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OPTIMIZATION OF TRUCK CHASSIS OF SUPPORT STIFFNESS TO IMPROVE THE FUNDAMENTAL NATURAL FREQUENCY

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ABSTRACT*: *To optimize the centre thickness and the number of stiffener locations to Maximize the Natural Frequency of Truck Chassis, keeping the Strength, Torsion, Stiffness and Weight with in a limit.

I. INTRODUCTION

Truck chassis is a major component in a vehicle system. This work involved static and dynamics analysis to determine key characteristics of a truck chassis. The Static characteristics include identifying location of High Stress Area and determining the Torsion Stiffness of the chassis.

The Dynamic characteristics of truck chassis such as the Natural Frequency and Mode shape were determined by using Finite Element Method.

II. CHASSIS FRAME

Automotive chassis is a skeletal frame on which various mechanical parts like engine, tires, axle assemblies, brakes, steering etc. are bolted. The chassis is considered to be the most significant component of an automobile. It is the most crucial element that gives strength and stability to the vehicle under different conditions. Automobile frames provide strength and flexibility to the automobile. The backbone of any automobile, it is the supporting frame to which the body of an engine, axle assemblies are affixed. Tie bars, that are essential parts of automotive frames, are fasteners that bind different auto parts together.

Automotive frames are basically manufactured from steel. Aluminium is another raw material that has increasingly become popular for manufacturing these auto frames. In an automobile, front frame is a set of metal parts that forms the framework which also supports the front wheels. It provides strength needed for supporting vehicular components and payload placed upon it. Automotive chassis is considered to be one of the significant structures of an automobile. It is usually made of a steel frame, which holds the body and motor of an automotive vehicle.

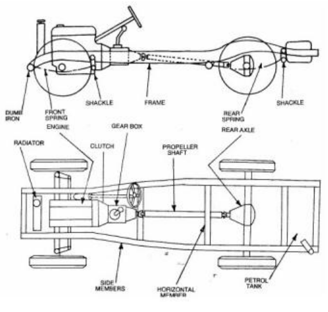

Figure 2.1: chassis frame

Since chassis frame forms the backbone of a heavy vehicle, its principal function is to safely carry the maximum load for all designed operating conditions. It must also absorb engine and driveline torque, endure shock loading and accommodate twisting on uneven road surfaces. To achieve a satisfactory performance, the construction of a heavy vehicle chassis is the result of careful design and rigorous testing.

Consequently, a modification to the chassis frame should only be attempted after consultation with the vehicle manufacturer or engineer experienced in commercial vehicle chassis modifications to ensure that the proposed modification will not be detrimental to the vehicle’s safety or performance. Because various manufacturers have individual design concepts and different methods of achieving the desired performance standards for the complete chassis, not all chassis components are interchangeable between various makes and models of vehicles. Due to the complexity of the variation in chassis design and because the major application of this section is for medium to heavy goods vehicles, the information supplied is orientated to suit the type of chassis used on these vehicles.

It should be noted that this ‘ladder’ type of frame construction is designed to offer good downward support for the body and payload and at the same time provide torsional flexibility, mainly in the region between the gearbox cross member and the cross member ahead of the rear suspension. This chassis flexing is necessary because a rigid frame is more likely to fail than a flexible one that can ‘weave’ when the vehicle is exposed to arduous conditions. A torsionally flexible frame also has the advantage of decreasing the suspension loading when the vehicle is on uneven surfaces.

