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Analysis of Vibration Characteristics of Transport Utility Vehicle by Finite Element Method

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ABSTRACT:

Present work deals with the design and analysis vibration characteristics for transport utility vehicle. The transport utility vehicle is designed using automotive industry standards. The dynamic behaviour of vehicle depends on the selection of overall dimensions, wheel base, track width, overall height and width that are decided using central motor vehicle rules. The selected dimensions for vertical and horizontal pillar members of the transport bus are modified to enhance the strength, stiffness and stability of the superstructure during travel. This increased stability enhances the ride comfort and passenger safety. Analysing the effect of utilizing manual meshing in complex areas of a transport utility vehicle for vibration analysis and passenger ride comfort has also been carried out. Modal analysis to evaluate the dynamic behaviour of transport utility vehicle model is also carried. Further with the use of finite element analysis deflection vehicle structure is evaluated. The outcomes from the analysis are compared with the behaviour of chassis mounted platform in dynamic conditions and are found in close correlation. The vehicle structure behaves as a single entity in dynamic situations, so surface model is prepared. Element selection for the finite element analysis is carried by considering plane stress condition. Two-dimensional quadrilateral shell elements are extensively used for meshing of the computer model of the vehicle structure. Complex areas in the optimised vehicle structure are meshed using relevant combination of quads & trias. The values of vector sum displacement and frequencies are found to be in good agreement with the experimental ones.

KEYWORDS:

Vehicle dynamics; Natural frequency; Displacement; Modal analysis; Transport utility vehicle

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1. Introduction

Transport utility vehicles are most common mode of day to day passenger transport. Passenger transport includes intercity and intracity travel. In many situations, the road quality, road type and the duration of travel on road, which the transport utility vehicle is expected to travel or travels is not in the control of driver. This signifies that the speed of driving is the sole parameter in the control of vehicle driver. When the transport utility vehicle along with passengers travels on rough or kuccha roads it is subjected to different levels of vibration from sprung and un-sprung masses. The lateral stability in this case is controlled by steering angle, yaw angle, wheelbase and the centre of gravity of the overall combination. During turning (right or left), lane changing, overtaking i.e. manoeuvring other vehicles the instantaneous rolling centre, instantaneous circular radius, slip angle, heading angle i.e. orientation of the vehicle and the wheelbase controls the dynamic stability. Much research on chassis and chassis mounted structures and their static, dynamic and vibration analysis has been carried through different perspectives by researchers. Vehicle response to vibration determines the direction and magnitude of displacement to which the passenger compartment is subjected. Increased levels of displacement lead to higher magnitudes of vibration, noise and passenger discomfort. Passenger ride comfort and quality for a transport utility vehicle can be better presumed when designer has thorough understanding of sources of excitation, fundamental mechanics of response of vehicle to vibration and human perception, tolerance and reaction for the vibrations. The system vibration transfer to the passenger is shown in Fig. 1.

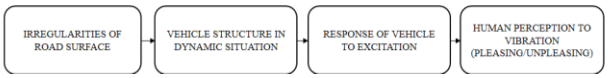

Fig. 1: Mechanics of human ride dynamic system

Irregularities of road surface include but not limited to pits, bumps, lumps, variable height of speed breaker and rumble strips etc. The displacement at the tyres due to road roughness is precepted as an acceleration input to the wheels/tyres. Magnitude of vibration transfer from un-sprung to sprung masses in the vehicle depends primarily on type of tyre, tyre-road interaction, co-efficient of friction between the tyre and road, tyre stiffness, air pressure inside the tyre. For the transport utility vehicles wherein there is a combination of longitudinal and cross members along with vehicle chassis the levels of vibration are significant. The amount or amplitude of vibrations generated during the tyre road interaction is so high that ride comfort of the passenger is affected to an extent that sometimes passenger avoids traveling through these vehicles. Various joints in transport utility vehicles ranging from temporary bolting, riveting to permanent welded joints are affected due to different vibration levels to which the vehicle is subjected. When subjected to vibrations these temporary joints become loose and generate noise, which is totally unpleasant and unwanted during the journey.

