the evolution of these structures is still slow and stable along the years [1]. Many researchers in the automotive industry have taken this opportunity to be involved in the chassis manufacturing technology and development. However, the automotive industry in Malaysia, especially in truck manufacturing, is still in the development phase and much relying on foreign technology. Such trend must be changed and to achieve this goal, research on truck chassis is thus important.

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As a truck travels along the road, the truck chassis is excited by dynamic forces induced by the road roughness, engine, transmission and more. Under such various dynamic excitations, the truck chassis tends to vibrate. If any of the excitation frequencies coincides with the natural frequencies of the truck chassis, then resonance phenomenon occurs. The chassis will undergo dangerously large oscillations, which may lead to excessive deflection and failure. The vibration of the chassis will also cause high stress concentrations at certain locations, fatigue of the structure, loosening of mechanical joints, creation of noise and vehicle discomfort. To solve these problems, study on the truck chassis dynamic characteristics is thus essential.

The dynamic characteristics of the truck chassis are also important for the mounting of components on the chassis. The existing truck chassis is normally emphasized on the strength of the structure to support the loading placed upon it. The dynamic aspects are normally not treated seriously. The mode shapes of the truck chassis at certain natural frequencies are very important to determine the mounting point of the components like engine, suspension, transmission and more. Therefore it is important to include the dynamic effect in designing the chassis.

Studies have been done to obtain an optimized chassis design for an off-road vehicle [1]. The torsion stiffness and modal parameters were determined experimentally and then used to validate the finite element model and finally the chassis was optimized to increase the structural stiffness. It was noted that the torsion mode dominated the natural frequency. Based on the basic of structure-borne Noise Vibration Harshness [2], the passenger seat and transmission can be mounted on the nodal point for the first torsion mode. Da Silva et al. [3] have performed the experimental modal analysis on a prototype chassis of an off-road competition vehicle. The Levy identification method was used to identify the basic modal parameters and the updated FEM model was used to create a flexible multi-body model to study vehicle ride, handling and comfort.

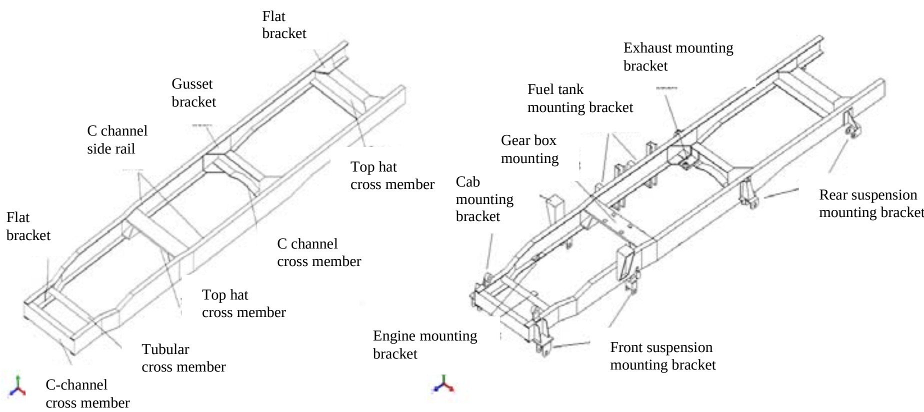
This paper presents the structural dynamic analysis of an existing truck chassis. The dynamic characteristics and mounting points of the truck chassis design are investigated using numerical method. The responses of the truck chassis under various static loading conditions are also presented.

2.0 TRUCK CHASSIS

The truck chassis used for the study has a narrow body with a gross weight of 4.5 ton and a payload of 2500 kg. It consists of 2 C-channel side rails and have 5 cross members along the 2 side rails as shown in Figure 1. There are some additional members like flat and gusset brackets located at the joint between side rails and cross members to strengthen the joints. Towards the middle is a top hat cross member to provide space for mounting of the gear box. The final 2 cross members are C-channel and top hat member. These are located exactly at the location where the rear suspension is mounted at the side rails. It is to strengthen the chassis frame as the suspension mounting point is a highly stressed area.

