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DYNAMIC ANALYSIS, UPDATING AND MODIFICATION OF TRUCK CHASSIS

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ABSTRACT

Reducing the weight and production cost of chassis is considered to be one of the most important area of research in automotive manufacturing industry. However with lighter vehicle, the chassis is easily subjected to large vibration mainly coming from the engine and road irregularities. This paper looks into the application of dynamic analysis for verification of the complex FE model of truck chassis. The dynamic characteristic of truck chassis such as the natural frequency and mode shape is determined by finite element method. Experimental measurement by modal testing is then carried out to determine the accuracy of finite element analysis. Initial result indicated some discrepancies with regard to natural frequencies at various modes. However for mode shape analysis, both methods produced almost the same shape. As for getting a better agreement between both methods of analysis, modal updating had been carried out by adjusting parametric value of Modulus Young and mass density. It had been pointed out that the error was reduced to ± 2% through the model updating verified FE model has been obtained. Finally, the modifications of the verified FE truck chassis model has been suggested by consider adding the stiffener to reduce the vibration through chassis as well as improve the strength of structure.

Keywords: dynamic characteristic, finite element, modal testing, model updating.

INTRODUCTION

Chassis is a main component in a vehicle system particularly for off-road vehicle. It integrates the main truck component systems such as the suspension, engine, cab and trailer. The use of chassis in off-road vehicles has almost the same appearance since the models developed in 20 or 30 years ago. This indicates that the evolution of these structures is slow and stable along the years (Filho *et al.*

2003). Many researchers in automotive industry have taken this opportunity to be involved in the chassis manufacturing technology and development.

The current trend in truck design involves the reduction of costs and increase in transportation efficiency. The pursuit of these objectives results in development of lighter truck. Chassis is one of the parts that are strongly influenced by these guidelines (Ferraro *et al.* 1998). Lighter chassis gives a vehicle that has structural resonance within the range of typical rigid body vibrations of the truck subsystems. On the other hand, vibration can be formed due to the dynamic forces induced by the engine, transmission and road irregularities. 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 (Brady 1997).

Modes of vibration are used to characterize resonant vibration in the structure. It is important to determine the modes as all structures including chassis have natural modes which often create excessive noise and vibration levels and premature failures if excited. Each mode is defined by its natural frequency, damping and mode shape. At or near a natural frequency, the operating deflection shape of a structure is usually dominated by a mode (Richardson 1997).

This paper focuses on determining the dynamic characteristics of truck chassis by using finite element analysis (FEA) and experimental modal analysis (EMA). By using the dynamic correlation technique, the accuracy of finite element representation of truck chassis can be measured. Modal Assurance Criteria (MAC) was used to compare the vectors and then observations were made to identify potential improvements. As for getting a high degree of confidence in the finite element model, model updating was then performed.

MATHEMATICAL MODELLING

Finite Element Model

To obtain poles and frequencies from the finite element model, an eigensolution is performed on the mass and stiffness matrices. The equation of motion for a multiple degree of freedom system is written in matrix form as:

$$[M]\{\ddot{x}\}+[C](x)+[K]\{x\}=\{F(t)\} \tag{1}$$

where $[M]$ is mass matrix, $[C]$ is damping matrix, $[K]$ is stiffness matrix, $\{F(t)\}$ is forcing vector and {x} is vector of displacements.

The so-called normal mode eigensolution is obtained using only the mass and stiffness matrix and assumes that the damping is either zero or proportional.

$$[[K]-\lambda[M]]\{\varphi\}=0 \tag{2}$$

The eigensolution provides eigenvalues, λ which is also known as frequencies and eigenvectors, φ which is also known as mode shapes.