Glass window pane tend to become loose in the space provided due to continuous vibration and contribute to the noise generation to large extent however these can be minimised. Other sources of vibration include imbalance of vehicle components such as engine, driveline, short and long shafts which arises during manoeuvring as the centre of gravity of the combined structure needs to be aligned. The type and design of transport utility vehicle depends upon passenger carrying capacity as per norms specified for corresponding number of axles, wheel base and track width. In this regard, the finite element model of bus is developed and its response in terms of acceleration and stress is obtained. Various methods for determining the torsional stiffness of semi-trailer chassis subjected to torsion provide an insight to the behaviour of platform structure under uniformly distributed nature of load.

Mathematical analysis of section properties of channel sections used in the construction of sub and super structure design is carried out using elementary concepts of strength of machine elements [5]. In order to analyse vibration characteristics of the transport utility vehicles thorough understanding of the behaviour of structures in transport utility vehicles is a prerequisite. Vibration assessment includes the determination of natural frequencies and mode shapes for various speed load situations to which the transport utility vehicle is subjected [3]. The location of sprung and un-sprung masses also contributes to the vehicle vibrations significantly. In transport utility vehicles tyre stiffness, tyre inflation pressure and components below the leaf springs constitute un-sprung masses while the overall structure along with passengers is treated as a sprung mass. To evaluate the response of such system, mathematical models including spring, mass and damper system have been utilized.

In present work, a computer aided model of transport utility vehicle is prepared using the combination of automotive industry standard and central motor vehicles rules. The selection of computer modelling technique is a critical step in preparation of computer aided model of the vehicle. Transport utility vehicle is a combination of thin sheets made of mild steel, square pipes as horizontal and vertical pillars. From the available computer modelling techniques of solid modelling, surface modelling and assembly modelling the surface modelling technique is selected for present work. As the transport utility vehicle in operation functions as a single entity, only surface modelling suffices the need. The thickness of steel sheet and square pipe is also very less as compared with their own lengths i.e. 6mm thick and 1500 to 5035 mm long sheets. In such situations, the stress/strain in the normal direction of load is zero, in finite element terminology this situation is termed as plane stress condition.

Surface modelling method satisfies both the situations needed for analysis. The decision of location of vertical and horizontal pillars is finalized based on the distances mentioned in automotive industry standard AIS 052. Overall height and width are kept in constrained dimensions of 3000mm and 10050mm. The mild steel sheet and pillar combination is fitted over the chassis which generally is provided by original equipment manufacturers. The detailed terminology of the transport vehicle geometry, its components and the computer aided surface model is shown in Fig. 2.

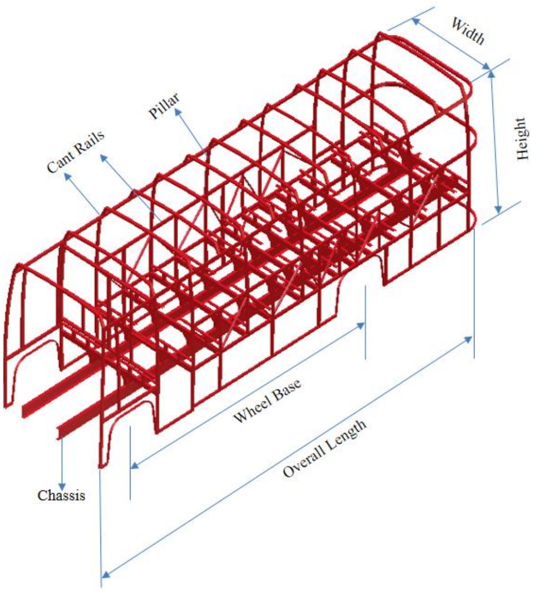

Fig. 2: Dimensional details of transport utility vehicle

2. Finite element analysis of the vehicle

For finite element analysis, the chassis, cant rails and pillars are presumed as integrated components and are treated as a single entity and hence modelled accordingly using surface modelling. This combined surface model of transport utility vehicle is evaluated for errors of the surface modelling. This surface model is then pre-processed for finite element analysis. Element selection is the foremost step in finite element analysis process. The selected element must conform to the physical situation of the loading to which the vehicle is subjected. Range of elements is available for meshing of the structure. One dimensional, two dimensional and three-dimensional elements are available in the software library. From the available elements, two-dimensional thin shell quadrilateral elements are selected for meshing of the computer aided surface model. During pre-processing the computer model is discretized by using the shell elements to simulate the plane stress condition [1]. Meshing of the vehicle structure is performed by selection of individual surfaces.