The material of the truck chassis is AISI 4130 alloy with quenched and tempered treatment. The properties of the material are listed below:

Chemical Composition by weight, %	=	0.30 C, 1.0 Cr, 0.90 Mn, 0.20 Mo
Modulus of Elasticity, E	=	207 GPa
Mass Density, ρ	=	7798 kg/m^3
Yield Strength	=	910 MPa
Tensile Strength	=	1030 MPa



(a) without mounting bracket (b) with mounting bracket

Figure 1: Truck chassis

3.0 FINITE ELEMENT MODEL

The truck chassis was modeled by using 10-noded tetrahedral (Tet-10) solid elements. Experimental and numerical studies on a simple hollow rectangular straight beam suggested that Tet-10 element can mesh the truck chassis geometry and give accurate results of the natural frequencies and the mode shapes. There are two types of analysis carried out; normal mode analysis to determine the natural frequencies and the mode shapes, and the linear static stress analysis to look into the stress distribution and deformation pattern of the chassis under static load. For normal mode analysis, the chassis model was meshed with 15648 Tet-10 elements and 32846 nodes, while for linear static analysis, 37014 Tet-10 elements and 77548 nodes were used.

The boundary conditions are different for each analysis. In normal mode analysis, free-free boundary condition will be applied to the truck chassis model, with no constraint applied to the chassis model. For static analysis, the boundary condition used is to represent the real operation environment of the truck chassis. The pinned boundary condition is applied to the suspension mounting-bracket of the chassis since these locations do not allow any translation but allow only rotation about the axis.

In normal mode analysis, no load is applied to the truck chassis model. In static mode, when the truck is stationary, the loads from the weights of the components like cab, engine, gear box, fuel tank, exhaust and payload is applied to the mounting brackets of the components. Meanwhile the pinned constraints are applied to the suspension mounting brackets. In a symmetry load case which simulates both the truck's front wheels hitting a hump simultaneously, the resulting load is applied to the front suspension mounting brackets. The pinned constraints are applied to the rear suspension mounting brackets. In asymmetry load case which simulates one of the front truck's wheel hitting a hump, the resulting load are applied to the front left suspension mounting brackets with other suspension mounting brackets constraint from any translation (pinned).

4.0 RESULTS AND DISCUSSION ON NORMAL MODE

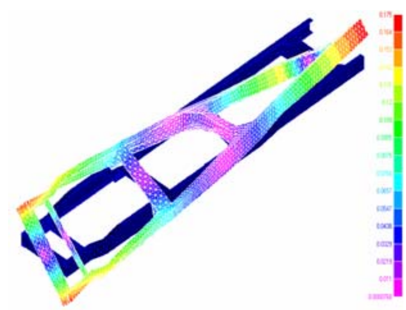
In normal mode analysis, the natural frequencies obtained are used to relate to the operating conditions of the truck while the mode shapes are used to determine whether the mounting locations of components on the truck chassis are suitable. A preliminary study was carried out in determining the suitable boundary conditions for the chassis. A free-free boundary condition was chosen as it is much simpler to test experimentally in this condition, if required. Thus, two types of free-free boundary conditions were tested. The first type is without any constraint and the other type is applying spring boundary condition (by default, $K = 10^{-13}$ N/mm) to the truck chassis. It was shown that both boundary conditions gave identical values of the natural frequency. Thus, in the following analysis, a free-free boundary condition was used and no external load was applied to the truck chassis model.

The natural frequencies and the corresponding modes shapes of the first 6 modes are shown in Figure 2. The contour shows the total translation values of the chassis under the vibration mode. The first 4 modes are global vibrations while local vibration starts at the fifth mode at 54.66 Hz. The dominant mode is a torsion which occurred at 12.68 Hz with maximum translation experienced by both ends of the chassis. The second mode is a lateral bending at 34.23 Hz and a vertical bending at 42.93 Hz for the third mode. At these two modes, the maximum translation is at the front part of the chassis. The forth mode is a lateral bending at 51.33 Hz with 3 nodal points. The maximum translation is experienced by the top hat cross member. The member also experienced maximum translation at fifth mode which is a localized bending mode. The top hat cross member is the mounting location of the truck gear box. The sixth mode is the torsion mode at 61.2 Hz with maximum translation at both ends of the chassis.

