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Design and Optimization of Variable Rectangular Cross Section Chassis for On-Road Heavy Vehicles

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ABSTRACT:

All the loads generated by other components of heavy vehicle are transferred to its chassis. Chassis related failures are few but the damages to the safety of occupant are huge; sometimes it leads to fatal accidents. In order to overcome this, the chassis has to be optimized based on static and dynamic loads by ensuring a uniform distribution of stress and strain. The shape and cross section of the chassis gives a resistance to the above mentioned loads. The cross section of the chassis structure of all on-road vehicles is uniform despite the variable loads. In this work, variable cross section chassis of an on-road heavy vehicle is designed by keeping optimum sections. Bending moment of the chassis has been mathematically related with section modulus of the chassis. Genetic algorithm based procedures have been used to optimize the height, width and thickness of the chassis cross section. Coding in C# language is used to automate the genetic algorithm procedures. For benchmark study, 3D models of optimized and existing chassis of an on-road heavy vehicle were developed. Finite element analysis reveals that the optimized chassis has less failure possibilities due to lower stress values and uniform distribution when compared to those from the model of existing chassis.

KEYWORDS:

Heavy vehicle chassis; Variable cross section; Optimization; Genetic algorithm

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ACRONYMS AND NOMENCLATURE

Z	Section Modulus (mm^3)
M	Bending moment (N-mm)
I	Mass moment of inertia (mm^4)
Y	Distance of the most distance point at the section from the neutral axis (mm)
F	Bending stress (N/mm^2)

1. Literature study

Chassis design for heavy vehicle applications are challenging and based on the loads acting on it. In heavy vehicles the primary load is vertical force due to vehicle pay load. To with stand this vertical load, the chassis has to resist the bending moment acting on it. All the chassis has to be designed for reliability and safety. In order to increase the reliability manufacturers may prefer more thickness chassis members. This adds more cost to the chassis and vehicle. Yilmazçoban et al [1] have taken Ford 3530 chassis for thickness optimization. Linear static analysis had carried out for 4mm, 5mm and 6mm thickness chassis with 16 ton distributed load. The study revealed that the 4mm thickness can be replaced in place of 6mm thickness chassis cross members. Kurdi et al [2] have studied the effect of cyclic loading in 36 ton GVW cargo truck with 12.35m x 2.45m chassis. Cargo load was applied as a uniformly distributed load that is equally divided over the contact surfaces of the cargo body with chassis. Sub-modeling method of finite element technique was used to get accurate results at high stress regions. Accelerometer and Dewe-soft data acquisition system are installed in two different locations to measure the acceleration of the truck. Accelerometer data were then converted as forces which were applied as cyclic loading to the sub-structure components of the chassis. The results revealed that the cargo static load has caused more stresses when compared to the cyclic load generated by road roughness.

Chinnaraj et al [3] have studied the braking and cornering cases to determine the dynamic behaviour of the heavy vehicle chassis. To find out the fully developed mean deceleration, 80% and 10% of initial velocity of the vehicle was considered. This gave the deceleration co-efficient 'g' value for braking load calculation. Cornering speed and radius were taken directly to find cornering loads. 16 strain gauges were installed in probable stress locations that have been found in linear analysis. Quasi-static numerical idealization approach was applied by using Ansys software to find out the dynamic stresses. The study has revealed that the deviation between experimental and simulation values were due to the residual stress in the chassis. Kim et al [4] have studied non-linear characteristics of multi utility vehicle frame by Virtual Proving Ground (VPG) approach. In VPG approach, except chassis and tyre, all the other components were assumed as rigid bodies including road surface where in other approaches all the components were considered as rigid bodies. Weak points were located by using graphical animation of stress distribution rather than the analysts experience and personal judgment. Using this approach, real conditions of driving can be simulated. Yoon et al [5] have developed Unified Chassis Control (UCC) system to prevent roll-over of the vehicle and improve lateral stability by integrating Active Front Steering and Electronic Stability Control. UCC have three modes of control for prevention of roll-over, manoeuvrability and lateral stability respectively. Manoeuvrability and lateral stability is improved by reducing the yaw rate error between the desired and actual based on steering input of the driver and side slip angle of the vehicle. Roll-over Index (RI) is used to detect the roll-over and indicating the risk. UCC was also evaluated by obstacle avoidance situation at high speed to ensure the roll-over resistance and lateral stability aspects. The study reveals that the UCC can reduce the steering effort, yaw rate error and rolling angle thus reduce the roll-over and improves lateral stability.

