A Study on the Vibrational Effects of Adding an Auxiliary Chassis to a 6-Ton Truck

Mousa Hadipour [1], Farshid Alambeigi [2], Ruhollah Hosseini [3], Reza Masoudinejad [4]

[1.] School of Mechanical Engineering, Yazd University, Yazd, Iran
[2.] School of Mechanical Engineering, K.N Toosi University of Technology, Tehran, Iran
[3.] School of Mechanical Engineering, Isfahan University of Technology, Isfahan, Iran
[4.] School of Mechanical Engineering, Shahr-e-Kord University, Shahr-e-Kord, Iran
mehran_fh68@yahoo.com

Abstract: Some of the truck producer involved in production of municipal-service vehicles make some modifications in the main chassis and permit overloading on it. With doing these modifications some special instructions must be considered such as: decreasing speed, strengthening the springs and chassis beams, and so on. Hence chassis vibrational behavior will change the dynamic behavior of automobile effectively, adding auxiliary chassis is an alternative trend to change the rigidity. But, the main question that must be answered is: Are natural frequencies of the modified chassis in suitable range? So, this paper investigates the vibrational characteristics of the chassis before and after strengthening. For this purpose, the modal analysis has been accomplished by the commercial finite element packaged ANSYS and natural frequencies and mode shapes have been determined. In addition, the relationship between natural frequencies and engine operating speed has been explained and, finally advantages of the modified chassis which leads to the increase of the rigidity with no much changes in natural frequencies, has been discussed. The results show that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range.

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1. Introduction

In Iran due to several reasons using truck-mounted sweepers is a preferred option in comparison to different types of compact street sweepers. Some of the companies involved in production of municipal-service vehicles install street sweeping equipments on trucks and launch their product to the market; by making some modifications in the main chassis of the truck and optimizing it for continental life in Iran. These equipments are shown in Table 1.

The mentioned equipments impose a considerable load on the truck and in order for the chassis to survive the loads some precautions must be taken. Also the chassis is bound to have the necessary rigidity for the loads. Adding double chassis known as auxiliary chassis and also changing the connecting bars are some usual tricks to enhance the rigidity. Such companies improve the stiffness and load bearing capacity of truck by installing an auxiliary chassis. This chassis is installed on the main chassis, stretching from the back-end up to the cabin of the truck. Using this method in addition to increasing chassis rigidity, chassis mass will increase and here we face the challenge: how do the car's dynamic behavior change with simultaneous increasing in mass and rigidity?

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Table 1 Installed equipment on the truck.

equipment	weight (kg)
water tank	1200
trash tank	1500
second motor of truck	800
first vertical brush	115
second vertical brush	115
width brush	156
oil tank	30
right suction pump	110
left suction pump	110
back door	165
others	100

This chassis is installed on the main chassis, stretching from the back-end up to the cabin of the truck. Using this method in addition to increasing

chassis rigidity, chassis mass will increase and here we face the challenge: how do the car's dynamic behavior change with simultaneous increasing in mass and rigidity?

The dynamic response of simple structures, such as uniform beams, plates and cylindrical shells, may be obtained by solving their equations of motion. However, in many practical situations either the geometrical or material properties vary, or the shape of the boundaries cannot be described in terms of known mathematical functions. Also, practical structures consist of an assemblage of components of different types, namely beams, plates, shells and solids. In these situations it is impossible to obtain analytical solutions to the equations of motion (Fahi and Walker, 2004). Using numerical solutions and finite element methods will solve this problem.

Automotive industry is one of the biggest users of the technology of modal analysis. Truck chassis forms the structural backbone of a commercial vehicle. The modal behavior of car chassis is a part of most necessary information for the inspection into car's dynamic behavior. In this essay the modal analysis of modified truck chassis (combination of main chassis and auxiliary chassis) has been studied (Forouzan and Hosseini, 2010).

As a truck travels along the road, the truck chassis is excited by dynamic forces induced by the road roughness, engine, transmission and more. Under such various dynamic excitations, the truck chassis tends to vibrate [3]. Whenever the natural frequency of vibration of a machine or structure coincides with the frequency of the external excitation there occurs a phenomenon known as resonance, which leads to excessive deflections and failure. The literature is full of accounts of system failures brought about by resonance and excessive vibration of components and systems [4].