Experimental Modal Model

The formulation of an experimental modal model is well documented and need not be developed for this purpose. The general equation for the frequency response matrix in terms of modal parameters is defined as:

$$[H(j\omega)] = \sum_{k=1}^{m}\left[\frac{\{U_k\}.\{L_k\}}{(j\omega-\lambda_k)} + \frac{\{U_k\}^*.\{L_k\}^*}{(j\omega-\lambda_k)}\right] - \frac{[LR]}{\omega^2} + [UR] \tag{3}$$

where $[H(j\omega)]$ is frequency response matrix, m is number of modes in database, $\{U_k\}$ is mode shape vector for *k*th mode, $\{L_k\}$ is row vector for modal participation factors, $[LR]$ is lower residual term, $[UR]$ is upper residual term and λ_k is the complex pole value for *k*th mode which can be defined as:

$$\lambda_k = -(\xi_k \omega_{nk}) + j\omega_{nk}\sqrt{1-\xi^2{}_k} \tag{4}$$

where ξ_k is damping factor for mode *k* and ω_{nk} is natural frequency of mode *k*.

Correlation Technique

Many tools are available for the evaluation of the correlation between FEA and test. A brief overview is given here and a more detailed treatment is contained in the references (Deweer and Langenhove 2001). The MAC is a commonly used method for assessing the degree of correlation between any two vectors and is formulated as:

$$MAC = \frac{\left|\phi_E^T \phi_A\right|^2}{(\phi_E^T \phi_E)(\phi_A^T \phi_A)} \tag{5}$$

where ϕ_E is experimental mode shape and ϕ_A is analytical mode shape. The value of the MAC ranges from 0 to 1.

Finite Element Model Updating

The purpose of model updating is to adjust the valued of selected parameters such that a reference correlation coefficient is minimized.

$$\{R_e\} = \{R_a\} + [S](\{P_u\} - \{P_o\}) \tag{6}$$

where $\{R_e\}$ is vector containing the reference system response, $\{R_a\}$ is vector containing the predicted system response, $\{P_u\}$ is vector containing the updated parameter values, $\{P_o\}$ is vector containing the given state parameter values and $[S]$ is sensitivity matrix.

METHODOLOGY

Figure 1 shows the flowchart of research methodology used in this study to produce a verified FE model. Each of the methods illustrated is briefly described below. The detailed of the theoretical development is contained in the references (He & Fu 2001).

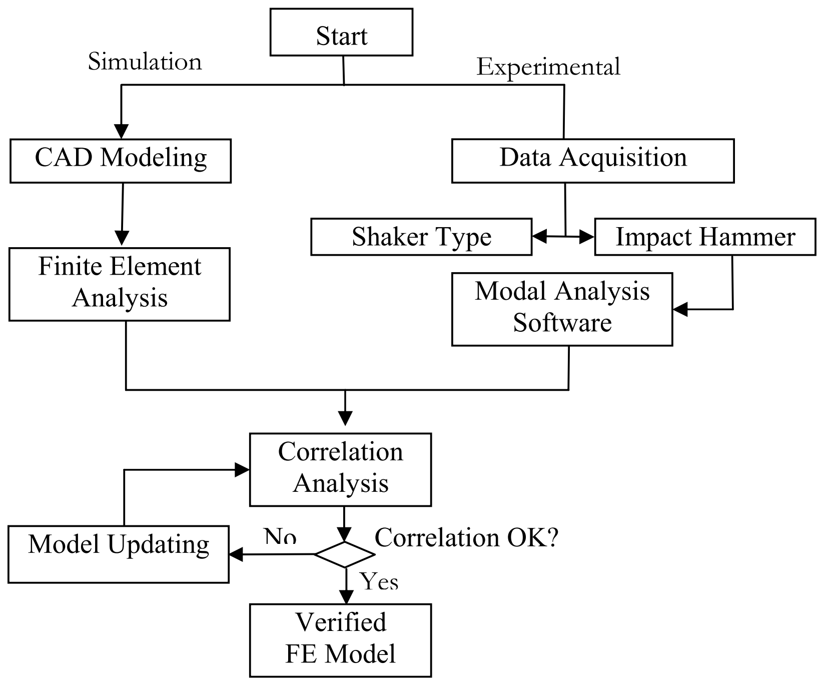

FIGURE 1 Research methodology flowchart.

