# Using MSC.Fatigue to Estimate the Fatigue Damage Caused to Vibrating Automotive Components

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*MSC.Software*

## Abstract

An earlier paper [1] demonstrated that vibration fatigue techniques [2] can be a powerful tool in the design of automotive components subjected to vibration loadings. This paper extends the work by utilizing new techniques in the software program MSC.Fatigue. In particular, new techniques now exist which enable principal stresses to be computed over the entire surface region of interest. Complete results are included in this paper.

Vibration testing of components using accelerated test tracks or laboratory simulators is widely used in automotive design, as is fatigue testing for reliability. Furthermore, there are many common features between these two disciplines. However, problems often arise when engineers who are skilled in one field have to use techniques and concepts more generally used in the other. One example of such a situation concerns the use of frequency domain descriptions of structural response, which are commonplace in vibration. Many engineering applications, such as offshore structures and wind turbines, have already seen the benefits of using frequency domain fatigue analysis for reliability assessment. The purpose of this paper is to assess the benefits of frequency domain fatigue analysis and compare it with more conventional time domain techniques. A typical automotive component has been analyzed using MSC.Nastran and MSC.Fatigue using both time and frequency domain methods. Probability density functions and fatigue lives computed using output from these two different approaches show good agreement.

## Background

Time domain fatigue methods consist of a number of steps. The first is to count the number of stress cycles in the response time history. This is done through a process called rainflow cycle counting. Damage from each cycle is determined, typically from an S-N curve. The damage is then summed over all cycles using linear damage summation techniques to determine the total life. The purpose of presenting these basic fatigue concepts is to emphasize that fatigue analysis is generally thought of as a time domain approach. That is, all of the operations are based on time descriptions of the load function. This paper shows that an alternative frequency domain fatigue approach can sometimes be more appropriate.

For example, a vibration analysis is usually carried out to ensure that potentially catastrophic structural natural frequencies or resonant modes are not excited by the frequencies present in the applied load. With an offshore platform, for instance, the objective is usually to make the structure so stiff that its first natural frequency is above the loading frequency. Sometimes this is not possible and vibration engineers then have to estimate the response at resonance caused by the loading. These analysis tasks are best performed in the frequency domain using Power Spectral Density's (PSD's) of input loading and stress response. This paper demonstrates the use of MSC.Fatigue for the design of such components. The software package MSC.Fatigue (used for the analysis) contains three major new advances useful to the designer. Firstly, fatigue damage can now be accurately calculated from PSD's of stress. This is the sort of information typically output by random vibration programs such as MSC.Nastran. Secondly, the program can rotate the response stress information onto principal stress planes (necessary if fatigue damage is being calculated). And finally, the program can calculate this information over the entire region of interest.

For more details about the theory of vibration fatigue see [2].

## Problem Description

The model used to demonstrate the benefits of frequency domain fatigue analysis is a simple engine component. The intercooler, shown in Figure 1[1], is mounted on an engine and is subjected to random engine vibration. The FEA model comprised of 3628 solid elements (3402 CHEXA, 218 CPENTA and 8 CTETRA), 6767 shell elements (6477 CQUAD4 and 290 CTRIA3), 784 spring elements (CELAS2), 132 rigid bar elements (RBAR) and 25 multi-point control rigid bar elements (RBE2), and 12351 nodes. The spring elements were used to model the rubber isolators and the constraints of the rubber hose between the intercooler and compressor.

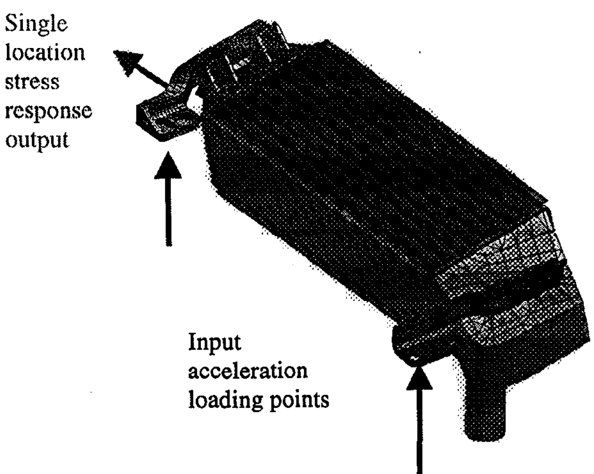


**Figure 1. The intercooler attached to the engine upper surface.**

Both frequency and time domain approaches have been used to establish the response of the component to the 3D acceleration load function shown in Figure 2.