The global vibrational characteristic of a vehicle is related to both its stiffness and mass distribution. The frequencies of the global bending and torsional vibration modes are commonly used as benchmarks for vehicle structural performance. Bending and torsion stiffness influence the vibrational behavior of the structure, particularly its first natural [5].

The mode shapes of the truck chassis at certain Natural frequencies are very important to determine the mounting point of the components like engine, suspension, transmission and more. Therefore it is important to include the dynamic effect in designing the chassis [3].

Fig. 1 A view of truck

Many researchers carried out study on truck chassis. Vasek et al. (1998) have analyzed a truck dynamically. In their method in addition to simulating truck with finite element packaged ANSYS and being sure that structure vibrational modes are in appropriate range, they vibrationally analyzed it. Yuan Zhang and Arthur Tang (1998), compare natural frequencies of a ladder chassis with finite element and experimental methods. Guo and Chen (2008), research into dynamic and modal analysis of a space chassis (complex 3-dimensional chassis) and analyses transient response using the principal of superposition.

This paper deals with a 6 ton truck chassis that includes natural frequencies and mode shapes. This chassis has been shortened by related companies for using in municipal service (street sweeper) and here we face the challenge and it raises the question: Are natural frequencies of the modified chassis in suitable range?

In the studied model unlike the most previous models, rivets and bolts have been modeled completely. Also shell element has been used for analysis. This element has better and more disciplined meshing in comparison with other elements and has the capability of gaining more accurate results with the same meshing containing the related 3-dimensional elements. Validity of the results has been verified by comparing the results of a similar model by the model proposed by Fui and Rahman (2007).

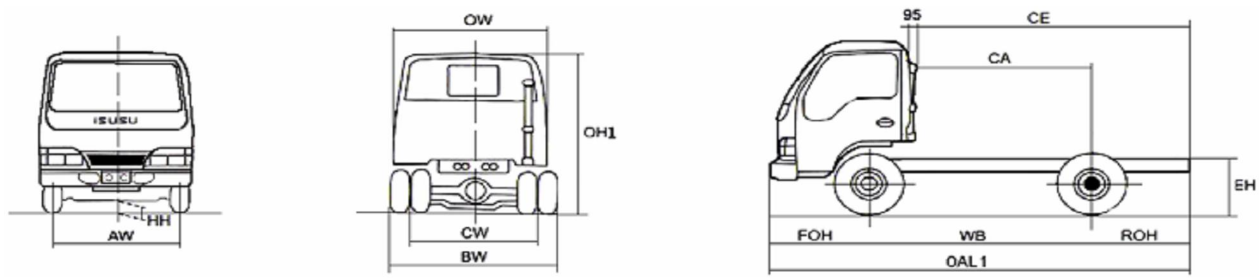


Fig. 2 Dimensional plane of truck

Table 2 Truck dimensions (in mm) related to Fig.1

EH	HH	OH1	CW	BW	AW	OW	CA	CE	ROH	FOH	WB	OAL1
830	210	2220	1650	2115	1680	1995	3195	4910	1700	1085	3815	**6600**

2. Truck chassis

In this article, a 6 ton truck chassis has been studied. This truck chassis is a ladder chassis and its longitudinal and cross connecting sections are channel shaped.

Automotive overviews are shown in Figs. 1and 2 and Table 2 illustrate dimensions of studied truck. Chassis material is JIS-SAPH41 with 7800kg/m3 density, 520MPa yield strength, and 590MPa tensile strength.

The auxiliary chassis is bolted by using eight structural connectors which are placed asymmetrically on the main chassis and improves the stiffness of the system. The length of the main chassis is 567 cm (considering the 660 cm long initial chassis, some of which is cut) and it is 82.6 cm wide while the auxiliary chassis which has the same width and is 367 cm long, is placed on it. The vertical distance between the main chassis and auxiliary chassis is 26 cm. Figures 3 and 4 shown a view of structural connector and a plan of auxiliary chassis, respectively.