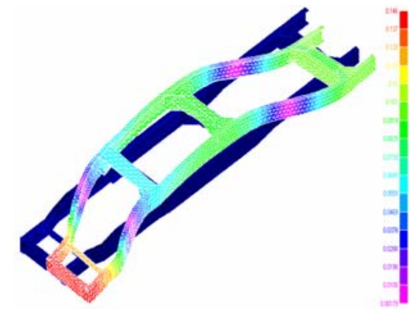
The dynamic characteristic of the truck chassis is very important when related to the operating environment of the truck. During the running of the engine, the chassis will be subjected to the forces from the engine, transmission system, exhaust and more. As the truck travel along the road, the chassis will be excited by dynamic forces induced by road roughness. Each of the above excitation forces has its excitation frequencies which when it coincidence with the natural frequencies will result in resonance phenomena.

Diesel engine is known to have the operating speed varying from 8 to 33 revolutions per second (rps) [4]. In low speed idling condition, the speed range is about 8 to 10 rps. This translates into excitation frequencies varying from 24 to 30 Hz. By considering the natural frequencies of the truck, it can be said that the truck can be excited at around the first natural frequency during the idling condition of the engine. The excitation from the transmission system is about 0 to

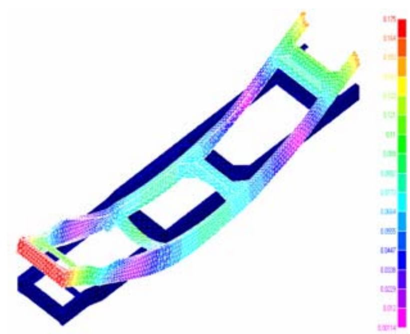
100 Hz [4]. The main excitation is at low speeds, when the truck is in the first gear. At higher gear or speed, the excitation to the chassis is much less.



Mode 1: 12.68 Hz
Global: torsion

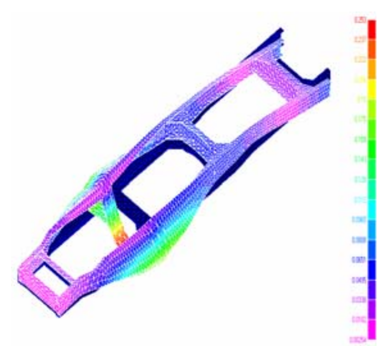


Mode 2: 34.23 Hz
Global: 2 nodal points lateral bending

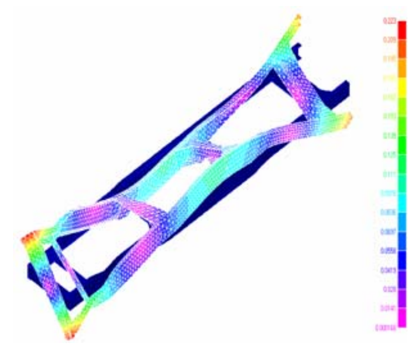


Mode 3: 42.93 Hz
Global: 2 nodal points vertical bending

Mode 4: 51.33 Hz
Global: 3 nodal points lateral bending



Mode 5: 54.66 Hz
Local: bending of top hat cross member



Mode 6: 61.66 Hz
Global: torsion

Figure 2: Natural frequencies associated to the operating environment of a truck chassis

The natural frequency of the truck chassis should not coincide with the frequency range of the axles because this can cause resonance which may give rise to high deflection and stresses and poor ride comfort. Excitation from the road is the main disturbance to the truck chassis when the truck travels along the road. In practice, the road excitation has typical values varying from 0 to 100 Hz. At high speed cruising, the excitation is about 3000 rpm or 50 Hz. This value is close to the fourth and the fifth mode natural frequency.