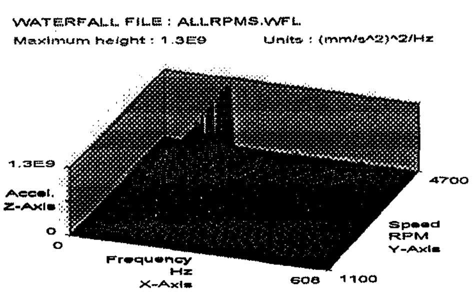


**Figure 2. The 3D loading (waterfall) plot used for the analysis.**

In both cases the load was applied, fully correlated with zero phase lag, to the brackets used to attach the intercooler to the engine. Initially only a single engine speed function was applied, i.e., a load function for one engine speed. If the engine speed varies then the load function also varies and this is shown in the waterfall plot. Since we are interested in finding the most damaging response conditions it is important to recognize that all engine speeds must be considered. This is where the flexibility of spectral techniques become particularly valuable since most of the computational work for a frequency domain analysis is done in computing the transfer function. Finding the response to many different inputs is then relatively straightforward, and computationally efficient, as will be illustrated later.

To assess the benefits of frequency domain techniques over time domain techniques, a single engine-speed loading slice was chosen to do the comparative analysis. The PSD at 2505 rpm is shown in Figure 3.

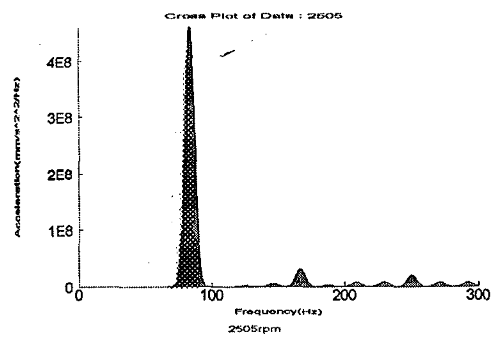


**Figure 3. The single loading plot initially used for the analysis.**

In general there are three analysis options. The first that shall be considered is the frequency domain, where an acceleration PSD is applied as the load input. The output is a stress response PSD from which fatigue life is determined (via a probability density function (PDF) determination of rainflow cycles and subsequent damage summation). For this approach, the input loading and response PSD take the general form shown in Figure 3.

The second is the time domain equivalent where an acceleration time signal input load is applied. The output is a stress response time signal from which rainflow cycle counting can be performed and the life assessed. A typical input loading for this situation is shown in Figure 4.

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**[1]Intercooler model courtesy of The Ford Motor Co.**

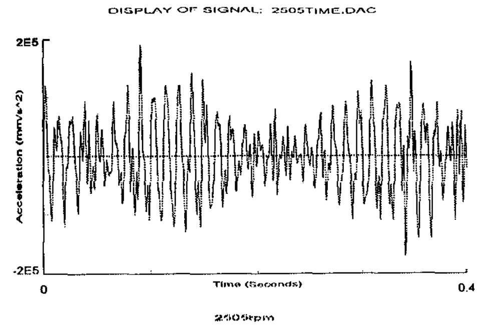


**Figure 4. A typical time history of acceleration.**

The third is also a time domain approach where the complex time signal is broken up into its harmonic components. An analysis from each acceleration harmonic is performed and damage summed due to all harmonics. A typical set of loadings for this approach is shown in Figure 5. Since this method is computationally equivalent to the second method, but much more inconvenient, it is not considered in this paper.

