# Vibration Induced Fatigue Assessment in Vehicle Development Process

Fatih Kagnici

***Abstract***—Improvement in CAE methods has an important role for shortening of the vehicle product development time. It is provided that validation of the design and improvements in terms of durability can be done without hardware prototype production. In recent years, several different methods have been developed in order to investigate fatigue damage of the vehicle. The intended goal among these methods is prediction of fatigue damage in a short time with reduced costs. This study developed a new fatigue damage prediction method in the automotive sector using power spectrum densities of accelerations. This study also confirmed that the weak region in vehicle can be easily detected with the method developed in this study which results were compared with conventional method.

***Keywords***—Fatigue damage, Power spectrum density, Vibration induced fatigue, Vehicle development

## I. INTRODUCTION

THROUGH the development of CAE studies in automotive industry, engineers can easily validate their design in the early phase of the vehicle product development cycle. Especially, improvement in the accuracy of correlation between FEA and test results provides replacing the definition of hardware prototype to analytical prototype. Durability is one of the main issues in vehicle design. Design engineers have to estimate service life of vehicle in order to overcome failure due to fatigue damage.

In the industry, there are several ways to predict fatigue damage of vehicles. Both advantages and disadvantages are listed in Figure 1. Design engineers should make a trade-off amongst these techniques in terms of time, cost and accuracy.

| | Type | Load Input | Stress Analysis | Fatigue Analysis | Advantages | Disadvantages |
|---|---|---|---|---|---|---|
| Multi-body Dynamic Simulation | Time domain | Road profile or measured accelerations | NA | NA | May be used to obtain force inputs for FEA from measured accelerations or road profiles | Complex tyre models |
| Remote Parameter Analysis | Quasi-static, time domain | Straingauge measurements | Static FEA | Rainflow counting + various fatigue life analysis methods | Can use remote measured straingauge data, economic FEA | Not suitable for complex dynamic response, rainflow on each stress point, loading results not suitable for code = not design independent |
| Co-variance method | Quasi-static, frequency domain | Measured /simulated input forces | Static FEA | Dirlik formula + various fatigue life analysis methods | Takes account of complex dynamic response, economic FEA | Requires stationary random input data, Dirlik formula approximations, loading not design independent |
| Random Vibration | Dynamic, frequency domain | Measured /simulated input forces | Eigen value FEA | Dirlik formula + various fatigue life analysis methods | Takes account of complex dynamic response, economic FEA | Requires stationary random input data, forces must be measured, Dirlik formula approximations, loading not design independent |
| Fatigue Domain Reponse Spectrum | Dynamic, fatigue/ frequency domain | Measured accelerations | Eigen value FEA | HdM cycle counting + Stress Life | Takes account of complex dynamic response, economic FEA, loading is design independent | Requires stationary random input data |
| Modal Superposition | Dynamic, time or frequency domain | Measured /simulated input forces | Eigen value FEA | Dirlik formula + various fatigue life analysis methods | Takes account of complex dynamic response, economic FEA | Forces must be measured, Dirlik formula approximations, loading not design independent |
| Direct Integration with Large Mass, Relative Inertial, La Grange Multiplier | Dynamic, time domain | Measured accelerations | Dynamic FEA | Rainflow counting + various fatigue life analysis methods | Takes account of complex and transient dynamic response, accelerations may be measured, can be design independent | Expensive FEA |

Fig. 1 Comparison of Fatigue Analysis Methods [1]

F. K. Author is with the Hexagon Studio Design and Engineering Company, Kocaeli, Turkey; (e-mail: fatih.kagnici@ hexagonstudio.com.tr).

Many studies have been carried out in order to predict vehicle body fatigue. The most popular technique is the hybrid superposition technique due to using dynamic model of vehicle in multi body simulation and Craig-Bampton method for modal and static superposition [2]. Although, this technique presents better results in order to predict damage, it has disadvantages in terms of cost and time. On the other hand, it is necessary for establishing a MBS model and full vehicle body finite element model for hybrid superposition. Then, the technique can be predictive for fatigue damage. General work-flow of that study is schemed in Figure 2.
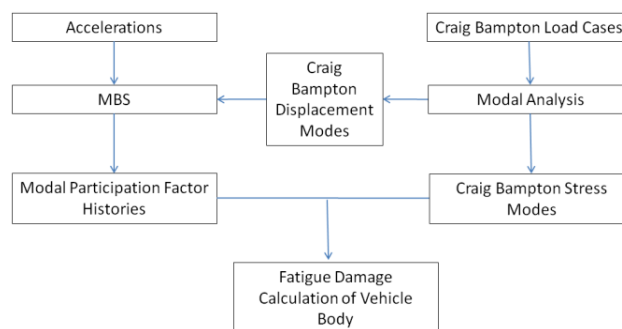