2.1 Types of chassis frame
The different types of automobile chassis include:

Ladder frame chassis

The history of the ladder frame chassis dates back to the times of the horse drawn carriage. It was used for the construction of ‘body on chassis’ vehicles, which meant a separately constructed body was mounted on a rolling chassis. The chassis consisted of two parallel beams mounted down each side of the car where the front and rear axles were leaf sprung beam axles. The beams were mainly channelled sections with lateral cross members, hence the name. The main factor influencing the design was resistance to bending but there was no consideration of torsion stiffness. A ladder frame acts as a grillage structure with the beams resisting the shear forces and bending loads. To increase the torsion stiffness of the ladder chassis cruciform bracing was added in the 1930’s. The torque in the chassis was retained by placing the cruciform members in bending, although the connections between the beams and the cruciform must be rigid. Ladder frames were used in car construction until the 1950’s but in racing only until the mid 1930’s. A typical ladder frame shown as below in Figure

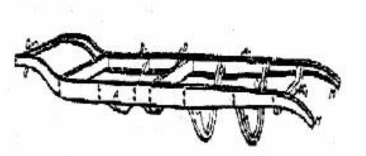

Figure 2.2: Ladder frame chassis

2.2 Natural frequency

All structures that are in a state of stable, static equilibrium have a definite configuration (shape). If such a structure that is at rest were subject to a force or disturbance, that configuration would change. If the structure is then released by removing the force, it would tend to return to its original equilibrium configuration. The following example illustrates this point.

A ball resting (let us say ‘comfortably’) at the bottom (A) of a concave surface is in a state of stable, static equilibrium. If it is displaced to a point A’ and released, it would move A’ A’’ towards A. It would approach A with a velocity, and therefore it would pass A this point and move further on the other Figure 1.1.1 Stable equilibrium side till it comes to a momentary halt at point A’’. If there are no frictional forces or air resistance. A’ and A’’ would be at the same level, and the ball would again move in the opposite direction, passing A to reach point A’ again. The ball will always have an acceleration towards the equilibrium state A. In the absence of friction or air resistance, this vibratory motion can go on indefinitely, the ball moving between A’ and A’’. The speed at which the ball passes A is the proportional to the frequency of this oscillation. If there are no external dynamic (time dependent) forces (as in this case). The motion is said to be free vibration. In this example. Although a force is required initially to displace the ball, once released it vibrates freely.

In this system, the motion of ball takes place along one path only. This is an example of a single degree of freedom system. This free vibration can take place only at a particular value of frequency, which is it natural frequency. The return period of vibration is the time taken to complete a full cycle.

In reality, nothing foes forever, except perhaps examinations! Any frictional force, which behaves like an opposition in parliament, causes energy loss during vibration, and the motion will eventually cease. Such forces that absorb energy are classified as damping forces, and the vibration in the presence of these forces is called a damped free vibration. As there will always be some energy loss, an undamped free vibration exists only in theory and all free vibrations in real life have an end. Nevertheless, in many practical situations the damping forces are not significantly large to alter the natural frequencies noticeably, and in such cases the frequencies may be calculated by using undamped model. However, the amplitude of vibration is sensitive to damping, and for this reason artificial dampers are sometimes used to control the vibration of structures.

2.3 Forced and Free Vibration

When a structure is subject to an excitation (dynamic force) it responds by translation or rotating which may be described as undergoing a change in its geometry. If the frequency of excitation is equal to one of its natural frequencies, then the amplitude of vibration may be very large (in the absence of damping the amplitude is indefinite) and this state is called resonance. In

General, discrete and continuous systems may vibrate in a combination of more than one pure natural mode (harmonics). The actual displacement would depend on initial conditions as well as on the natural frequency and modes. In the case of forced vibration, the dynamic displacement of the system would depend on the system properties, its initial condition, and the applied dynamic force. It is interesting to compare this vibration behaviour with human behaviour. Just as we may have a behaviour pattern (that could be described as a personality – whether acquired or cultivated is a different matter) with certain ways of reacting when we experience different feelings (for example the way we smile when we are happy), physical systems posses natural frequencies and modes. However, the way react to a particular situation depends on also whether the day started off well and the nature of the situation, while the way a physical system responds to an excitation depends in initial conditions and the properties of the excitation force. In a forced vibration analysis, the dynamic displacement (response) of a system is expressed as the sum of the transient response, which depends on the natural frequencies and modes as well as initial conditions, and the steady-state response, which is a function of the dynamic force and system properties.