FINITE ELEMENT MODELING

Figure 2 shows the complete finite element of the truck chassis model before meshing analysis. The 10-node tetrahedral element was chosen in the meshing analysis. Based on the previous finding (Rahman *et al.* 2003), it was found that this element gave a closer result to the actual condition. The final chassis model consists of 24,322 nodes and 12,087 elements and the material employed was steel. During the model construction, the following consideration had been taken into account so as to simplify the analysis:

i. All brackets were excluded from the model.
ii. The connections between longitudinal rail and cross members were considered perfect.

iii. The material was considered isotropic in its elastic phase.

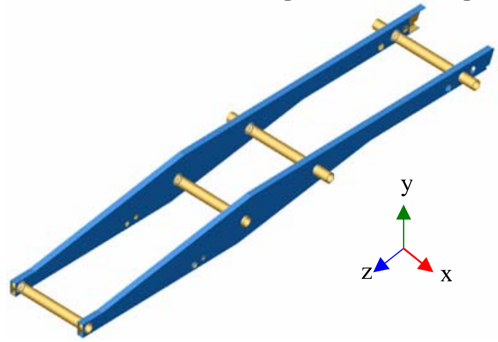

FIGURE 2 Meshing model of truck chassis.

Normal Mode Analysis

In the prediction of analytical dynamic characteristics of truck chassis, the normal modes analysis has been performed using commercial FEM software. 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 to stimulate this free-free boundary condition.

The frequency range of interest was set between 10 to 200 Hz. The reason for setting the starting frequency at 10 Hz is to avoid the solver from calculating rigid body motions which have the frequency of 0 Hz. Under the study, only the next four fundamental frequencies were observed, as these frequencies were critical to the truck chassis dynamic behaviour. Figure 3 to 6 show the typical mode shape of the truck chassis at 43.7, 64.8, 99.1 and 162.3 Hz where the chassis experienced 1st torsion mode, 1st bending mode, 2nd torsion mode and 2nd bending mode respectively. The results show that the chassis experience a global vibration as the whole structure follow to vibrate. The contour illustrated the translation value where it is unitless or without any unit since any force is applied in normal mode analysis.

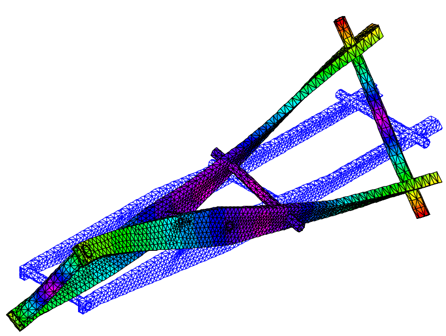

FIGURE 3 FEA 1st mode at 43.7Hz.

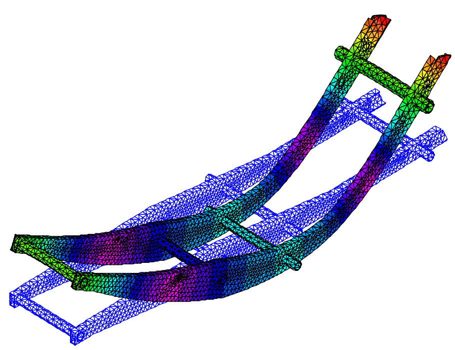

FIGURE 4 FEA 2nd mode at 64.8Hz.

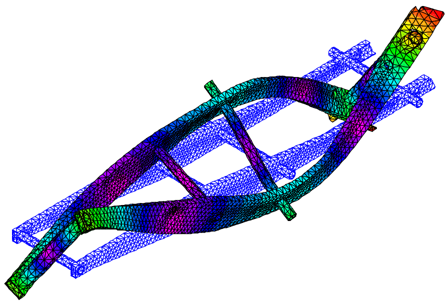

FIGURE 5 FEA 3rd mode at 99.1Hz.