Fig. 2 General work flow of conventional fatigue damage calculation of a vehicle body [3]

In this study, a new fatigue damage calculation methodology will be presented in order to predict the fatigue damage of full vehicle body. Power spectral densities of the system and frequency response function of the finite element model are the inputs for random vibration induced fatigue. The most important advantage of this method using in vehicle product development are cost and time reduction because it removes multi-body simulation out of the workflow.

## II. THEORY OF VIBRATION INDUCED FATIGUE

It is often easier to obtain a PSD of stress rather than a time history due to using finite element based in the frequency domain [4]. It can be applicable if the system is linear and the loading is random, stationary and Gaussian. Many different methods have been studied in vibration fatigue calculation. According to Bishop [5], Dirlik method was found to be robust method among the other techniques.

Another study was performed by Qiang and Hongyan [6]. They compared the frequency based fatigue techniques against time domain fatigue. In addition, they recommended using Dirlik's method for the prediction of fatigue damage

World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:6, No:4, 2012

assessment of vehicle components.

In 1985 Dirlik stated a different solution using the Monte Carlo technique [7]. Dirlik's empirical formula has been shown to be effective in terms of accuracy [8].

The formula is given below.

$m_0$, $m_1$, $m_2$ and $m_4$ are the moment of areas of PSD.

$$N(S) = E[P] \cdot T \cdot p(S) \tag{1}$$

Where,

$S$, stress cycles of range (N/mm$^2$)

$E[P]$, Expected number of peaks

$$E[P] = \sqrt{\frac{m_4}{m_2}} \tag{2}$$

T, Period time (sec)

$$p(S) = \frac{\dfrac{D_1}{Q} \cdot e^{\frac{-Z}{Q}} + \dfrac{D_2 \cdot Z}{R^2} \cdot e^{\frac{-Z^2}{2 \cdot R^2}} + D_3 \cdot Z \cdot e^{\frac{-Z^2}{2}}}{2 \cdot \sqrt{m_0}} \tag{3}$$

$$D_1 = \frac{2 \cdot (x_m - \gamma^2)}{(1 + \gamma^2)} \qquad D_2 = \frac{1 - \gamma - D_1 - D_1^2}{(1 - R)} \tag{4}$$

$$D_3 = 1 - D_1 - D_2 \qquad \gamma = \frac{m_2}{\sqrt{m_0 \cdot m_4}} \qquad x_m = \frac{m_1}{m_0} \cdot \sqrt{\frac{m_2}{m_4}}$$

$$Q = \frac{1.25 \cdot (\gamma - D_3 - D_2 \cdot R)}{D_1} \tag{5}$$

$$R = \frac{\gamma - x_m - D_1^2}{1 - \gamma - D_1 + D_1^2} \qquad Z = \frac{S}{2 \cdot \sqrt{m_0}}$$

The damage is calculated using Palmgren-Miner Rule [6]:

$$D = \sum_i \frac{n_i}{N(S_i)} = \frac{S_t}{K} \int S^m P(S) dS \tag{6}$$

Where,

$n_i$, the actual counted number of cycles

$S_t$, total number of cycles in required time

## III. VIBRATION INDUCED FATIGUE ASSESSMENT IN VEHICLE BODY

Vehicle is subjected to random excitation from the wheels on the road. In other words, random acceleration signals occur in a period. In addition, these signals are linear and commonly Gaussian which are vital criteria for the vibration induced fatigue. With the aid of the FFT, PSD of these signals can be obtained.

Although, there have been different studies with vibration induced fatigue, application to vehicle body was unique which is stated by Wang X. et al. [9] in 2010. They applied the PSD function of the accelerations which are collected from shock towers.

In this study, a new designed minibus vehicle is used for the investigation of the vibration induced fatigue. Firstly, accelerations of the wheel hubs were collected. Then, frequency response analysis was performed in order to get modal stresses. Finally, fatigue damage analysis was performed with these inputs.