2.4 Stiffness and Natural Frequencies

The stiffness of a structure is an indication of its resistance to deformation under loading. In case of a structure that has been displaced from its equilibrium state, the restoring actions increase with stiffness. The natural frequencies of a structure therefore increase with its stiffness. This is why tall, slender structures having low stiffness have long return periods and low natural frequencies. We can now ask, what would happen if the stiffness of a structure corresponding to deformation in a given mode is extremely small or zero? Thant is, what would happen if there is little or no restoring action? To answer this question, let us go back to relationship between the natural frequency and stability of the ball in the bowl. The explanation given for the models in Figures holds for any elastic structure. If the natural frequency is zero the return period would be infinity, which means if subject to a disturbance the structure would never return to its original equilibrium state. By definition this is a critical state of equilibrium. Therefore one can conclude that if a natural frequency of a structure is zero, the equilibrium corresponds to a critical state in that mode. And static load that causes a natural frequency of a structure to vanish is therefore a critical load. We will now focus on some other examples to appreciate the significance of the above statements.

III. EXISTING HEAVY VEHICLE CHASSIS

A typical ladder chassis, BS2 VIKING 222 is chosen for investigation. This chassis is used by the Tamil Nadu State Transport Corporation in passenger buses. The material and geometrical properties of the chassis are

Material	Structural Steel
Volume	$1.1489e^5$ cubic metres
Mass	901.86 kg
Chassis Type	Ladder
Young's Modulus	$2e^5$ MPa
Poisson's Ratio	0.3
Density	7850 kg/m^3
Tensile Yield Strength	250 MPa

Table: 3.1 Chassis properties

The 2D drawing of the BS2 VIKING 222 chassis with dimensional details is given below.

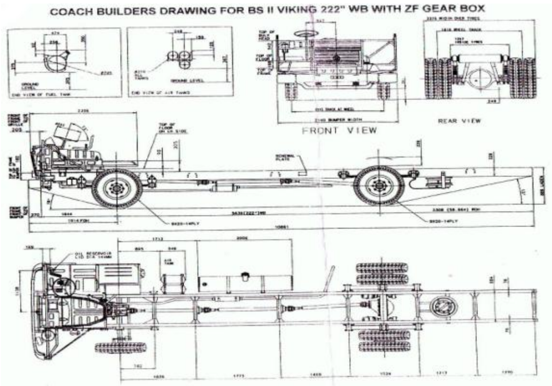

Figure: 3.1 Line drawing of chassis frame

This chassis is modelled in Pro-Engineer 2.0, CAD modelling software and analysed with Ansys Workbench 11, FEA tool. Modal analysis is carried out and the mode shapes for respective natural frequencies are generated.

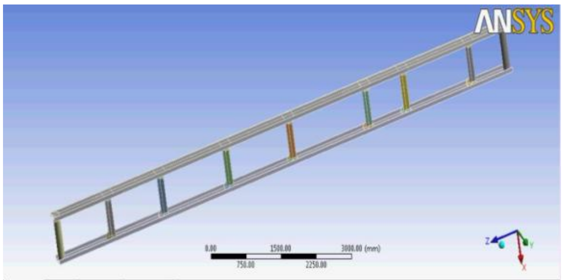

Figure: 3.2 Modelling of chassis in pro-e

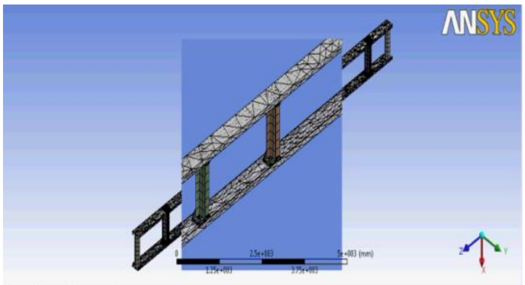

Figure: 3.3 Tetrahedron meshing

NODES 36438
ELEMENTS 15245

Figure: 3.4 Nodes and elements

Supports are given at the place where axles are meant to be attached at the chassis frame as follows,

