Comparative Stress Analysis of Automotive Chassis for Multiple Thickness Variations

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Abstract: This study examines the stress analysis of a ladder-type low-loader truck chassis with C-beams that can support 7.5 tons of weight. The commercial software program CATIA version 5 was used to assist in the Finite Element Method (FEM) study. Reducing the thickness of the structural components or altering the design are the only ways to lower the production costs of truck chassis. It is essential to evaluate the truck chassis's stress distribution before manufacture to improve the design and guarantee dependability. The thickness of the side and cross members, as well as the cross member's position concerning the rear end, were altered to reduce the amount of stress at crucial locations on the chassis frame. The numerical study showed that moving the cross member can be a valuable substitute for altering the thickness when it is not possible. Analytical computations and the calculated outcomes were contrasted. It was found that, notwithstanding some magnitude fluctuations, the maximum deflection values derived from the numerical analysis were in agreement with theoretical expectations.

Keywords: Evaluation of stress, forecasting fatigue lifespan, utilizing finite element techniques, etc.

I. INTRODUCTION

The truck chassis serves as the vehicle's structural cornerstone, combining important components such as the axles, suspension, motor, cab, and trailer. Static, dynamic, and cyclic loads are among the forces it can resist from road imperfections while supporting the weight of the cabin and its cargo. Predicting the service life of the chassis components and assessing fatigue behavior need stress analysis. The point of greatest stress, also known as the critical point, is frequently the source of possible failure and is identified with the aid of this study. The amount of stress present at this moment is a crucial factor in determining how long the chassis will last. To attach main components like the engine, suspension system, and gearbox efficiently and optimally, it is essential to know where this vital point is. This crucial stress zone may be found using the widely recognized Finite Element Method (FEM) [1,2]. Using a safety factor, which offers a buffer over the theoretical strength limitations, engineers may accommodate any design uncertainties [3]. A structure's fatigue life is influenced by several elements, including the cyclic nature of the stress, component geometry, surface polish, material properties, residual stress levels, internal defect size and distribution, loading direction, and grain size, according to Jadav Chetan S. et al. [4].
Variations in profile thickness were used in this study to assess various chassis constructions. To evaluate the strength, structural analysis was done on frames that were 4 mm, 5 mm, and 6 mm thick. Cross members were also repositioned and their thickness changed in regions that were determined to be high-stress locations. CATIA V5R10 was used for finite element analysis and vehicle chassis modeling.

Five distinct scenarios were investigated to assess how thickness affected the distribution of chassis stress:
Case 1: The thickness of the side members is fixed at 04 mm.
Case 2: The thickness of the side members was raised to 05 mm.
Case 3: The side member thickness rose to 06 mm.
Case 4: The fourth cross member was moved to a distance of 02520 mm from the back end.
Case 5: The fifth cross member's thickness was changed to 05 mm.

II. LITERATURE REVIEW

Roslan Abd Rahman et al. [1] used the commercial finite element program ABAQUS to perform a stress study on a heavy-duty vehicle chassis. The goal was to pinpoint important stress locations where fatigue life may be improved by design changes. The chassis was made of ASTM low alloy steel A710 C (Class 3), which has a tensile strength of 620 MPa and a yield strength of 552 MPa. Element 86104 and node 16045, which are important locations close to a bolt hole, had the highest stress ever measured, 386.9 MPa. This area was determined to be a likely place where structural breakdown would begin.

Cicek Karaoglu et al. [2] used the commercial finite element program ANSYS version 5.3 to do a stress study of a heavy-duty vehicle chassis with riveted joints. The study looked at how connection plate size and side member thickness affected the distribution of stress. While the connection plate thickness was altered in two ways—by employing local plates (8–12 mm) and by changing the base plate thickness from 7 mm to 10 mm—the side member thickness was adjusted between 8 mm and 12 mm. Additionally, the connecting plate's (L) length was changed from 390 mm to 430 mm. Using an ideal connection plate length is the most practical way to reduce stress concentrations when using local plates to increase the side member thickness is not viable because of the additional weight.

Mohd Azizi Muhammad Nor et al. [3] used CATIA V5R18 for modeling to do a stress study of a real-world low-loader structure made of I-beams for a 35-ton trailer. When a simply supported beam was subjected to an evenly distributed load, the analysis showed that the sites of maximum stress and maximum deflection nearly matched the theoretical predictions. However, a disparity between the 3D finite element analysis (FEA) and the theoretical 2D analysis was noted. The largest deflection, 7.79 mm, was found between BC1 and BC2 boundary conditions.

Jadav Chetan S et al. [4] have out an exhaustive review of several fatigue analysis techniques used on car frame structures.