### A. Acceleration Data Acquisition from the Vehicle

In order to catch the dynamic behavior of the new designed vehicle, acceleration in Z direction data was collected from previous version of the minibus on the defined track. (Figure 3)



Fig. 3 Instrumented test vehicle

In this process, track length was also evaluated. Data acquisition properties of the test vehicle and track are listed in Table 1.

TABLE I
DATA ACQUISITION PROPERTIES

| Track Type | Duration (Min) | Length (Km) | Speed (Km/h) |
|---|---|---|---|
| Asphalt | 4 | 3 | 70 |
| Damaged Asphalt | 6 | 5 | 50 |
| Country Road | 12 | 9 | 50 |
| Autobahn | 10 | 15 | 100 |
| Bump | 4 | 1.2 | 20 |
| Pave | 7 | 1.3 | 40 |

In order to simulate the durability behavior, the short representative samples were extracted from collected accelerations. In total, signals were processed in 60 seconds duration to establish a duty cycle event for the fatigue damage prediction. The signals are shown in Figure 4.

Here, the first graph shows the right front wheel hub accelerations in Z direction. Second graph shows the left front wheel hub accelerations in Z direction. Third graph shows the right rear wheel hub accelerations in Z direction. Final graph shows the left rear wheel hub accelerations in Z direction.

X Axis represents the duration in seconds. Y axis represents the accelerations in g units.

World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
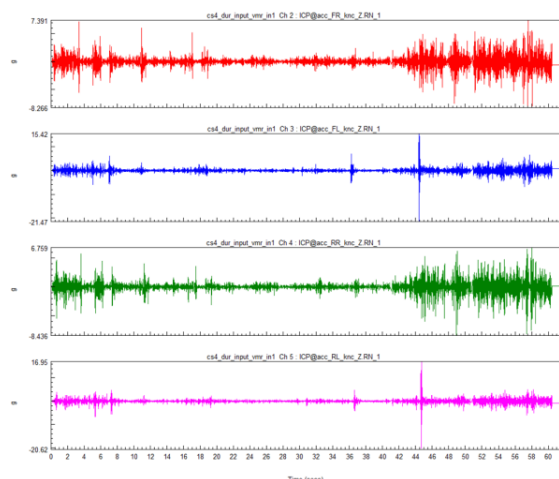Vol:6, No:4, 2012

Fig. 4 Wheel hub accelerations vs time

### B. Finite Element Model and Frequency Response Analysis

HYPERMESH software was used to establish finite element model of the vehicle which includes vehicle body, front sub-frame, rear and front suspensions. Power-train, baggage area, seats, passengers and the other parts were modeled as mass elements. (Figure 5)
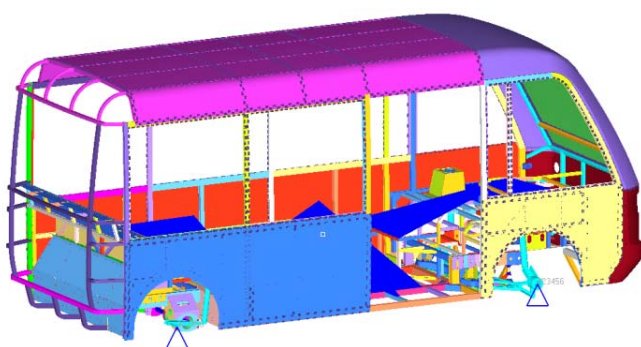


Fig. 5 Finite element model of vehicle body

Frames and sheet metals were modeled as shell QUAD elements. In addition, some components of the rear axle were modeled as solid HEXA elements and welds were modeled as nugget elements.

Moreover, spring and dampers were modeled as BEAM elements. Connection between suspensions to vehicle body was modeled with rigid RBE2 element.

Mass of the other non-modeled components has been added to the center of mass of the components.

The modal frequency response analysis was performed with unit acceleration input for four-wheel excitation in order to extract modal stresses of vehicle body by RADIOSS software.