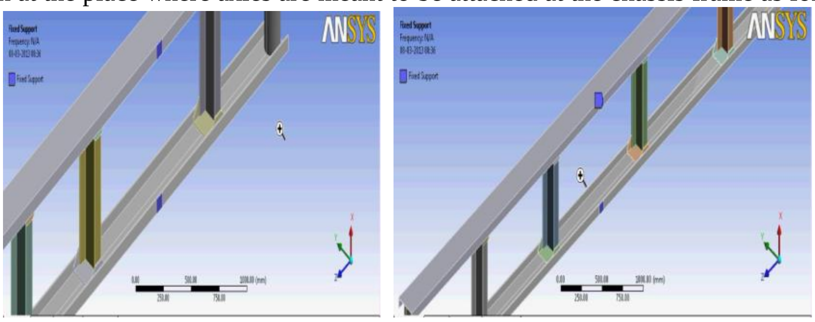

Figure: 3.5 Constraints

Modal analysis is carried out with 12 modes within a range of 0 to 1000 Hz and the natural frequencies of different mode shapes were obtained. The natural frequency of vertical bending is obtained at 54.781Hz and the mode shape obtained is shown below

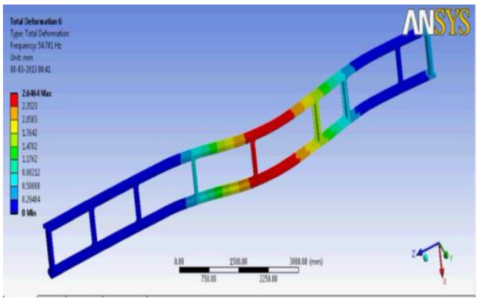

Figure: 3.6 Natural frequency of the existing chassis frame

IV. CHASSIS REINFORCEMENT

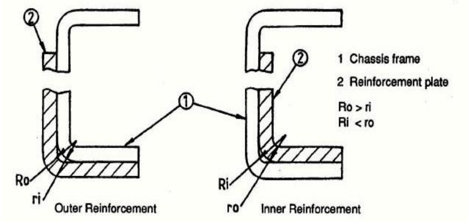

Figure: 4.1 Chassis reinforcement

Chassis reinforcement is a method of increasing the stiffness of the chassis frame by shifting the natural frequency to considerable extent. This is done by placing additional stiffening members to the side rails of the chassis frame. The geometrical stiffness can be increased to different levels with respect to the position and length of the reinforcement plates added to the frame. Some of the methods used for reinforcement are given below.

4.1 Frame Rail Reinforcement

The following requirements apply to frame rail reinforcements:

- Reinforcements should not be terminated within a distance 2H from the centre of a spring hanger (H = the frame rail depth) unless contrary practice is adopted by the vehicle manufacturer. Typical details for terminating reinforcement are shown in Figure 4.2.