4.1 Mounting Location of Components on the Chassis

Mounting of vibration components of the truck on the nodal point of the chassis is one of the vibration attenuation methods to reduce the transmission of vibration to the truck chassis. Figure 3 shows the mounting location of the engine and transmission system on the chassis which is along the symmetry axis of the chassis first torsion mode. At such location, the excitation from the engine can be reduced for the first torsion mode.

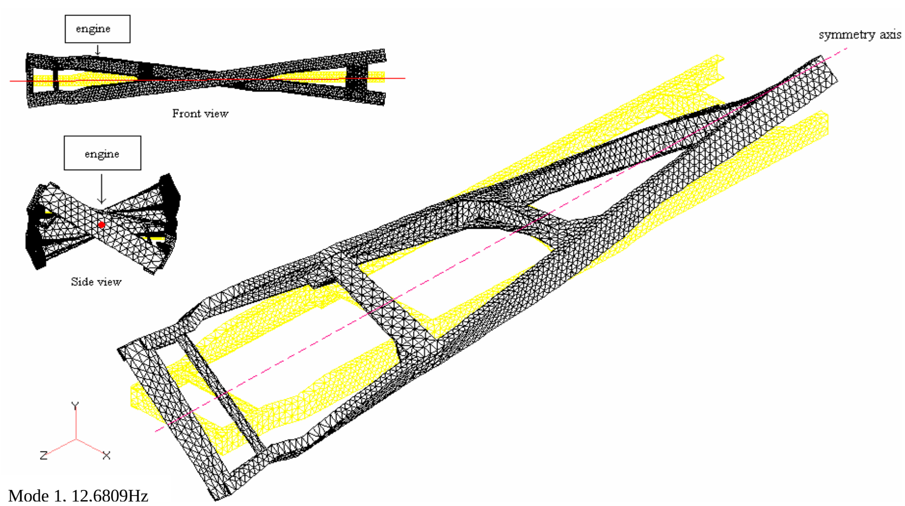


Figure 3: Mounting location of engine and transmission on the chassis

The first vertical bending mode with the location of nodal points and the truck wheel center location is shown in Figure 4. It is apparent that the wheel center is located not on the nodal point, but close to the nodal point. The distance between the nodal point and the wheel center is about 306 mm for front wheel and 127 mm for rear wheel. This is mainly due to the configuration of the static loading on chassis which determines the mounting of the suspension system. The mounting location of the suspension is suitable because the excitation from the suspension input motion can be reduced for the vertical bending mode.

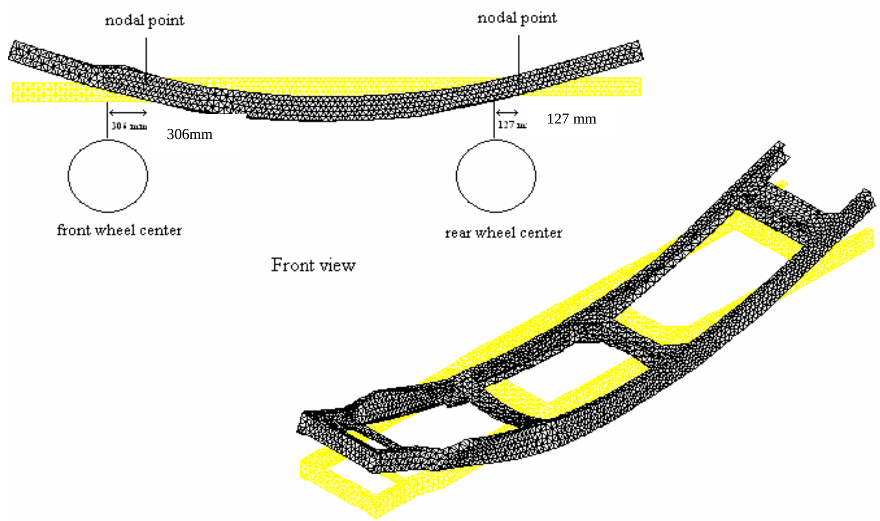


Figure 4: Location of nodal point for vertical bending mode and the truck wheel center location

5.0 RESULTS AND DISCUSSION ON STATIC ANALYSIS

5.1 Truck Components Loading

This simulation is based on the condition of the truck being stationary. The ladder frame chassis was treated as a simply supported beam and loads were due to the weight of components applied to the beam. The support loads from the axles were distributed through spring hangers. The axle's reaction loads were obtained by resolving forces and taking moments from the weights and positions of the components.

