Vehicle Chassis Analysis: Load Cases & Boundary Conditions For Stress Analysis

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ABSTRACT

The current work contains the load cases & boundary conditions for the stress analysis of chassis using finite element analysis over ANSYS. Finite element model of the vehicle chassis is made. Shell elements have been used for the longitudinal members & cross members of the chassis. The advantage of using shell element is that the stress details can be obtained over the subsections of the chassis as well as over the complete section of the chassis. Beam elements have been used to simulate various attachments over the chassis, like fuel tank mountings, engine mountings, etc. Spring elements have been used for suspension & wheel stiffness of the vehicle. Impact loads have been measured (in terms of 'g') experimentally by using accelerometers on the front & rear axles. Input spectrum for Power Spectrum Analysis has been obtained by using FFT Analyzer for the secondary roads at the driving speed of 30 kmph. The results of finite element analysis have been checked by experimental methods too, & very good resemblance has been found between both the methods.

1. INTRODUCTION

After years of steady, predictable model changes, the automobile industry is in the midst of the most intense product changeover in its history. To accomplish the need to design a moderate car, the structural engineer will need to use imaginative concepts. The demands on the automobile designer increased and changed rapidly, first to meet new safety requirements and later to reduce weight in order to satisfy fuel economy requirements. Experience could not be extended to new vehicle sizes, and performance data was not available on the new criteria. Mathematical modeling was therefore a logical avenue to explore. Most recently, the finite element method, a computer dependent numerical technique, has opened up a new approach to vehicle design.

2. LOAD CALCULATIONS

Whilst adequate durability under dynamic conditions is a design requirement for the vehicle structures, the static load cases cannot be disregarded. The values for the individual load cases are taken from the expected service conditions of the particular vehicle. The worst-case loading conditions (distribution of the load) as well as overloading must be considered for the static load case. The factors usually applied to the static load case, especially for those vehicles with a long overhang containing concentrated loads (e.g., rear engine buses). Such loads result in high bending moments over the rear axle. The various dynamic conditions considered here for the determination of the axle loads are considered below.

2.1 Loads on the Grades

The influence of the grade on the axle loads is worth considering. On primary and secondary loads, grade angle θ is as high as 6.842 deg. For the front axle, taking moment about front axle wheel at ground considering fig. (1), we get,

$$M_r \times L = M_t \cos\theta \times L_f - M_t \sin\theta \times H \qquad (1)$$

The above equation gives the weight on the rear axle on grades, through which increase in weight (ΔM_g) on the rear axle due to grade can be calculated. Similarly the loads shift on negative grade can also be calculated.

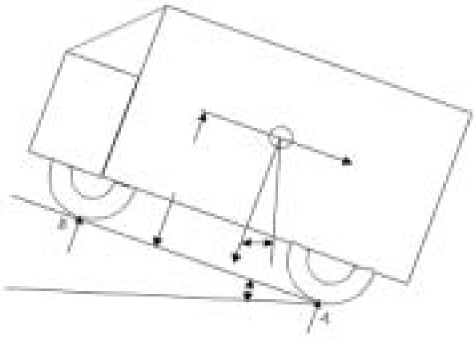


Fig. (1) Vehicle on Grades

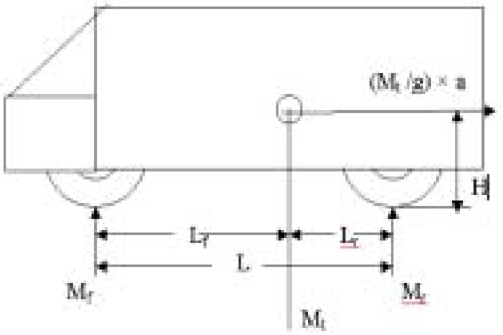


Fig. (2) Acceleration of Vehicle

2.2 Low Speed Acceleration

Assuming the vehicle is accelerating on the level ground from the low speed, then the inertia force will act against the direction of acceleration, the load on the axles are calculated by taking moments. Taking moments about 'A' again as shown in the fig (2),

$$M_f \times L = M_t \times L_r - (a \times M_t \times H) / g \qquad (2)$$

Through the above equation the change in loads due to acceleration (ΔM_a) on wheels can be calculated.

