APPLICATION OF DYNAMIC CORRELATION TECHNIQUE AND MODEL UPDATING ON TRUCK CHASSIS

Izzuddin bin Zaman @ Bujang* , Roslan Abd. Rahman+

* Faculty of Mechanical Engineering ,College University Technology Tun Hussein Onn (KUiTTHO)
+ Faculty of Mechanical Engineering, University Technology of Malaysia (UTM)

Abstract
Truck chassis is a major component in a vehicle system. It is often identified for refinement in order to develop vehicles with reduced cost and weight. Nowadays the process of chassis design in the automotive industry has been significantly refined with the high capabilities of advanced computer aided design and engineering tools,. The application of FEA such as structural modification and optimization is used to reduce component complexity, weight and subsequently cost. Because the level of model complexity can be high, the opportunity for error can also be high. For this reason, some form of model verification is needed before design decisions made in the FEA environment can be implemented in production.
This paper looks into the application of dynamic correlation techniques for verification of the FEA models of truck chassis. The dynamic characteristics of truck chassis such as the natural frequency and mode shape were determined using finite element method. Experimental modal analysis was carried out to validate the FE models. Initial results from both analysis show that the truck chassis experienced 1st torsion mode for 1st natural frequency, 1st bending mode for 2nd natural frequency, 2nd torsion mode for 3rd natural frequency and 2nd bending mode for 4th natural frequency. However there is a small discrepancy in terms of frequency. Thus, the model updating of truck chassis model was done by adjusting the selective properties such as Modulus Young and Poisson ratio in order to get better agreement in the natural frequency between both analysis. Finally, the modifications of the updated FE truck chassis model was suggested such as by considering adding the stiffener. The purpose is to reduce the vibration as well as to improve the strength of the truck chassis.

Keywords: Modal parameters, Finite element, Modal testing, Model updating

1.0 INTRODUCTION

Chassis used in off-road vehicles have almost the same appearance since the models developed in 20 or 30 years ago. This indicates that the evolution of these structures is still slow and stable along the years [1]. Many researchers in automotive industry have taken this opportunity to be involved in the chassis manufacturing technology and development. Malaysia as one of the developing country had invested large amount of money in automotive industry. However, the automotive industry in Malaysia especially in truck manufacturing is still in the development phase and much relying on foreign technology.

Nowadays, the current trend in truck design involves the reduction of costs and increase in transportation efficiency. The pursuit of both these objectives results in lighter truck, which uses less material and carries less dead weight. One of the parts in the truck that is strongly influenced by these guidelines is chassis [2]. The consequence of a lighter chassis is a vehicle that has structural resonance within the range of typical rigid body vibrations of the truck subsystems. On the other hand, the vibration can be formed due to dynamic forces induced by the road irregularities, engine, transmission and more. Thus under these various dynamic excitation, chassis will tend to vibrate and can lead to ride discomfort, ride safety problems, road holding problems and also to cargo damage or destruction [3].

This paper focused on the dynamic correlation techniques which used to measure the accuracy of finite element representation of the truck chassis. Treating the chassis independently, analytical and experimental models were developed using Finite Element Analysis (FEA) and Experimental Modal Analysis (EMA) techniques. Experimental modal surveys were conducted and the frequencies and mode shapes were compared to those extracted from the FEA models. Technique such as the Modal Assurance Criteria

(MAC) was used to compare the vectors and the observations were made about the potential for improvements. Model updating was then performed to achieve a high degree of confidence in the FEA.
In truck chassis development, the structural modification is one of the important stages. It is done through modifying the dynamic behavior of the chassis which result in reducing the vibration effect and improve the strength of truck chassis. The most common method used in structural modification is adding stiffener.