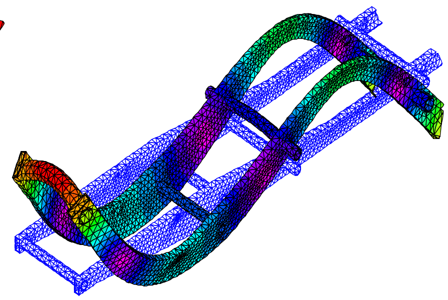

FIGURE 6 FEA 4th mode at 162.3Hz.

EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis is a method where model parameters such as natural frequency, mode shape and damping ratio were extracted from the structures experimentally. For this case, the chassis was divided into 22 grid points where at these points, Frequency Response Functions (FRF) were measured in the range of 0-200 Hz to identify the modal characteristics. This 22 grid points were chosen to give adequate spatial resolution to describe the global structural mode shapes.

Two excitation methods were implemented in the experimental test. The first testing was done with a fixed input location (in y-direction), with uniaxial accelerometers moved from point to point on the structure. This test is known as a shaker test. Figure 7 shows the experimental setup for shaker test. The boundary conditions were similar to the FEM model where the free-free boundary condition was applied. There are some significant effects when using this method such as the locations of the accelerometer could affect the dynamics of the structure (Maia *et al.* 1997). This is referred to as "mass loading". The change of modal frequencies values which depend on the location of the accelerometers make this method unacceptable.

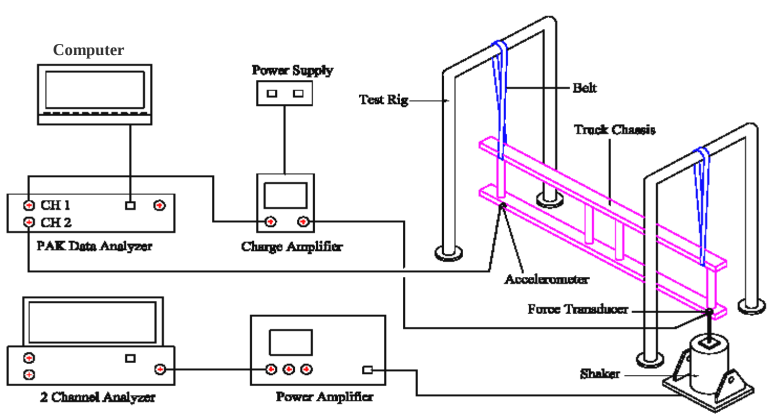

FIGURE 7 An experimental set-up for shaker test.

The second tests known as impact hammer test was performed by connecting the uniaxial accelerometer to a reference point and exciting the structure at all other points with the modal impact hammer. This method provided better results for the case of negligible mass loading. Figures 8 and 9 show the superimposed FRF at all points for both experimental methods.

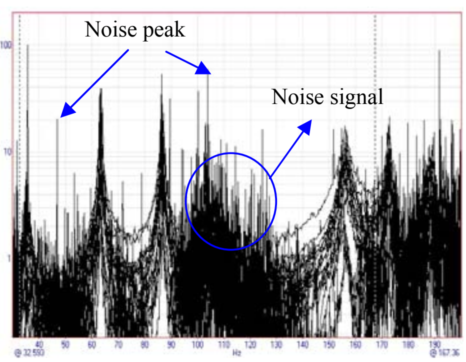

FIGURE 8 FRFs by shaker test.

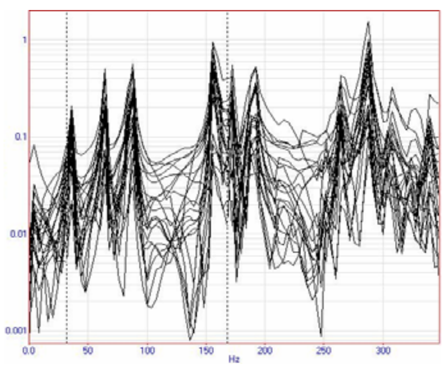

FIGURE 9 FRFs by impact hammer.