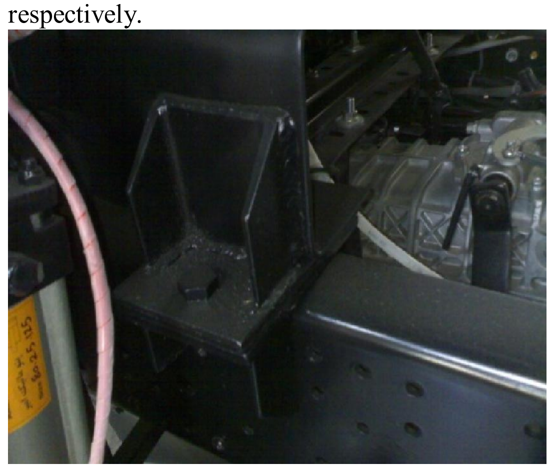


Fig. 3 A view of structural connector

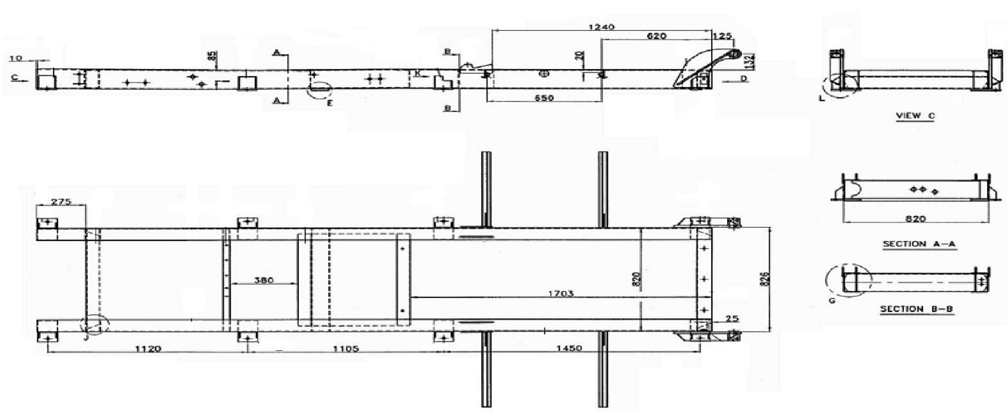


Fig. 4 A plan of auxiliary chassis

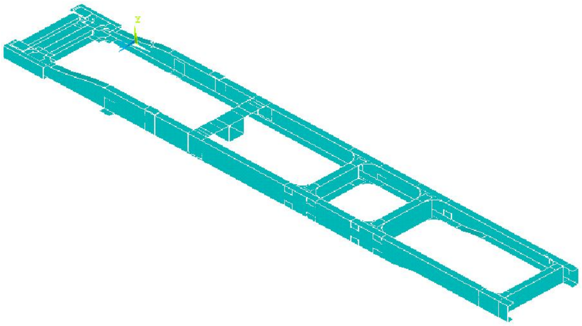


Fig. 5 Main chassis model.

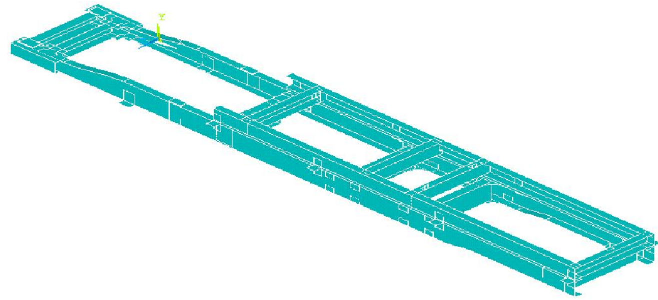


Fig. 6 Modified chassis model.

3. Finite Element model

Truck chassis has been modeled with 4-node shell element in ANSYS. Numerical studies on simple hollow rectangular beam show that this element is suitable for creating and meshing the model and it yields accurate results. The element used has 4 nodes with 6 degrees of freedom and is appropriate for linear and nonlinear deformations and also large deflections. There are approximately 70000 elements in the model that has proved suitable in comparison with other cases, so that the error in each case is less than one percent. In Fig. 5 and Fig. 6 main chassis model and modified chassis model have been shown.

Model with appropriate accuracy and with considering bolts and riveted joints effects has been simulated. Meshes and constraints have been shown in Fig. 7. The boundary conditions are different for each analysis. In normal mode analysis, free-free boundary condition will be applied to the truck chassis model, with no constraint applied to the chassis model [3]. A free-free boundary condition has been chosen as it is much simpler to test experimentally in this condition, if required (Forouzan and Hosseini, 2010).