The constraint points of the loading are the hub-suspension connections which are shown in Figure 6. In addition, unit acceleration excitations are also applied these points only in Z direction.
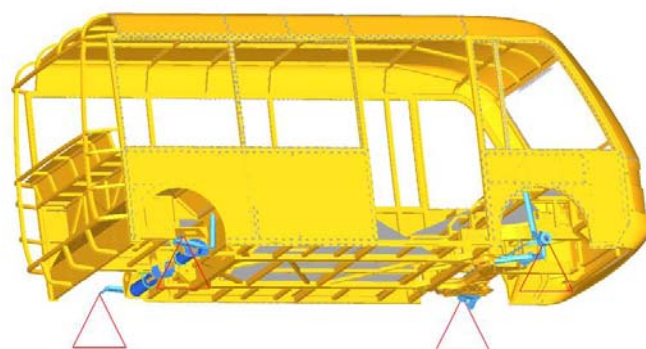


Fig. 6 Constraint and excitation points

Many studies have been performed for defining frequency interval in order to obtain better result for applicable result file size. An optimized result was obtained for the value of frequency interval is 0.5 Hz. Moreover, frequency range of excitation distributed from 0.5 to 100 Hz because excitation signals from the road oscillates the vehicle in the range from 0 to 45 Hz [9].

### C. Fatigue Damage Analysis of the Vehicle

Vibration fatigue induced damage analysis was performed with the inputs which are result file of the frequency response analysis and PSD of acceleration signals by the software NCODE Design Life. PSD functions are obtained with converting of the acceleration signals in time domain to frequency domain via Fast Fourier Transform (FFT) which are shown in Figure 7. The order of the rows in the figure is same as Figure 4. In these graphs, X axis represents the frequency in Hz and Y axis represents the PSD $((mm/s^2)^2 /Hz)$
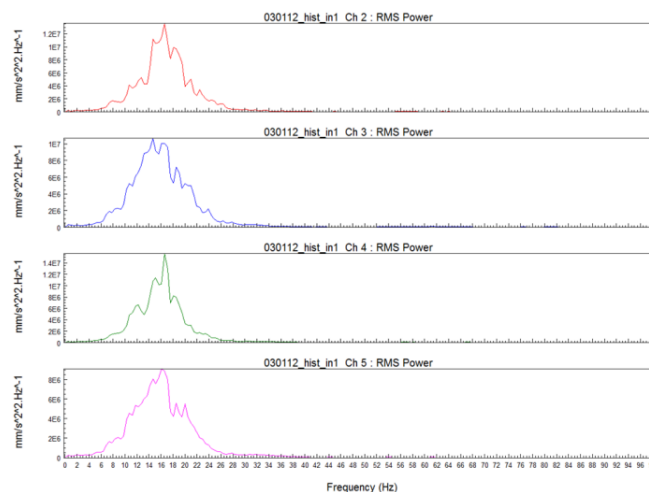


Fig. 7 PSD of accelerations

Fatigue material which was selected for the whole vehicle, was generated by the software NCODE whose ultimate tensile strength has 300 Mpa in order to investigate worst case in terms of fatigue damage. The S-N curve of the material is shown in Figure 8.
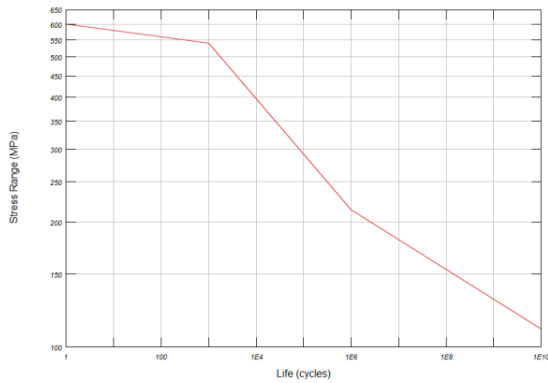
World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:6, No:4, 2012

Fig. 8 S-N Curve of the fatigue material

## IV. RESULTS

In this study, fatigue damage results of the vehicle are compared between two methodologies which are vibration induced fatigue methodology and the Craig-Bampton method with MBS fatigue calculation methodology mentioned in the introduction part.

### A. Fatigue Damage Analysis of the Vehicle via Craig-Bampton Method with MBS

For calculating the fatigue damage of vehicle via Craig-Bampton method with MBS , full vehicle was modeled in the software ADAMS/Car and Craig-Bampton modes were extracted by the software RADIOSS. Then, the time histories of the deformation for Craig-Bampton modes and modal stresses were processed in the software NCODE Design life in order to calculate fatigue damage. This method is frequently used by automotive manufacturers. The main advantages of this method are taking into account of non-linear suspension characteristics with MBS and easy calculation of the fatigue life because of having deformation time histories of the vehicle. On the other hand, the disadvantages are listed below.
- Necessity of having the MBS software
- Necessity of having ability of using the MBS software
- Taking lots of time for establishing the MBS model
- High costs of using the MBS software

Establishment time of the MBS model was approximately 5 days. In addition, it consumed 2 days for solving problems of the software while it was simulating and 1 day for simulating. Also, one colleague worked for it during these 8 days in total. So, it costs overall 8 days of staff salary who is qualified for MBS and software license price.