Karaoglu et al [6] have carried out the finite element analysis of truck chassis with riveted joints. The cross member and side rail member were joined by rivets with connection plate. The thickness of the side rail member were varied from 8mm to 12mm and the connection plate thickness also varied from 7mm to 10mm. Length of the connection plate varied from 390mm to 430mm to study the stress variations. Analysis revealed that the increase of the side rail member thickness can reduce the stresses but weight of the chassis may increase. To avoid this, the thickness can be increased locally. Connection plate length increase also reduces stress values. Wannenburg et al [7] presented Fatigue Equivalent Static Load (FESL) methodology after verifying in two different cases like tanker and haul dumper. FESL follows initial static analysis by finite element method to find the peak stress locations then measurement of strain values at those locations in a vehicle operational cycle using strain gauges. FESL was calculated by using equivalent stress and peak stress. Finally fatigue life was measured in finite element method with the help of S-N curve. The proposed FESL methodology yielded good results by eliminating more costly and time consuming durability test methodologies.

Thompson et al [8] have studied the NASCAR Winston cup race car chassis to understand the influence of structural members on torsion stiffness. Increasing the torsion stiffness with less weight addition and not much variation in centre of gravity, 24 different designs were taken for study. The sensitivity study was carried out by considering twist angle and rate of change of twist angle. Modified locations of the chassis members were decided from the understanding of sensitivity analysis. 1/20 scale model was made using rapid prototyping for better visibility and placement of members. Final design was selected by improved torsion stiffness with slight increase in weight. Authors [9-11] have used the genetic algorithm procedures to optimize the cross section of an on-road heavy vehicle chassis. They have used the rectangular cross section instead of open C-channel cross section. The ranges of the parameters were taken from the standard thickness of plates and historical data. C++ language programming was used to automate the genetic algorithm based optimization procedures. Applications of finite element modelling and analysis of vehicle chassis is widely reported by many researchers [12-15]. The chassis frame design must be assessed for static and dynamic stress states for applied static and moving loads expended on field [16-18].

In this research work, the height, width and thickness of chassis cross section were optimised through genetic algorithm procedures which have been applied by using the mathematical relationship with section modulus. The objective function considered for optimizing the section modulus is based on the bending moment equation. 3D models of existing and optimised chassis structures were developed and meshed for performing finite element analysis. The results from linear static, model and buckling analyses of these two chassis structures were compared.

2. Genetic algorithm based optimization

In most of the on-road vehicles the cross section of the chassis structure is uniform (see Fig. 1) despite the variable loads. The variable section chassis concept (see Fig. 2) is based on the basic principle of high and low section modulus respectively for more and less load bearing locations of the chassis. The cross section parameters P1 to P3 represents thickness, width and height respectively as shown in Fig. 3. The section modulus in terms of P1 to P3 is given by,

$$Z = \frac{(P_2 P_1^3 + P_1 P_3^3)/6 + P_2 P_1 (P_1 + P_3)^2/2}{(2P_1 + P_3)/2} \quad (1)$$

