STRESS ANALYSIS OF HEAVY DUTY TRUCK CHASSIS USING FINITE ELEMENT METHOD

O Kurdi, R Abd- Rahman, M N Tamin

Faculty of Mechanical Engineering
Universiti Teknologi Malaysia
81310 UTM Skudai, Johor
E-mail: ojokurdi@yahoo.com
roslan@fkm.utm.my

Abstract

One of the most important steps in development of a new truck chassis is the prediction of fatigue life span and durability loading of the chassis frame. The age of many truck chassis in Malaysia are of more than 20 years and there is always a question arising whether the chassis is still safe to use. Thus, fatigue study and life prediction on the chassis is necessary in order to verify the safety of this chassis during its operation. Stress analysis using Finite Element Method (FEM) can be used to locate the critical point which has the highest stress. This critical point is one of the factors that may cause the fatigue failure. The magnitude of the stress can used to predict the life span of the truck chassis. In this study, the stress analysis is accomplished by the commercial finite element packaged ABAQUS.

Keywords: Stress Analysis; Finite Element Method; Truck Chassis

1. Introduction

The major challenge in today's ground vehicle industry is to overcome the increasing demands for higher performance, lower weight, and longer life of components, all this at a reasonable cost and in a short period of time. The chassis of trucks is the backbone of vehicles and integrates the main truck component systems such as the axles, suspension, power train, cab and trailer. Since the truck chassis is a major component in the vehicle system, it is often identified for refinement.

Many researchers carried out study on truck chassis. C. Karaoglu and N. S. Kuralay investigated stress analysis of a truck chassis with riveted joints using FEM. Numerical results showed that stresses on the side member can be reduced by increasing the side member thickness locally. If the thickness change is not possible, increasing the connection plate length may be a good alternative [1]. M. Fermer, G. McInally and G. Sandin investigated the fatigue life of Volvo S80 Bi-Fuel using MSC/Fatigue [2]. F. A. Conle and C.-C. Chu, did research about fatigue analysis and the local stress-strain approach in complex vehicular structures [3]. Structural optimization of automotive components applied to durability problems has been investigated by W. G. Ferreira et al [4]. M. Fermér and H. Svensson studied on industrial experiences of FE-based fatigue life predictions of welded automotive structures [5].

R. R. P. Filho et al have investigated and optimized a chassis design for an off road vehicle with the appropriate dynamic and structural behavior, taking into account the aspects relative to the economical viability of an initial small scale production. The design of an off-road vehicle chassis has optimized by increasing the torsional stiffness, maintenance of center of gravity, total weight of structure and simpler geometry for reduction of production cost [6]. The integration of computer aided design and engineering software codes (Pro/Engineer, ADAMS, and ANSYS) to simulate the effect of design changes to the truck frame has been studied by C. Cosme et al [7].

M. Chiewanichakorn et al [8] investigated the behavior of a truss bridge, where an FRP deck replaced an old deteriorated concrete deck, using experimentally validated finite element (FE) models. Numerical results show that the fatigue life of the bridge after rehabilitation would be doubled compared to pre-rehabilitated reinforced concrete deck system. Based on the estimated truck traffic that the bridge carries, stress ranges of the FRP deck system lie in an infinite fatigue life regime, which implies that no fatigue failure of trusses and floor system would be expected anytime during its service life.

N. Ye and T. Moan have investigated the static and fatigue behavior of aluminium box-stiffener/web frame connections using Finite Element Analysis (FEA) to provide a connection solution that can reduce the fabrication costs by changing the cutting shapes on the web frame and correspondingly the weld process meanwhile sufficient fatigue strength can be achieved [9]. FE based fatigue was used to locate the critical point of probable crack initiation and to predict the life in a door hinge system [10]. In this study, stress analysis of heavy duty truck chassis loaded by static force will be investigated to determine the location of critical point of crack initiation as a preliminary data for fatigue life prediction of this truck chassis.

2. Finite Element Analysis of Truck Chassis

2.1. Truck definition and classification

Generally, truck is any of various heavy motor vehicles 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.