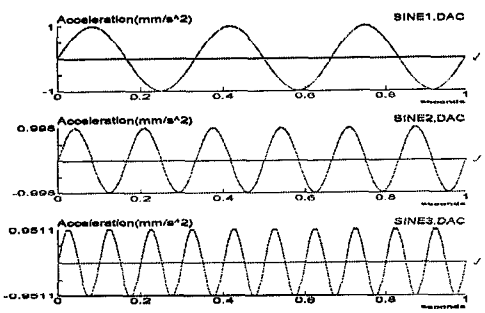


**Figure 5. A typical set (3 of perhaps 40) of harmonics derived from a time history.**

## Frequency Analysis

In order to perform a frequency domain analysis, an eigenvalue analysis was required to determine the frequencies to use as dynamic excitation. This can be accomplished automatically in a modal frequency response analysis or manually with a modal analysis. Once completed, a unit amplitude load function was applied, fully correlated with zero phase lag, to both fixing point brackets. The modal frequency response analysis (MSC.Nastran SOL 111) was then used to compute the transfer functions for all locations on the aluminum bracket, based on the excitation frequencies. Only the aluminum bracket is considered in the analysis since this is the critical component that was failing during test.

The single engine speed PSD load function was then multiplied by the transfer functions to compute the required stress and acceleration responses. A typical stress response PSD, obtained from the middle of the aluminum bracket, is shown in Figure 6.

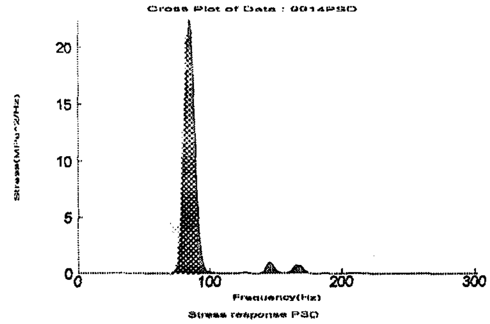


**Figure 6. A typical PSD stress response output from the single critical location.**

The Dirlik method was used to evaluate the PDF of stress ranges, from which a fatigue life estimate was obtained. The complete frequency domain analysis took approximately 2 hours of CPU on a typical workstation. Further analysis runs at different rpm's take a trivial amount of computing time because the same transfer function can be used, i.e., the finite element analysis does not have to be repeated for each rpm.

## Transient Analysis

As a comparison, a transient time history analysis was carried out using, as an input function, a time history of acceleration, the equivalent of the acceleration load input PSD for the single engine speed of 2505 rpm. A short sample from this input time history is shown in Figure 4.

Forty seconds of this time history were used with a time interval between points of 0.0008 seconds, thereby giving 50,000 time steps. The modal transient response analysis (MSC.Nastran SOL 112) was then used to compute the stress and acceleration responses based on the imposed time step interval. Responses were calculated for the same locations used in the frequency domain analysis. A short sample of the computed time history of stress response is also shown in Figure 7. This time history analysis took approximately 6 hours. To do this for twenty other rpm levels, the analysis runs would take twenty times as long because the computational effort required would be the same for each analysis run. This is computationally inefficient compared to the frequency technique.

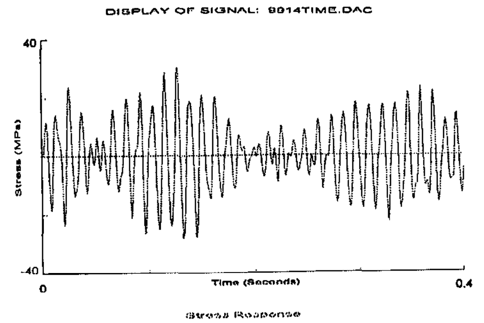


**Figure 7. A typical stress time history response output at the critical location.**

## Time and Frequency Comparison

The fatigue life was computed using typical material S-N data for aluminum. The fatigue life obtained for the frequency domain approach was 8% higher than the time domain result in terms of equivalent stress. In Figure 8, the PDF of rainflow cycles (obtained using Dirlik) is superimposed on top of the rainflow cycle count from the time domain.