For practical calculations, it was recommended that the load on the chassis frame, including its own weight, is concentrated at a small number of points. These point loads were statically equivalent to the actual distributed load carried by the vehicle [5]. The weight of components mounted was considered as point loads acting on the chassis. The pay load of 2500 kg was divided into six equivalent forces acting on positions where the cargo is mounted. Table 1 shows the weights and forces of the components and their positions along the chassis. The axle reaction forces acted vertically upward and applied at the center position between the suspension mounting brackets.

The result shows that the front axle reaction force is 854 kg (8376 N) and the rear axle reaction force is 2396 kg (23507 N). These reaction force values are below the maximum capacity of the axle-front axle (2200 kg) and rear axle (3000 kg), compared to the truck specifications. The calculation result is as expected because the axle reaction forces from the calculations are only due to the weight of the components and the pay load when the truck is stationary. In actual case, when the truck is moving along the road, the reaction force should be higher due to road roughness or in some cases, like the truck hitting humps or holes.

Table 1: Weights and forces of components and positions along the chassis

No.	Components	Weight (kg)	Load (N)	Position from origin (mm)
1	Cab	125	1226	4183
2	Engine	50	490	3875
3	Engine	100	981	3523
4	Cab	125	1226	3216
5	Gear box	50	490	2873
6	Pay load	417	4088	2873
7	Fuel tank	40	392	2433
8	Pay load	417	4088	2150
9	Chassis weight	200	1962	2150
10	Fuel tank	40	392	2023
11	Exhaust	20	196	1805
12	Pay load	417	4088	1710
13	Pay load	417	4088	1080
14	Pay load	417	4088	450
15	Pay load	417	4088	0

Figure 5 shows the results of stress distribution and deformation pattern of the chassis under the weight loading of components mounted on it. The stress distribution is almost uniform; with the highest stressed areas are at the rear suspension brackets, where the maximum stress is about 128 MPa. The rest of the chassis structure has very low stress value, about 50 MPa. The stress contour shows that all the cross members and side rails take the bending load evenly. Every member plays a role in strengthening the chassis.

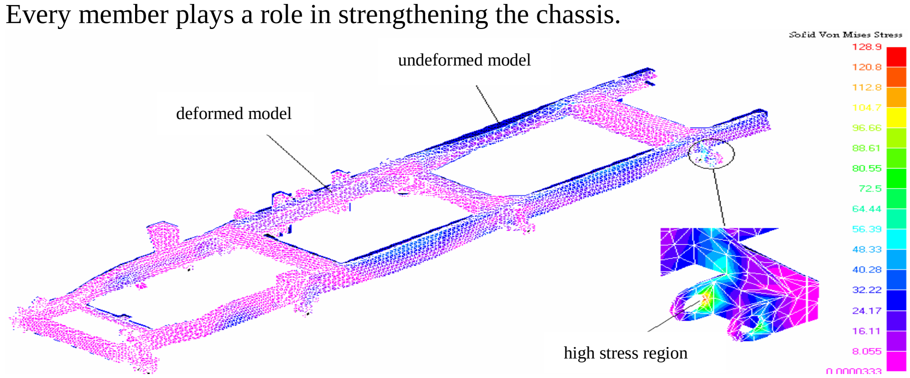


Figure 5: Stress contour and deformation pattern of the chassis under truck components loading

Figure 6 shows the deformation contour and the deformation pattern of the chassis under static case. The highest deformation happens at the side rails where the cargo payload is applied. The maximum translation is 0.768 mm. The deformations of the cross member and mounting brackets is low. The deformation contour shows that the side members act as the load bearing structure especially when the payload is placed upon it. The other components like the cab, engine, gear box, fuel tank and exhaust do not have significant effect to the stress and deformation of the chassis compared to the payload.