There are two classifications most applicable to Recreational Vehicle tow trucks. The first one is the weight classes, as defined by the US government, ranging from Class 1 to Class 8 as listed in Table 1. The second is classified into a broader category:

- Light Duty Truck
- Medium Duty Truck
- Heavy Duty Truck

Table 1:Vehicle Manufacturer Truck Classification

Category	Class	GVWR[1]	Representative Vehicles
Light	1	0 - 27 kN 0 - 6,000 lbs.	pickup trucks, ambulances, parcel delivery
	2	27 - 45 kN (6,001 - 10,000 lbs.)	
	3	45 - 62 kN (10,001 - 14,000 lbs.)	
Medium	4	62 - 71 kN (14,001 - 16,000 lbs.)	city cargo van, beverage delivery truck,
	5	71 - 87 kN (16,001 - 19,500 lbs.)	wrecker, school bus
	6	87 - 116 kN (19,501 - 26,000 lbs.)	
	7	116 - 147 kN (26,001 to 33,000 lbs.)	
Heavy	8	147 kN and over (33,000 lbs. and over)	truck tractor, concrete mixer, dump truck, fire truck, city transit bus

Notes:

1. Gross Vehicle Weight Rating (GVWR): weight specified by manufacturer as the maximum loaded weight (truck plus cargo) of a single vehicle.

2.2. Model of truck chassis

In this work, the truck chassis model used is the Hino model. The model is depicted in Figure 1. The model has length of 12.350 m and width of 2.45 m. The material of chassis is ASTM Low Alloy Steel A 710 C (Class 3) with 552 MPa of yield strength and 620 MPa of tensile strength. The other properties of chassis material are tabulated in Table 2.

Table 2: Properties of truck chassis material [11]