2.0 FINITE ELEMENT MODEL DESCRIPTION

The truck chassis was generated using commercial FEM software. The 10-node tetrahedral elements was chosen in the meshing analysis as this element gave a closer result to the actual condition [4]. The final chassis model consists of 24322 nodes and 12087 elements. The material employed was steel. Figure 1 and 2 show the complete finite element of the truck chassis model under study before and after meshing. During the model construction, the following consideration had been taken into account in order to simplify the analysis:

i. All brackets were excluded from the model.
ii. The connections between longitudinal rail and cross members were considered perfect.
The material is considered isotropic in its elastic phase

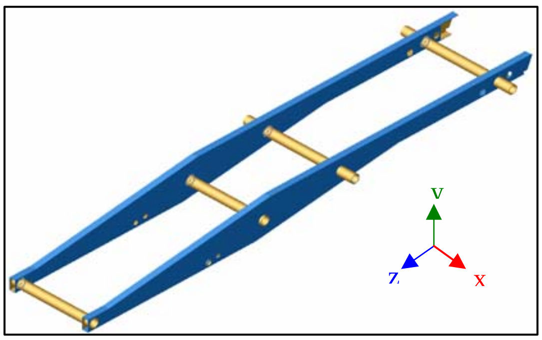

Figure 1: FE model of truck chassis

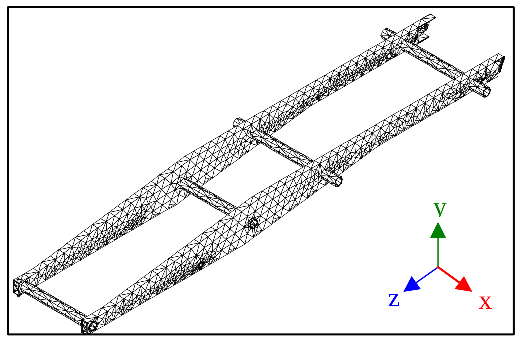

Figure 2: FE meshing mdel of truck chassis

The free-free boundary condition was adopted in order to obtain the chassis's natural frequencies and mode shape vectors. Neither constraints nor loads were assigned in an attempt to stimulate this free-free boundary condition. Thus the frequency range of interest was set between 10 to 200 Hz. The reason for setting the starting frequency at 10 Hz was to avoid the solver from calculating rigid body motions [5]. Under this study, only the next four fundamental frequencies were observed, as these frequencies are critical to the truck chassis dynamic behaviour. Figures 3 until 6 represent typical mode shape of the truck chassis at 35.2, 64.8, 99.1 and 162.3 Hz. The contour shows the translation value of the chassis under vibration modes.

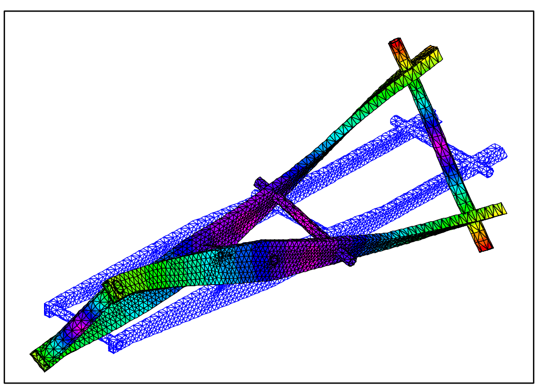
Figure 3: FEA first mode shape @ 43.7 Hz
(1st torsion mode)

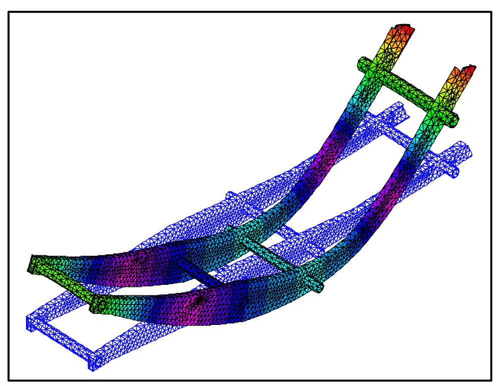
Figure 4: FEA second mode shape @ 64.8 Hz
(1st bending mode)

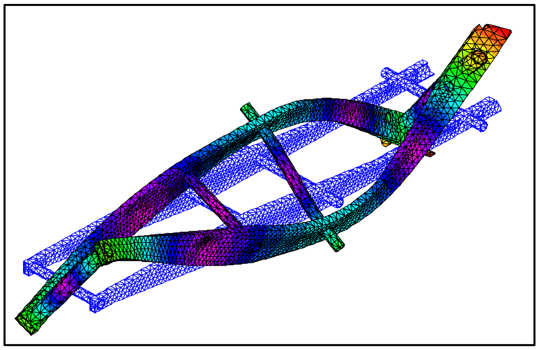

Figure 5: FEA third mode shape @ 99.1 Hz
(2^{nd} Torsion mode)