### B. Comparative Results between two Fatigue Damage Analysis Methods of the Vehicle

Here are the comparative results of between 2 fatigue damage analysis methods which are used in vehicles. Fatigue damage results of panels between two methods are shown in Figure 9. The upside figure belongs vibration induced fatigue damage analysis results and the other one belongs the Craig-Bampton with MBS fatigue damage analysis results.
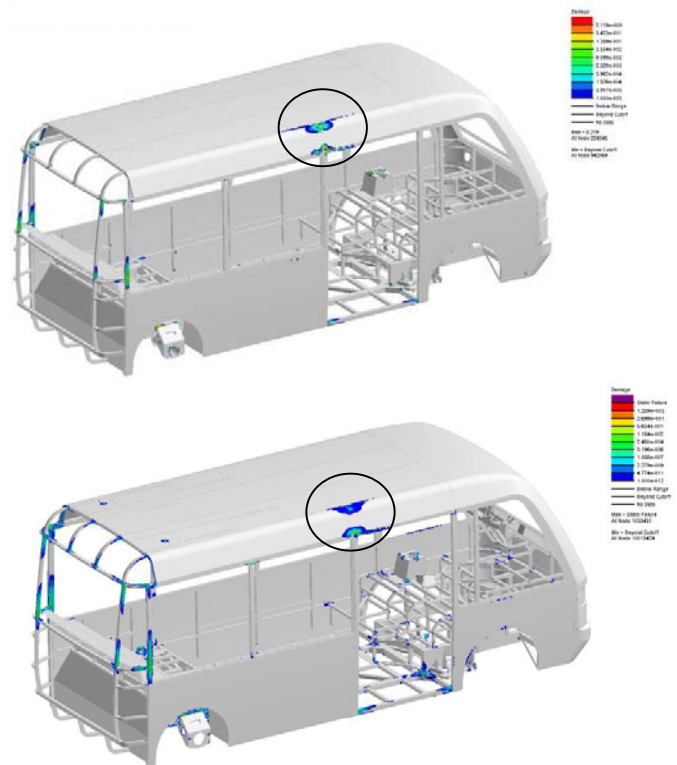


Fig. 9 Comparative fatigue damage results of the vehicle panels

According to Figure 9, panels at the top of middle doors side are found critical region for fatigue damage analysis results. In other words, the results indicate the weak region mainly concentrated around the panels which are at the upper side of the middle doors. Vibration induced fatigue analysis results were found to be similar with the results of the MBS method in vehicle panels within the values of $1e^{-10}$ to 10

Figure 10 shows the comparison of fatigue damage results in the carcass structure of vehicle. Results have some differences depending on the loading of the occupants. According to the vibration induced fatigue damage results, weak regions are indicated mainly in the middle sides of the frames due to finite element modeling of occupants mass in the vehicle. On the other hand, main weak regions according to results of Craig-Bampton method with MBS are found at the back side of the carcass structure because of modeling of the occupants in the multi body simulation. However, some differences can be figured between two methods for the carcass structure, the results can be correlated with modifying the modeling between finite element models and multi body simulation models.
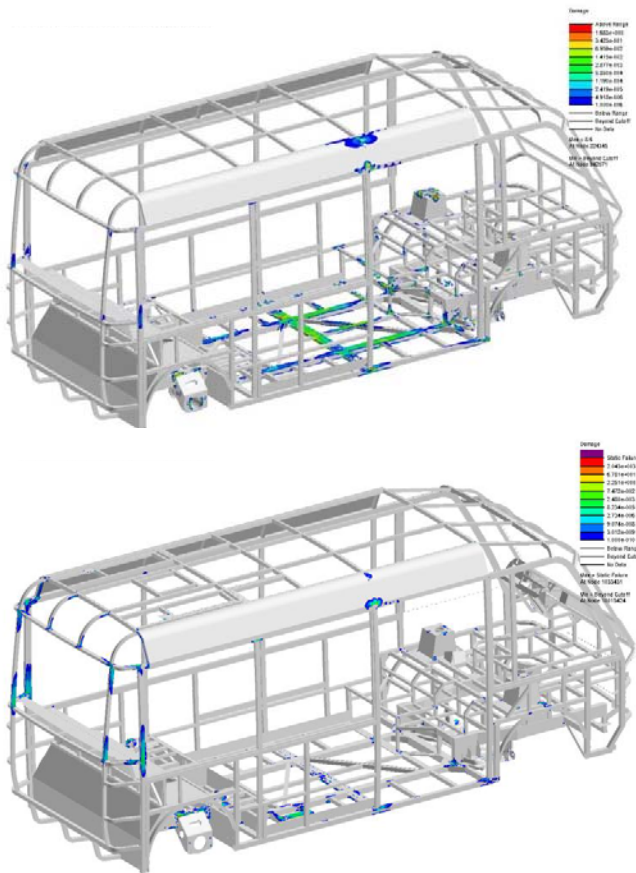
World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:6, No:4, 2012

