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Non-Linear Thermal Modal Analysis for Hot End Exhaust System

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Abstract: In this paper, we will discuss the need of non-linear material properties for performing modal analysis in hot condition. To stay competitive in the brisk rising automotive industry, automakers have to employ an integrated approach to use both simulation/modeling and hardware testing to reduce product development cost and time. Influence of temperature effects on exhaust hot end system are compared with linear and non-linear material properties for natural frequencies and mode shapes by performing modal analysis in hot condition. For these, temperature distribution are mapped from conjugate heat transfer results and then analyzed in finite element analysis with structural boundary conditions in hot condition. Modal analysis theory of exhaust system is discussed with appropriate explanations.

Key words: Non-linear, Temperature effects, Modal analysis, Exhaust system, Finite element method.

1. INTRODUCTION

The automotive exhaust environment is somewhat of an enigma because they are not clearly definable and conditions are not consistent on which to base material selection. Generally, an automotive exhaust gas consists of carbon monoxide mixture, oxides of nitrogen, unburned hydrocarbons and moisture. Temperatures run the gamut from ambient to 1000° C. Internal combustion engine cause pulsation due to high velocity of gas stream and operational is cyclic. The problem is further exacerbated by the different levels of expertise on the part of customizing and servicing the engine, which results in change in operating conditions expectation and anticipated temperatures.[3] The requirement of simulation in concept and design phases has been a major leap through in product development cycle time. Determining the natural frequency of exhaust system is a major criterion to understand the deformable shapes at system resonance frequency.

Designing of hot end exhaust system are getting complicated due to the stringent requirement of NVH. Thermal designs require study on known investigations of thermal and fluid behavior by numerical and experimental measurements to evaluate and detect the critical mode shapes under duress. Thus validating hot end exhaust system becomes daunting task to engineers. [1]

As engine discharges gases at high temperature such as 700° C – 850 ° C, tail gas heating effects will be experienced by hot end exhaust system. As a result of exhaust gas heating, thermal stresses will be high as hundreds of MPa which will lead to thermal fatigue and structural failure. [2]

Lot of research has been done, for example Case 1 - Shi & Yang considered flat plate as research subject. [5] The field of observation was considered under the thermal loads for structural vibration and transient temperature field. Case 2 - Yang and Guo performed a similar research by considering thermal stress and mechanical load for structural vibration characteristics of an engine piston. [4] The end result evidently showed severe effects on the characteristics of structural vibration were observed due to transient heating.

The traditional modal analysis techniques are linear ignoring the non-linear inherent characteristics such as material, contact and geometry which may cause qualitative errors in stiffness requirements on response analysis. Most approximate method is comparing the results of modal analysis with linear and non-linear material properties in hot condition with pre-defined stress.

2. METHODOLOGY

This paper includes hot end exhaust system as subject, which was pre-processed using Hypermesh software with tetrahedral elements for manifold, outlet cone, substrate, mat and mounting bracket, quad elements for shell. Figure 1 shows the schematic representation of the system. Conjugate heat transfer results performed in Star CCM+ are coupled with modal analysis in ABAQUS software for the cases mentioned below. The model has 0.2 million DOF, ensuring detailed modeling in order to capture stresses accurately in critical areas.

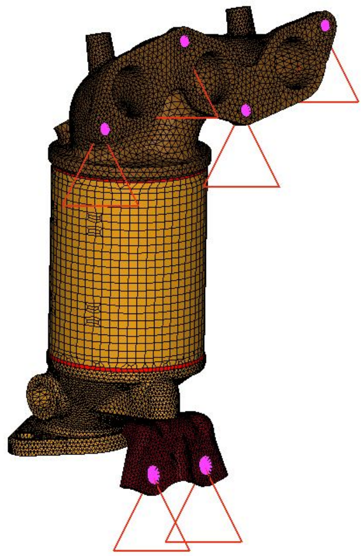

Figure 1 - Finite Element Model

3. STRUCTURE THERMAL STIFFNESS

The temperature effect on structural stiffness is mainly from two aspects:

- First aspect:
  - The heating up temperature can change the material elastic modulus and lead to the initial stiffness matrix changes.
  - Take the structure initial stiffness matrix after heating up used in calculations.[5]

$$K_T = \int \Omega B^T D_T B D\Omega \qquad (1)$$

Where,

B = Geometric matrix

D^T= Elastic modulus which is related to material elastic matrix E & Poisson's ratio μ.

