Dynamic Analysis & Correlation for Exhaust System

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ABSTRACT

This paper emphasis on the systematic procedure for the FEA dynamic analysis for automotive exhaust system, including the application of real road excitations, identification of the potential failure location, consideration of system non-linearity, selection of MSC.Nastran SOL, and the comparison with test results.

INTRODUCTION

The ordinary dynamic analyses are straightforward, such as the normal modal analysis for linear system, and the frequency response analysis for a linear system with free end excitation. The dynamic analysis for the automotive exhaust system, however, will be much more complex due to the nonlinear component properties, multiple random base excitations (enforced motions). These system dynamic properties (stiffness and damping) will change with frequency, amplitude, temperature, etc. One realization of excitation may not be repeatable and the response will be unpredictable correspondingly. It will be extremely challenging to compare the FEA results to the test under this situation.

MSC.Nastran normal mode analysis (SOL 103) can be used to identify the system nominal natural frequency and the critical locations using the strain energy distribution. MSC.Nastran direct frequency response (SOL 108) can update the frequency response with the changed stiffness and damping.

The multiple base excitations can be simulated in MSC.Nastran by using large mass method. It can provide both the restraint and multi-directional excitations at the same location.

If the excitations are deterministic acceleration, the amplitude/phase or real/imaginary verse the frequency can be directly input. If the excitations are the random or pseudo-random, their power spectral density (PSD) should be input and output. For the fatigue calculation under random excitation, three standard deviations should be included for accuracy consideration.

The variable non-linear properties, such as stiffness and damp for isolator and decoupler, can be input to MSC.Nastran by using CBUSH, PBUSH, PBUSHT in MSC.Nastran SOL 108.

NORMAL MODAL ANALYSIS

The MSC.Nastran SOL 103, normal modal analysis, can be used to identify the natural frequency, mode shape and the strain energy distribution. The average dynamic stiffness is used as its nominal value. The modal frequency was confirmed with tested frequency and can used to compare with the excitation frequency to identify any potential resonance. The modal strain energy distribution identified the critical location where was observed in this exhaust system. The model should be built to be reasonably detail to cover most modes.

Table 1 Modal Frequencies Comparison

MODES	Simulation (Hz)	Test (Hz)
1	3.29	
2	4.1	4.0
3	4.33	
4	6.46	6.0
5	6.92	
6	8.2	7.5
7	9.1	
8	11.2	11.5,
9	14.5	14.75
10	19.9	19.5
11	25.8	25.5
12	26.8	26.0
13	29.4	31.0
14	37.3	
15	55.5	