The same situation is valid for the sub-frame structure. (Figure 12) According to result of vibration induced fatigue damage analysis, the weak regions are seen at the edge of sub-frames. On the other hand, the weak regions are found mostly on the other side of sub-frame according to Craig-Bampton with MBS method.



Fig. 12 Comparative fatigue damage results of the sub-frame

The best correlation between the fatigue damage results of two methods can be seen in the rear suspension brackets in Figure 13. The left upper side of the bracket is found to be weak region for both fatigue damage analysis methods results. It is very significant as it can be seen from the figures. Also, this region is the most damaged area in total fatigue damage assessment of whole vehicle according to Craig-Bampton with MBS fatigue damage analysis results. In addition, this area has high fatigue damage values according to vibration induced fatigue damage analysis results.



Fig. 10 Comparative fatigue damage results of the vehicle carcass structure

As it seen from the Figure 9 and 10, the weak regions inside vehicle are shock tower region, sub-frame component and rear suspension connections.

Figure 11 shows the comparison of the fatigue damage analysis for the shock tower. The right one refers the results of vibration induced fatigue damage analysis and the left one refers the MBS method.
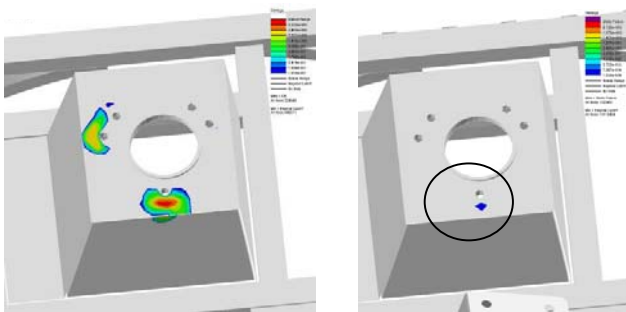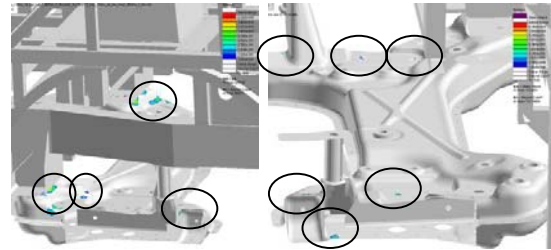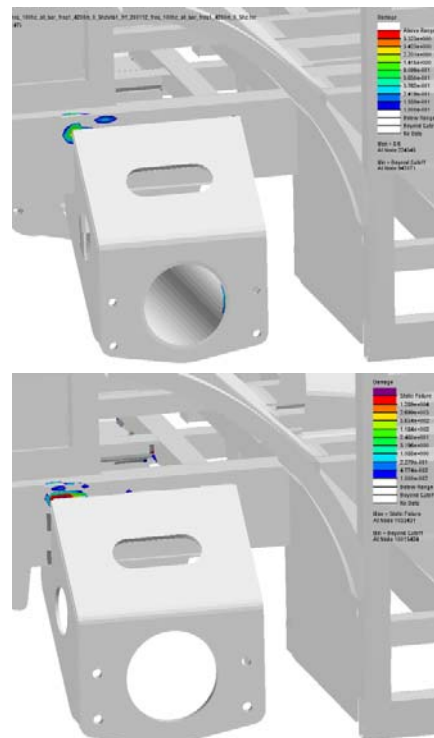


Fig. 11 Comparative fatigue damage results of the shock towers

At the same scale for damage results within the values of 1 $e^{-2}$ to 10, the weak regions are more significant in the result of vibration induced fatigue damage analysis. According to result of the Craig-Bampton with MBS method, weak region seems to be a single element.