- Second aspect:
  - The thermal stress caused by temperature gradients, additional initial stress stiffness matrix is needed.
  - Take the structure stiffness matrix.

$$K_\Omega = \int \Omega B^T \Gamma\, G D\Omega \qquad (2)$$

Where,

G = Shape function matrix

Γ = Stress matrix

In summary, the structure thermal stiffness matrix is:

$$K = K_T + K_\Omega \qquad (3)$$

4. MODAL ANALYSIS THEORY

- The basic equation for typical un-damped modal analysis is classic Eigen value problem.
- According to mode theory, the structure will be typically seen as a system constituted by the mass point, rigid body, damper and discrete it as finite number of elastic coupling rigid bodies.
- Therefore, an infinite multi-degree of freedom system turns into limited multi-degrees of freedom system.
- When the linear time-invariant system requirements are met, the system's general motion mathematical model can be expressed as:

$$M\ddot{x}+C\dot{x}+Kx = f(t) \qquad (4)$$

Where,

M, C, K: The mass matrix, damping matrix and stiffness matrix

x: The exhaust pipe vibration displacement vector

f (t) : The exhaust pipe load vector

- Modal analysis method is to replace the physical coordinates of modal coordinates that each principal mode corresponded, so that the differential equation decoupling to be independent differential equations in order to obtain the system modal parameters.
- The vibration of the engine exhaust pipe is a random vibration, which basically belongs to linear time-invariant systems. It can be assumed that M is a constant matrix. The structural damping of exhaust pipe has little effect on the natural frequencies and therefore external load and damping are not considered.
- Thus equation shown above becomes :

$$K- \omega^2 M\Phi = 0 \qquad (5)$$

Where,

M- $\int_\Omega \rho N^T N d\Omega$ is the structure overall quality matrix

- When the order of matrix K and M is n, the ω^2 in formula shown above is the n times real coefficient equation and the system degree of freedom vibration characteristics (natural frequencies and mode shapes) problem is to solve the matrix Eigen value ω.

5. CASE STUDIES

- Case 1 - Modal analysis in cold condition includes linear material properties parameters
- Case 2 - Modal analysis in hot condition includes nonlinear material properties under maximum nodal temperature at each component.
- Case 3- Modal analysis in hot condition including nonlinear material properties, pre-tension force, thermal stress and mechanical stress

5.1. CASE 1

Modal analysis in ambient temperature with linear material properties

Table 1 - Linear material properties

Material	Young modulus MPa	Poisson ratio	Density Ton/mm^3
FCD 550	1.7e05	0.25	7.2e-09
SS409	2.08e05	0.3	7.7e-09
SS439	2.00e05	0.3	7.69e-09
Mild steel	2.00e05	0.3	7.85e-09

5.1.2. BOUNDARY CONDITION

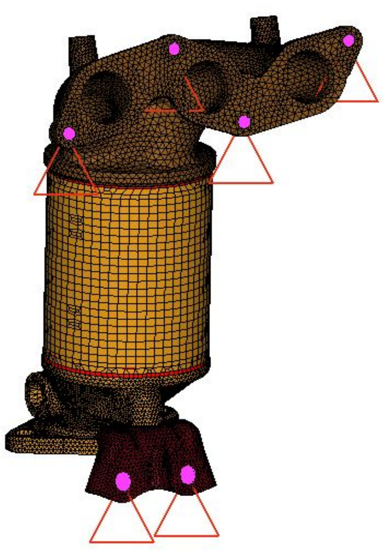

Figure 1 - Manifold inlet flange and Hot end bracket bolt holes are constrained in all DOF