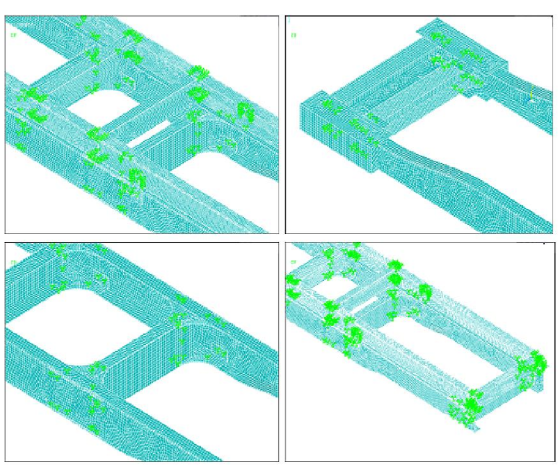


Fig. 7 Meshed chassis.

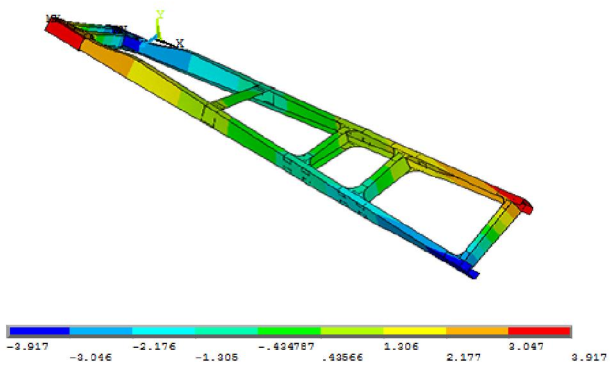
4. Modal analysis and results

Modal analysis has been performed after creating the chassis finite element model and meshing in free-free state and with no constraints. The results have been calculated for the first 30 frequency modes and show that road simulations are the most important problematic for truck chassis. In this analysis we have made use of subspace method in ANSYS.

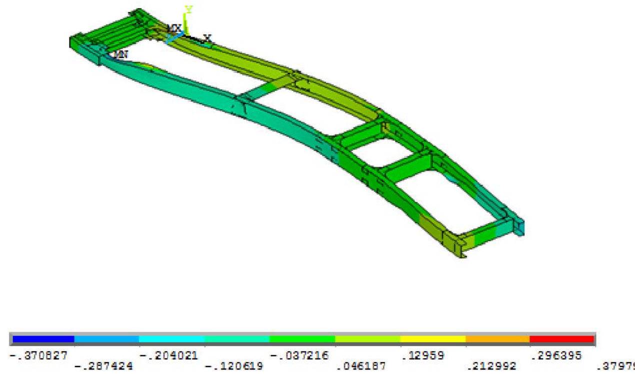
Since chassis has no constraints, the first 6 frequency modes are vanished. 3 modes are related to the chassis displacement in x, y and z directions and 3 modes are related to chassis rotation about x, y and z axes. In Fig.8 related natural frequencies and mode shapes for chassis with maximum displacement in y direction in each mode, have been shown.

The first, second and sixth modes are the global vibrations, while the others are local vibrations. Local vibration starts at the third mode at 29.612 Hz. The dominant mode is a torsion which occurred at 7.219 Hz with maximum translation experienced by both ends of the chassis. The second mode is a vertical bending at 17.153 Hz. At this mode, the maximum translation is at the front part of the chassis. The third and fourth modes are localized bending modes at 29.612 Hz and 33.517 Hz. The maximum translation is experienced by the top hat cross member. The member also experienced big translation at fifth mode which is a localized torsion mode. The top hat cross member is the mounting location of the truck gear box. The sixth mode is the torsion mode at 38.475 Hz with maximum translation at both ends of the chassis.