In case of complex areas proper combination of tri and quad elements is made. With the aid of manual meshing the mesh size in complex and critical areas is excellently controlled. Total number of shell quad & triangular elements after meshing with minimum element size is 146723. The element size is decided according to the overall dimensions of the structure. Element size is the vital parameter for achieving convergence over the stress values [1]. The meshed model of the platform is shown in Fig. 3. The thickness applied to the elements is 6 mm and at the lapping surfaces where the two surfaces are fitted over each other, the thickness at this section is 12mm. All the components are assumed to be made of mild steel. Hence material properties applied to element are Poisson's ratio of 0.3 and Young's modulus of 210 GPa. The aspect ratio of the elements achieved in present analysis is 98.5%, which is in accordance with the requisites of finite element procedures. Warpage of the elements in complex areas is 0.8%. Distortion of the elements is primarily due to combination of various geometries in design. The taper provided on all the lateral members is the prime factor for war page. Other element quality parameters as skew ness, Jacobian etc. are kept within acceptable limits considering the rectangular shape of the sections.

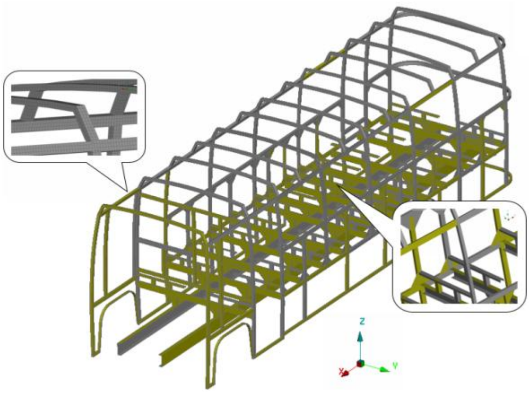

Fig. 3: Meshed model of the vehicle with modified dimensions

3. Boundary conditions, loading and vibration analysis of vehicle

During braking action, major component of load is to simulate the dynamic behaviour of transport utility vehicle under load in static and dynamic condition the longitudinal channel i.e. chassis is assumed to be rigid [1]. On this vehicle chassis components of transport utility vehicle are mounted e.g. fuel tank, battery, driver compartment, engine and mounts, front and rear axles are attached. For simulation of the rigid chassis all the degrees of freedom of the nodes on the bottom surface are constrained. This provides similar condition when the structure is mounted on automobile chassis. By constraining all the degrees of freedom, a condition of rigidity is achieved. These constraints provide a rigid base for analysis of the transport utility vehicle. When the load is applied on nodes, the utility vehicle model behaves as a single entity under the action of load at all the load locations. The boundary conditions and loads applied on the structure are shown in Fig. 4. The magnitude of loads applied depends upon the total payload carrying capacity of the transport utility vehicle under consideration. The distribution of load on the transport utility vehicle in present analysis is kept symmetric on both left and right-hand sides of the model. For applying the load on the structure, the upper nodes on the structure and outer longitudinal member are selected. Modal analysis of the transport utility vehicle is carried to evaluate frequency response i.e. natural frequencies and mode shape during the design phase itself. Each mode shape is a list of displacements at various places and in various directions. The mode shapes are unique for a structure and deflection of a structure at a specified frequency results from the excitation of more than one normal mode. Modes of vibration are functions of the whole structure under consideration. The mode shape of the transport utility vehicle describes how every point of the vehicle moves when it is excited at any point. Present finite element model is solved for deflection and frequency. The node shapes are shown in Fig. 5 to 10.