5.1.3. MODE PLOTS

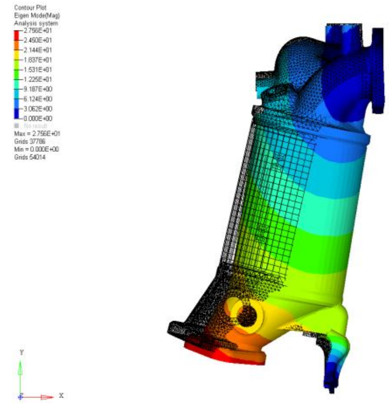

Figure 3 - Mode 1- 830 Hz – Longitudinal mode

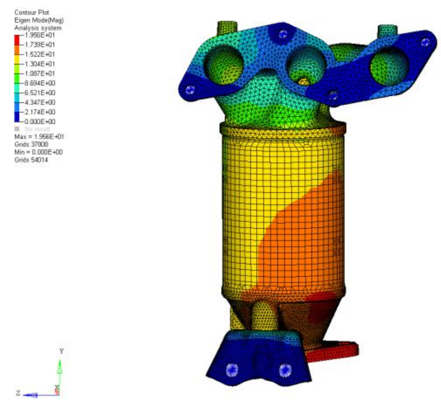

Figure 4 - Mode 2- 936 Hz – Vertical mode

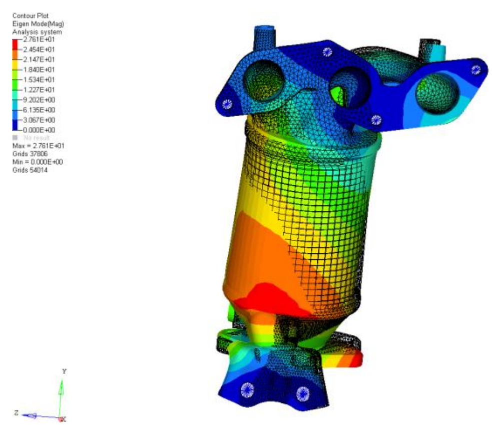

Figure 5 - Mode 3- 1849.3 Hz – Bending mode

5.1.4. OBSERVATIONS

- Critical areas are concerned with bracket having mode shapes like bending, twisting in first, second & third order vibration shapes

5.2. CASE 2

Modal analysis in hot condition includes nonlinear material properties under maximum nodal temperature at each component.

5.2.1. MATERIAL PROPERTIES

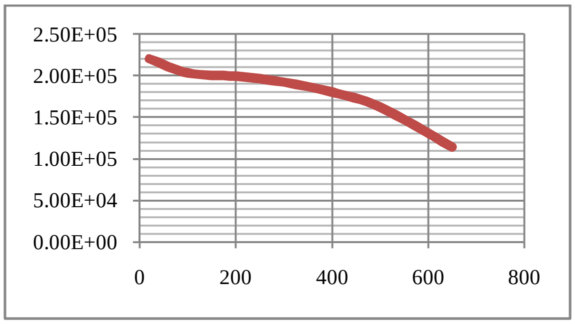

Figure 6 – Y axis - Young Modulus (MPa) Vs X axis - Temperature (°C)

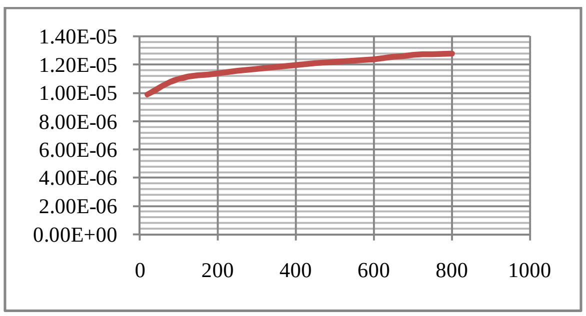

Figure 7 – Y axis – Thermal expansion co-efficient (µm/m °c) Vs X axis - Temperature (°C)