Table 1 tabulates a list of frequencies modes of the truck chassis extracted from both finite element model and experimental test. From the theoretical standpoint, each test mode frequency should match with each finite element frequency. However, in this case all the mode frequency obtained from the test is not equivalent with each of FE mode frequency. It is noted that each FEA frequency is slightly higher than its matching tests frequency, indicating that the stiffness of the FE model is greater than the real structure. Meanwhile, the typical mode shapes for the first four modes of truck chassis obtained by impact hammer describes the same shape obtained by shaker test as well as in the FEA.

TABLE 1 Natural frequencies obtained by EMA and FEA.

Mode	Impact Hammer		Shaker		FEA modes Freq. (Hz)
	Natural Freq. (Hz)	Damping (%)	Natural Freq. (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

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 may be attributed to the possibilities error in experimental data such as the measurements could have been carried out at an imperfect set-up, noise in the data, the existence of inherent model parameter errors and also the model structure errors (Ariffin *et al.* 2003).

By using FEMtools software, the correlation analysis was carried out in three steps. Firstly, a geometric correlation was performed. The test geometry matched perfectly with the FE model. Thus at this point, a node pair table can be created instantly where no translation and rotation values were needed. The test modes were then transformed to the FE model geometry using the previous created node pair table. At this stage, only the real measured degrees of freedom (DOFs) of the truck chassis were selected to continue the correlation analysis. Lastly, a MAC matrix was performed and the result would tell how good the FE modes correlate with the test modes. The high MAC values (> 75 %) would show which FE mode shapes resemble to which test mode shapes.

Table 2 shows a comparison of natural frequencies between FEA and EMA model and also the MAC value where the FEA frequency for mode 1 and 3 show a slightly larger error than its matching tests frequency. For a mode shape correlation, it was observed that the first 3 modes have the MAC value above 95 % which indicate that the test and FEA shapes were very similar. The fourth pair of modes had a MAC value above 90 %, which still indicated that the shapes were similar.

TABLE 2 Mode pairs with frequency difference.

Mode	FEA modes Freq. (Hz)	EMA modes Freq.(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

MODEL UPDATING

In order to bring the FE model into a better agreement with the experimental data, the model updating analysis was needed. It is an important step in validation process that modify the values of parameters in FE model in order to create a reliable finite element model suitable for the further analysis (Deweer and Langenhove 2001). At this stage, 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 to decide the parameters in FE model that have significant influence to the change of the modal properties of truck chassis. After several iterations by sensitivity analysis, the following parameters were selected for finite element model updating which are the dynamic modulus of truck chassis, E and the mass density of the truck chassis, ρ.

These test modal parameters were used as reference data in the model updating analysis. 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 finite element analysis and experimental modal analysis mode shapes was again quantified based on Modal Assurance Criterion (MAC). Table 3 shows a comparison between the natural frequencies from the first FE model, the updated FE model and the experimental results. It can be seen that the updated FE model shows a better results where the error between FE model and experimental was reduced within ± 2%.

TABLE 3 Comparison between natural frequencies before and after model updating.

Mode	EMA Freq. (Hz)	First FE Freq. (Hz)	First FE Error (%)	Updated FE Freq. (Hz)	Updated FE 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

From Table 4, for the mode shape comparison, it is noticed that the model updating did not significantly improve the values of MAC. There was a small increase for the first mode but a decrease for modes 2, 3 and 4. This may be due to several factors. One of the reasons is the experimental mode shape obtained was only in one degree of freedom since the accelerometer used was a single axial. In the FE model, the mode shapes obtained was based on three degrees of freedom calculation. 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 (Deweer and Langenhove 2001).

TABLE 4 MAC diagonal values before and after model updating.

