

Durability Analysis of Engine Parts under Engine Loads and Random Road Vibration Using Direct Transient Methods

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ABSTRACT

Some of the engine components are connected both to the chassis and to the engine. These parts are subject to vibrational loads coming from the road and the engine. Engine loads are generally considered to be deterministic, whereas the road load is random and conventional methods such as steady state forced frequency analysis, or PSD approach usually focus on engine and road loads separately. Since phase information is lost during PSD conversion, it is not possible to superimpose the results of these analyses without making additional assumptions such as creating random phase data for stress results, which would be a questionable technique. Consequently, these approaches generally result in delivering incomplete information for fatigue evaluation. In this study, both of the engine and road loads are considered simultaneously. Direct transient analysis technique is utilized in order to obtain overall stress values on the parts. Up to 40% difference in the stress values are seen in the selected bracket of interest, and it is concluded that ignoring one of the sources of vibration, either the road or the engine, may result in misinterpretation of fatigue durability of the part.

INTRODUCTION

It is common practice to utilize methods such as Steady-State Forced Frequency Response Analysis, or PSD Response Analysis to investigate the fatigue durability of parts under loads coming from the engine or the chassis (road). These methods are usually fast and effective with linear models in the determination of stresses that occur on the parts. A difficulty that arises with the PSD method, which is the common way of dealing with random vibration, is that since the phase information is lost during PSD conversion, the calculation of Von Mises stresses necessitates some additional assumptions to be made [1],[2], such as creating random phase data for stress results, which would be a questionable technique.

Some of the engine components, however, are connected both to the chassis and the engine. Consequently, these parts are subject to vibrational loads coming from the road and the engine. Ignoring one

of the sources of vibration – usually the weaker one – , and fixing the part in space, instead of applying the necessary vibrational load is a common CAE approach. This is usually done because of the limitations of finite element solvers, to be able to utilize the conventional linear methods such as PSD response or steady-state forced response methods.

In this study, a bracket, which connects the heater rail system to the chassis, is investigated. Since the bracket is directly connected to the chassis, it is subjected to the road loads in the first place. In addition to this, bracket is also subjected to the engine loads since the heater rail system has connections to the engine in some places. Direct transient analysis technique (with the extended ability to cover nonlinear models) is utilized in order to obtain overall Von Mises stress values on the part in time domain, considering both sources of vibration. The peak stress results are compared to the cases, in which one of the sources of vibration – either the engine or the load – is ignored. It is concluded that ignoring one of those sources of vibration, may result in misinterpretation of fatigue durability of the part.

1. PROBLEM DEFINITION

In this study, the fatigue durability of a bracket, which is connected to the chassis and the heater rail system in a inline 4 cylinders diesel engine, is investigated. Engine speed range is 900 rpm – 4500 rpm. 2nd, 4th, and 6th order engine loads are considered to be critical.

Finite element model of the whole system is seen in Figure 1, whereas the bracket of interest can be seen in Figure 2. Heater Rail system is connected to the chassis at point 1 and to the engine at point 2.

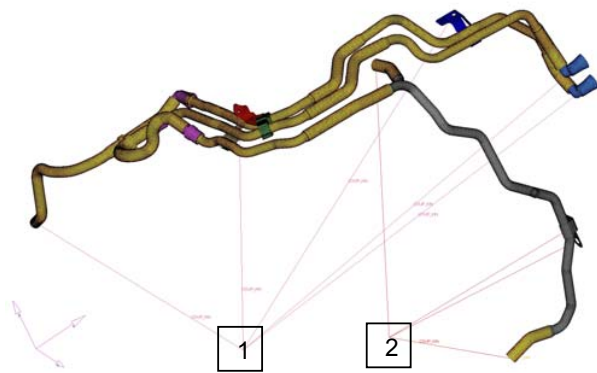


Figure 1: Heater Rail System

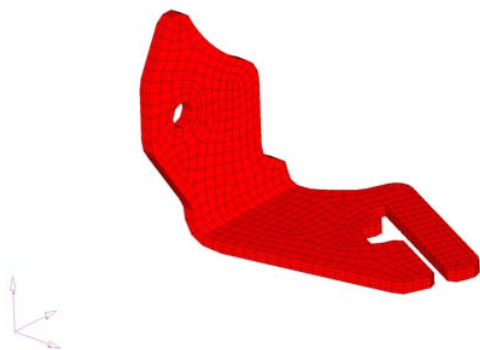


Figure 2: Bracket to be investigated

FE MODEL DETAILS - Pipes are modeled with first-order quadrilateral shell elements. Brackets are modeled with either first-order hexahedral solid elements or first-order quadrilateral shell elements. Pipe-to-pipe connections are made with spider rigid elements and pipe-to-bracket connections are modeled with rigid elements with single dependent nodes.