N.K. modified the tractor-trailer chassis model in several ways. Ingole et al. [5]. These included (1) changing the cross-sectional areas of the cross members, (2) changing the cross-sectional areas of both cross and longitudinal members, (3) introducing additional cross-sectional area variations for both components and (4) changing the cross-member positions while taking into account the variable cross-sectional areas of both cross and longitudinal members. Maximum stress of 75 MPa and a weight of 751.82 kg were displayed by the original chassis. Case 4 produced the largest weight decrease of the suggested examples, shedding almost 112 kg in comparison to the other three. As a result, Case 4's adjustments are suggested. In Case 3, the maximum stress ranged from 25 MPa to 66 MPa, and an 88 kg weight reduction was accomplished.

III. BASIC CALCULATION FOR CHASSIS FRAME

The model used for Case 1 is Eicher 10.75. 200 mm × 55 mm × 5 mm "C" channel pieces are used to form the chassis side components.

A wheelbase of 3515 mm, a rear overhang of 1305 mm, and a front overhang of 1005 mm are all measured. Steel is used in the chassis' construction.

Table1. Physical and mechanical traits of truck chassis steel

Stiffness Constant	00200GPa.
Lateral Strain Ratio	000.266.
Bulk Density	007860 Kg/m^3.
Proportional Limit	00250 MPa.
Congruity	Isotropic Continuum

The truck has a 7.5-ton rated capacity, which is equivalent to 7500 kg or 73,575 N. Taking into account an extra 1.25%, the effective capacity rises to 9375 kg or 91,968.75 N. The body and engine weigh 2 tons, which is the same as 2000 kg or 19,620 N.

Therefore, the entire load on the chassis is:

Weight of Body and Engine + Capacity = 91,968.75 N + 19,620 N = 111,588.75 N is the total load.

Because the chassis contains two beams, the weight is dispersed equally:

111,588.75 N divided by two is 55,794.375 N for the load per beam.

A single beam's Uniformly Distributed Load (UDL), assuming the load is evenly distributed over a 5825 mm span, is:

UDL = 9.578 N/mm = 55,794.375 N / 5825 mm

Figure1 Generic Chassis

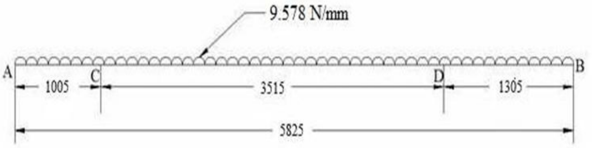

Figure 2 Chassis as a simply supported beam with overhang.

$$Y = \frac{wx(b-x)}{24EI}\left[x(b-x) + b^2 - 2(c^2 + a^2) - \frac{2}{b}\{xc^2 + a^2(b-x)\}\right]$$

Stress produced on the beam is as under

$$\sigma = \frac{Mmax}{Z}$$

=0123.83MPa.

Deflection of chassis

=01.08456mm.

FE ANALYSIS OF CHASSIS

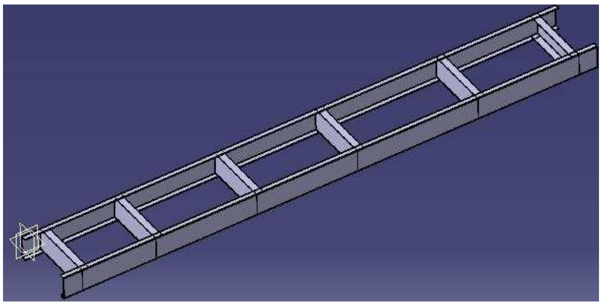

Figure 3 Chassis Blueprint Case1.

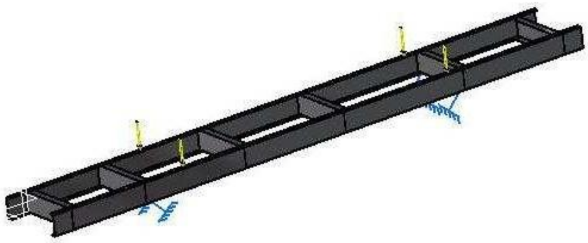

Figure 4 External Constraints Case1.