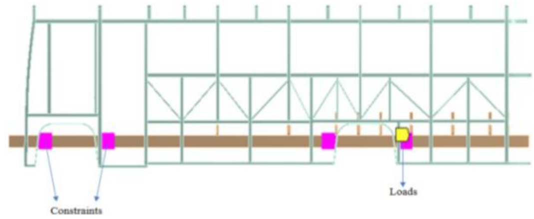

Fig. 4: Boundary conditions applied to meshed model

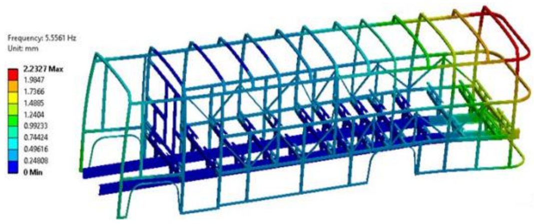

Fig. 5: First mode shape of transport utility vehicle

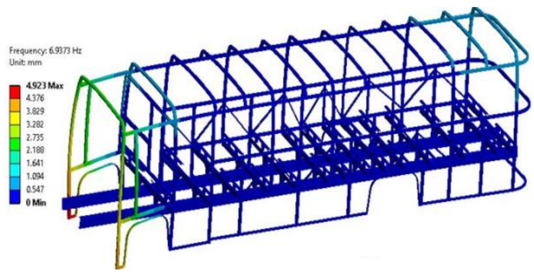

Fig. 6: Second mode shape of transport utility vehicle

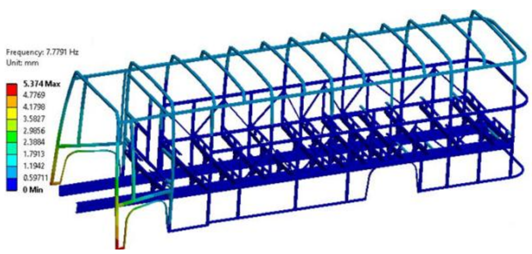

Fig. 7: Third mode shape of transport utility vehicle

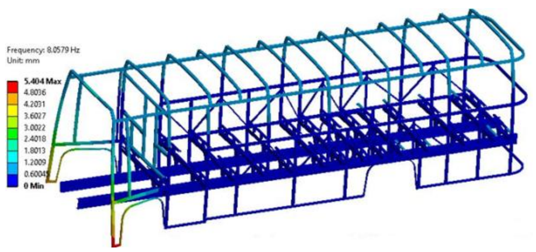

Fig. 8: Fourth mode shape of transport utility vehicle

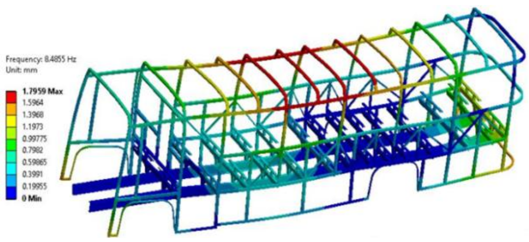

Fig. 9: Fifth mode shape of transport utility vehicle

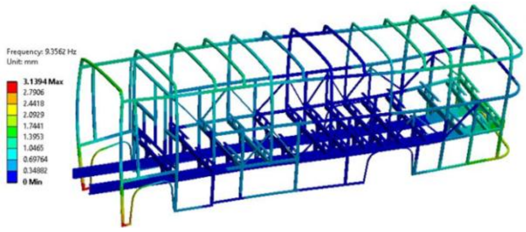

Fig. 10: Sixth mode shape of transport utility vehicle

The modal analysis is carried to evaluate frequency and deflection of the transport utility vehicle for six mode shapes and frequencies. The displacement of the transport utility vehicle along X Y and Z axes are represented in Figs. 11 to 13.