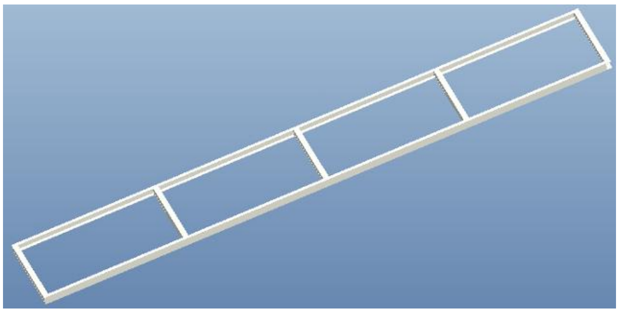

Fig. 1: Uniform cross section chassis

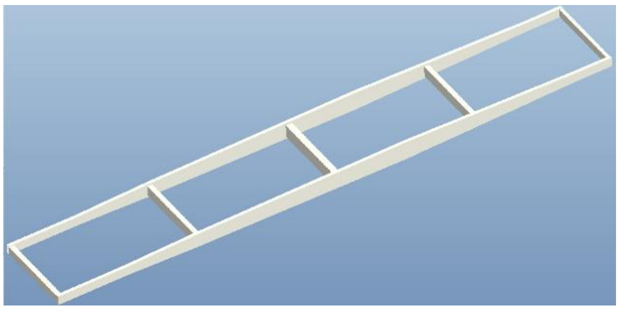

Fig. 2: Variable cross section chassis concept

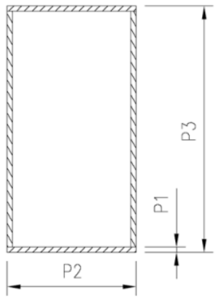

Fig. 3: Rectangular cross section

The ranges for the rectangular cross section parameters of the chassis are taken as the nearest values of the standard cross sections which are used in commercial chassis, as given in Table 1. The coding is compiled in C++ language and checked for logical and syntax errors. The parameter ranges and accuracy were assigned as an input to the computer program. Based on the mathematical model relationship with the parameters the optimum result is printed as an output file. The number of parameters, number of bits representing each parameter, number of bits in a string, initial spool size, ranges of each parameter, mathematical model and number of iterations can be changed by user based design parameters. The number of bits representing each parameter is 2, 4 & 5 for thickness, width and height respectively. The length of string and ranges has been set based on the standard thickness availability of plate materials and existing chassis width and height. A spool size of 30 is defined as initial population. The number of iterations which repeated the genetic algorithm process has been set as 1000. The optimum result that has gone through the genetic algorithm operators like initial population, reproduction, cross over and mutation. The optimum results have been chosen from the spool of results that are closest to the optimum based on the mathematical model defined in the objective function. After the first optimum result, keeping parameters P1 & P2 as the same and the parameter P3 is optimized in various locations along the length of the chassis based on the vehicle loading.

Table 1: Parameter ranges with accuracy

Parameter	Range (mm)		Accuracy (mm)
	Min	Max	
P1	6	12	2
P2	50	125	5
P3	75	230	5

3. 3D modelling

TATA 1613 Turbo truck having the gross vehicle weight of 15660 kg chassis has been taken for the comparative study. It's a ladder type chassis with riveted and bolted connections. C-channels are used to make the side rails of the chassis. The cross members are used to improve the torsion stability by joining two parallel side rail channels [18]. The two C-channel constructions at the backside where the body rest on the frame. These channels are fixed together by using "U" clamps. The front and rear axle loads are transferred to the chassis by 1450 mm and 1600 mm semi-elliptical leaf springs respectively. The other technical specifications of the model are provided in Table 2. The 3D models of the chassis are designed by using Pro-E Wildfire 5 software. The existing chassis uses C-channels and its 3D model is shown in Fig. 4. New optimized chassis uses variable rectangular cross section based on the optimum thickness, width and height obtained from genetic algorithm at various locations. The LH and RH side rails are made through welding of the top and bottom plates. The cross members' locations are kept as the same. The 3D model of optimised chassis is shown in Fig. 5.

Table 2: Technical specifications of the vehicle

Major component	Details
Chassis	Ladder type heavy duty frame
Engine	CUMMINS 6BT 5.9 TC
Gear box	TATA GBS-40
Rear axle	TATA RA - 108RR
Front axle	Heavy duty forged I beam, reverse Elliot type
Service brake	Full air S-CAM
Parking brake	Spring actuated
Suspension	Semi-elliptical leaf spring
Wheel	Tyre: 10x20" - 16 PR, RIM: 7x20"
Fuel tank	225 litres
Cab/cowl	All steel semi-forward control
Wheelbase	3625 mm
Track front and rear	1933 mm & 1809 mm
Overall length	6100 mm
Front overhang	1185 mm
Min. turning circle radius	7250 mm
Max. front axle weight	5460 kg
Max. rear axle weight	10200 kg
Max. permissible GVW	15660 kg