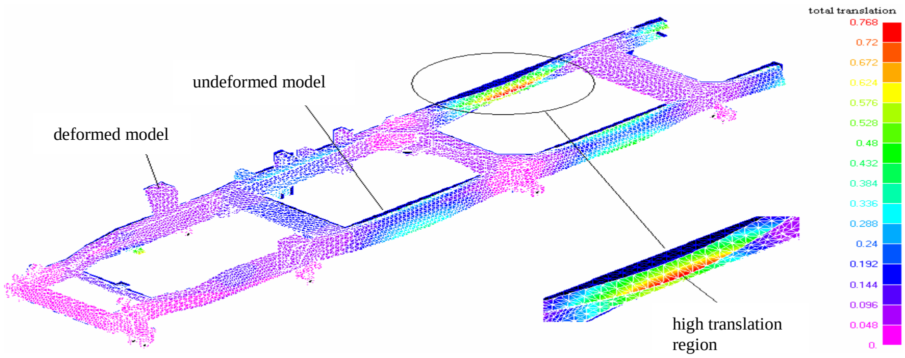


Figure 6: Deformation contour and the deformation pattern of the chassis under truck components loading

5.2 Asymmetrical Loading

This simulation is based on the condition whereby one of the truck's front wheel rest on a hump, thus causing torsion to the chassis. Figure 7 illustrates the stress contour and deformation pattern of the chassis under asymmetrical loading. The front suspension brackets with constraint experience highest stress, about 480 MPa. The other region has less stress value, below 150 MPa. Figure 8 shows the deformation contour and deformation pattern of the chassis under asymmetrical loading. The front part of the chassis where the asymmetry load is applied experiences the highest translation, whose magnitude depends on the height of hump.

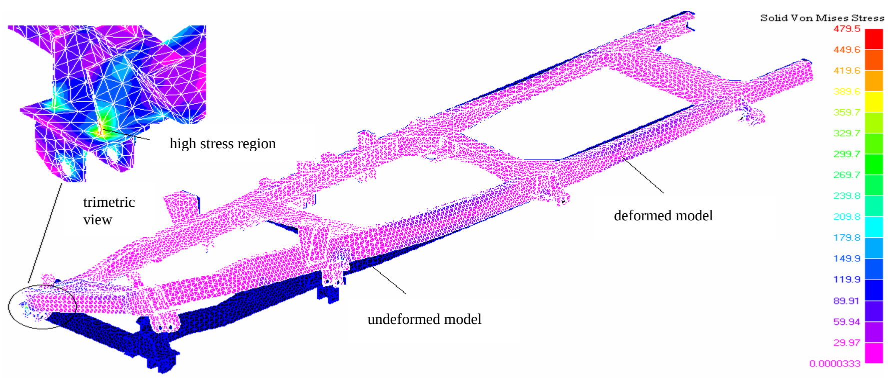


Figure 7: Stress contour and deformation pattern of the chassis under asymmetrical loading

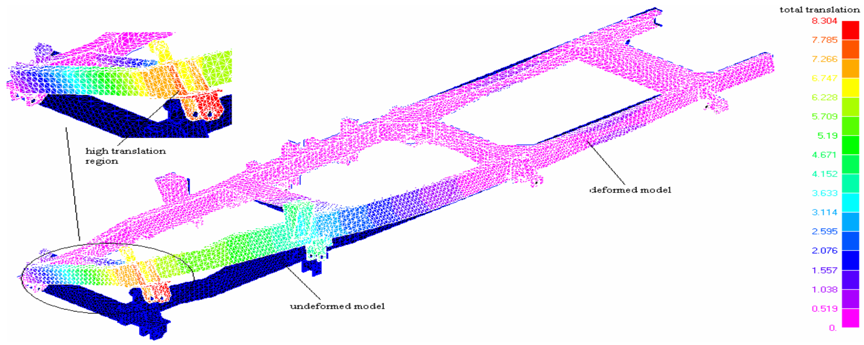


Figure 8: Deformation contour and the deformation pattern of the chassis under asymmetrical loading