Fig. 13 Comparative fatigue damage results of the rear suspension brackets

World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:6, No:4, 2012

## V. CONCLUSION

In this study a new methodology was developed in order to predict the fatigue damage of vehicle and compared with the conventional method which is used frequently in automotive sector.

According to comparison of results, the weak regions in the vehicle were easily detected with both methods. The main structure of the new method is schemed in Figure 14.
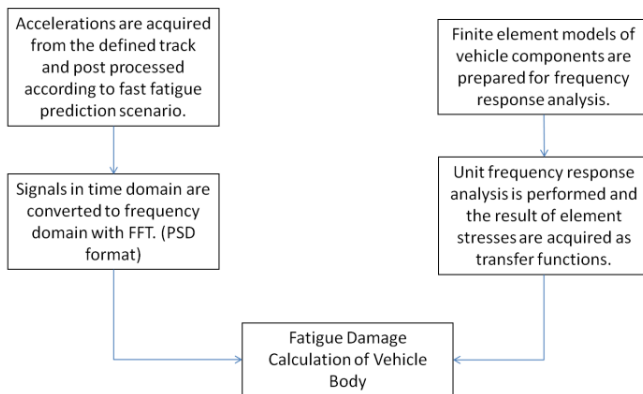


Fig. 14 Vibration induced fatigue damage calculation methodology in vehicle product development

Main advantages of the new method are time and cost reduction which are very important in automotive product development process.

For the future works, the headings listed below can be studied;

- PSD from 3 axis accelerations signals
- Comparing the results with vehicle durability test in proving ground
- Improvement in finite element models according to test results

### ACKNOWLEDGMENT

### REFERENCES

[1] Wannenburg, J., *A Study of Fatigue Loading on Automotive and Transport Structures,* Doctorate Thesis in Mechanical Engineering, University of Pretoria, Pretoria, August 2007.
[2] Fischer, P., Witteveen, W., Schabasser, M., "Integrated MBS-FE-Durability Analysis of Truck Frame Components by Modal Stress", *ADAMS User Meeting*, Rom, 2000
[3] Kim, H. S., Yim, H. J., Kim, C. B., "Computational Durability Prediction of Body Structures in Prototype Vehicles", *International Journal of Automotive Technology*, Vol. 3, 2002, pp.129-135.
[4] A. Halfpenny, "A Frequency Domain Approach for Fatigue Life Estimation from Finite Element Analysis", *International Conference on Damage Assesment of Structures*, Dublin, 1999.
[5] Bishop NWM, Hu Z, Wang R, Quarton D., "Methods for Rapid Evaluation of Fatigue Damage on the Howden HWP330 Wind Turbine", *British Wind Energy Conference*, York, 1993.
[6] Qiang R., Hongyan W., "Frequency Domain Fatigue Assessment of Vehicle Component under Random Load Spectrum", *J. Phys.: Conf. Ser.* vol. 305, no. 012060, 2011.
[7] Dirlik T., *Application of Computers in Fatigue Analysis*, PhD Thesis, University of Warwick, UK, 1985.
[8] Bishop, N. W. M., Lack, L. W., Li, T., Kerr, S. C., "Analytical Fatigue Life Assessment of Vibration Induced Fatigue Damage"*, MSC 1995 World Users' Conf. Proc.*, Paper No. 18, May, 1995.
[9] Wang X., Zhao G., Tang C., "Analysis of Frequency Response and Fatigue of Car Body by Random Loading", *3rd IEEE International Conference on Computer Science and Information Technology*, Chengdu, China, July 9-11, 2010.

**Fatih Kagnici** received his M.S. degree from Graduate School of Istanbul Technical University, Istanbul, Turkey in 2010. He is currently in the doctoral course of Graduate school of Istanbul Technical University and works for Hexagon Studio Design & Engineering. His research interests are mathematical modeling and optimization of systems in automotive industry.