2.3 Braking

The vehicle under consideration is the N_2 category vehicle. For the N_2 category of the vehicle, the following conditions apply (IS: 11852 (Part-3) – 1987), under P-F type of test, mean retardation (D) ≥ 6 m/sec^2 . Due to this retardation the change of the axle loads is, (considering fig. (2), with inertia due to retardation in opposite direction),

$$\Delta M_{br} = (H \times M_t \times D) / (L \times g) \tag{3}$$

2.4 Steady State Cornering

Lateral forces act on the at the Roll Center of vehicle during cornering. A line joining the roll centers of the suspensions is roll axis. The lateral separation between the suspensions causes them to develop roll resisting moments proportional to the difference in the roll angle between the body and the axle.

$$K_\varphi = 0.5 \times K_s \times s^2 \text{ Kgf mm/rad} \tag{4}$$

Taking moment about the roll axis, considering fig. (3),

$$M_\varphi = [\{(M_t \times H_l \times v^2)/(Rg)\} \times \cos\varphi + M_t \times H_l \times \sin\varphi] \times \cos\varepsilon \tag{5}$$

Also, $M_\varphi = (K_{\varphi f} + K_{\varphi r}).\varphi$ (6)

During turning, the vehicle rolls due to centrifugal forces. The transfer of loads on front axle due to rolling can be calculated as follows:

$$M_{\varphi f} = [K_{\varphi f}.M_t.H_l(v^2/(Rg)) / (K_{\varphi f}+K_{\varphi r} - M_t.H_l)] + M_f.H_f.(v^2/(Rg)) \tag{7}$$

Also, $M_{\varphi f} = \Delta F_{zf}.T$

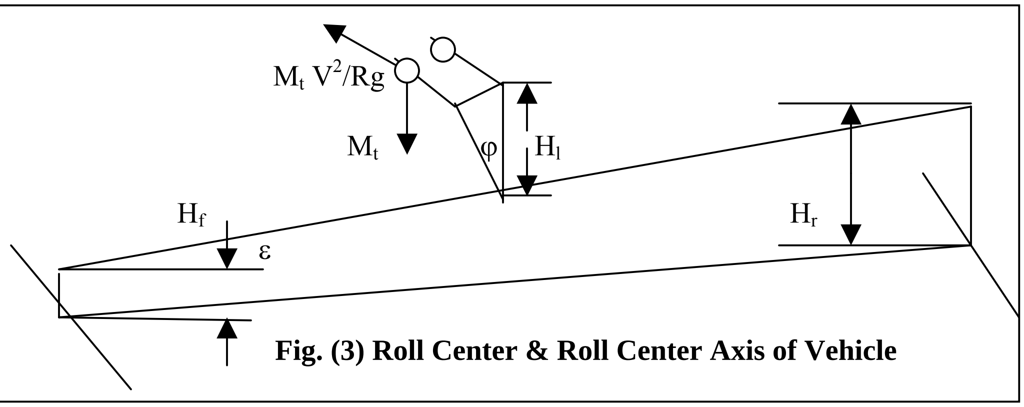


Fig. (3) Roll Center & Roll Center Axis of Vehicle

3 FINITE ELEMENT MODEL OF A VEHICLE CHASSIS

In Finite element model shell elements have been used for the longitudinal members & cross members of the chassis. The advantage of using shell element is that the stress details can be obtained over the subsections of the chassis as well as over the complete section of the chassis. Beam elements have been used to simulate various attachments over the chassis, like fuel tank mountings, engine mountings, transmission mounting, etc. Spring elements have been used for suspension stiffness of the vehicle. The vehicle model is fixed at the wheels. The model is tested with the experimental results determined for the opposite wheels at the bumps. The diagonally opposite wheels of the vehicle are lifted to full deflection of suspension, fig (5), & the stress is measured at six locations. The measured stresses & the stresses calculated in ANSYS for the vehicle model at these six locations are almost similar, as shown in fig (4 & 6).