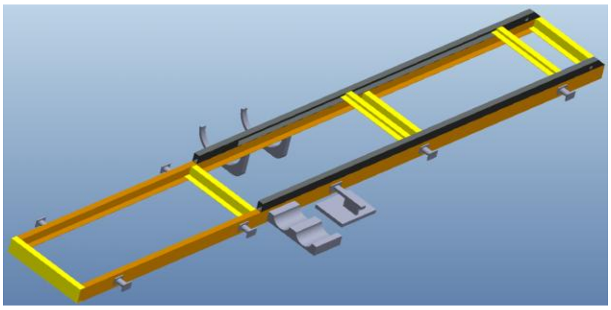

Fig. 4: 3D model of existing chassis

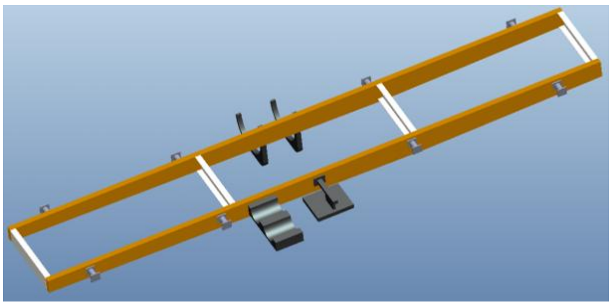

Fig. 5: 3D model of optimized chassis

4. Finite element simulation and results

Linear static analyses of existing and optimized chassis are carried out separately. Preceding the finite element meshing, the 3D models have been cleaned by removing free edges, small fillets, and small radius and datum planes to ensure proper geometry for meshing. The 3D models are converted in to IGS format to carryout meshing. The continuity of the finite element model is ensured by connecting all the elements for uniform distribution of loads and stresses. The front and rear leaf spring mounting locations were fixed and other loads are applied at corresponding locations. After the chassis structures are analyzed, the stress plots are made for comparative study. Figs. 6 to 8 respectively show the finite element model, its boundary conditions and VonMises stress plots from the linear static analyses of existing and optimised chassis structures. From the results, the peak displacement and stress values are summarised in Table 3. From the results, it is well clear that the peak stress and peak displacement of the optimised chassis are significantly reduced when compared to those from existing chassis.

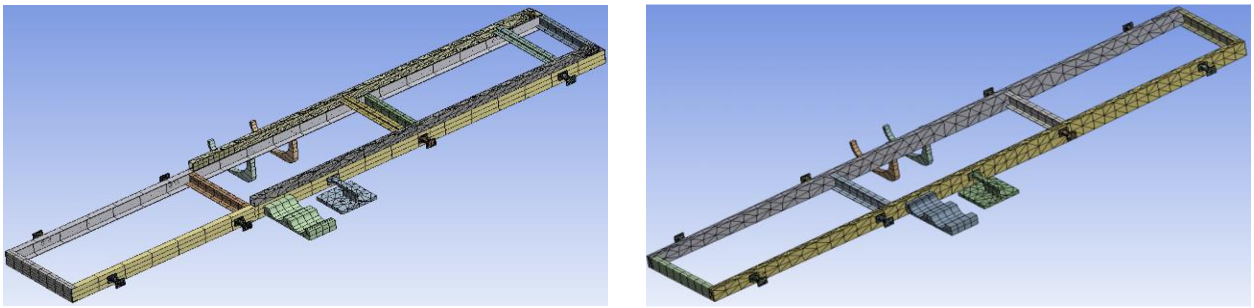

Fig. 6: Finite element model of existing (Left) and optimised (Right) chassis structures

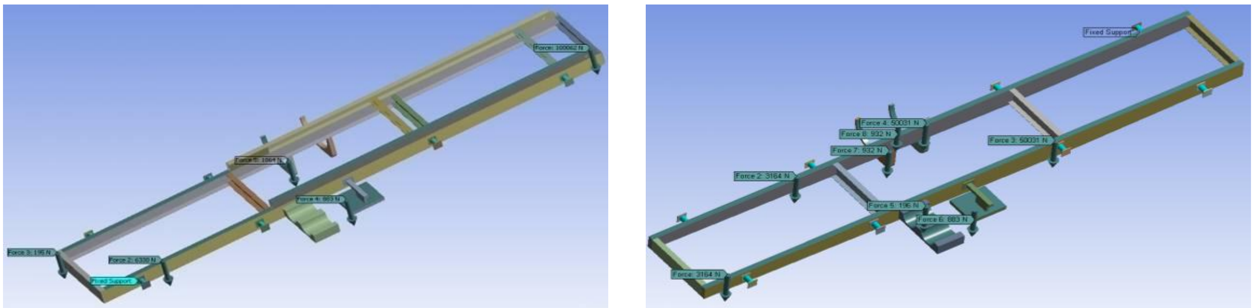

Fig. 7: Boundary conditions for existing (Left) and optimised (Right) chassis finite element models