	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

Figure 10 and 11 illustrated the parameters E and ρ that were updated. The results of model-based updating showed the dynamic modulus of welds (the connection between cross member and longitudinal rail) in the FE model truck chassis for structural elements had reduced as much as 50% which has the nominal values between 78 to 80 GPa. It also found that the mass density increased locally between 2.0×10^4 to 2.50×10^4 kg/m^3 or approximately by 100% from the initial values.

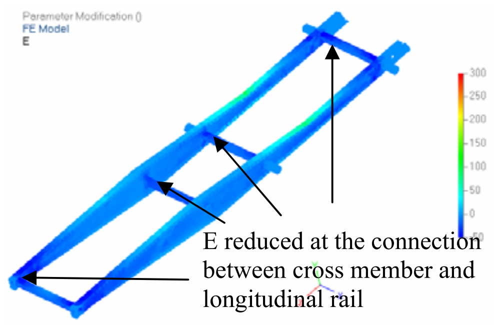

FIGURE 10 Parameter E changes.

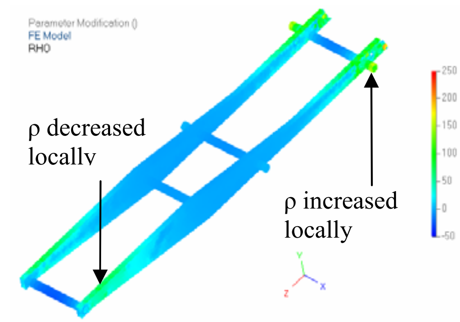

FIGURE 11 Parameter ρ changes.

STRUCTURAL MODIFICATION

After the model updating analysis completed, the FE model were then transfered to the FE software for further analysis in the structural modification. At this stage, the FE truck chassis model obtained can be assumed for represent the real chassis structure. Thus, any modification on the FE model will give an approximately the same result as to real structure. The additional cross member with diameter 80 mm and thickness 10 mm was added at the rear of truck chassis and the center cross member was replaced with K-member as shown in Figure 12. 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 exerted by the torsional loads.

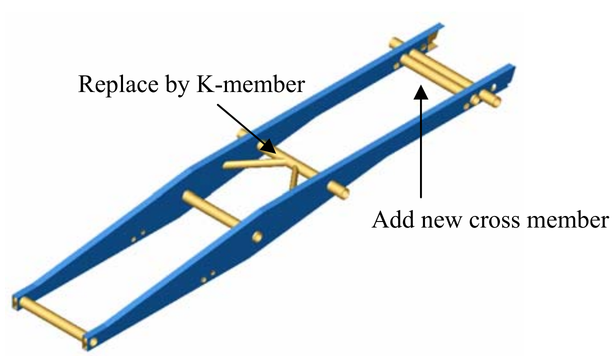

FIGURE 12 Modification on truck chassis by adding stiffener.

The only first mode of truck chassis which experienced torsion was analyzed. This is because 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 in order to shift the natural frequency away from the operating frequency range and at the same time minimize the torsional displacement. Table 5 shows the result of displacement and natural frequency of the first mode before and after the modification analysis. As observed from the table, the result shows that the modification of truck chassis has successfully minimized the displacement due to torsional vibration by 3.28%. This result indicates reduction of displacement after modification particularly at rear part of truck chassis.

TABLE 5 Maximum displacements of chassis in the first mode.

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

DISCUSSION

Generally, preliminary FE modelling is necessary to ensure a better understanding of the behavior of truck chassis as well as to aid the selection of a reasonable test grid and digital data acquisition parameters in modal testing. Initial results from the FEA found that none of the fundamental natural frequencies obtained was within the frequency range of operating condition.

In the experimental modal analysis, some of the problems were encountered particularly with reference to mass loading or known as a shaker test. Although the chassis structures were relatively heavy compared to the mass of the accelerometer, mass loading was still significant especially for modes with high participation from local areas (Ewins 1984). However, these conditions normally happened in the higher modes of excitation. The first four mode shapes as discussed earlier was not affected by the local vibration. Somehow these difficulties can be overcome by using the roving impact hammer method. Besides that, there are other problems encountered during the FRF measurement. It is noticed that, the shaker test produced an unwanted portion or noise signal in the FRFs plot which is shown in Figure 8. This occurred due to the inability of the shaker to excite the chassis properly close to supporting belt, particularly around the center of the chassis and near the cross member area.

