Structural Analysis of Front axle beam of a Light Commercial Vehicle (LCV)

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Abstract **—There has been exhaustive efforts to develop the front axle design by studying the noise and vibration analysis at static and dynamic loading conditions. The model selected is that of a light commercial vehicle (LCV) which has a gross vehicle load of around 5-10 tons. The front basically is of drop forged steel type depending upon the extent of total load the LCV experiences. The collapse of LCV axle (front) while dynamic and static loading conditions is of huge apprehension to both goods and human lives, hence it becomes essential to scrutinize the structural integrity of the axle to endure characteristic such loading which can build up stresses in the same being consequential to fracture and finally failure. Stressed regions due to vehicle static load, braking torque, and during turning is established and the front axle beam is investigated to find out its factor of safety and maximum deformation under the mentioned conditions. The present work aims to determine the load capacity of the front rigid axle of a LCV and determine its behaviour at static and dynamic conditions. This paper analysis the static, transient and modal analysis of the front axle beam. The geometry of axle is created in Pro-E WildFire5.0 software which is imported to ANSYS14.5. A fine congregate finite element model (meshed) is generated using the software to assess the strength and capability of the product to survive against all forces and vibrations.**

Keywords – *Front Axle, static, modal, dynamic analysis.*

I. INTRODUCTION

The front dead axle supports the weight of front part of the vehicle, facilitates steering, absorbs shocks which are transmitted due to road surface irregularities and also absorbs torque applied on it due to braking of vehicle. Front axle is made of I-section in the middle portion and circular or elliptical section or I section at the ends. The special x-section of the axle makes it able to withstand bending loads due to weight of the vehicle and torque applied due to braking. A typical LCV front axle consists of main beam, stub axle, and swivel pin or kingpin. The wheels are mounted on stub axles. The front axle beam is subjected to bending loads due to vertical forces due to mass present above in static condition of the vehicle, while driving a truck around a corner results in multiple forces such as twisting forces on kingpin or steering knuckle, axial forces between Pad and spring interface along the length of the beam and unsymmetrical vertical loads due to centrifugal action. Worst situation arises while a cornering truck is braked to stop giving rise to turning moment on Pad and a retarding force acting on the surface of the Pad in the sense of vehicle motion. Current article carries on the numerical simulation towards the front axle to understand comprehensively the automobile front axle's stress, the strain distribution and the vibration frequency under different varieties of conditions, and provides the scientific theoretical base for the designer, thus improves design quality, shortens design cycle and reduces design cost.

II. LITERATURE REVIEW

Various experiments and numerical methods were adopted by Leon et al. to obtain the stress analysis of a frontal truck axle beam. The results obtained by finite element method were verified experimentally using photo stress.

Based on an experimental and numerical analysis of a tractor's front axle carried out by Mahanty et al, redesign was carried out for the front axle for weight optimization and easy manufacturability. Five different models were proposed based on ease of manufacturing and weight reduction. The results obtained by finite element method were analyzed by thirteen different certification test load conditions.

Another survey done by Topac et al. 9 deals with a premature failure that occurs prior to the expected load cycles during the vertical fatigue tests of a truck rear axle housing prototype. In these tests, crack mainly originated from the same region on test samples.

III. METHODOLOGY

During the front axle modeling process, according to the front axle structural characteristics and the subsequent mesh divide ease, it guarantees the front axle's structure characteristic as well as convenience following finite element analysis, and carries on the partial simplification to the front axle structure, thus establishes front axle's finite element computation shell model. Based on the front axle structural characteristics and bear loading conditions to make appropriate assumption, simplify the front axle entity model into reasonable mechanical model, and choice five static analysis conditions, namely the front axle static full load,
Impact load, Emergency braking, Maximum side force and corner sideslip to make loading analysis, and set the loading analysis results as finite element modal's loaded load, and come within the linear elastic to calculate respectively the front axle's tensile strength and yield strength in these five conditions. Moreover, it uses the first and fourth strength theory to collate respectively front axle's tensile strength and yields strength in a variety of conditions, and with rigidity standard to check its stiffness. Meanwhile, combine the finite element Modal Analysis method to conduct Front axle's

Modal Analysis. And through Block Lanczos to extract the front 6 stages natural frequency and mode of vibration of front axle to analyze the coupling conditions among the natural frequency, external stimulation, human resonance frequency and mode of vibration in each stage, to provide the theory reference for the front axle structure system's vibration characteristic analysis, the vibration error diagnosis and forecast and automobile's structural dynamic performance optimization design.