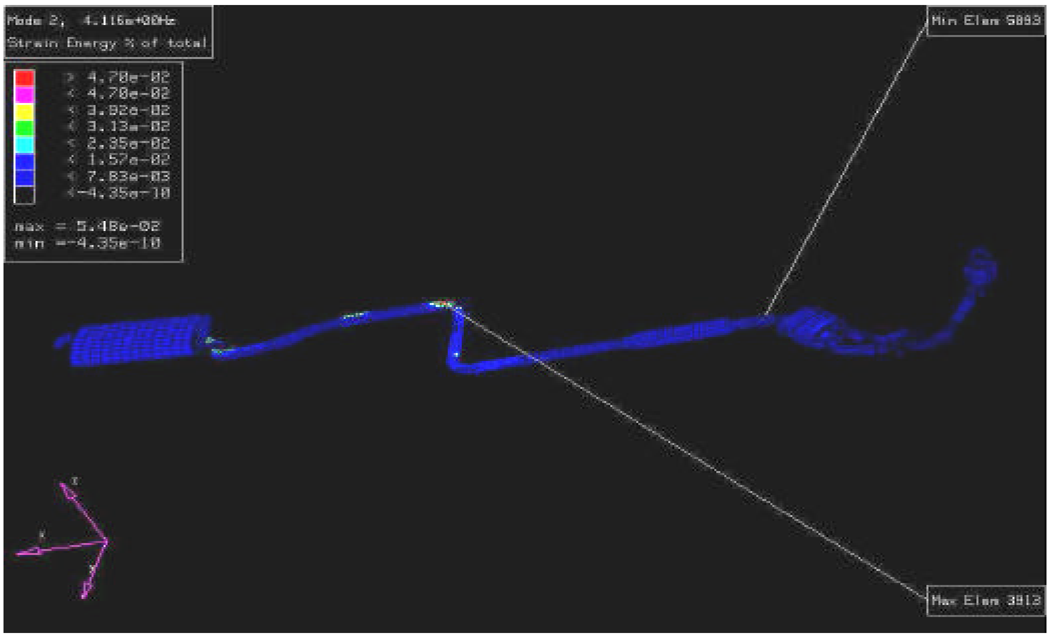

Figure 2 FEA Modal Strain Energy % for Modes 2, 3, 4, 5, 6, 7 (4.1 – 9.1 Hz)

EXCITATION

Total 18 channels' accelerations have been input in MSC.Nastran. All excitations are stable in engineering point of view, and their peaks are normally distributed. The excitation frequencies concentrate on a range from 5 – 15 Hz, and will excite certain modes of this range.