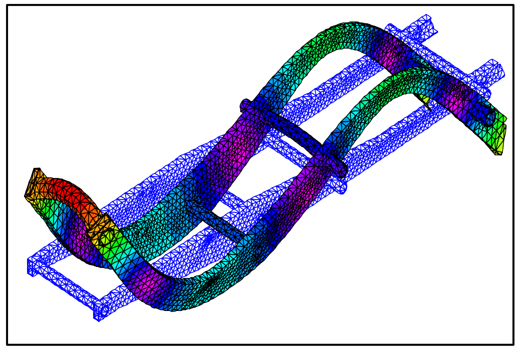

Figure 6: FEA fourth mode shape @ 162.3 Hz
(2^{nd} Bending Mode)

3.0 EXPERIMENTAL MODAL ANALYSIS

This section presents the experimental data acquired in order to identify the modal characteristics of truck chassis. The chassis was divided into 22 grid points where at these points, frequency response functions were measured in the range of 0-200 Hz. These 22 grid points were chosen to give adequate spatial resolution to describe the global structural mode shapes

In this study, two excitation methods were implemented in the experimental test. The first testing was done by using a shaker fixed at one input location , and a roving uni-axial accelerometers moved from point to point on the structure. Figure 7 shows the experimental setup for the shaker test. The boundary conditions were similar to the FEM model. The nature of the structure presented difficulty with this method, as the location of the accelerometer affected the dynamics of the structure significantly [6]. This is referred to as "mass loading". The modal frequencies changed values depending on the location of the accelerometers making this method unacceptable. The second tests which known as impact hammer test was performed by attaching the uni-axial accelerometer to a reference point, and excite the structure at all other points with the roving impact hammer. This method provides better results with negligible mass loading. Figure 8 and 9 show the superimposed FRF at all points for both methods.