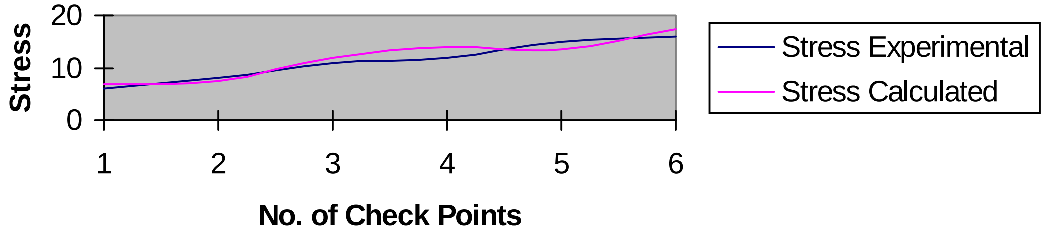


Fig. (4) Graph Showing Measured & Calculated Stresses

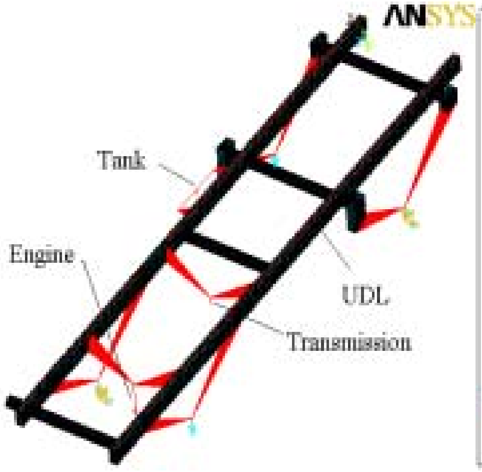


Fig. (5) Finite Element Model

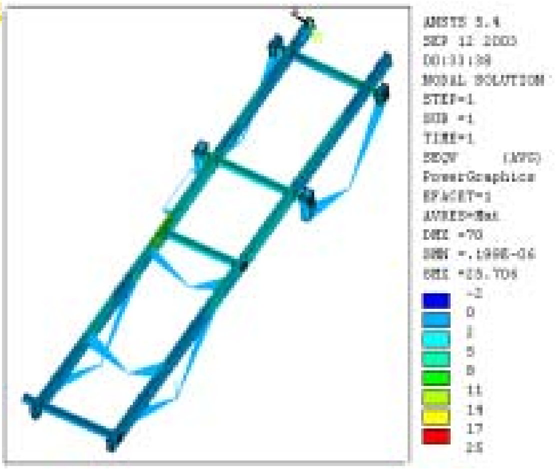


Fig. (6) Stress Validation.

4 BOUNDARY CONDITIONS & RESULTS

The punishing treatment received by the vehicle bodies in service, together with the great variety of use and abuse, means the combinations of many load cases have to be considered in finding the worst one, for the realistic use in the analysis. Before moving directly to these conditions, it will be a fair practice to visit the simplified representation of the structure, and the actual representation of the load over it. The structure can be simply assumed as beam, with uniformly distributed load due to cabin & body over its length, and the point loads at the engine, transmission and various heavily loaded accessories. It is fixed at its four wheels or suspensions.

4.1 Diagonally Opposite Wheels on Bumps

The cross members & side members are severely stressed when the diagonally opposite wheels of the vehicle are on the bump. Here, the suspensions whose wheels are on bumps are given the maximum displacement (permissible for the suspension) in the vertical direction. The non-bump wheels spring mounting locations are made constrained in the vertical direction as shown in the fig. (5). Heights of the bumps are taken such that the diagonally opposite suspensions undergo the maximum deflection. The maximum stress is coming near the fuel tank mountings, as the fuel tank will add to the rigidity due to its own section modulus. The diagonally opposite bumps will twist the chassis, due to which the rear most cross member will undergo the high torsional stresses as can be seen from the fig. (6).