IV. MATERIAL DESCRIPTION

The front axle is made of structural steel which has the properties as mentioned below:

Poisson's Ratio: 0.3
Young's Modulus: 210 GPa
Yield Stress: 250 MPa
Allowable shear stress: 600 MPa
Tensile Ultimate Strength: 460 MPa
Density: 7850 kg/m3

V. FRONT AXLE MODEL DESCRIPTION

By solid modeling technique of PRO/E, the complex forging curved surface and a 3D solid model of front axle beam is established. The shape of the front axle beam is based on an I-beam and as it is forged, it is practical to stay with the I-beam shape. The forging tools would press upon the I-beam from one side each. Holes for interfaces are thereafter created by machining. The front axle beam has defined interfaces in order to support a high level of modularization.

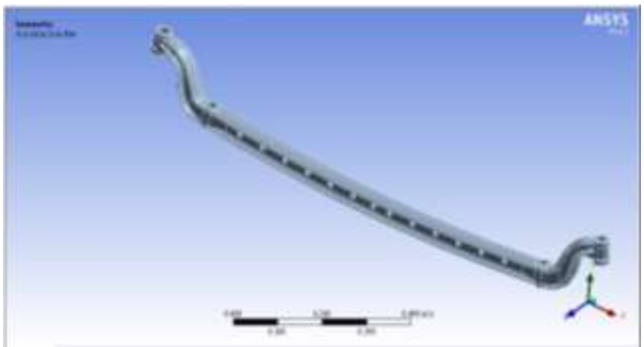


Fig. 1 Solid front axle beam

There are two main interfaces covered in this article, the kingpin interface and the spring seat, which can be seen in figure 1. The kingpin interface connects the beam to the spindle/stub axle, and consists of two contact surfaces for tapered roller bearing raceways. The spring seat consists of a contact surface and single hole to connect the hooks which hold the leaf spring in place.

VI. SOFTWARE SIMULATION OF STATIC AND DYNAMIC MOTION OF FRONT AXLE

A simplified geometry model of front axle is imported into ANSYS and then the ideal analytical model is established by the powerful finite element modeling technique of ANSYS software. The chosen problem is considered as 3-D solid model as shown in Fig1. The current model was meshed using SOLID45 and SOLID92 Tetrahedral elements available in ANSYS. Finite Element model (MESH) of the front axle beam is shown in Fig 2. The model consists of 69009 nodes and 38998 elements. Appropriate boundary conditions are incorporated in the analysis. The solid is defined by ten nodes having three degrees of freedom (UX, UY and UZ) at each node translations in the nodal x, y and z directions. The element has Plasticity, Creep, Swelling, Elasticity, Stress stiffening, large deflection, large strain, Adaptive descent, Initial stress import capabilities.

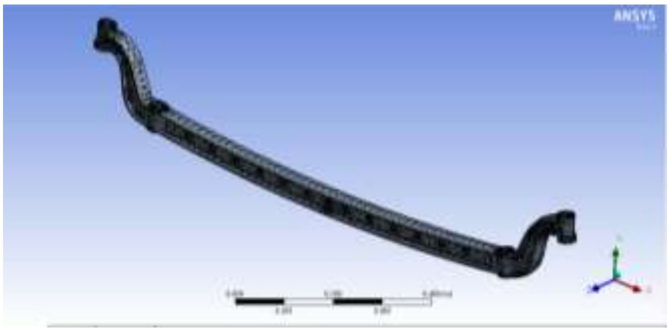


Fig. 2 Finite Element model of front axle beam (Mesh)

A. Stress analysis of front axle under static loading:

Light commercial vehicles have a medium capacity of 5 tonne to 10 tonne of gross vehicle weight. In this case of rigid front axle, boundary conditions for static loading is given by arresting all DOFs of king pin hole for any kind of movement, the beam therefore acts as a simply supported beam supported near its ends and loaded at intermediate points - the spring seats. While the axle itself represents distributed load, we will be justified in considering this also at concentrated at the spring seats, as this simplifies the calculations, and the resulting error is insignificant. Static Analysis of front axle housing made up with Structural steel is performed the analytical results are obtained. A simplified diagram of all kinds of load is shown (Fig. 3). The magnitudes of these forces were determined by experimental and theoretical methods.