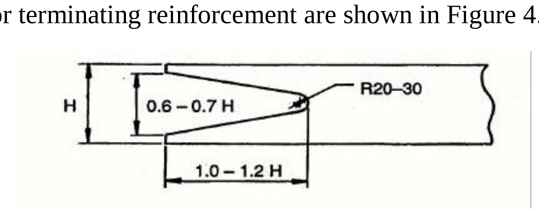

Figure: 4.2 Frog mouth reinforcement

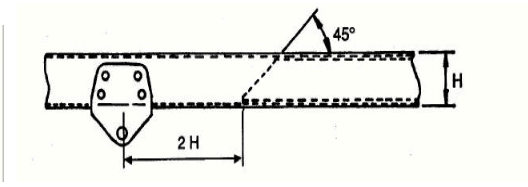

Figure: 4.3 Tapered reinforcement

- It is preferable that additional reinforcement should extend at least a distance of 2H forward of the rearmost front spring hanger bracket and rearward past the rearmost rear spring hanger bracket by a distance of 2H. Note: Allowance must be considered for associated components that would be displaced by the reinforcement section.
- Each end of reinforcement should be tapered at 45 degrees, or alternatively, a 'frog-mouth' tapering may be used. Typical frog-mouth details are shown in figure.
- The thickness of each reinforcement must not exceed the thickness of the main frame rail.
- The reinforcement section should be either angle or channel. Reinforcements may be located inside or outside the main chassis frame rail.
- All reinforcements must be securely attached to the main frame rail. It is recommended that reinforcements be fastened by bolting. Existing bolt holes should be used where possible.
- Inside radius Ro of outer reinforcement curvature must be smaller than outside radius ri of chassis frame curvature (Outside radius ro of inner reinforcement curvature must be larger than inside radius

Ri of chassis frame curvature). (Refer to Figure).

- Multi-Section Reinforcement. When a chassis is to be upgraded over its entire length, it is often difficult to fit a full length reinforcement section due to the installation of other chassis components. A satisfactory method of overcoming these difficulties is by the use of multiple sections of reinforcement. When this method of chassis reinforcement is utilized such reinforcements must be securely attached to each other by either overlapping or by bolting or by butt welding.

V. POSITIONING OF THE REINFORCEMENT ACROSS THE FRAME

Based on the axis of the axle where the chassis frame is connected to the axles the reinforcement plates are positioned across the side rails of the frame.

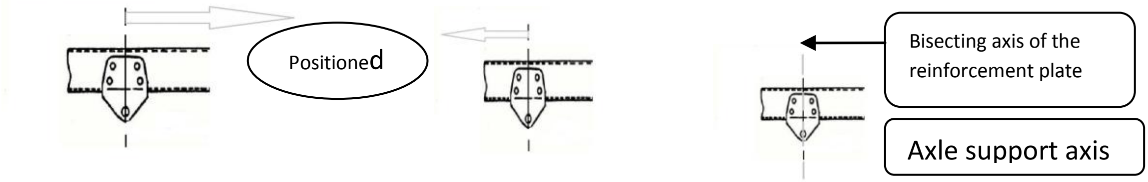

Figure: 5.1 Aligned towards the centre of the chassis frame from the support axis Aligned away from the centre of the chassis frame from the support axis. Figure: 5.2 Aligned away from the centre of the chassis frame from the support axis Aligned on the support axis with the middle axis. 5.3 Aligned on the support axis with the middle axis

For finding the position of the reinforcement where maximum stiffness can be obtained, reinforcement plates of same cross section as that of the BS2 VIKING222 chassis and with the same thickness is placed as shown in above diagrams and the results were obtained as follows. Note : the length of the reinforcement plates is taken as 2H, where H is the height of the side rail.

POSITIONING OF REIN PLATE			
Position	**Frequency(Hz)**	**Mass**	**Deflection**
Outer	55.176	7.3509	2.6452
Center	55.975	7.3507	2.6709
Inner	56.563	7.3507	2.6755

Table: 5.1 Positioning of rein plate

Now, it can be witnessed that the natural frequency shift is maximum when the reinforcement plates are placed inwards through which it had shifted from 54.781Hz to 56.563Hz. Therefore the positioning inwards of reinforcement plates is taken into consideration for further analysis.