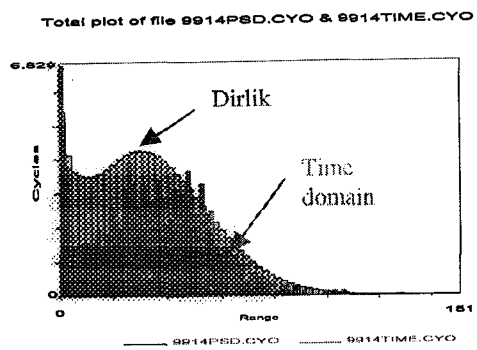


**Figure 8. The rainflow cycle counts obtained from the transient analysis, and compared with that obtained from Dirlik.**

This results in a difference in fatigue life of up to about a factor of two, which is fully acceptable given the statistical nature of fatigue. In order to get a closer agreement, a longer time history analysis would be needed. The results therefore demonstrate the computational elegance and efficiency that can be obtained by working in the frequency domain.

Also, an alternative comparison between the time and frequency domain analysis approaches was obtained by Fast Fourier transformation of the stress time history output from the transient analysis. The result in Figure 9 confirms the close agreement between the frequency domain and transient analyses (compared to Figure 6).

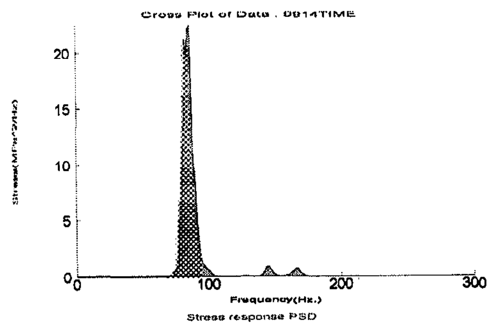


**Figure 9. The PSD obtained after applying an FFT to the time history in Figure 7**

One last comparison was made to actual measurement before going forward with the global fatigue analysis to give more confidence. The only measured response available was a waterfall plot of vertical intercooler rms acceleration response, shown in Figure 10 for all rpm's. This clearly shows the expected second order response of the engine as well as several much smaller higher order harmonics.

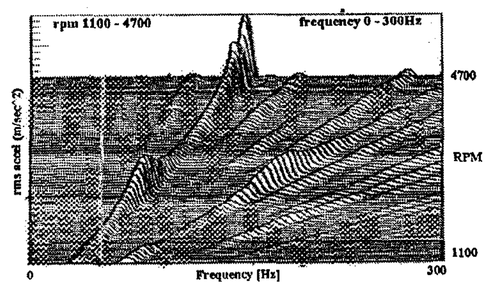


**Figure 10. The measured intercooler acceleration rms response.**

Earlier the point was made that further PSD responses, at different rpm's, are easily obtained using the same transfer function. This was done and a comparison made with the measured data. Figure 11 shows the same type of information, but obtained with a transfer function approach using MSC.Nastran random vibration analysis.

Comparing the two plots shows that the higher order responses are only reasonably well correlated but this was to be expected since the model was not optimized for this kind of dynamic analysis. However, a very good correlation can be observed for the important, and dominant, second order response. Peaks in these responses correspond to natural frequencies in the system.

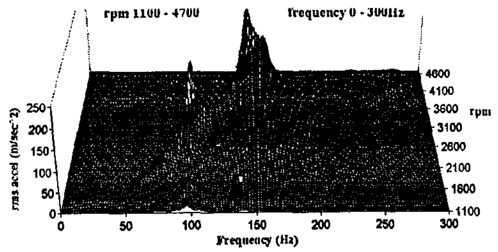


**Figure 11. MSC.Nastran generated acceleration rms response.**

There are many potential advantages associated with using an FEA solver instead of measured data. Firstly, computer generated data is likely to be much less expensive than measured data since each measured response plot is likely to cost many tens of thousands of dollars.