Total number of nodes in the model: 84584

Total number of elements in the model: 80090

Number of nodes in the bracket (output set): 2344

Number of elements in the bracket (output set): 1545

2. NORMAL MODES ANALYSIS

A modal analysis is performed first in order to obtain the critical modes of the bracket. Range for modal analysis (0-900Hz) is chosen to cover twice the excitation frequency, which is 6th order engine load at 4500 rpm corresponding to 450 Hz.

The system has 113 modes between 0-900 Hz. Critical modes of the bracket are at 18.6Hz, 29.4Hz, 58.7Hz, 352.8Hz, 402.5Hz, 410.1Hz, 566.3Hz and 803.4Hz.

Strain energy density contours for the critical modes of the bracket are shown in Figures 3 – 10.

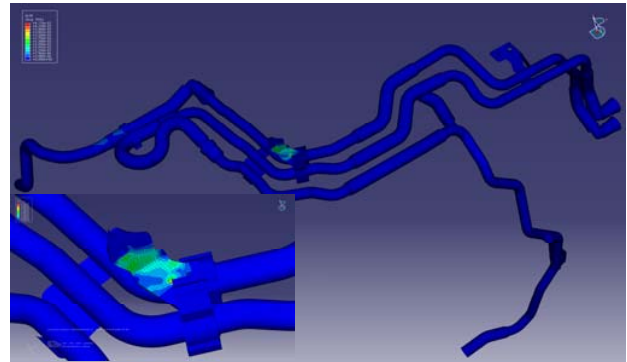


Figure 3: Mode at 18.6 Hz – Strain Energy Density Contour

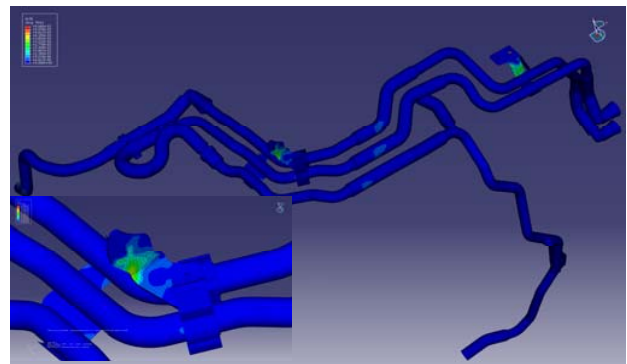


Figure 4: Mode at 29.4 Hz (to be excited at ~900rpm, 2nd order) – Strain Energy Density Contour

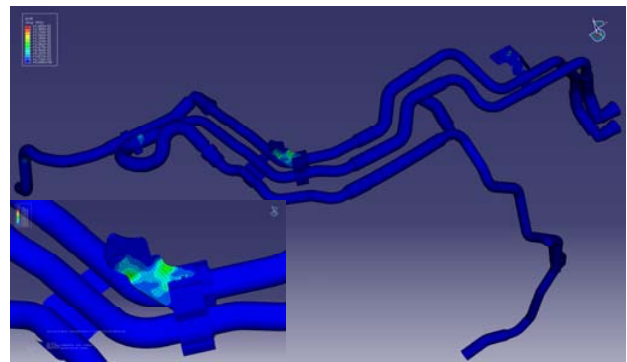


Figure 5: Mode at 58.7 Hz (to be excited at ~900rpm - 4th order or harmonics of 2nd order, ~1800rpm - 2nd order) – Strain Energy Density Contour

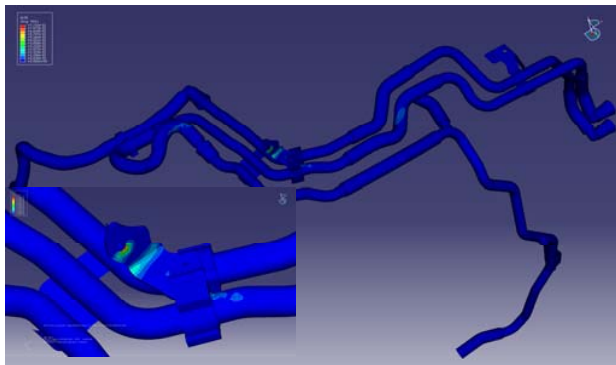


Figure 6: Mode at 352.8 Hz (to be excited at ~3500rpm - 6th order, ~1800rpm harmonics of 6th order) – Strain Energy Density Contour