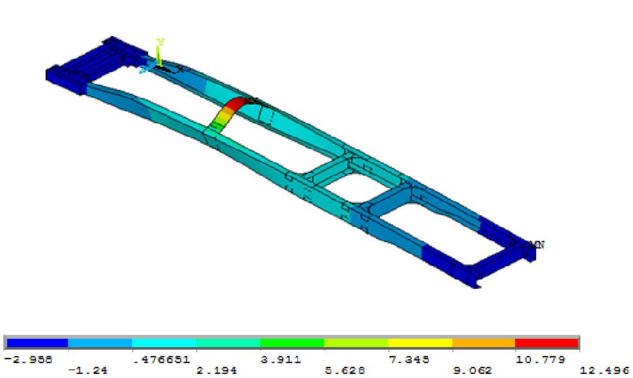
Found natural frequencies from modal analysis of truck chassis, are used for determining the suitable situations for truck parts in working conditions.
Figure 8 shows that twisting mode with the frequency of 7.219 Hz is the prevailing mode. Results of first 12 analyzed modes for main chassis are shown in Table 3.



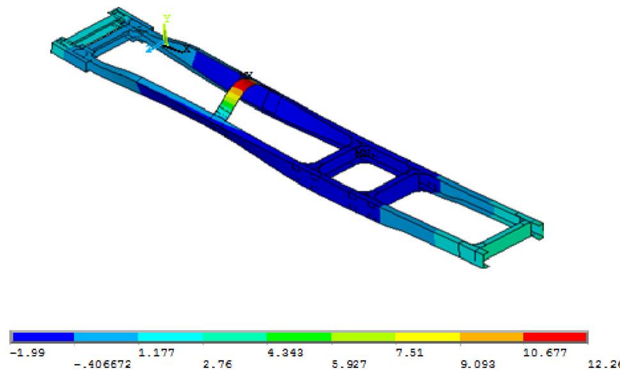
1st mode of frequency



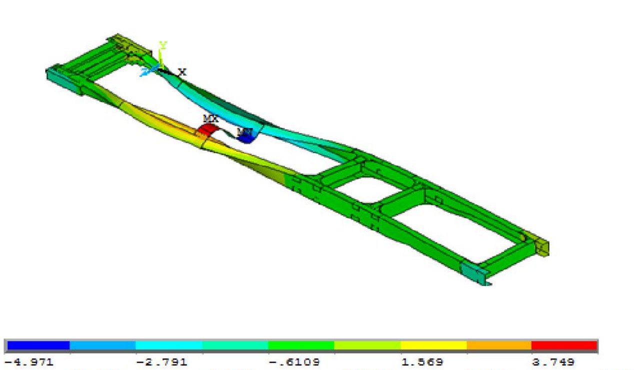
2nd mode of frequency



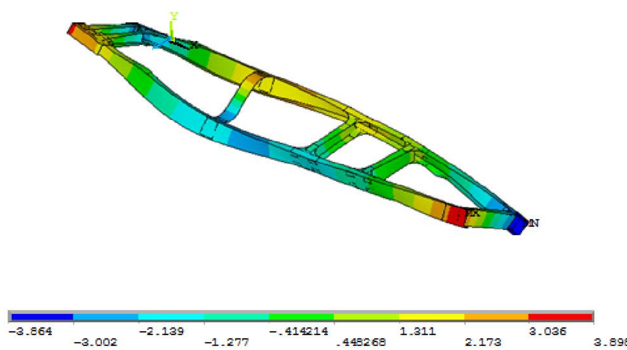
3rd mode of frequency



4th mode of frequency



5th mode of frequency



6th mode of frequency

Fig. 8 Mode shapes and natural frequencies of truck chassis.

Because the analysis is in free-free state, the first 6 modes that have zero frequency aren't considered and mode numbers 7 to 12 in table 3 represent the first 6 modes of frequency.

It is necessary to notice that in usual, the first 6 modes of frequency (mode numbers 7 to 12 in Table 3 that have been shown in Fig. 8) play the main role in dynamic behavior of chassis and since the increasing noise effects and limited energy of motor to generate these frequencies, the effects of higher frequencies can be ignored.

Diesel engine is known to have the operating speed varying from 8 to 33 revolutions per second (rps) (Johansson and Edlund, 1993). In low speed idling condition, the speed range is about 8 to 10 rps. This translates into excitation frequencies varying from 24 to 30 Hz [3]. The main excitations are at low speeds, when the truck is in the first gear. At higher gear or speed, the excitations to the chassis are much less. The natural frequency of the truck chassis should not coincide with the frequency range of the axles, because this can cause resonance which may give rise to high deflection and stresses and poor ride comfort. Excitation from the road is the main disturbance to the truck chassis when the truck travels along the road.