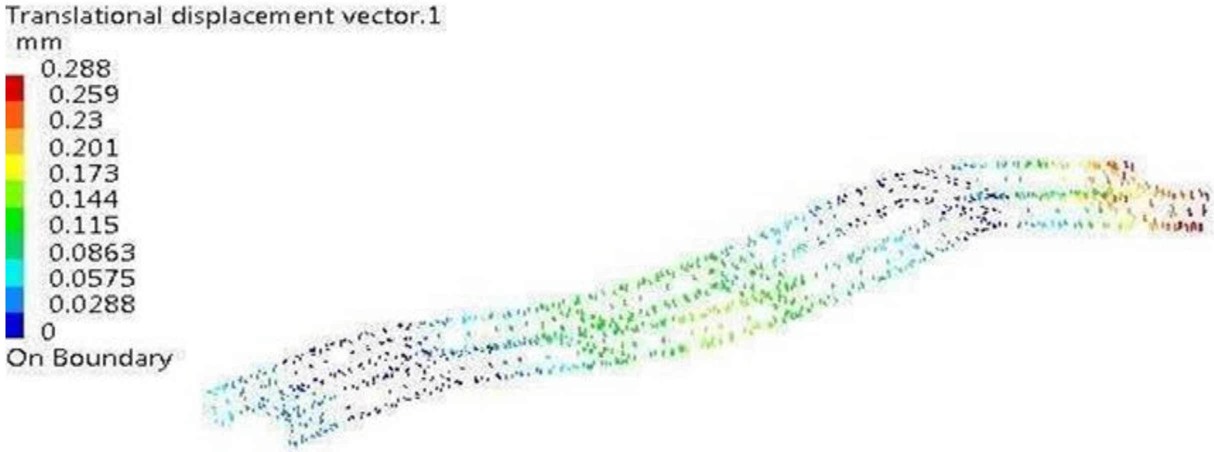

Figure 5 Dislocations of Case 1.

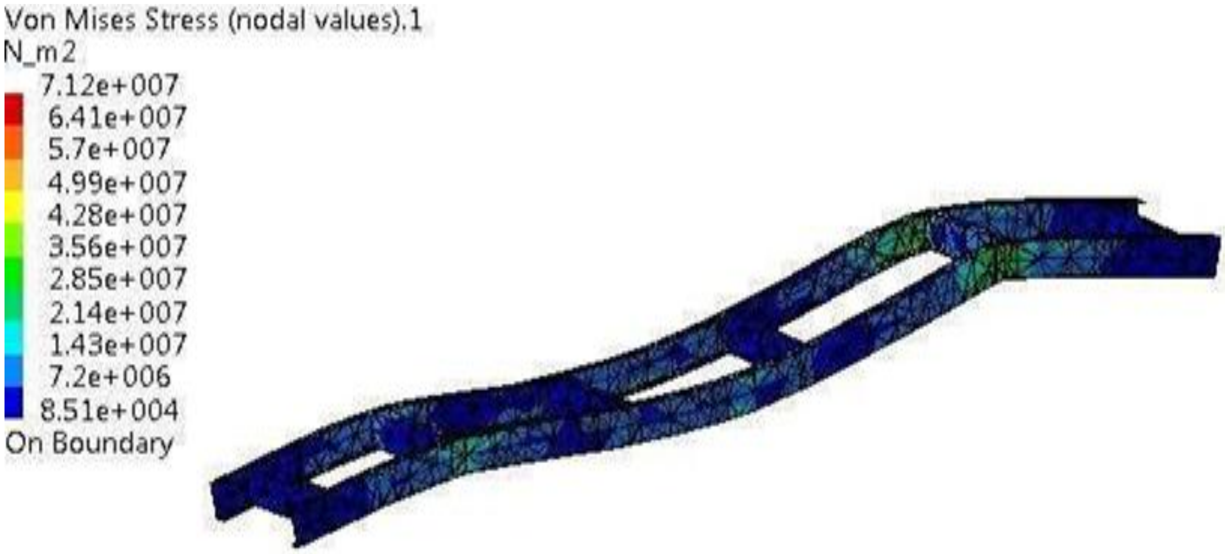

Figure 6 Von Mises Stress of Chassis for Case 1.