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

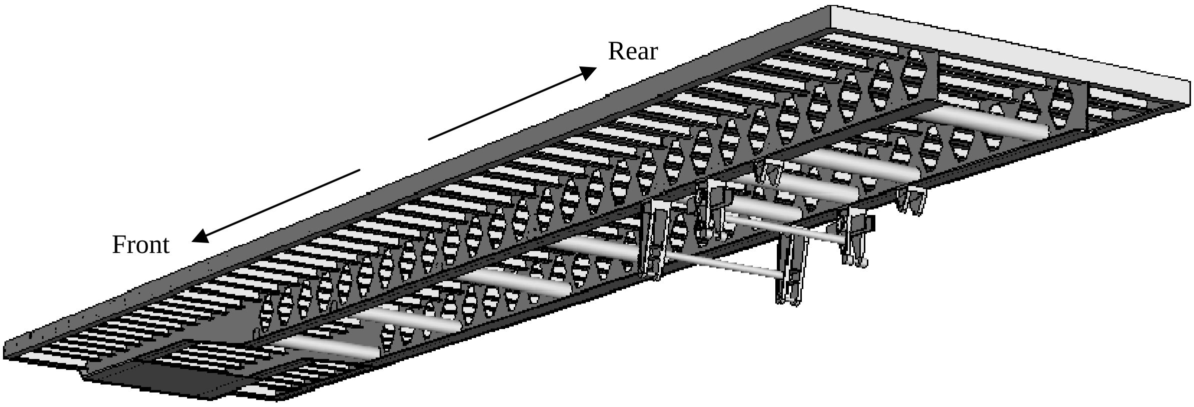


Figure 1. Model of Truck Chassis

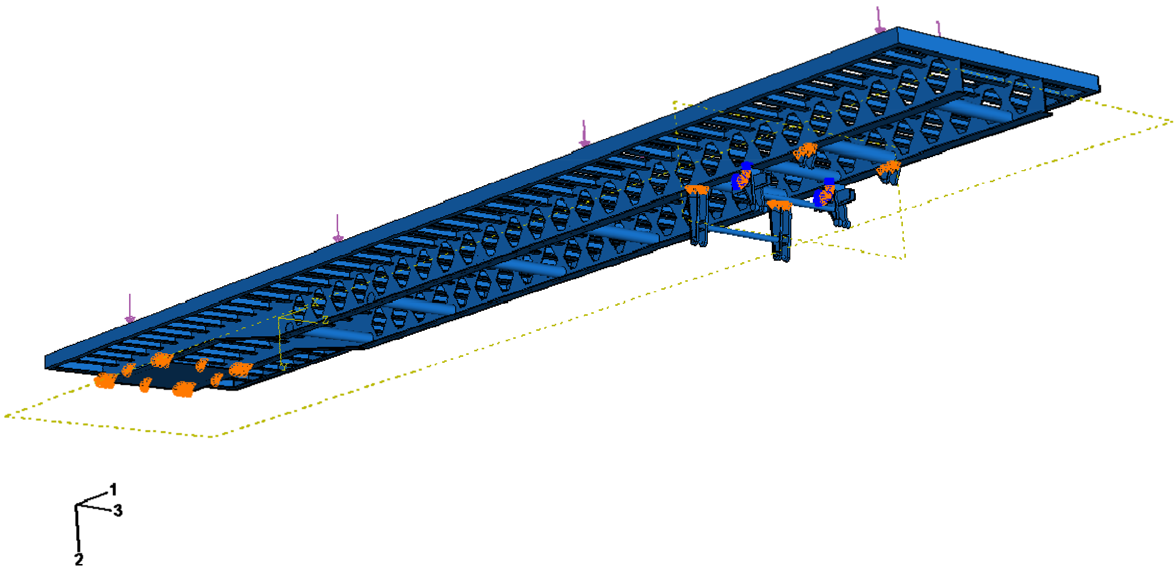


Figure 2. Static load (pressure = 67564.57 N/m^2)

2.3. 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 36.000 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:

$$p = \frac{F}{A} = \frac{36.000\,\text{kg}\;x\;9.81\,\text{m/s}^2}{5.227\,\text{m}^2} = 67564.57\ \text{N/m}^2 \qquad (1)$$

Where:

p = pressure (N/m^2)

F = force (kg. m/ s^2)

A = total contact area (m^2)

9.81 is a constant of gravity.

2.4. Boundary conditions

There are 3 boundary conditions (BC) of model; the first BC is applied in front of the chassis, the second and the third BC are applied in rear of chassis, there are shown in Figure 3. The type of BC 1 is pinned (the displacement is not allowed in all axes and the rotation is allowed in all axes) that represent the contact condition between chassis and cab of truck as shown in figure 4(a). The BC 2 represents the contact between chassis and upper side of spring that transfer loaded weight of cargo and chassis to axle.