Table 3 Natural frequencies for main chassis.

Mode number	Natural frequency (Hz)
1	0
2	0
3	0
4	0
5	0
6	0
7	7.2195
8	17.153
9	29.612
10	33.517
11	35.161
12	38.475

In practice, the road excitation has typical values varying from 0 to 100 Hz. At high speed cruising, the excitation is about 3000 rpm or 50 Hz.
Mounting of vibration components of the truck on the nodal point of the chassis is one of the vibration attenuation methods to reduce the transmission of vibration to the truck chassis [3].

The mounting location of the engine and transmission system is along the symmetrical axis of the chassis's first torsion mode where the effect of the first mode is less. However, the mounting of the suspension system on the truck chassis is slightly away from the nodal point of the first vertical bending mode. This might due to the configuration of the static loading on the truck chassis.

In the following modal analyzing has been used for the modified chassis, in order to compare the main chassis and modified chassis. In Fig.9 related natural frequencies and mode shapes for chassis with maximum displacement in y direction in each mode, have been shown. Similarly the first, second and sixth modes are the global vibrations, while the others are local vibrations. Local vibration starts at the third mode at 27.610 Hz. The dominant mode is a torsion which occurred at 9.642 Hz with maximum translation experienced by both ends of the chassis. The second mode is a vertical bending at 16.271 Hz. Similar to the main chassis analysis, at this mode, the maximum translation is at the front part of the chassis. The third and fourth modes are localized bending modes at 27.610 Hz and 31.013 Hz. The maximum translation is experienced by the top hat cross member. The member also experienced big translation at fifth mode which is a localized torsion mode. The sixth mode is the torsion mode at 39.042 Hz with maximum translation at both ends of the chassis. Figure 9 shows that twisting mode with the frequency of 9.462 Hz is the prevailing mode.

Table 4 Natural frequencies for modified chassis.

Mode number	Natural frequency (Hz)
1	0
2	0
3	0
4	0
5	0
6	0
7	9.4623
8	16.271
9	27.610
10	31.013
11	37.929
12	39.042

Results of first 12 analyzed modes for modified chassis are shown in Table 4. As a reminder it is mentionable that because the analysis is in free-free state, the first 6 modes that have zero frequency aren't considered and mode numbers 7 to 12 in Table 4 represent the first 6 modes of frequency.

