

Analysis of Dynamic Crack Development and Crack Surface Connection in Single Region Boundary Element Method

¹Mohammad Najafiyan, ²Zabihola Bagheri, ²Rohullah Rabiee,
²Hadi Doust Mohammadi and ²Alireza Ghasemi

¹Department of Civil Engineering, Khomein Branch, Islamic Azad University, Khomein, Iran

²Islamic Azad University, Khomein Branch, Iran

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 be used to predict the life span of the truck chassis.

Key words: Stress Analysis • Dynamic crack • Crack surface • Element method

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. McNally 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.

MATERIALS AND METHODS

In the following, both a two and three-dimensional beam models with a non-propagating surface crack are presented. For both models the crack surfaces are assumed to be planar and smooth and the crack thickness negligible. The beam material properties are considered linear elastic and the displacements and strains are assumed to be small. The region around the crack is discretized into conventional finite elements. The breathing crack behavior is simulated as a full frictional contact problem between the crack surfaces, which is an inherently non-linear problem. Any possible sliding is assumed to obey Coulomb's law of friction and penetration between contacting areas is not allowed. The non-linear dynamic problem is discussed for the two-dimensional model and the corresponding quasi-static for the three-dimensional model. Both problems are solved utilizing incremental iterative procedures. For completeness reasons, the contact analysis in three dimensions and the formulation and solution of the non-linear dynamic problem are presented below.

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

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

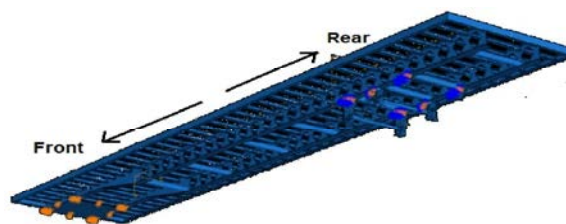


Fig. 1: Model of Truck Chassis

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,

Table 2: Properties of truck chassis material

Modulus Elasticity E (Pa)	247 x 10 ⁹
Density ρ (kg/m ³)	7800
Poisson Ratio	.6
Yield Strength (MPa)	535
Tensile Strength (MPa)	641

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.

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:

$$P = \frac{F}{A} = \frac{36,000 \text{ kg} \times 9.81 \text{ m/s}^2}{5.224 \text{ m}^2} = 67564.57 \text{ N/m}^2 \quad (1)$$

Where:

p = Pressure (N/m²)
 F = Force (kg. m/ s²)
 A = Total contact area (m²)
 9.81 is a constant of gravity.

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

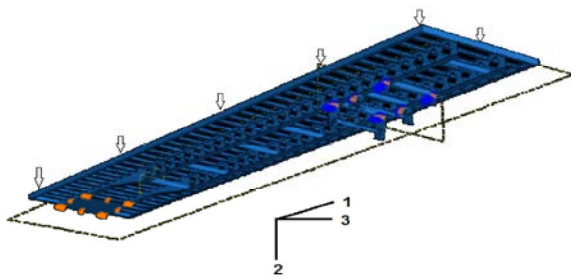


Fig. 2: Static load (pressure = 75534.27 N/m²)

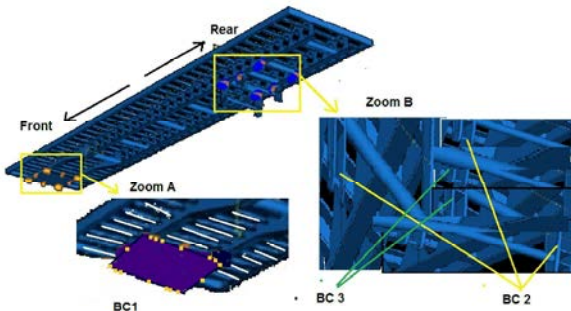


Fig. 3: Boundary conditions representation in the model

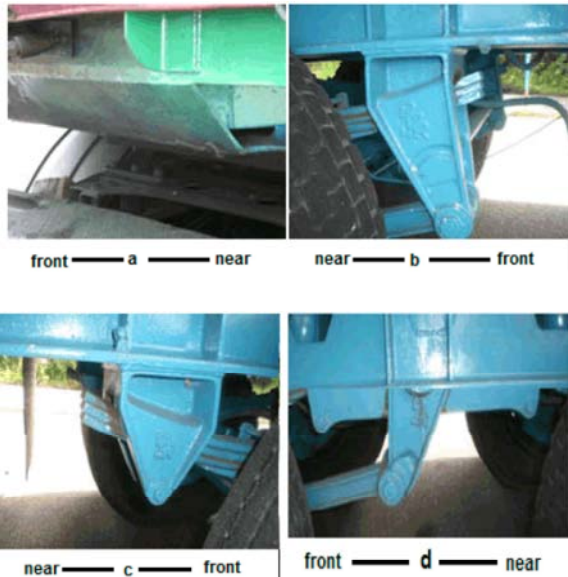


Fig. 4: Boundary conditions representation in the object, 4(a). BC 1, 4(b). and 4(c). BC 2, 4(d). BC 3

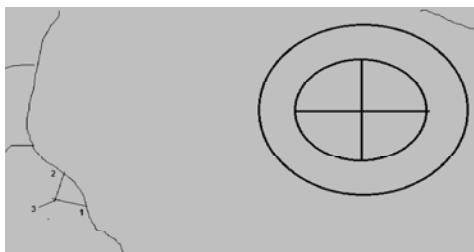


Fig. 5: Frictionless surface to surface contact on BC 3

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.