4.2 Combination Load Case-I

The combination of various load cases over the roads punishes the vehicle very hard. One such condition occurs when the vehicle moves over the grade of 12%, with acceleration of ‘a’ m/sec^2 and over the road bank of 250 mm & vehicle is turning towards left. Considering ΔM_g & ΔM_a as increase in load on rear axle due to grade & acceleration . Taking moment about rear left wheel,

$$M_{rr} = \{(\Delta M_g + \Delta M_a + M_r) \times \text{Cos}\alpha \times T_l + (\Delta M_g + \Delta M_a + M_r) \times \text{Sin}\alpha \times H\}/T + \Delta F_{zrr} \quad (8)$$

Similarly weights on the other wheels can also be calculated.

4.3 Combination Load Case-II

The second load combination for which chassis should be analyzed is for the negative grade of 12%, braking retardation of 6 m/sec^2 and the road banking of 250 mm, vehicle is turning towards left too. Taking moment about front left wheel, we get,

$$M_{fr} = \{(\Delta M_g + \Delta M_{br} + M_f) \times \mathrm{Cos}\alpha \times T_l + (\Delta M_g + \Delta M_{br} + M_f) \times \mathrm{Sin}\alpha \times H\}/T + \Delta F_{zfr} \quad (9)$$

Similarly loads on the other wheels can be calculated.

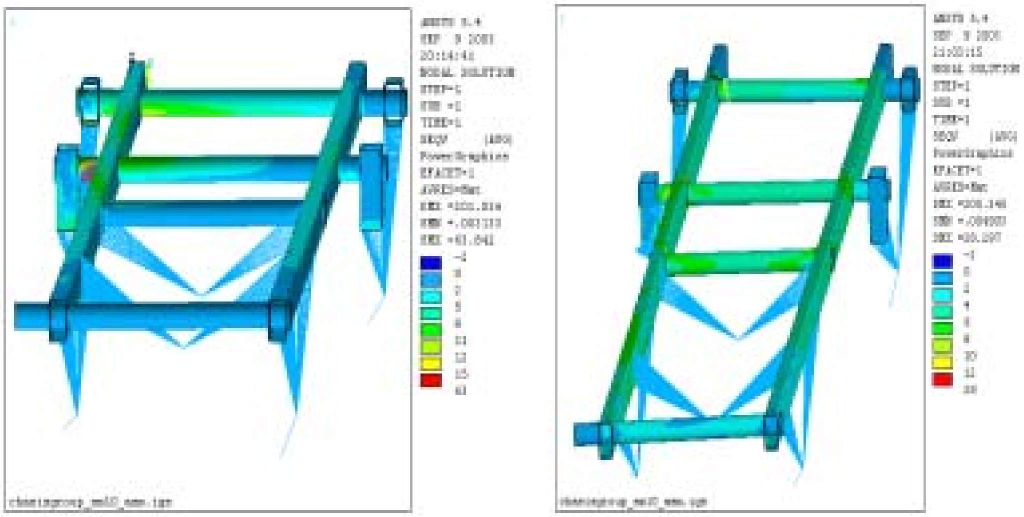


Fig. (6) Stresses for the Combination Load Case II

4.4 Road Bump Load Case

The impact forces experienced by the vehicle at the road bumps can be determined by mounting an accelerometer at the front and the rear axles such that it measures the vertical acceleration. Then the vehicle is passed over the bumps at the speed range between 10-60 Km/hr. The Eicher Motors Ltd. with the heavy vehicle 18 tones GVW has conducted few such tests. The results are expressed in the fig. (7) & stresses in fig. (8). It can be noticed that the 'g' increases to '3' times on front wheels, & '2.5' times for the rear wheels.

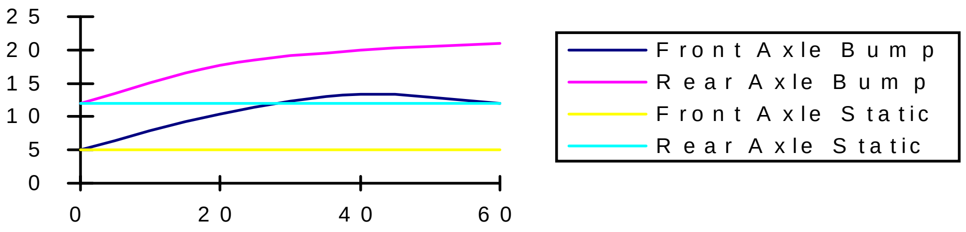


Fig. (7) Loads at Road Bumps.