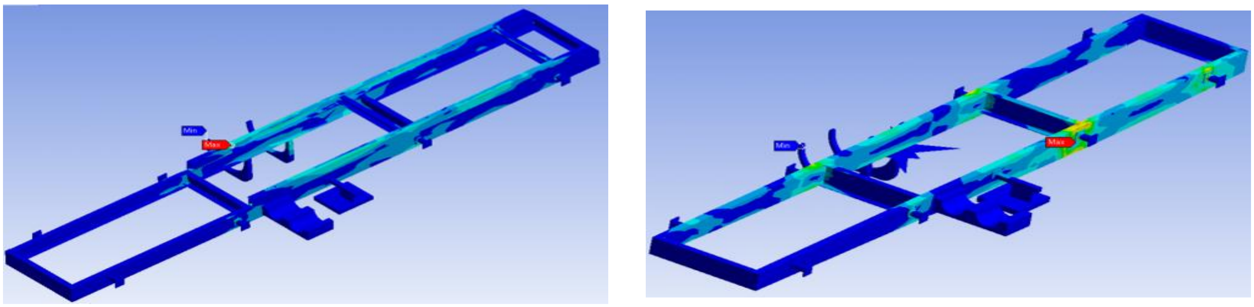

Fig. 8: VonMises stress plots from existing (Left) and optimised (Right) chassis finite element analysis

Table 3: Static analysis results comparison

Results	Chassis and Location	Body rest location	Rear suspension mounting	Area between rear suspension mounting
VonMises Stress (MPa)	Existing	82.85	46.02	36.82
	Optimized	28.52	15.85	12.68
Max. Shear Stress (MPa)	Existing	43.73	24.36	19.46
	Optimized	15.72	8.74	6.99
Displacement (mm)	Existing	-	-	1.60
	Optimized	-	-	0.64

The finite element models of the existing and optimised chassis were run for modal analysis. First 16 modes have been requested in the analysis run. Ignoring first 6 rigid body modes, the next 10 modal frequencies of the existing and optimised chassis are summarised in Table 4. The bending and torsion mode shapes of the existing and optimised chassis are shown in Fig. 9 and Fig. 10 respectively.

Table 4: Modal frequencies comparison

Mode No	Chassis modal frequency (Hz) Existing	Optimized
7	5.7897	17.453
8	15.061	24.971
9	18.533	27.967
10	25.211	29.226
11	30.169	36.409
12	34.399	44.495
13	36.184	48.889
14	37.857	49.971
15	42.828	50.202
16	45.566	51.981

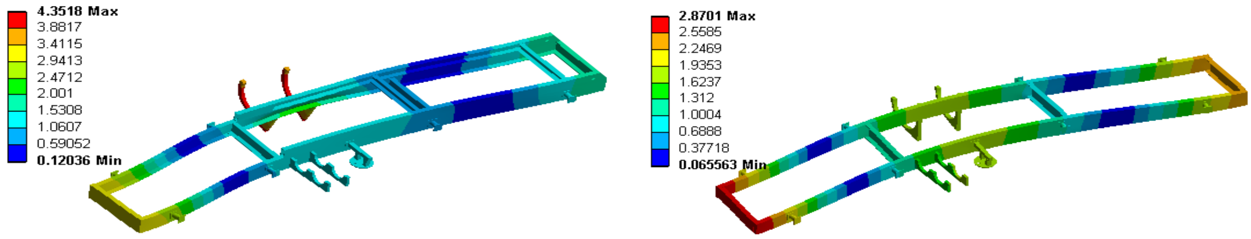

Fig. 9: Bending mode shape of existing (Left) and optimised (Right) chassis structures