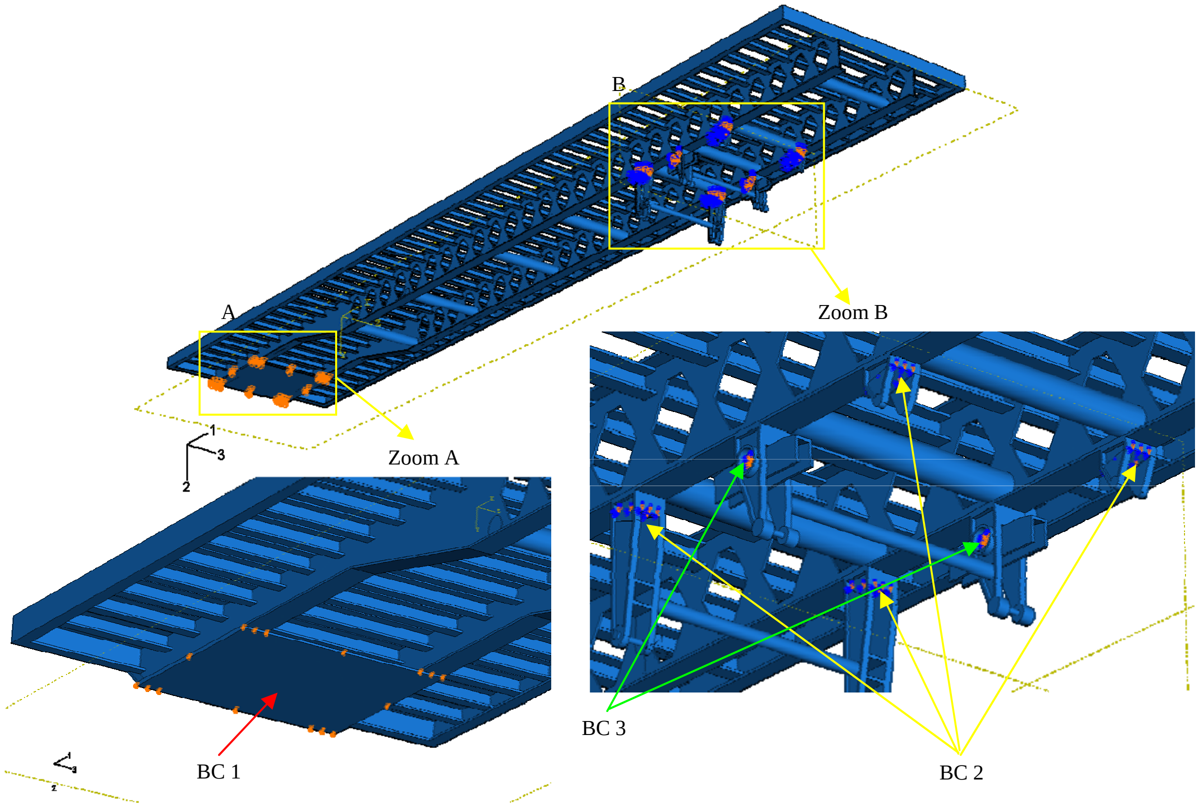


Figure 3. Boundary conditions representation in the model

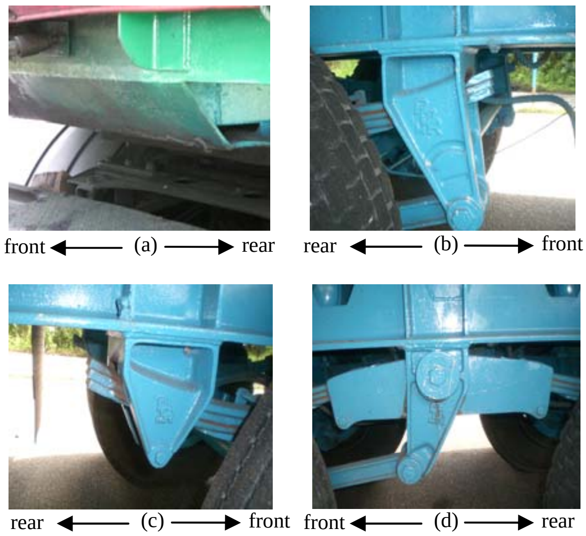


Figure 4. Boundary conditions representation in the object, 4(a). BC 1, 4(b). and 4(c). BC 2, 4(d). BC 3

The contact condition of BC in the object is shown in figure 4. In the BC 2, the displacement in axis 2 and the rotation respect to all axes is zero.
In the position where the BC 3 applied, there is a contact between inside surface of opening chassis and outside surface of bolt. In ABAQUS, this contact is called interaction. In this case, the type of the interaction is frictionless surface to surface contact. In the BC3, the displacement and the rotation is zero in all axes on all of bolt's body. This condition is called fixed constrain. The detail interaction condition on surface of BC 3 is shown in figure 5. The bolt in BC 3 was assumed perfectly rigid. This assumption was realized by choosing a very high modulus young value of the bolt properties.

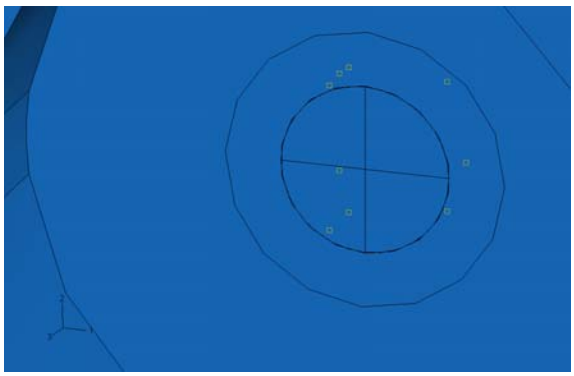


Figure 5. Frictionless surface to surface contact on BC 3

2.5. Element and Nodes

The meshed truck chassis model has 101466 elements and 37697 nodes. The element shape is tetrahedral and element type is 3D stress. In order to get a better result, locally finer meshing is applied in the region which is suspected to have the highest stress.