VI. SELECTION OF THICKNESS FOR REINFORCEMENT PLATE

Based on the chassis modification requirement, the reinforcement plates are checked for different thickness values which are kept equal or below the thickness of chassis frame used for analysis. Different natural frequency shifts obtained due to the reinforcement plates of different thickness are given below,

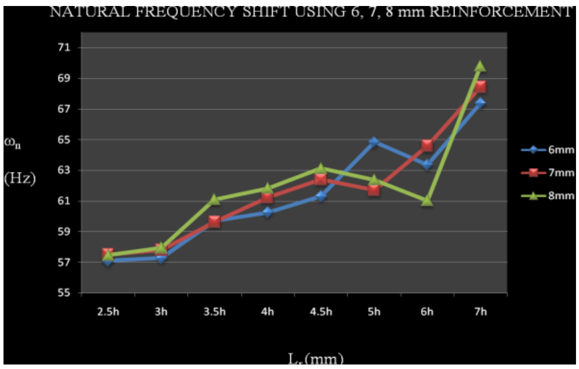

Figure: 6.1 Natural frequency shift using 6, 7, 8 mm reinforcement plates

Natural Frequency shift in Hz for different Lr in terms of H								
Plate thickness in mm	**2.5H**	**3H**	**3.5H**	**4H**	**4.5H**	**5H**	**6H**	**7H**
6	57.105	57.286	59.716	60.251	61.3	64.833	63.355	67.369
7	57.514	57.795	59.624	61.225	62.395	61.703	64.589	68.426
8	57.465	57.943	61.105	61.819	63.12	62.381	61.022	69.782

Table: 6.1 Natural Frequency shift in Hz for different Lr in terms of H

	6mm Full length		**7mm full length**		**8mm full length**	
Rein plate length	**Mass**	**Deflection**	**Mass**	**Deflection**	**Mass**	**Deflection**
2.5h	9.0871	2.6756	11.433	2.6814	11.972	2.6774
3h	10.905	2.6746	13.72	2.6802	14.67	2.6812
3.5h	12.722	2.7095	16.006	2.7109	16.761	2.728
4h	14.539	2.7097	18.293	2.721	19.156	2.7289
4.5h	16.357	2.7147	20.599	2.7251	21.55	2.7343
5h	18.174	2.7515	22.866	2.6982	23.945	2.7056
6h	21.809	2.7012	27.439	2.7016	28.734	2.6455
7h	25.444	2.6745	32.013	2.658	33.523	2.6636

Table: 6.2 6,7,8mm Full length

VII. TAPERED REINFORCEMENT Vs FROG MOUTH REINFORCEMENT

With the results of positioning and thickness selection, the one among the tapered and frog mouth reinforcements can be chosen for the next step of analysis. These reinforcements are used to provide more structural stiffness and saves material than ordinary c-shaped reinforcements as given below,

	Natural Frequency shift in Hz								
Reinforcement plate geomet	**Plate length in mm**	**2.5**	**3**	**3.5**	**4**	**4.5**	**5**	**6**	**7**
	Tapered	54.7	55.149	55.762	57.459	58.603	59.912	60.621	60.569
	Frog Mouth	55.939	57.04	60.636	61.503	61.788	62.155	61.904	61.16

Table: 7.1 Natural Frequency shift

From the above results, it is clear that thickness of 8 mm brings the maximum shift on natural frequency and is chosen for further analysis.

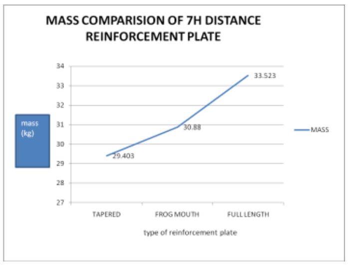

Figure: 7.1 Mass comparison of 7H distance reinforcement plate

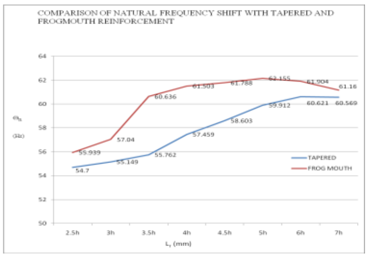

Figure: 7.2 Comparison of natural frequency shift with tapered and frog mouth reinforcement

From the above graph, It can be seen that the tapered and frog mouth plates consume lesser mass with the same length as that of normal reinforcement plates. Now the shift of natural frequencies by the different reinforcement plates is given below.