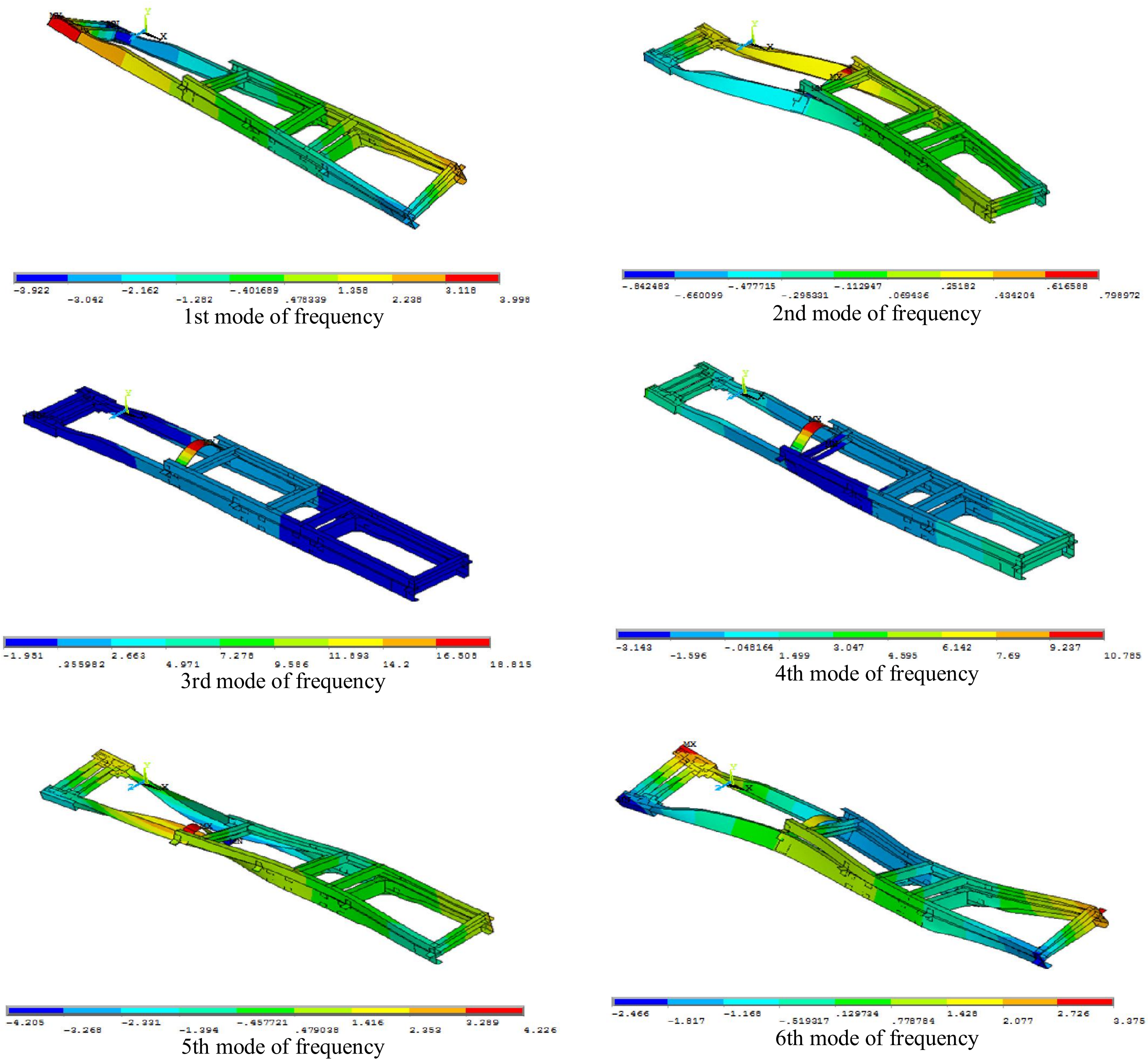
In general the natural frequency can be calculated using the equation (1);

$$\omega_n = \sqrt{\frac{K}{m}} \qquad (1)$$

where K and m stand for stiffness and mass respectively.

The equipment installed on the chassis of municipal-service truck increase the chassis mass which leads to the decrease of natural frequencies. Regarding the fact that the chassis stiffness and mass have both increased and considering equation (1) and noticing that the frequencies are in normal range and it has no considerable change, one can state that the mass increase due to the addition of auxiliary chassis is equally efficient on the natural frequencies as the increase in structural stiffness and thus eventually one can hardly find a change in natural frequencies.
By changing the place of gasoline tank and mounted equipments and other such considerations, attempt has been made to prevent coinciding the excitation force frequencies and natural frequencies. Otherwise resonance phenomenon occurs and the chassis undergoes destructive vibrations and if these two frequencies coincide, this phenomenon may lead to the structural failure of the chassis. As a reminder it

is mentionable that validity of the results is verified by comparing the results from a similar model with the model proposed by Fui and Rahman (2007). Also their results confirm the results of this paper.



1st mode of frequency

2nd mode of frequency

3rd mode of frequency

4th mode of frequency

5th mode of frequency

6th mode of frequency

Fig. 9 Mode shapes and natural frequencies of the strengthened truck chassis.

5. Conclusion

The article has looked to changes of chassis dynamic behavior caused by change in usage and adding auxiliary chassis with finite element method. First six frequency modes of the main chassis and modified chassis that determine their dynamic behavior are below 40 Hz. For the main chassis frequencies vary from 7.219 to 38.475 Hz and for the modified chassis they vary from 9.462 to 39.042 Hz. For the first two modes and sixth mode, the truck chassis experienced global vibration. The global vibrations of the truck chassis include torsion and vertical bending with 2 nodal points. The local bending vibration occurs at the top hat cross member where the gearbox is mounted on it.

Since chassis mass increases due to the installed equipment, the natural frequencies fall out of the natural range that can be compensated with increasing the chassis stiffness. Auxiliary chassis can increase the chassis stiffness. Using this method, we can prevent resonance phenomenon and unusual chassis vibration and place the natural frequencies in natural range.

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