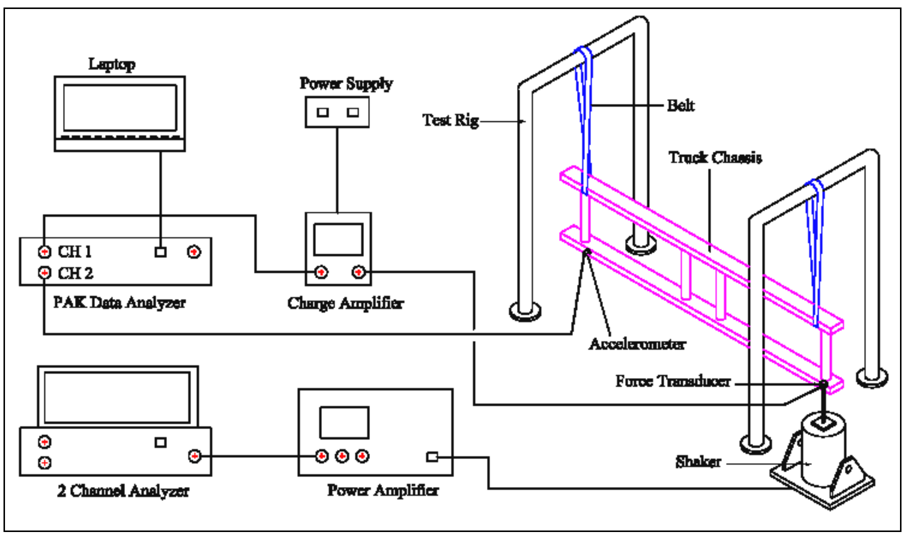

Figure 7: A typical experimental set-up for impact hammer test

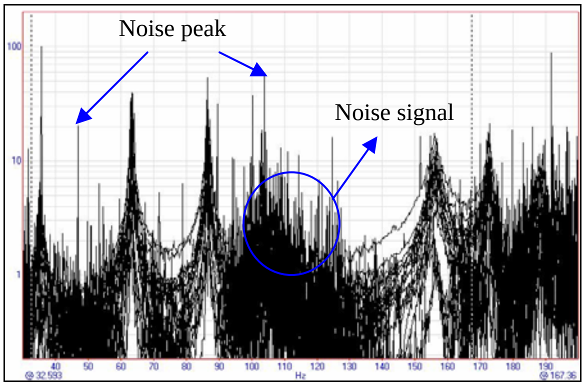

Figure 8: Superimposed FRFs by shaker test

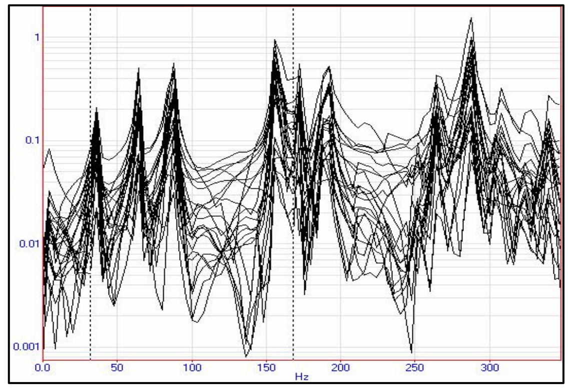

Figure 9: Superimposed FRFs by impact hammer

Table 1 shows a list of frequencies modes of the truck chassis below 200 Hz that was extracted from both of finite element model and experimental test.

Mode	Impact Hammer		Shaker		FEA modes Frequency (Hz)
	Natural Frequency (Hz)	Damping (%)	Natural Frequency (Hz)	Damping (%)	
1	35.2	2.8758	35.7	0.0300	43.7
2	63.4	0.8148	63.4	0.1213	64.8
3	86.8	0.7553	86.6	0.1417	99.1
4	157.0	0.6556	156.4	0.3492	162.3

Table 1: Natural frequencies obtained by EMA and FEA

Theoretically, each of the test mode frequency should match with one of the finite element. In this case, all the mode frequency obtained from the test is counterpart with each of FE mode frequency. Notice that each FEA frequency is slightly higher than its matching tests frequency, indicating that possibly the stiffness of the FE model is greater than the stiffness of the real structure.

4.0 CORRELATION OF FEA AND EMA

Correlation is a process to evaluate how close the FE model resembles the reality or in other words, how good the FE model agrees with the experimental model. The result from impact hammer test was chosen for correlation as it gave good coherence results as compared to shaker test. Discrepancies will always exist between the FE model and the EMA model. This because there are possibilities error in experimental data such as noise exist in the data and the measurements were carried out at an imperfect set-up. The model parameter errors and model structure errors can also contribute to the source of discrepancies [7].

The correlation analysis was executed in three steps. Firstly, a geometric correlation was performed. The test geometry matches perfectly with the FE model because it was derived from the finite element model. Thus, at this point a node pair table can be created on the spot. No translation and rotation values need to be specified. If the test geometry would not have been taken from the FE model but separately have been created by the test engineers, a calculation of translation and rotation would be necessary to be able to put the two models on top of each other. Then the test modes were transformed to the FE model geometry using the previous created node pair table. At this point, only the real measured DOFs of the truck chassis are selected to continue the correlation analysis. Lastly, a Modal Assurance Criterion (MAC) matrix was

performed and the result will tell how good the FE modes correlate with the test modes. The high MAC values (> 75 %) show us which FE mode shapes resemble to which test mode shapes [8].
Table 2 shows natural frequencies comparison between FEA and EMA model and also the MAC value. It is observed that the FEA frequency for mode 1 and 3 show a slightly bigger error than its matching tests frequency. For a mode shape correlation, notice that the first 3 modes have the MAC value above 95 % which indicate that the test and FEA shapes are very similar. The fourth pair of modes has a MAC value above 90 %, which still indicating that the shapes are similar. Figure 10 show the MAC-matrix graph after the correlation analysis.