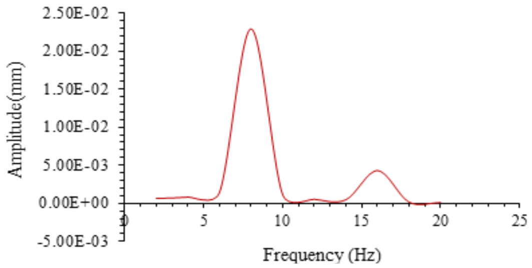

Fig. 11: Displacement of transport utility vehicle in X-direction

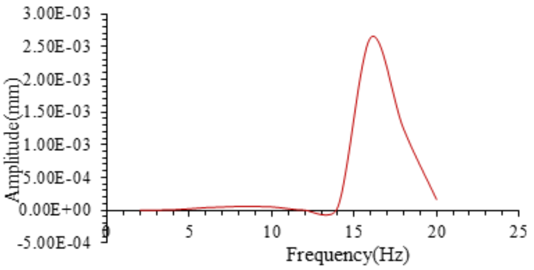

Fig. 12: Displacement of transport utility vehicle in Y-direction

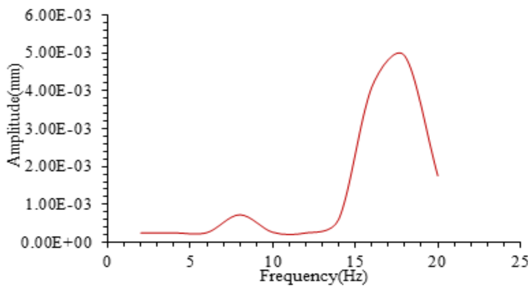

Fig. 13: Displacement of transport utility vehicle in Z-direction

4. Results and discussion

The modal analysis carried in above process has undergone through six steps. Also, the results obtained are verified for through variable element sizes i.e. convergence criterion is verified. Number of nodes and elements are finalized using the convergence concept. This leads to the optimal solution with optimum element size. The element size considered in the present analysis is 6mm (quad). The results obtained are summarized in Table 1. The displacement of the transport utility vehicle along X Y and Z axes are represented in Fig. 11 to 13.

Table 1: Summary of results obtained from modal analysis

Mode of vibration	Frequency (Hz)	Displacement (mm)	Location of deflection
1st	5.5561	2.2327	Horizontal bar and vertical pillar in rear portion
2nd	6.9373	4.9320	Vertical pillar in front portion
3rd	7.7791	5.374	Vertical pillar in front portion
4th	8.0579	5.404	Vertical pillar in front portion
5th	8.4855	1.7959	Horizontal square pipes in mid portion
6th	9.3562	3.1394	Vertical pillar in front lower portion

From the deflection plots in Figs. 5 to 10 it is observed that the deflection pattern for a specific mode of vibration in the horizontal and vertical structural members of the transport utility vehicle is uniform and symmetric [3]. The maximum deflection for six consecutive modes of vibration occurs at the front mid and rear portions. The frequency range of vibration varies from 5.5 Hz to 9.4 Hz which is in exact correlation from the experimental analysis of similar structures as seen from available literature [3]. In first mode of vibration, the deflection range is from 0.24 to 2.23mm and frequency is 5.55 Hz. The maximum deflection magnitude is observed in rear portion of the transport utility vehicle. The range of the other nodes is open in Table 2. In second, third, fourth and sixth mode of vibration, maximum deflection is observed in front portion of the transport utility vehicle.

In fifth mode, the deflection range is 0.199 to 1.79mm far less as compared with all the other modes and this deflection is observed at the exterior top portion of the transport utility vehicle. Further this deflection is observed in the mid portion with the corresponding frequency of 8.48 Hz [2]. From Fig. 11, it is observed that maximum displacement in X-direction occurs in frequency range of 5 to 10Hz which is again corresponding to the fundamental frequency of natural vibration. This indicates that the transport utility vehicle structure developed is stable to withstand the vibrations in dynamic situations. From Fig. 12 and 13, it is observed that, the peak displacement for Y and Z-directions is observed in between frequency range of 15 to 20 Hz and corresponding displacement is observed in less significant areas for vehicle design [4]. At the other locations on the transport utility vehicle the displacement varies from 0.16 mm to 1.18mm.

Table 2: Deflection range for 2nd, 3rd, 4th & 6th mode of vibration

Mode	Deflection range (mm)
2nd	0.457 to 4.93
3rd	0.597 to 5.37
4th	0.600 to 5.40
6th	0.348 to 3.13

5. Conclusion

The present design of transport utility vehicle is stable enough to withstand the level of vibration to which it is subjected. This concentrated loading is observed in all-terrain vehicles especially for general transportation and defence purpose. The overall stiffness of the transport utility vehicle is increased with the use of mild steel material and the optimal location of centre of gravity of the overall combination is achieved. The overall transfer of magnitude of vibrations from road to passengers is lowered as observed from the modal analysis. Ride comfort level and safety of the passenger is enhanced with the use of mild steel as material and present configuration.

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