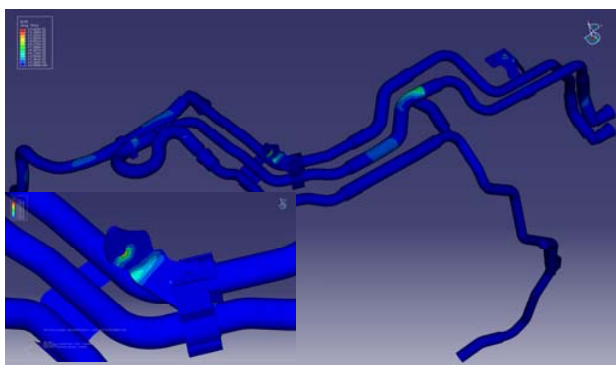


Figure 7: Mode at 402.5 Hz (may be excited at ~4000rpm - 6th order) – Strain Energy Density Contour

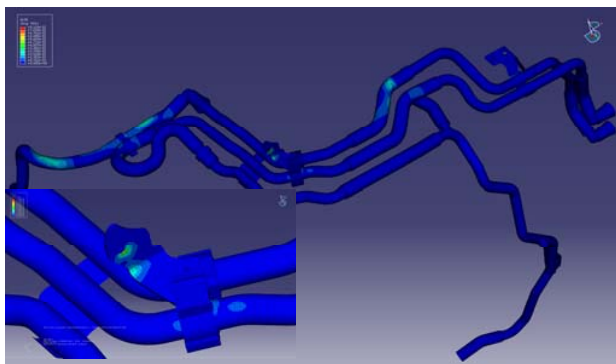


Figure 8: Mode at 410.1 Hz (may be excited at ~4000rpm - 6th order) – Strain Energy Density Contour

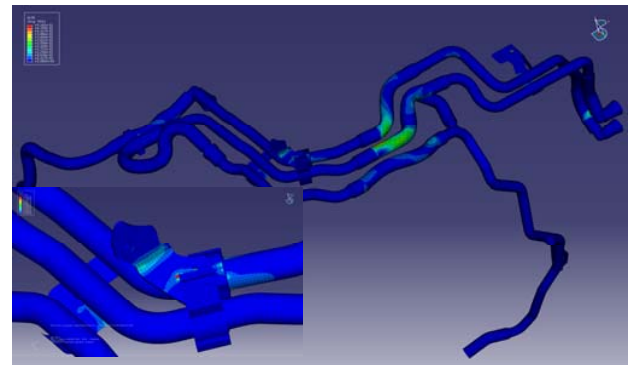


Figure 9: Mode at 566.3hz – Strain Energy Density Contour

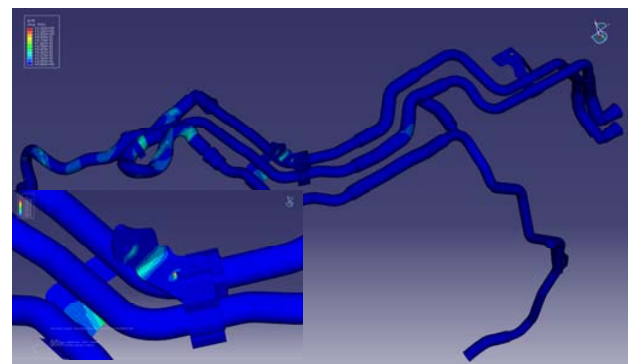


Figure 10: Mode at 803.4hz – Strain Energy Density Contour (may be excited at ~4000rpm - harmonics of 6th order)

3. LOADS AND BOUNDARY CONDITIONS FOR TRANSIENT DYNAMIC ANALYSIS

3 different sets of loads and boundary conditions are applied during the analyses. These are:

1. Deterministic engine vibrations in X, Y and Z directions together with random road vibrations in X,Y and Z directions. Rotational degrees of freedom are fixed at the points of connection to the chassis and the engine.
2. Deterministic engine vibrations in X,Y and Z directions only. Rotational degrees of freedom are fixed at the point of connection to the engine. All degrees of freedom are fixed at the point of connection to the chassis.
3. Random road vibrations in X,Y and Z directions only. Rotational degrees of freedom are fixed at the point of connection to the chassis. All degrees of freedom are fixed at the point of connection to the engine.