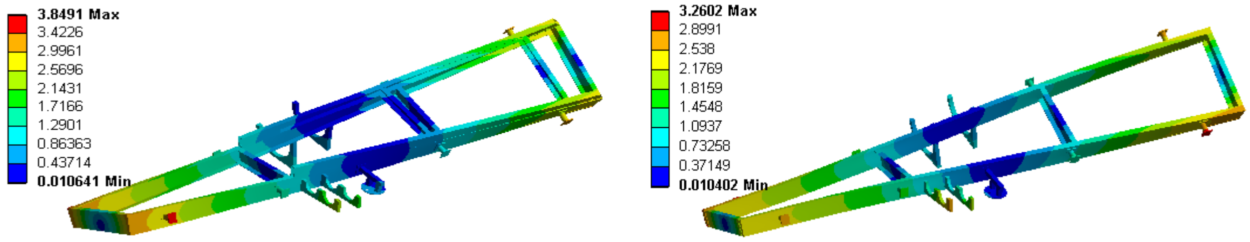

Fig. 10: Torsion mode shape of existing (Left) and optimised (Right) chassis finite element models

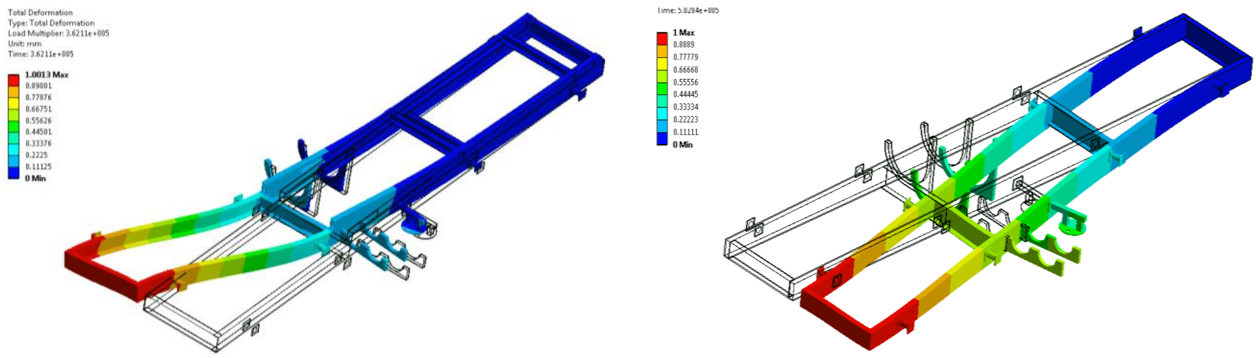

Fig. 11: First buckling mode shape of existing (Left) and optimised (Right) chassis finite element models

Linear buckling analysis of existing and optimized chassis is undertaken using fixed boundary conditions at the rear surface and a unit compressive load is applied at the front surface. The critical buckling load factor for this unit load is then calculated as 362110 and 502837 for the first mode (see Fig. 11) of the existing and optimised chassis respectively.

5. Conclusions

In this work, heavy vehicle chassis is optimized based on section modulus by applying genetic algorithm procedures. The height, width and thickness of rectangular cross section were optimized at 7 different locations along the length of the chassis. The program developed in C# was used to automate the genetic algorithm process. 3D models of the existing ladder type chassis and new optimized variable rectangular cross section chassis were built. The linear static analysis is carried out in ANSYS software. The results revealed that the existing ladder chassis encounter the more stresses (VonMises = 82.8 MPa) at the body rest location due to sudden variation in sections created by fixing the two C-channels one over the above. The new optimized chassis transfers the load and stresses uniformly to the entire structure due to its optimum section modulus. The max stress encountered by optimised chassis (VonMises = 28.5 MPa) also lesser than the existing chassis. The modal analysis results revealed that the bending and torsion stiffness got improved in optimized chassis over existing chassis. The linear buckling analysis revealed that the critical load factor was also improved by 39% in optimized chassis over the existing one. Due to the reduced number of components, permanent welding will reduce the failure possibilities and preferred for mass production. Future scope of work will include the dynamic analysis of the chassis like cornering, acceleration and braking and fatigue analysis to ensure safety by avoiding weaker sections.

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