Mode	FEA modes Frequency (Hz)	EMA modes Frequency (Hz)	Error (%)	MAC (%)
1	43.7	35.2	24.29	98.4
2	64.8	63.4	2.22	97.2
3	99.1	86.8	14.11	96.3
4	162.3	157.0	3.43	93.8

Table 2: Mode pairs with frequency difference

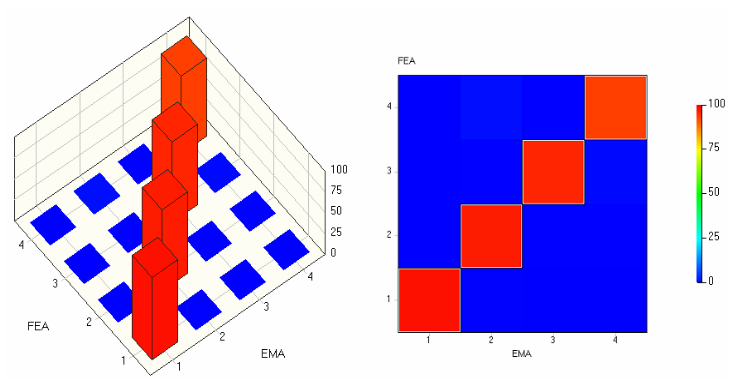

Figure 10: MAC-matrix before model updating

5.0 MODEL UPDATING

The calculated natural frequencies from the FEA did not match with the experimental especially for mode 1 and 3. Consequently, a model updating was requested. Model updating is a step in model validation process that modifies the values of parameters in FE model in order to bring the FE model prediction into a better agreement with the experimental data [9]. In other word, the finite element model was tuned to match the experimental data in order to create a reliable finite element model suitable for further analysis. The test data was used as the target and the FE parameters were updated. Before the model updating can be carried out, sensitivity analysis was performed using FEMtools software [10] in order to decide the parameters in FE model which have significant influence to the change of the modal properties of truck chassis. After several sensitivity analysis, the following parameters were selected for finite element model updating:

i. The dynamic modulus of truck chassis, E
ii. The mass density of the truck chassis, ρ

Modal based methods use these test modal parameters as reference data to be used in the model updating procedure. Parameters E and ρ were selected as local updating variables. Local updating refers to the individual modification of parameters associated with finite elements such as the material or geometrical properties or nodes. They may relate to simplifications used in the FE model. Correlation between FEA and EMA mode shapes was again quantified based on MAC. Table 4 shows a comparison between the natural frequencies from the first FE model, the updated one and the experimental results. It can be seen that the updated model shows better results where the error between FE model and experimental was reduced within ± 2%.

Mode	EMA (Hz)	First FE		Updated FE	
		(Hz)	Error (%)	(Hz)	Error (%)
1	35.2	43.7	24.29	35.8	1.64
2	63.4	64.8	2.22	62.4	-1.58
3	86.8	99.1	14.11	87.7	0.99
4	157.0	162.3	3.43	156.5	-0.31

Table 4: Comparison between natural frequencies before and after model updating

As for the mode shape, Table 5 shows that the model updating did not significantly improve the values of MAC. There was a small increase for the first mode but a slightly decrease for mode 2, 3 and 4. This may be due to several factors. The experimental mode shape was only in one degree of freedom since the accelerometer used was a single axial. At the same time, the mode shapes of the FE model were calculated in three degrees of freedom. Therefore, this difference gives an imperfect mode shape. The MAC values can even be more unsatisfactory if correlation was allowed up to ten modes since higher modes have complex mode shapes [8].

	Degree of Correlation MAC Diagonal Values	
	Before Updating	After Updating
Mode 1	98.4	98.5
Mode 2	97.2	96.9
Mode 3	96.3	96.2
Mode 4	93.8	92.3

Table 5: MAC diagonal values before and after model updating

Figure 11 and 12 illustrate the parameters E and ρ that were updated. After the model-based updating analysis, it is noticed that the results show the dynamic modulus of welds in the FE model (the connection between cross member and longitudinal rail) has reduced by 50% which has the nominal values between 78 to 80 GPa while the other area are kept more or less the same as their initial values. It was also found that the mass density was increased locally from 2.0×10^4 to 2.50×10^4 kg/m^3 .