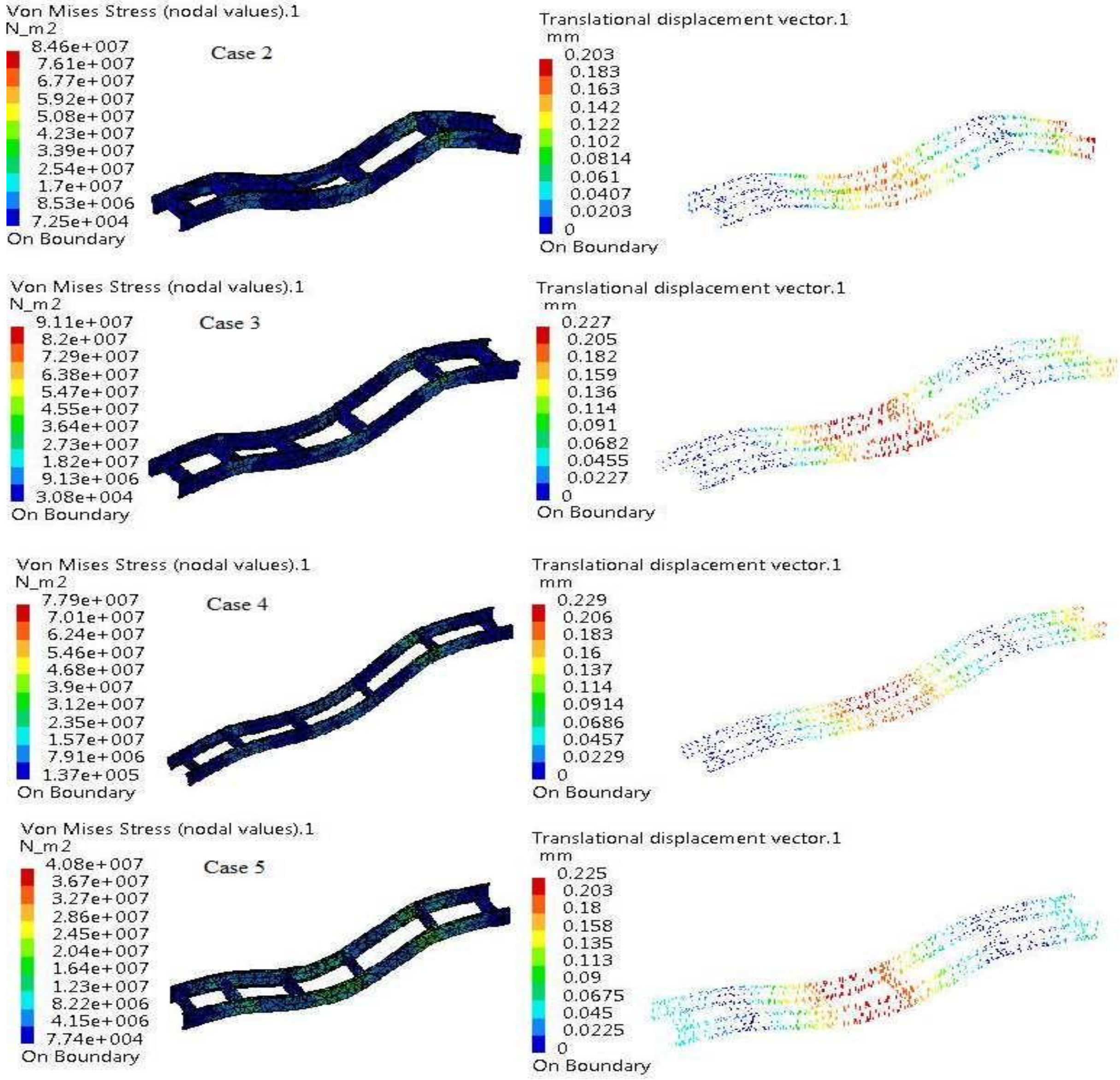

Figure 7 Stress Intensity (Von Mises formulation) of Cases 2, 3, 4, and 5.

IV. RESULT AND DISCUSSION

Table 2 Contrast of Results

Sr No.	Analytical Method		FE Analysis		
	Displacement (mm)	Stresses (N/mm^2)	Displacement (mm)	Stresses (N/mm^2)	Weight (Kg)
Case 1	001.08450	00123.830	000.2880	0071.20	00141.480
Case 2	001.02710	00100.830	000.2030	0084.60	00165.060
Case 3	000.97800	00085.570	000.2270	0091.10	00188.640
Case 4	001.02710	00100.830	000.2290	0077.90	00165.060
Case 5	000.93000	00100.830	000.2250	0040.80	00167.980

V. CONCLUSION

- The studies were carried out under static and structural loading conditions. For a material thickness of 4 mm, the finite element (FE) analysis found a maximum stress of 123.83 MPa, which matches the analytically predicted maximum shear stress. The highest displacement obtained by numerical simulation was 0.288 mm. The observed discrepancies are due to model simplifications, numerical computation errors, and poor mesh quality.

- A weight gain of 2.92 kg is obtained by simply increasing the thickness of the cross element, as demonstrated by a comparison between Cases 5 and 2. The displacement goes up by 0.022 mm, however, this change also results in a 43.8 N/mm² decrease in stress.
- The overall weight of the construction is unaffected by changing the cross member's position alone, as illustrated by a comparison between Cases 4 and 2. On the other hand, it raises displacement by 0.026 mm and decreases stress by 6.7 N/mm².
- A comparison of Cases 3 and 2 shows that a 23.58 kg increase in weight causes a 6.5 N/mm² increase in stress and a 0.024 mm increase in displacement.
- Therefore, it turns out that adjusting the thickness of the cross member at the key stress point reduces stress and deflection in the chassis more effectively than changing the thickness of the side members or moving chassis parts.

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