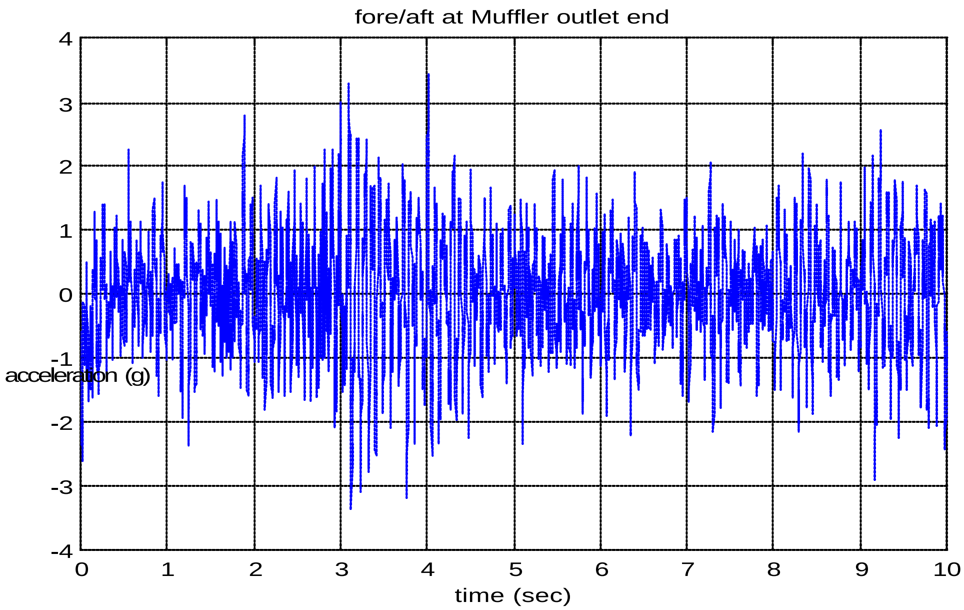

Figure 3 Excitation Time History at Muffler Outlet in fore/aft

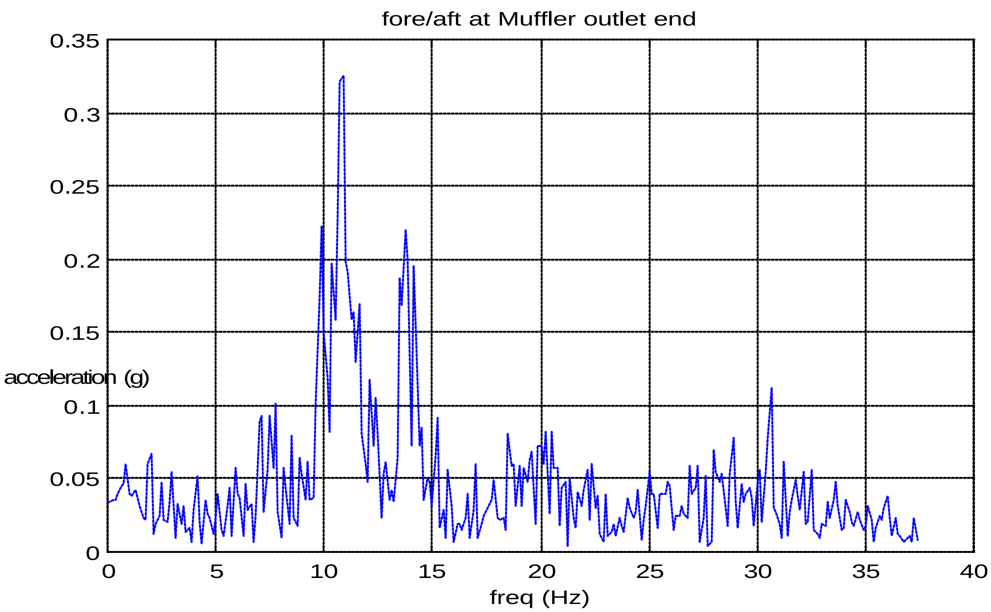

Figure 4 Frequency Domain Excitation at Muffler Outlet in fore/aft

NON-LINEARITY PROPERTIES

This is the most challenging step in the dynamic analysis. Usually, the test data for the stiffness and damping for isolator and decoupler will not available or sufficient. Different stroke level will result in quite different stiffness/damp curves. A special test has been conducted to acquire the spring rate and damping at special amplitude level for both isolator and decoupler. The CBUSH element has been used to input the dynamic stiffness and damping. Without test data, a FE model with equivalent material properties for isolator and decoupler can be used to automatically provide the component stiffness/damping.