TAPERED			FROM MOUTH	
Rein plate length	**Mass**	**Deflection**	**Mass(kg)**	**Deflection(mm)**
2.5h	7.8526	2.6141	9.3294	2.6746
3h	10.247	2.6169	11.724	2.6809
3.5h	12.642	2.6246	14.118	2.7259
4h	15.036	2.649	16.513	2.7288
4.5h	17.43	2.6602	18.907	2.7197
5h	19.812	2.6685	21.302	2.7063
6h	24.614	2.6391	26.091	2.6653
7h	29.403	2.5919	30.88	2.6043

Table: 7.2 Tapered and Frog mouth

From the above results, maximum shift of natural frequency is obtained at frog mouth reinforcement plate and hence it is considered for further analysis.

VIII. RELOCATION OF REINFORCEMENT PLATE FOR MAXIMUM STIFFNESS

Considering all the results obtained in the above steps, the frog mouth reinforcement plate is checked out with different positions across the axis of the support of the axle at the frame. For example positioning three quarters of the complete length of the reinforcement towards the centre of the chassis frame and the remaining one quarter is placed to face away from the centre of the chassis frame.

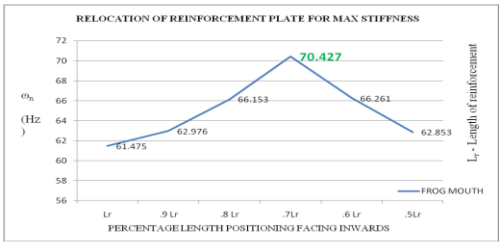

Figure: 8.1 Relocation of reinforcement plate for max stiffness

percentage positioning	frequency (Hz)	Mass(kg)	Deflection(mm)
L_r	61.475	30.88	2.6066
0.9 L_r	62.976	30.88	2.6728
0.8 L_r	66.153	30.88	2.7193
0.7L_r	70.427	30.88	2.747
0.6 L_r	66.261	30.88	2.7673
0.5L_r	62.853	30.88	2.7301

Table: 8.1 Percentage of positioning

Thus the maximum shift of natural frequency is achieved when 70% of the reinforcement plate length is placed facing towards the centre from the axis of support. The mode shape obtained for vertical bending of this positioning of the reinforcement plate is shown as follows,

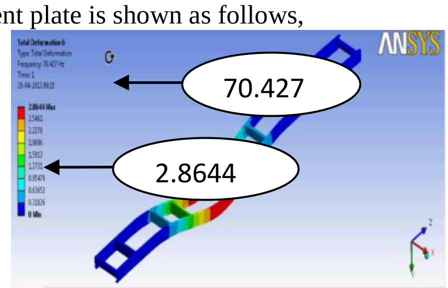

Figure: 8.2 Natural frequency of the chassis frame after the frame reinforcement

IX. CONCLUSION AND DISCUSSIONS

An updated truck chassis FE model to evaluate the critical characteristics like vibration, strength and weight of the truck chassis was proposed.

The major areas of concern in the truck chassis were found to be coincidence of a structural resonance at 52 Hz, experienced the torsional and bending mode. Modifications to shift natural frequencies were proposed by increasing thickness of the chassis centre section by 2 mm and additional of stiffeners member located at centre of the base plate by 10 mm, to understand their influence. The achieved natural frequency shift by 13 % higher than the original value, the increased torsion stiffness by 25 % and the reduced total deflection by 16 % were observed.

The observations indicate the importance of choosing chassis centre thickness and the stiffener location to be optimized with respect to the performance matrices ‘natural frequency’. The overall weight of new truck chassis can be major constraints as it directly related to the cost.

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