5.2.2. BOUNDARY CONDITION

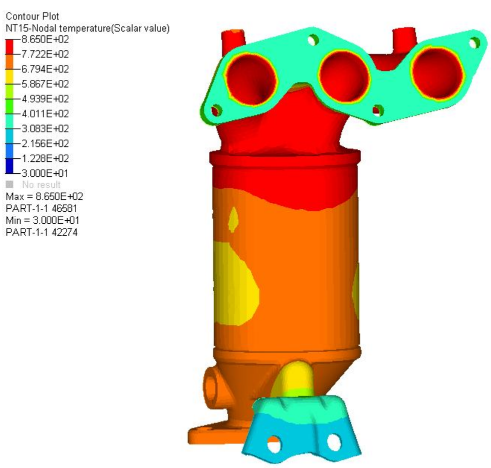

Fig 8 – Conjugate heat transfer results

5.2.3. MODE PLOTS

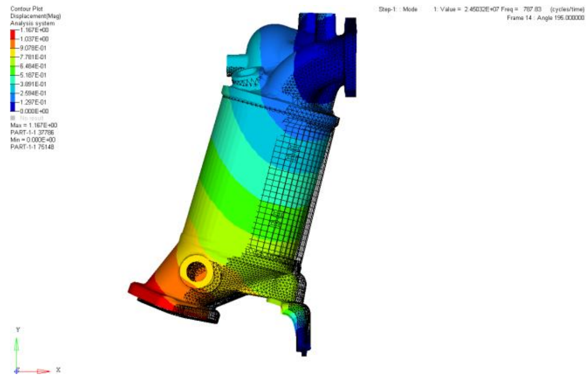

Figure 9 - Mode 1- 787.9 Hz – Longitudinal mode

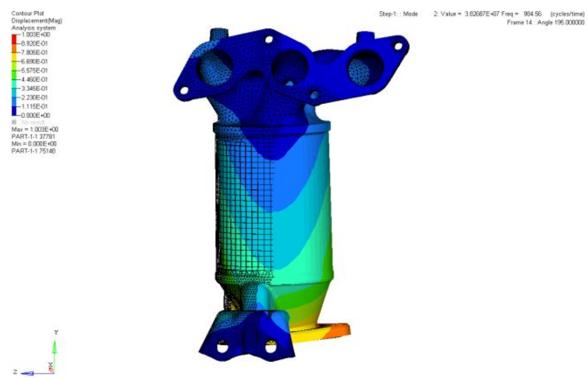

Figure 10 - Mode 2- 861.3 Hz – Lateral mode

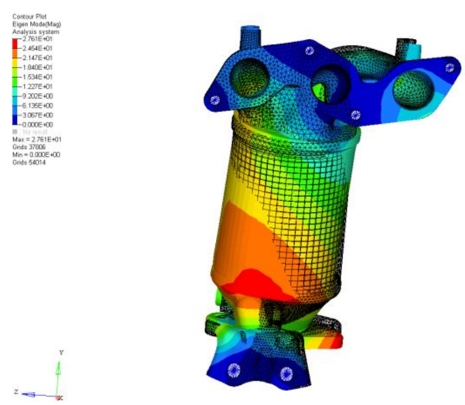

Figure 11 - Mode 3 – 1358.3 Hz – Torsion mode in Y-axis

5.2.4. OBSERVATIONS

- With temperature effect there is a change in the material stiffness hence mode shapes in hot condition varies differently for first, second & third order frequencies decreasing as whole when compared to mode shapes in cold condition.