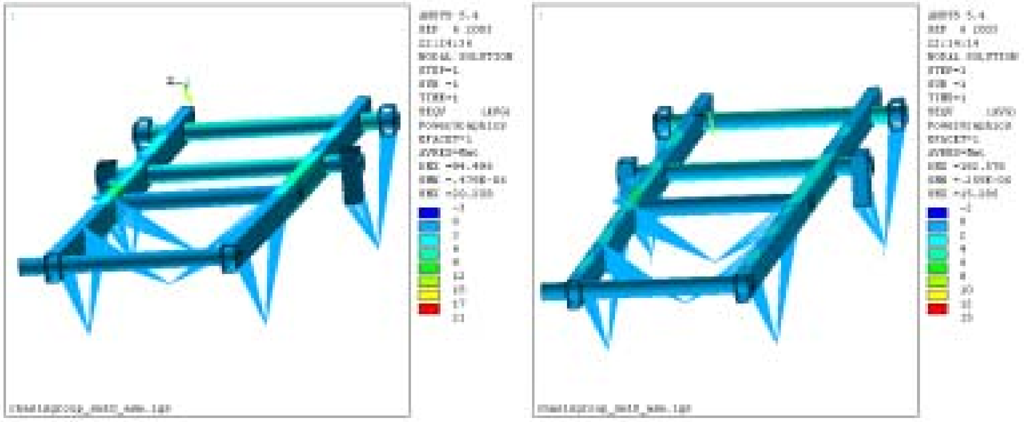


Fig. (8) Stresses at the & Front Wheel Road Bump

4.5 Power Spectrum Density Analysis

Consider the dynamic response of a bus owing to four random loadings, acting simultaneously at the front and the real tyres as shown in fig (10) .The response of a bus due to this random loadings can be studied using PSD Analysis of the chassis structure. The FFT analyzer has been used to measure the road undulations at four locations above the vehicle chassis at vehicle speed of 30 Km/hr. The measured spectrum is shown in the fig. (9). The PSD analysis has been done for the first five mode shapes only, & then final response spectrum has been taken by mode combination method. As the vehicle bears random input excitation from the road, therefore the stresses as shown in fig. (11) & displacement are most varying over the chassis structure in this case.