In the correlation analysis, the first 4 modes have MAC value above 0.90 indicating that the test and FEA shapes are similar. The result shows that the natural frequency of FEA is higher than EMA model particularly for mode 1 and 3 where they show a large error. This could be due to the FE model having a high stiffness as well as low mass as it was designed based on several assumptions. First assumption is that the brackets were excluded from the model which 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 that 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.

Based on the problems stated above, series of trial changes to the FE model had been made by setting Modulus Young and mass density as the parametric changes. The correlation to the test was continuously checking until acceptable levels were achieved. In this case, 60 iterations were needed for the result to converge. The frequency correlation and the MAC correlation were improved by changing the Modulus Young and mass density. The Modulus Young of chassis was reduced to 50% at the connection of cross member and longitudinal rail in order to represent the weld. So at the end of this stage, the verified FE truck chassis model was obtained. Therefore, any dynamic analysis such as force vibration analysis or torsional analysis on the verified FE model will give an approximately the same result as to real structure.

In the structural modification analysis, the existing model has been modified by adding the stiffener to the chassis. The modified truck chassis has reduced the displacement in the torsion mode about 3.28%. At the same time it stiffens the chassis structure and increased its natural frequency. This is due to the deformations associated with this mode are prevented by the stiffener and hence resulted the increased in the natural frequency. Therefore it can be said that to reduce vibration and deformation by torsional mode, the dominant natural frequency and mode shape have to be identified for effective stiffening.

CONCLUSIONS

In conclusion, the dynamic characteristic of truck chassis could be determined using FE analysis when the right element and method were used. However, due to model complexity, large error could be expected. Therefore, for the model to be useful, it need to be verified using EMA. In the dynamic analysis, the FE model proved to have a strong correlation with the EMA in the mode shape. However, for natural frequencies, the FE model presented an average of 11% higher frequencies than the real chassis. This could be attributed to some assumption made in the FE model. For the case of low level of correlation, the model updating could be performed by adjusting the selected test parameters. In this case, the selected test parameters were Modulus Young and mass density which were considered the important variables for welded joints in the modeling. Through model updating analysis, the error between both results has been reduced to ± 2%.

REFERENCES

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

Brady, R.N. 1997. *Heavy Duty Trucks: Powertrain, Systems and Service*. Prentice Hall, New Jersey.

Deweer, J. & Langenhove, T.V. 2001. Identification of the Best Modal Parameters and Strategies for FE Model Updating. *SAE Technical Paper*, SAE Paper 2001-01-1439.

Ewins, D.J. 1984. *Modal Testing: Theory and Practice*. Research Studies Press Ltd., Hertfordshire, England.

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

Filho, R.R.P., Rezende, J.C.C., Leal, M.d.F. & Borges, J.A.F. 2003. Automotive Frame Optimization. *Congresso SAE Brasil,* SAE Paper 2003-01-3702.

He, Jimin & Fu, Zhi-Fang. 2001. *Modal Analysis*. Butterworth Heinemann Publication, Oxford, England.

Maia, N.M.M., Silva, J.M.M., He, J., Lieven, N.A.J., Skingle, G.W., Lin, R.M., To, W.M. & Urgueira, A.P.V. 1997. *Theoretical and Experimental Modal Analysis*. Research Studies Press Ltd., Somerset, England.

Rahman, R.A., Zubair, M. & Amin, N. 2003. Finite Element Modeling, Correlation and Model Updating of Stiffened Plate. *Jurnal Mekanikal*.

Richardson, M.H. 1997. Is It a Mode Shape, or an Operating Deflection Shape. *Sound and Vibration Magazine 30th Anniversary Issue*.