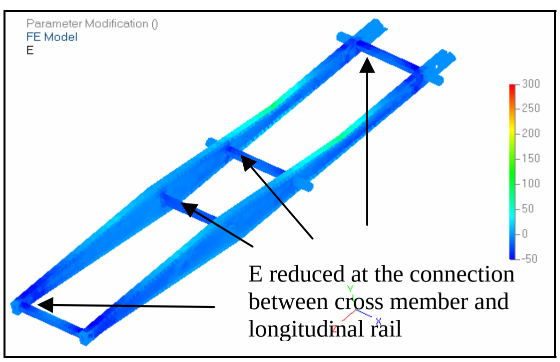

Figure 11: E changes as result of updating

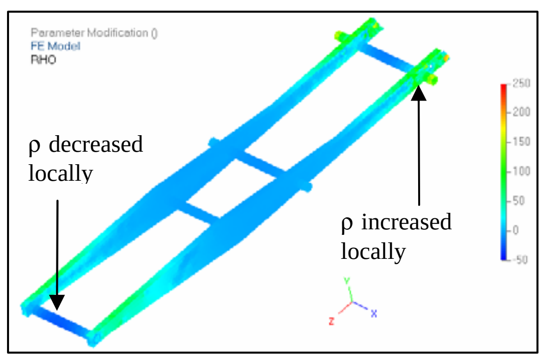

Figure 12: RHO changes as a result of updating

6.0 STRUCTURAL MODIFICATION

Structural modification is important to improve the dynamic behaviour of the truck chassis. After the model updating analysis, the FE model were then transfered to the FE software for structural modification. Additional cross member with diameter 80 mm and thickness 10 mm had been added at the rear of truck chassis and the center cross member was replaced with K-member as shown in Figure 13. The main purpose of the analysis is to investigate the stiffness effect against the dynamic behavior of truck chassis as well as to reduce the vibration when it is exerted by the torsional loads.

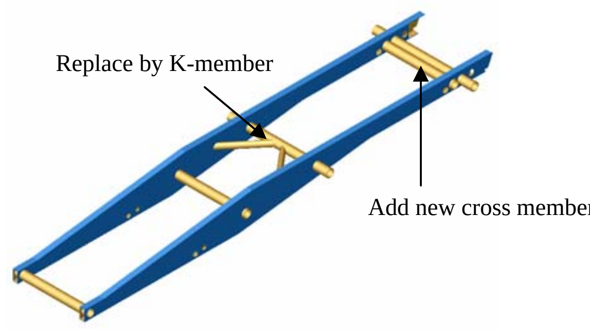

Figure 13

In this study, the first mode shape of truck chassis which experienced torsion mode was analyzed. The first mode at 35.7 Hz is a predominant natural frequency and present almost within the engine operating range. Thus structure modification is essential to shift the natural frequency away from the operating frequency range and at the same time minimize the torsional displacement. Table 6 shows the result of displacement and natural frequency of the first mode before and after the modification analysis. The result shows that the modification of truck chassis has successfully minimized the displacement by 3.28% with minimal changes in natural frequency. This result indicates reduction of displacement after modification particularly at rear part of truck chassis.

Before Modification		After Modification		Displ. Reduction (%)
Nat. Freq. (Hz)	Max. Displ.	Nat. Freq. (Hz)	Max. Displ.	
35.8	0.183	35.9	0.177	3.28

Table 6: Maximum displacement of chassis in the first mode

7.0 DISCUSSION

Some of the problems were encountered during the testing, particularly with reference to mass loading or known as a shaker test. Although the chassis structure is relatively heavy compare to the mass of the accelerometer, mass loading was still significant especially for modes with high participation from local areas [11]. However, these conditions normally happened in the higher modes of excitation. The first four mode shapes as discussed earlier is not affected by the local vibration. Somehow these difficulties can be overcome by changing to the roving impact hammer method. Besides that, there are other problems encountered in the shaker test during the FRF measurement. It is apparent that, the shaker test produced an unwanted portion or noise signal in the FRFs plot as shown in Figure 8. This maybe occurred due to the inability for the shaker to excite the chassis close to supporting belt, particularly around the center of the chassis and near the cross member area [12].