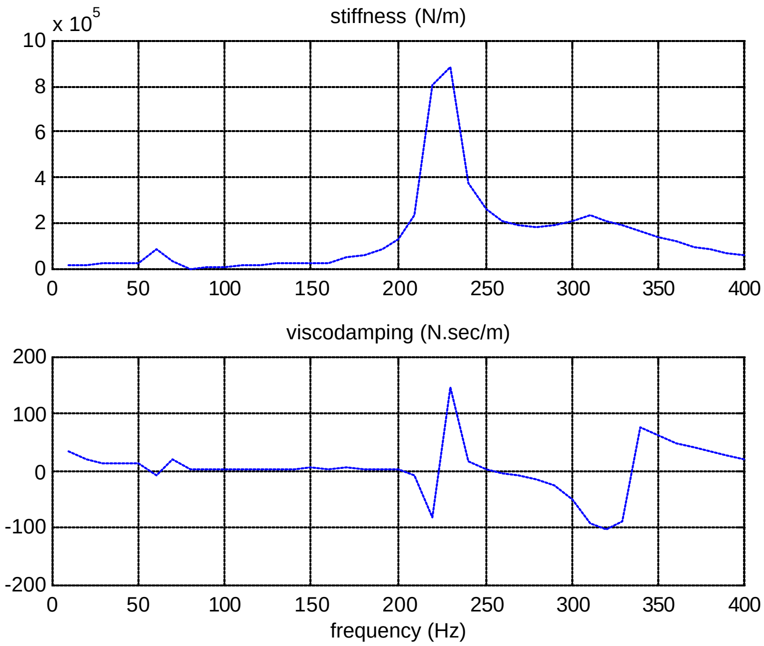

Figure 5 Isolator Stiffness/Damping verse Frequency

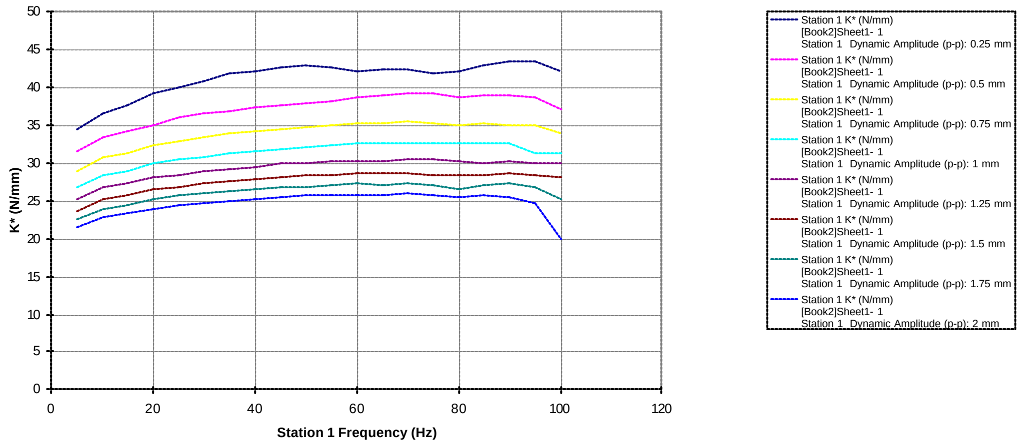

Figure 6 Isolator Stiffness verse Amplitude

FREQUENCY RESPONSE ANALYSIS

The MSC.Nastran SOL 108 is recommended to use for direct frequency response. It can update the changed stiffness and frequency at different frequency step. One large mass has been attached to each hanger arm location and engine mounting area. All the restraints at the excitation directions should be released.

The output frequency resolution should be selected to allow more than 5 point near any peak above the half-power bandwidth. The element/node selection for output can based on modal strain energy distribution or only the interested area/component.

If the excitation is random signal and will use the corresponding stress response for fatigue analysis, then the root mean square value should be expanded to 3 standard deviations. The damages caused by 1d (68.1%), 2d (27.3) or 3d (4.3%) stress will be accumulated to cover any unknown factor in the real world.

The stress frequency response for the critical area is used for comparison between FEA and testing. Shown in Figures 7 to 10, the simulated stress frequency response matches reasonably with the test data. The comparison will be improved with updated stiffness and damping information.