For the two types of loading, it was shown that the suspension mounting brackets experienced the highest stress when; the truck components loading (128 MPa) and asymmetrical loading (490 MPa). The maximum stress of 490 MPa is below the yield strength of the chassis material which is 910 MPa. The additional hump loading on the chassis increased the stress. As for the other regions of the chassis, low stresses were obtained. Thus, the chassis structure and joints are sufficiently strong to withstand the loading applied to it.

6.0 IMPROVEMENT ON THE CHASSIS

Some suggestions are proposed to improve the vibration response and stress distribution on the chassis structure. From the normal mode analysis, the top hat cross member shows high translation in the fifth mode as illustrated in Figure 2. Also, the local vibration mode starts at the top hat cross member. At 54.66 Hz, the major disturbance to the chassis is from the road excitation.

The bending deformation will cause concentration of high stress at the joint where the top hat cross member is attached to the side rails. Moreover, the top hat cross member is the location where the gear box is mounted. The gear box excitation can cause the top hat cross member to vibrate easily and excessively. The excessive vibration also affects the tolerance between the mounting of the gear box and causes loosening of joints like bolts and nuts. Besides that, the drive shaft misalignment problem occurs due to the vibration of the top hat cross member where the gear box is mounted.

Thus, it is suggested to stiffen the top hat cross member by adding thickness to the top hat cross member. The overall weight of the chassis will be slightly increased. Additional member (stiffener) to the joint between top hat cross member and side rail can also be fitted to stiffen it.

The normal mode analysis enables the locations of nodal points of the vibration modes to be determined. To effectively reduce the excitation from the suspension input motion, the wheels suspension should be located exactly or nearer to the nodal points of the vertical bending mode. By doing so, no or little energy will be transferred from the suspension motion to the chassis structure. Thus the suspension excitation source can be reduced.

The static analysis shows concentration of stress at the sharp edges of the chassis. Even though the stress value at the sharp edges of the chassis is not so high compared to the stress value on the suspension mounting brackets, this concentration of stress can be reduced by smoothening the sharp corner. The sharp corner can be eliminated by introducing fillet in the design. Besides that, the welded joints are also one of the highest stress concentration areas. The imperfect welding during the fabricating process are the weak areas where failure of the chassis is likely to occur due to fatigue. It is suggested that the chassis to be designed and manufactured using less welding joints. The welding joints can be replaced by mechanical joints like rivet.

7.0 CONCLUSION

The paper has looked into the determination of the dynamic characteristic (the natural frequencies and the mode shapes) of the truck chassis, investigating the mounting locations of components on the truck chassis and observing the response of the truck chassis under static loading conditions. The first six natural frequencies of the truck chassis are below 100 Hz and vary from 12.68 to 61.64 Hz. For the first four modes, the truck chassis experienced global vibration except for the fifth mode. The global vibrations of the truck chassis include torsion, lateral bending and vertical bending with 2 and 3 nodal points. The local bending vibration occurs at the top hat cross member where the gearbox is mounted on it.

The mounting location of the engine and transmission system is along the symmetrical axis of the chassis's first torsion mode where the effect of the first mode is less. However, the mounting of the suspension system on the truck chassis is slightly away from the nodal point of the first vertical bending mode. This might due to the configuration of the static loading on the truck chassis.

For the linear static analysis, the stress distribution and deformation profile of the truck chassis subjected to two loading conditions: truck components loading and asymmetrical loading had been determined. Maximum stress occurred at the mounting brackets of the suspension system while the maximum translation occurred at the location where the symmetry and asymmetry load is acting. The maximum stress of the truck chassis is 490 MPa while the maximum translation is 33.6 mm. These values are acceptable as compared to the yield strength of the chassis material and the tolerance allowed for the chassis.

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