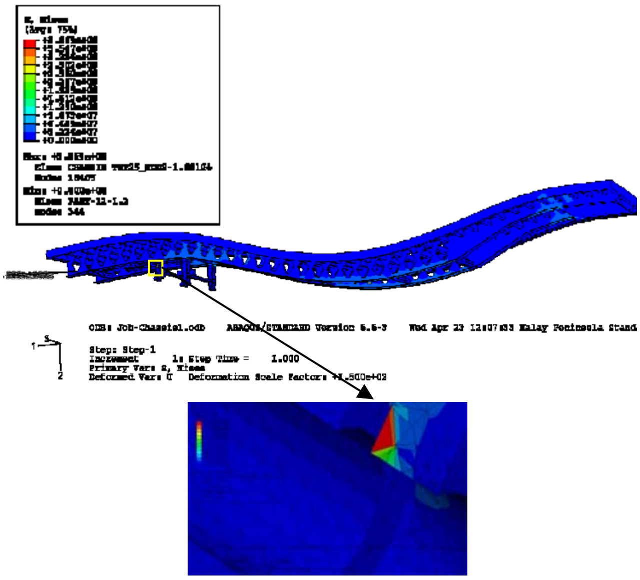
3. Results and discussion

3.1. Von Misses stress

The location of maximum Von Misses stress is at opening of chassis which is contacted with bolt as shown in Figure 6. The stress magnitude of critical point is 386.9 MPa. This critical point is located at element 86104 and node 16045. The internal surface of opening of chassis was contacted with the very stiff bolt. The BC 3 is also a fixed constraint, thus it cause a high stress on it. Based on static safety factor theory, the magnitude of safety factor for this structure is 1.43. The formula of Safety Factor (SF) is defined by [11]:

$$SF = \frac{\text{significant strength of material}}{\text{corresponding significant stress, from normal load}} \quad (2)$$

J. Vidosic [12] recommends some value of safety factor for various condition of loading and material of structures. He recommends the value of 1.5 to 2 for well known materials under reasonably environmental condition, subjected to loads and stresses that can be determined readily. Based on this result, it is necessary to reduce the stress magnitude of critical point in order to get the satisfy SF value of truck chassis. The truck chassis can be modified to increase the value of SF especially at critical point area.

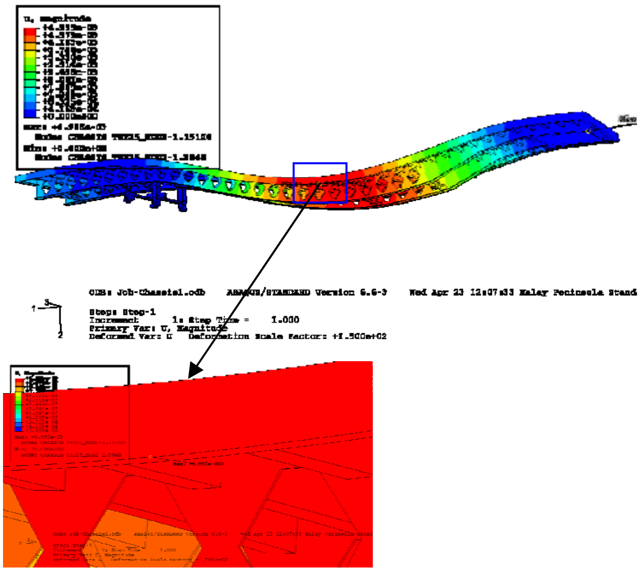


Detail location of maximum Von Misses Stress

Figure 6. Von Misses stress distribution and critical point location

3.2. Displacement

The displacement of chassis and location of maximum displacement is shown in Figure 7. The magnitude of maximum displacement is 4.995 mm and occurs at middle of chassis. Maximum deflection is occurred at the middle of BC 1 and BC 2.