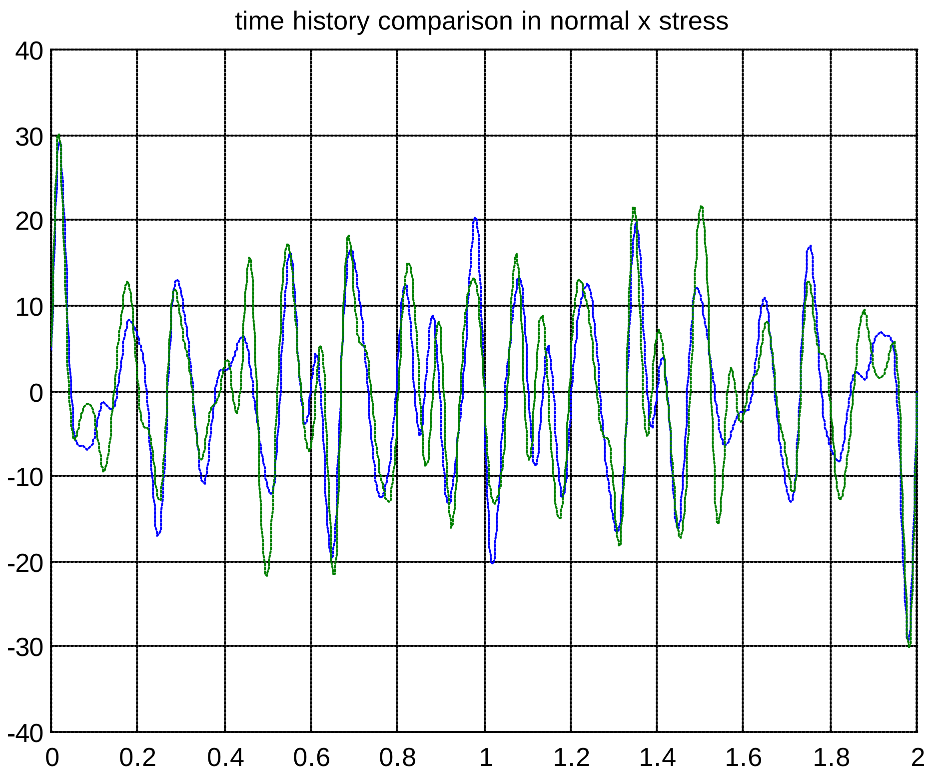

Figure 7 Stress Time History Comparison in Normal X

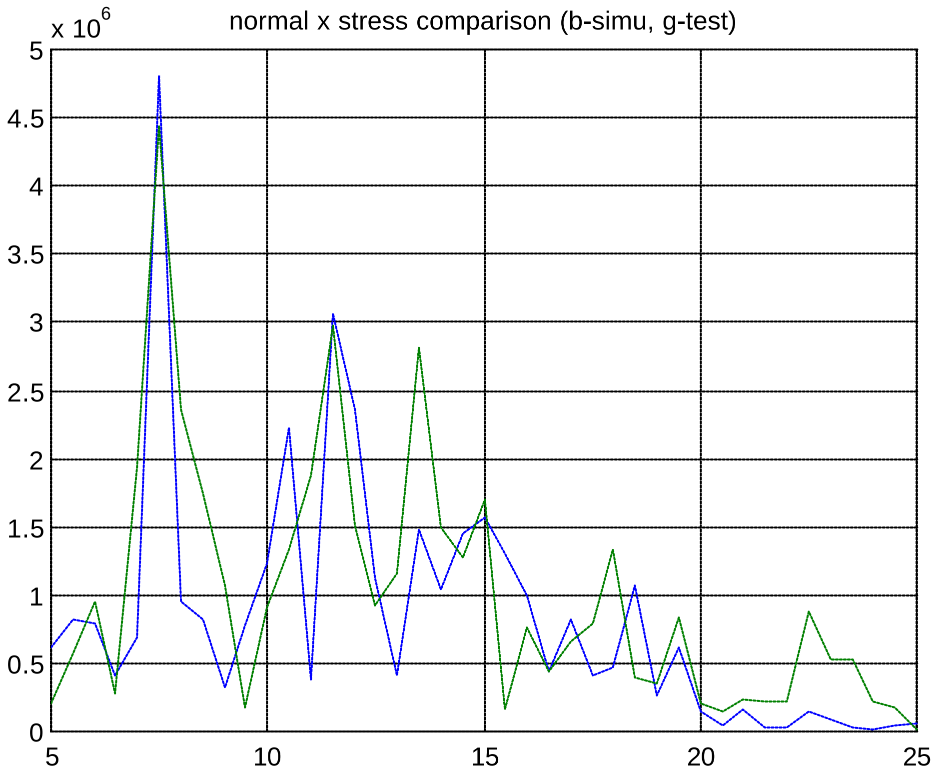

Figure 8 Stress Frequency Domain Comparison in Normal X

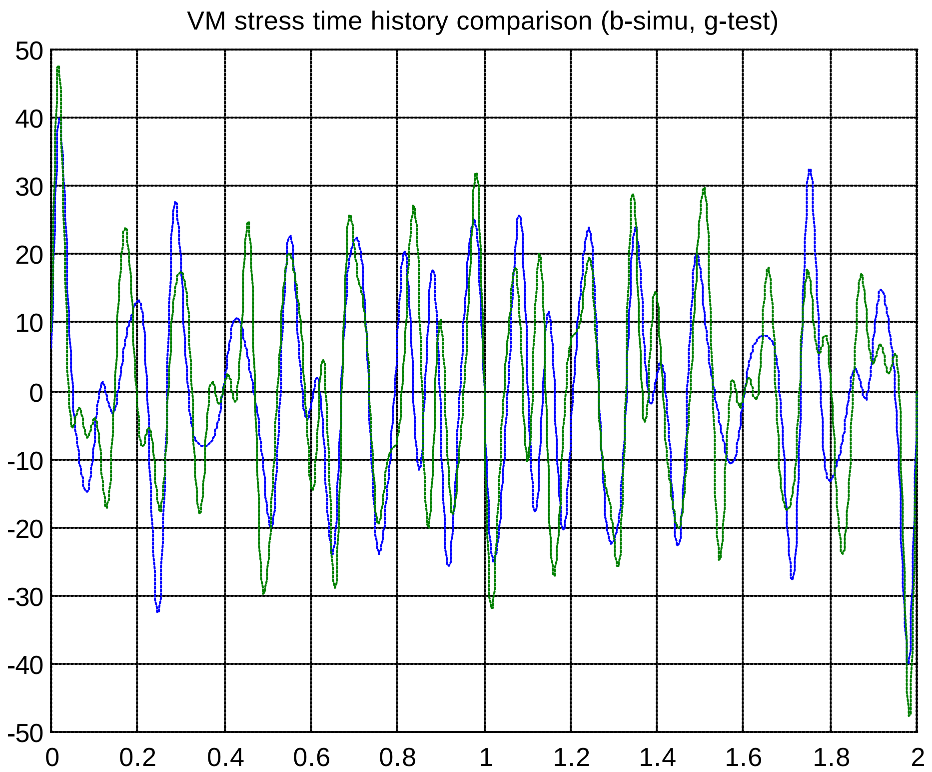

Figure 9 VM Stress Time History Comparison

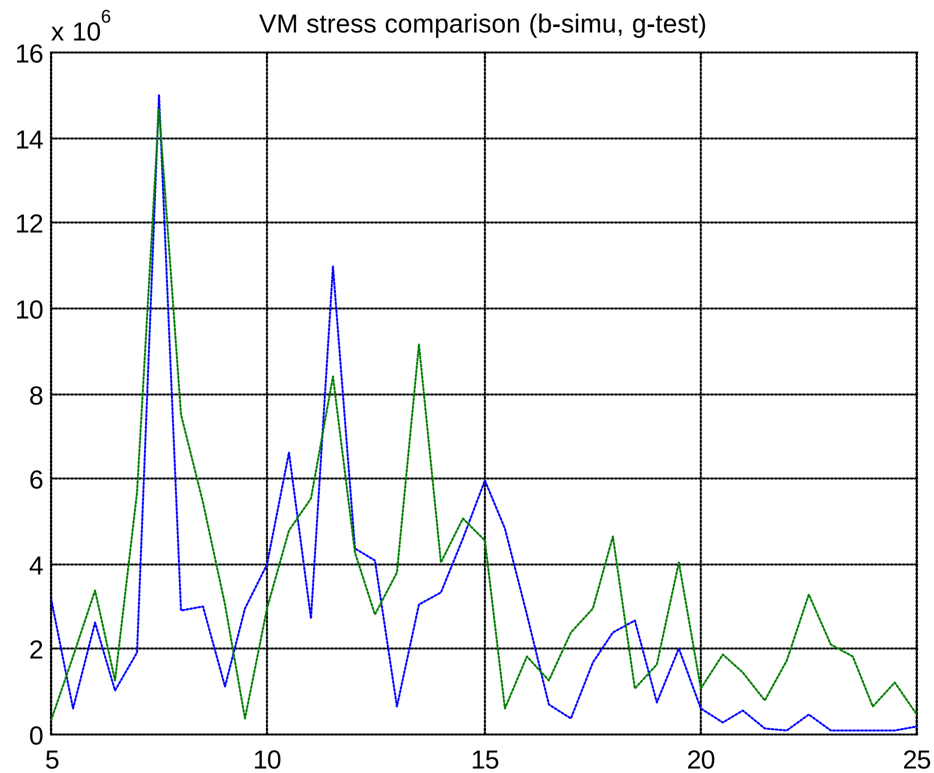

Figure 10 VM Stress Frequency Domain Comparison

CONCLUSION

The analysis performed here is real and a daily practice for the author. The modal analysis method to identify the critical location has been confirmed with many failure observations. The modal frequency comparisons with the test data or excitation have been well used to evaluate the FE model and to identify the potential resonance. The systematic procedure of frequency response analysis for this exhaust system can be used for any system's dynamic analysis. Again, this simulation can help the lab test or simulation to setup the equivalent excitation from which the same damaging stress response as the strain gauge data can be duplicated.

This paper gives certain insight into random excitation, non-linear dynamic properties, and normal mode analysis and frequency response analysis. This systematic methodology will be applicable for any dynamic situation.

REFERENCE:

(1) MSC/NASTRAN Basic Dynamic Analysis User's Guide, version 69, The MacNeal-Schwender Corporation, Los Angeles, CA, July, 1997

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(3) MSC/NATRAN Advanced Dynamic Analysis User's Guide, version 70, The MacNeal-Schwender Corporation, Los Angeles, CA, 1997