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.

1Element 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.

RESULTS AND DISCUSSION

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]:

$$SF = \frac{\text{Significant strength of material}}{\text{corresponding significant from normalload}} \quad (2)$$

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

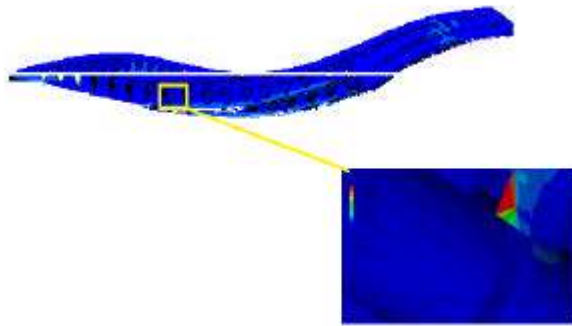


Fig. 6: Von Mises stress distribution and critical point location

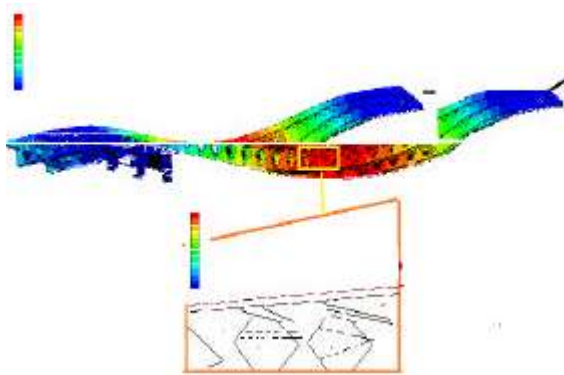


Fig. 7: Displacement distribution and the maximum displacement location

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.

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.

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 approximation result shows that the displacement of this simple beam is located in the midpoint of beam with magnitude of:

$$\delta_{\max} = \delta(L/2) = \frac{PL^2}{48EI} = \frac{76543 \times 9.21^2}{42(312 \times 10^9) \times (5.4 \times 10^{-3})} = 5.32 \text{ mm} \quad (3)$$

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.

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.

REFERENCES

1. Karaoglu, C. 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.
2. Fermer, M., G. McNally and G. Sandin, 1999. Fatigue Life Analysis of Volvo S80 Bi-Fuel using MSC/Fatigue, Worldwide MSC Automotive Conference, Germany.
3. Conle, F.A. and C.C. Chu, 1997. Fatigue Analysis and the Local Stress-strain Approach in Complex Vehicular Structures, International journal of fatigue (Int. j. fatigue).
4. Ferreira, W.G., F. Martins, S. Kameoka, A.S. Salloum and J.T. Kaeya, 2003. Structural Optimization of Automotive Components Applied to Durability Problems, SAE Technical Papers.
5. Fermér, M. and H. Svensson, 2001. Industrial Experiences of FE-based Fatigue Life Predictions of Welded Automotive Structures, Fatigue & Fracture of Engineering Materials and Structures, 24(7): 489-500.
6. Filho, R.R.P., J.C.C. Rezende, M. De F. Leal and J.A.F. Borges, 2003. Automotive Frame Optimization, 12th International Mobility Technology Congress and Exhibition, Sao Paulo, Brasil, pp: 18-20.

7. Cosme, C., A. Ghasemi and J. Gandevia, 1999. Application of Computer Aided Engineering in the Design of Heavy - Duty Truck Frames, International Truck & Bus Meeting & Exposition, Detroit, Michigan, 17: 15 -17.
8. Chiewanichakorn, M., A.J. Aref and S. Allampalli, 2007. Dynamic and Fatigue Response of a Truss Bridge with Fiber Reinforced Polymer Deck, International Journal of Fatigue, 29: 1475-1489.
9. Ye, N. and T. Moan, 2007. Static and Fatigue Analysis of Three Types of Aluminium Box-Stiffener/Web Frame Connections, International Journal of Fatigue, 29: 1426-1433.
10. Bekah, S., 2004. Fatigue Life Prediction in a Door Hinge System Under Uni-Axial and Multiaxial Loading Condition, Master Thesis, Ryerson University, Toronto, Ontario, Canada.
11. Juvinall, R.C. and K.M. Marshek, 2006. Fundamental Machine Component Design, John Wiley & Son, Inc. USA.
12. Vidosic, J.P., 1957. Machine Design Project, Ronald Press, New York,
13. Gere, J.M., 2001. Mechanics of Material, Brooks/Cole, USA.