Detail location of the maximum displacement

Figure 7. Displacement distribution and the maximum displacement location

For validation purpose, the region between BC 1 and BC 2 of chassis where the highest stress occurred is approximated by one dimensional simple beam loaded by concentrated force at mid point. The uniform distributed pressure on this region is replaced by a single concentrated force at mid point. The magnitude of the single force is obtained by multiplying the magnitude of pressure with the total area where the pressure is applied. The result agrees well with this approximation. The result agrees well with this approximation. The approximation result shows that the displacement of this simple beam is located in the midpoint of beam with magnitude of:

$$\delta_{\max} = \delta(L/2) = \frac{PL^3}{48EI} \quad (3)$$

$$= \frac{71145.5(8.1^3)}{48(207x10^9)x(8.6x10^{-4})}$$

$$= 4.43 \text{ mm}$$

The maximum displacement of numerical simulation result is 4.99 mm. The result of numerical simulation is bigger 11.2 % than the result of analytical calculation. The difference is caused by simplification of model and uncertainties of numerical calculation.

4. Conclusion

Numerical simulation result shows that the critical point of stress occurred at opening of chassis which is contacted with bolt. The magnitude of highest stress is critical because the value of SF is below than the recommended value. Since fatigue failure started from the highest stress point, it can be concluded that this critical point is an initial to probable failure. Thus, it is important to take note to reduce stress magnitude at this point. The location of maximum deflection agrees well with the maximum location of simple beam loaded by uniform distribution force

Acknowledgements

The authors would like to thank the Ministry of Science, Technology and Innovation (MOSTI) for funding this research via project no. 04-01-06-SF0109. Our gratitude also goes to the members of Sound and Vibration Laboratory and Computational Solid Mechanics Laboratory of Universiti Teknologi Malaysia for their helpful discussions throughout the completion of this work.

References

1. C. Karaoglu and N. S. Kuralay, Stress Analysis of a Truck Chassis with Riveted Joints, Elsevier Science Publishers B. V. Amsterdam, the Netherlands, 2000, Volume 38, 1115 – 1130.
2. M. Fermer, G. McInally and G. Sandin, Fatigue Life Analysis of Volvo S80 Bi-Fuel using MSC/Fatigue, Worldwide MSC Automotive Conference, Germany, September, 1999.
3. F. A. Conle and C.-C. Chu, Fatigue Analysis and the Local Stress-strain Approach in Complex Vehicular Structures, International journal of fatigue (Int. j. fatigue), 1997.
4. W. G. Ferreira, F. Martins, S. Kameoka, A. S. Salloum, and J. T. Kaeya, Structural Optimization of Automotive Components Applied to Durability Problems, SAE Technical Papers, 2003.
5. M. Fermér and H. Svensson, Industrial Experiences of FE-based Fatigue Life Predictions of Welded Automotive Structures, Fatigue & Fracture of Engineering Materials and Structures 24 (7), 2001, 489–500.
6. R. R. P. Filho, J. C. C. Rezende, M.de F. Leal, J. A. F. Borges, Automotive Frame Optimization, 12th International Mobility Technology Congress and Exhibition, Sao Paulo, Brasil, November 18 – 20, 2003.
7. C. Cosme, A. Ghasemi, and J. Gandevia, Application of Computer Aided Engineering in the Design of Heavy – Duty Truck Frames, International Truck & Bus Meeting & Exposition, Detroit, Michigan November 15 – 17, 1999.
8. M. Chiewanichakorn, A. J. Aref, S. Allampalli, Dynamic and Fatigue Response of a Truss Bridge with Fiber Reinforced Polymer Deck, International Journal of Fatigue 29 (2007) 1475–1489.
9. N. Ye and T. Moan, Static and Fatigue Analysis of Three Types of Aluminium Box-Stiffener/Web Frame Connections, International Journal of Fatigue 29 (2007) 1426–1433.
10. S. Bekah, Fatigue Life Prediction in a Door Hinge System Under Uni-Axial and Multiaxial Loading Condition, Master Thesis, Ryerson University, Toronto, Ontario, Canada, 2004.
11. R. C. Juvinall and K. M. Marshek, Fundamental Machine Component Design, John Willey & Son, Inc., USA, 2006.
12. J. P. Vidosic, Machine Design Project, Ronald Press, New York, 1957.
13. J. M. Gere, Mechanics of Material, Brooks/Cole, USA, 2001.