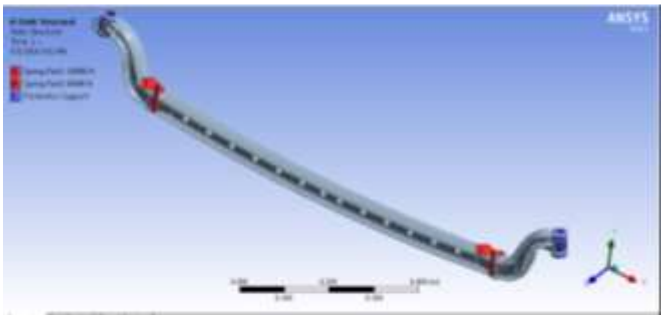


Fig. 3 Boundary conditions for static loading

B. Stress analysis of front axle under dynamic loading:

When a dynamical system is excited by a suddenly applied non-periodic excitation, the response to such excitation is called transient response, since steady state oscillations are generally not produced. In transient analysis, front axle of

combine is considered as a beam that was subjected to two dynamic right and left bodyworks loads (Fig. 4).
The beam was modeled with a 3-dimensional element, some geometrical attributes of cross-section area of axle is used as real constants of this element. After this step, the dynamic loads were applied on the model. A dynamic loading condition consists of two load steps.

In the first load step, the force increased during the rise time, the value of force at the end of the first step, reached the maximum value. During the second load step, value of load remained constant at maximum value. Rise time and maximum value of two dynamic loads applied on the axle were calculated by considering some assumptions.
In dynamic loading conditions, the vertical acceleration of lumped mass of the vehicle body due to the road surface roughness can be six times as much as the acceleration of gravity. This means that the maximum dynamic loads can be increased six times as much as the corresponding loads in static loading conditions.

This issue approximately verifies the mentioned assumptions and calculations for obtaining the rise time and maximum value of dynamic loads applied on the front axle. In this study to increase the design factor of safety, the maximum value of dynamic load is considered six times as much as the applied static load. Subsequently, the defined boundary conditions applied to the model were analyzed using post processing. The results of analysis were used to identify value of maximum induced stress.

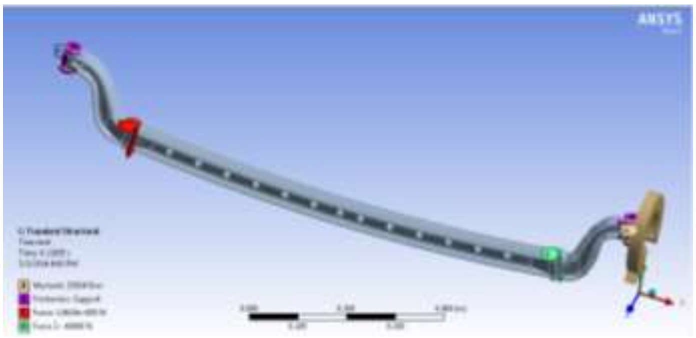


Fig. 4 Boundary conditions for dynamic loading

C. Modal analysis of front axle:

The eligibility of the static strength of front axle cannot prove that it will never break. In reality, the front axle is loaded with kinds of stimulations, which result in breakages such as resonance, fatigue etc. It is very significant to analyze dynamic characteristic for the chosen modified design of front axle. Modal analysis is used to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed.

For performing modal analysis, finite element model of optimized front axle was used. The applied boundary condition in this analysis was as similar as boundary conditions applied on the static analysis. After obtaining the solution, the results of analysis can be used to determine the natural frequencies and their appropriate mode shapes.

Modal analysis of front axle is performed by Block Lanczos method. Because of the small difference of mass and stiffness between the original and the modified structure, there is small difference of the natural frequency.

The first modal shape is the bending vibration in the horizontal plane and the second modal shape is the bending vibration in the vertical plane. The dynamic characteristics are much better because the natural frequencies are all bigger than the excitation frequency range of 0.33 ~ 28.3Hz from the ground level. Harmonic response is also obtained by providing the same boundary conditions for modal analysis. Fig 5 shows the above mentioned conditions.