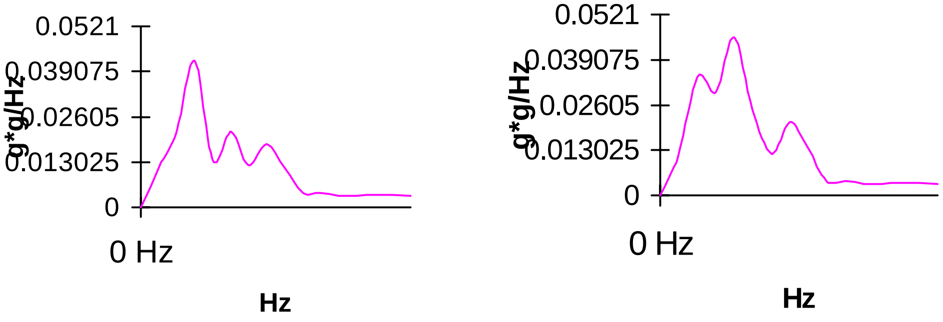


Fig. (9) PSD Over the Left & Right Wheels of the Vehicle

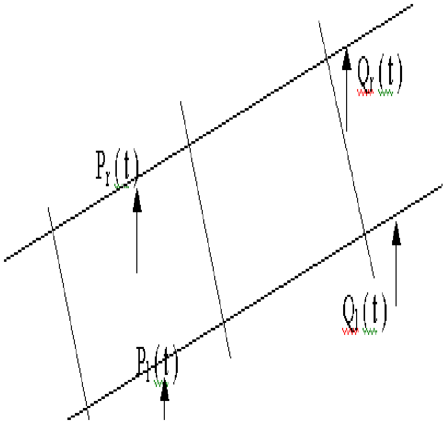
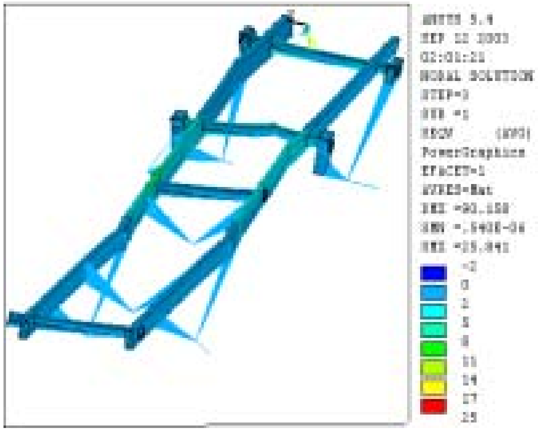


Fig. (10) PSD Excitation on Wheels



Fig, (11) PSD Stress Analysis

5. CONCLUSION

It is necessary to use the detailed model of the structure for analysis of the vehicle chassis. It becomes even more necessary, when the center of gravity of the vehicle is towards left or right of the central plane of the vehicle. Here lot of practical works has been done before finalizing the boundary conditions & load cases. Loads for various load cases have been calculated, then checked with the measured loads & then loads for load combination cases have been calculated on all the wheels.

The finite element model shows the possibilities to fulfill the requirements regarding sufficient model description as well as efficiency. The finite element model has been tested to the experimental results available for some of the cases.

The same finite element model can be used for the fatigue analysis of the chassis, if the adequate data's are available on the repetitions of the loads creating stresses on the chassis above the endurance limit of the chassis material. The same model can also be used to analyze the lateral vibrations responses.

It is impossible to cover all the conditions for the analysis of the vehicle on road conditions, however the above-mentioned boundary conditions can be used as the starting analysis for the stresses in the vehicle.

7. ACKNOWLEDGEMENT

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NOMENCLATURE

Sr. No.	Notation	Description	Unit
01	a	Acceleration	m/sec^2
02	ΔF_{zfr}	Load Transfer on Front Right Wheel (due to cornering)	Kgf
03	ΔF_{zfr}	Load Transfer on Rear Right Wheel (due to cornering)	Kgf
04	g	Acceleration due to Gravity	m/sec^2
05	H	Height of the Center of Gravity of the Vehicle From the Ground	m
07	H_f	Height of the Roll Center at the Front Axle	m
08	H_l	Height of the CG above Roll Center of the Vehicle	m
09	H_r	Height of the Roll Center of the Rear Axle	m
10	H_u	Height of the Bump	mm
11	K_s	Vertical Stiffness of the Suspension	Kgf/mm
12	K_{φ}	Roll Stiffness of the Suspension	Kgf mm/rad
13	$K_{\varphi f}$	Roll Stiffness of Front Suspension	Kgf mm/rad
14	$K_{\varphi r}$	Roll Stiffness of Rear Suspension	Kgf mm/rad
15	L	Wheel Base of the Vehicle	m
16	L_f	Distance of Center of Gravity of the Vehicle From Front Axle	m
17	L_f	Distance of Center of Gravity of the Vehicle From Front Axle	m
18	M_f	Weight on the Front Axle of the Vehicle	Kgf
19	M_r	Weight on the Rear Axle of the Vehicle	Kgf
20	M_t	Total Weight of the Vehicle	Kgf
21	M_{fl}	Weight on the Front Left Wheel	Kgf
22	M_{fl}	Weight on the Front Left Wheel	Kgf
23	M_{rl}	Weight on the Rear Left Wheel	Kgf
24	M_{rr}	Weight on the Rear Right Wheel	Kgf
25	ΔM_{br}	Change in Axle Load due to Braking	Kgf.
26	R	Radius of the turn	m
27	s	Lateral Seperation Between Suspensions	mm
28	T	Track Width of the Vehicle	m
29	V	Velocity	m/sec
30	θ	Grade Angle	Degrees
31	α	Road Bank Angle	Degrees

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