In the correlation analysis, it is observed that all the first 4 modes have MAC value above 0.90 which indicating that the test and FEA shapes are similar. Somehow the result of natural frequency of FEA is higher than EMA model particularly for mode 1 and 3 which has large error. This maybe because the FE model has a high stiffness as well as a low mass since it was design based on several assumptions. The first assumption that the brackets were excluded from the model explains why the FE model is lighter than the actual model. Besides that, the blend radii and fillets that are not represented in the model in an effort to minimize geometric complexity have also contributed to the low mass model. Second assumption is the connections between longitudinal rail and cross members were considered perfect. This consideration represents in a correct way the welded joints. However in the actual model where the weld is not perfect, this consideration can make the model stiffer than the real system [8].

Based on problem stated above, *trial changes* to the FE model had been made by setting modulus young and mass density as the parametric changes to better represent the weld, and continue checking correlation to the test until acceptable levels are achieved. In this case, it needed until 60 iterations for the result to converge. In the updating process, the frequency correlation and the MAC correlation is improved at the same time by changing the modulus young and mass density. The modulus young of chassis had been reduced to 50% at the connection of cross member and longitudinal rail in order to represent the weld.

Normally, the first natural frequency of truck chassis experienced the torsional mode [1]. This mode is always defined as the critical mode as it easy to occurred and located near the working frequency. Sometimes, the torsional mode can cause fatigue failure due to the bumps or road irregularities. Thus as for that, structure modification is required in order to strengthen the truck chassis structure and at the same time reduce the torsional effect. In the structural modification analysis, the existing model was modified by adding stiffener to the chassis. The modified truck chassis has reduced the displacement for the torsion mode about 3.28%. At the same time it stiffen the chassis structure, thus increased its natural frequency. However, there will be some increased in weight. Therefore it can be concluded that to reduce vibration and deformation by torsional mode, the dominant natural frequency and mode shape have to be identified for effective stiffening.

8.0 CONCLUSION

The application of dynamic correlation technique was performed for verification of the finite element model of truck chassis. The experimental data was used to validate a finite element model representing the real structure. The result indicating that the FE model shows a good correlation with the experimental model for the mode shape but not for the natural frequencies as the FE model presented an average of 10% higher frequencies than the real chassis. This fact is due to the perfection of the model and the imperfection of the real structure. Thus, the model updating was performed to reduce this error by adjusting test modal parameters. The structural modification of truck chassis was useful to reduce a deflection by 1st torsion mode through placing the stiffener in the right position and right place.

REFERENCES

[1] R.R.P.Filho, J.C.C.Rezende, M.d.F.Leal, J.A.F.Borges, “Automotive Frame Optimization”, SAE Paper 2003-01-3702, Congresso SAE Brasil, 2003

[2] L.C.Ferraro, V.L.Veissid, C.A.M.Freitas, “A Study of Vibrational Behavior of Medium Truck Considering Frame Flexibility with the use of Adams”, International ADAMs User Conference, 1998

[3] Brady, R.N., “Heavy Duty Trucks: Powertrain, Systems and Service”, Prentice Hall, New Jersey, 1997

[4] R.A.Rahman, M.Zubair, N.Amin, “Finite Element Modeling, Correlation and Model Updating of Stiffened Plate”, Jurnal Mekanikal, 2003

[5] MSC.Nastran for Windows: Quick Start Guide, MSC.Software Corporation, United States of America, 2000

[6] N.M.M.Maia, J.M.M.Silva, J.He, N.A.John Lieven, G.William Skingle, R.M.Lin, W.M.To, A.P.V.Urgueira, “Theoretical and Experimental Modal Analysis”, Research Studies Press Ltd., Somerset, England, 1997

[7] LMS Virtual. Lab Rev 2, LMS International, Belgium

[8] J.Deweer, T.V.Langenhove, “Identification of the Best Modal Parameters and Strategies for FE Model Updating”, SAE Paper 2001-01-1439, SAE Technical Paper, 2001

[9] A.K. Ariffin, F.R.Romlay, M.J.M.Nor, “Model Updating of Center Member Bar”, The 3rd International Conference on Numerical Analysis in Engineering, Pulau Batam, Indonesia, 2003.

[10] FEMtools Software Version 2. 2.1.1 Dynamic Design Solution

[11] D.J.Ewins, “Modal Testing: Theory and Practice”, Research Studies Press Ltd., Hertfordshire, England, 1984.

[12] ME’sope VES: Demo Guide, Vibrant Technology Inc., Central of Amerika, 2003.