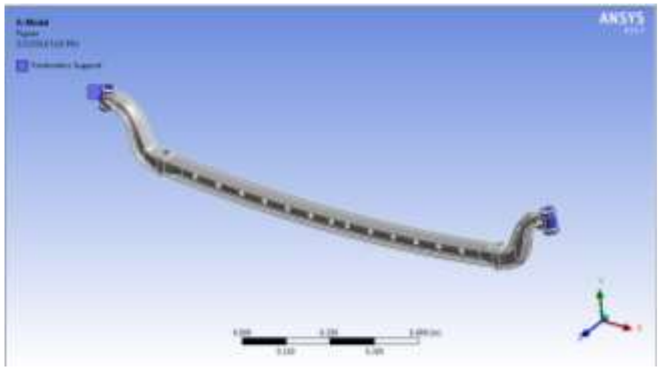


Fig. 5 Boundary conditions for modal analysis

VII. Results and Discussions

The equivalent Von Mises stress distribution of the front axle combine under static loading is provided from the finite element analysis. The maximum Von Mises stress appears on the upper part and near to the right and left spring seats.
Calculated value of factor of safety is very low and obviously this value decreases under cyclic loading conditions of field operation. The analysis results show that front axle can basically meet performance requirements, but in the state of the front axle design of the emergency braking condition, it may cause the rupture of front axle and coil spring sleeve welded region because of inadequate tensile and yield strength, or result in other serious faults such as fail to change direction. And the front axles will easy to have coupling with stimulation frequency of rough road and it also maybe couple with the body's 1^{st} and 2^{nd} resonant frequency, and then cause resonance and thus reduce front axle's fatigue endurance and affect normal driving and driving safety. Therefore, we should improve the front axle in order to use in the modified vehicles. The finite element analysis method and process towards this kind of commercial vehicle in the paper can be used to provide a theoretical reference on how to improve the structure of the front axle. In addition, the analysis method and process are also applicable to analyze and design other commercial vehicles or cars. According to the natural frequency values and excitation frequency range, it was concluded that the natural frequency values were much higher than those of the excitation frequency. Therefore, the

condition of resonance was not encountered. The first six mode shapes of the front axle are given in Fig.

A. Static Analysis –

Maximum deformation: 0.28897 mm
Maximum Von-mises Stress: 319.46 MPa
Maximum Strain: 1.8789e-003
Maximum Shear Stress: 136 MPa

TABLE I
MODAL ANALYSIS

B. Transient Analysis –

Mode	Frequency[Hz]
1.	77.285
2.	204.97
3.	286.78
4.	333.06
5.	355.17
6.	512.84

Maximum deformation: 0.27571 mm
Maximum Von-mises Stress: 351.09 MPa
Maximum Strain: 2.1697e-003
Maximum Shear Stress: 145.65 MPa

The results of the transient analysis showed that the maximum Von-Mises stress in the optimized front axle was 351 MPa. The value of maximum bending stress due to transient loading conditions was lower than the yield stress strength. Because the transient dynamic loads were applied on the axle suddenly, a permanent deformation could not take place. Therefore, it is concluded that the optimized front axle of combine has enough strength under transient loading conditions. The designed front axle is strong enough to be installed on the LCV and the optimized front axle has enough strength under static, harmonic and transient loading conditions.

Results and figures:

A. Static Loading –

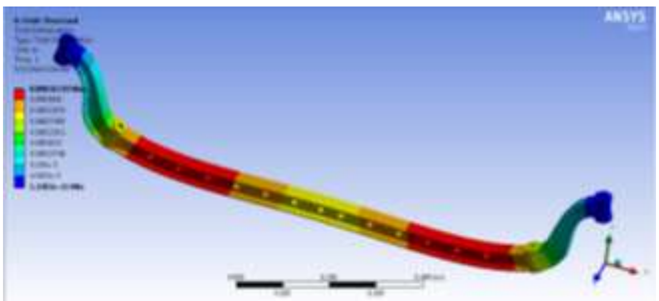


Fig. 6 Deformation of the front axle beam

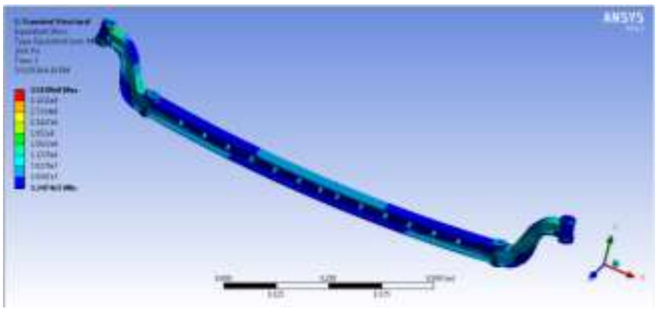


Fig. 7 Von-Mises Stress of the front axle beam

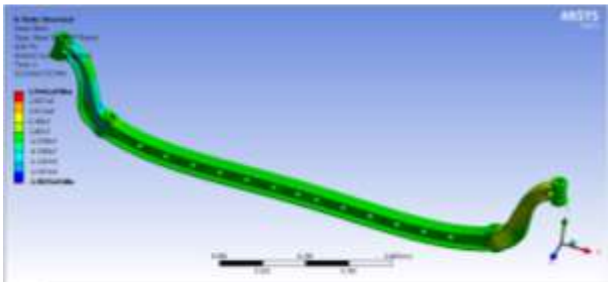


Fig. 8 Shear Stress of the front axle beam

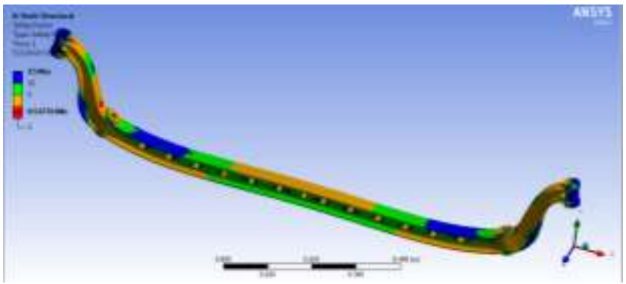


Fig. 9 Factor of safety

B. Transient Loading –

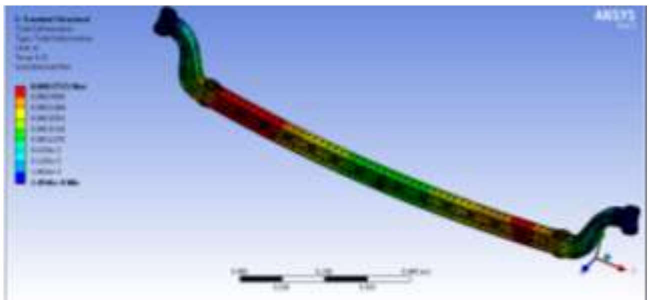


Fig.10 Deformation of the front axle beam

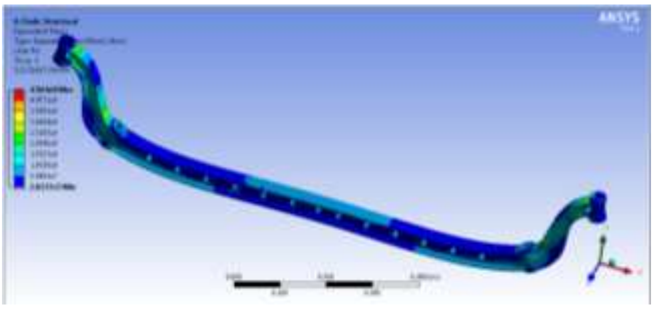


Fig. 11 Von-Mises Stress of the front axle beam

C. Modal Analysis (Mode Shapes of Front Axle) –

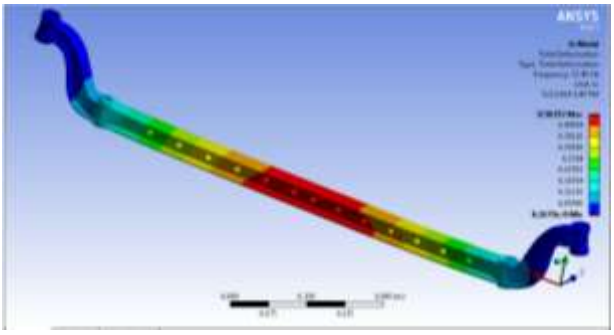


Fig. 12 Mode shape 1

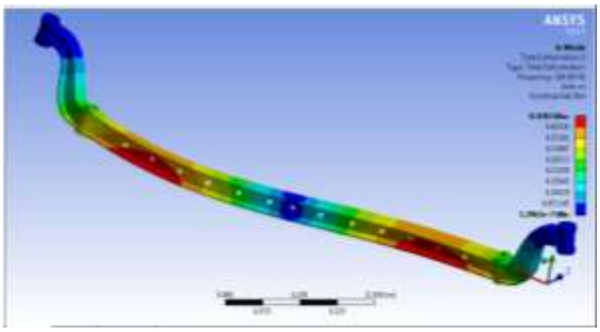


Fig. 13 Mode shape 2

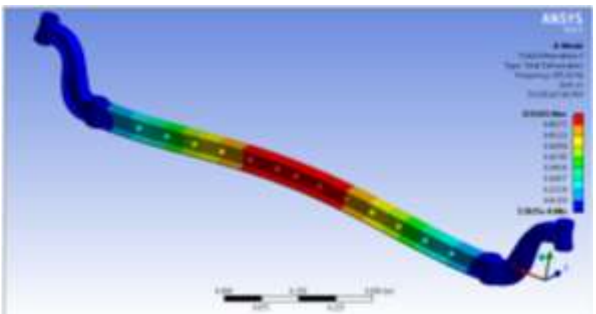


Fig. 14 Mode shape 3

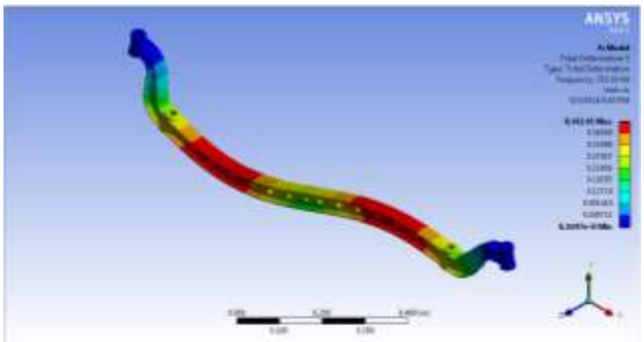