The second advantage comes from the fact that stress information is also relatively easy to obtain with the FEA solver, but it is much harder to obtain measured stress data at the critical locations. How are the critical locations determined for a start?

The third, and much more important advantage comes from the fact that by using an FEA solver, response data can be obtained very early in the design cycle, long before a prototype has been built. This then leads to the possibility of designing, and optimizing, for fatigue in a way that has not been previously possible.

## Global Fatigue Analysis

Once confidence was established in the frequency domain procedures, the global fatigue analysis could proceed. The goal was to compute fatigue damage over the entire aluminum bracket for all rpm's. The benefits of using MSC.Fatigue become immediately apparent in the ability to do this quickly and easily. Critical locations are easily identified for each rpm. For instance, Figure 12 shows a contour plot of the life in seconds plotted on log scale for 2505 rpm.

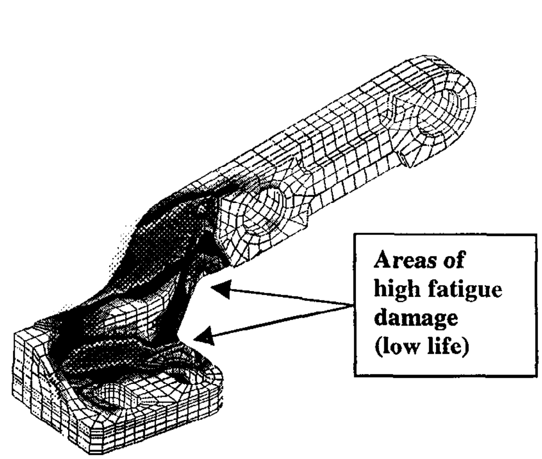


**Figure 12. Contour plot of fatigue life at 2505 RPM.**

The previous analyses were localized at a pre-selected location. This location was selected based on previous measurements, not because it was the most critical area. In fact, it is near impossible to identify the critical location for general dynamic systems such as these since the critical location can be different for each rpm input load level, and can change from frequency to frequency within a single rpm load level.

In addition to this, the previous localized analyses were done in a single component direction and rms or PSD responses extracted at element centroids. This is a limitation of all FEA solvers. MSC.Fatigue resolves stresses around to principal directions and has the ability to calculate fatigue life at element centroids or at surface nodes. Clearly most fatigue problems begin on the surface with crack initiation. This affords a much clearer understanding by using more representative stresses at the correct areas of the model.

## Damage Summation

Clearly, to fully determine fatigue life of the intercooler bracket, all rpm's must be considered. Simply summing the damage from each rpm would lead to overly conservative answers since equal weight would be given to each. It is highly likely that the engine will be driven at some rpm levels for more time than at others. Combining the damage with a "customer usage factor" is a more realistic way to assess the total damage. Figure 13 shows fatigue damage at each rpm for a particular critical location.

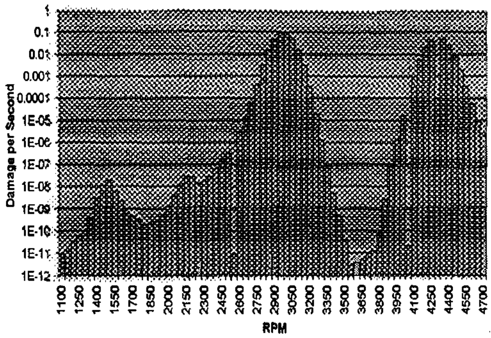


**Figure 13. Damage at critical node 10313.**

This shows most of the fatigue damage done between 2750 rpm and 3050 rpm.

In principle, it is possible to build up a "customer usage factor" as shown in Figure 14, say, for a typical driver.