Fig. 15 Mode shape 4

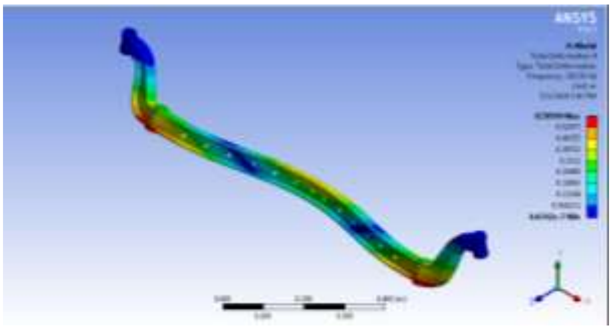
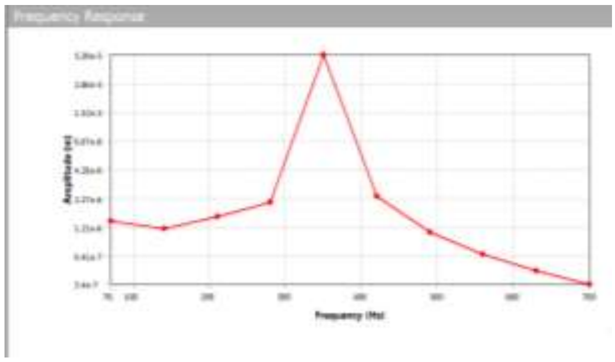
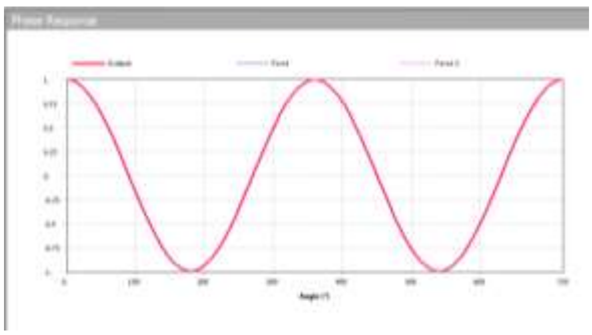


Fig. 16 Mode shape 5

D. Harmonic Response –





VIII. CONCLUSIONS

The static strength and dynamic characteristics are analyzed by ANSYS software. Analytical results can confirm the modified design and help to avoid expensive and time-consuming development loops and also allow the number of high-cost test carriers to be substantially reduced, so design periods is shortened. In the meantime it can be used to determine testing positions of the load spectrum and examine testing results. On the premise of the load spectrum, fatigue strength analysis and optimization design can be carried out to lighten the weight of front axle finally.

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