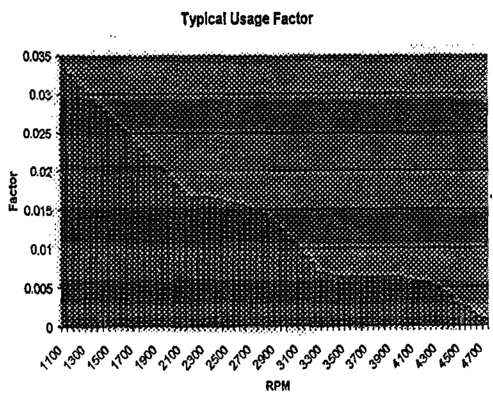


**Figure 14. A typical customer usage factor.**

The actual fatigue damage at each rpm is obtained by multiplying the fatigue damage at each rpm by the customer usage factor to redistribute the damage. The results can then be summed to produce a final fatigue contour plot at all locations shown in Figure 15, thus identifying the actual critical location when all rpm's levels are considered. This is easily done with result post-processing tools in MSC.Fatigue, also found in most graphical pre- and post-processing systems such as MSC.Patran.

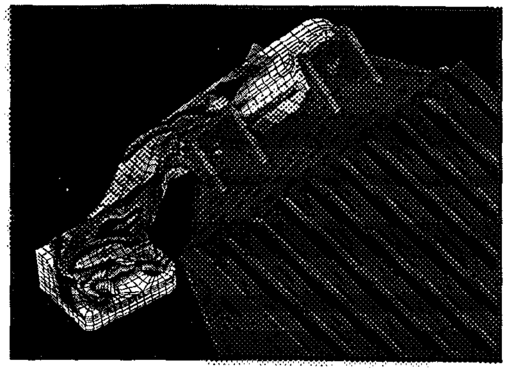

**Figure 15. The combined fatigue damage done at all rpm levels.**

## Spectral Fatigue Damage Editing

As a final study, MSC.Fatigue allows the designer/analyst to take a response PSD, such as the one shown in Figure 16, and make modifications to its shape and monitor the change in fatigue life. In this example the effect is shown of reducing one of the resonant peaks in the system by about 98%. In this way a fatigue designer can find the structural changes necessary to achieve a desired fatigue life. This highlights an FEA based example of using the vibration fatigue tools, although it could also, just as easily, apply to test-based analysis.

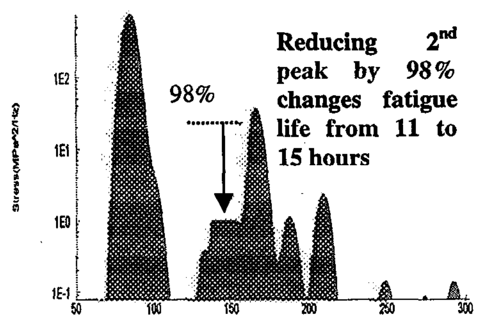


**Figure 16. The PSD plot after the second peak has been removed.**

## Conclusions / Benefits

Advantages of using vibration fatigue tools conveniently fall into two categories, these being test and design. For test based analysis, condition monitoring is already used as a means of identifying changes in any system transfer function. Vibration fatigue tools are an obvious addition that can be coupled with such monitoring. System (fatigue) analysis was demonstrated earlier as a tool that the fatigue designer can use to determine structural changes required to achieve a certain design life.

In addition, by utilizing vibration fatigue tools the designer now has the option to perform a complete analysis, including fatigue design, within the FEA environment. This has resulted from the development of robust and generally applicable tools for estimating fatigue damage from PSDs of stress.

Fatigue assessment in the frequency domain is now therefore an option for the designer/analyst. Using these tools the designer can now aim to design for fatigue rather than just check at a late stage in the design cycle. This also allows the possibility of performing optimization studies to assess how to overcome various dynamic problems, for instance by increasing damping by placing bushings at attachment points or other possible solutions.

## References

[1]. Analytical Fatigue Life Assessment of Vibration Induced Fatigue Damage, NWM Bishop, LW Lack, T Li and SC Kerr, Paper presented to MSC World Users Conference, Universal City, CA, May 1995.

[2]. Vibration Fatigue Analysis in the Finite Element Environment. NWM Bishop, An Invited Paper presented to the XVI ENCUENTRO DEL GRUPO ESPAÑOL DE FRACTURA, Torremolinos, Spain, 14-16 April 1999.