5.3. CASE 3

- ❖ Case 3- Modal analysis in hot condition including nonlinear material properties, pre-tension force, thermal stress and mechanical stress

5.3.1. MATERIAL PROPERTIES

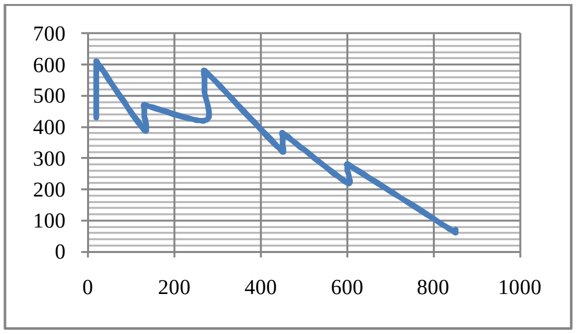

Figure 12 – Y axis – Yield stress (MPa) Vs X axis - Temperature (°C)

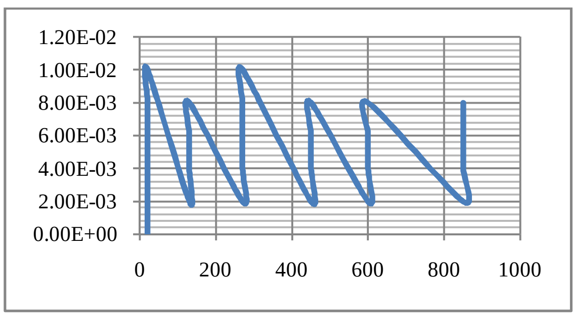

Figure 13 – Y axis – Plastic strain Vs X axis - Temperature (°C)

5.3.2. BOUNDARY CONDITION

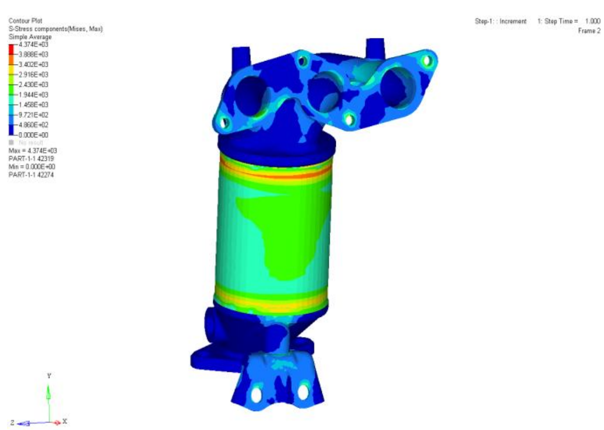

Figure 14 – Thermo-mechanical stress results

Mode Plots:

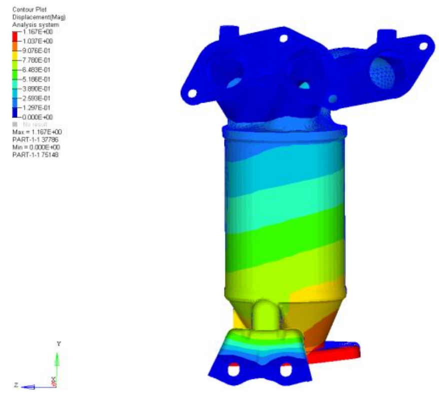

Figure 15 - Mode 1- 653.6 Hz – Vertical mode

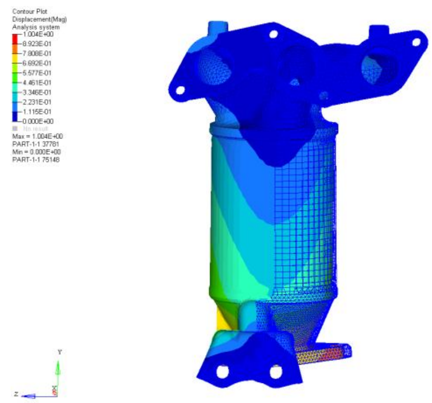

Figure 16 - Mode 2- 832.1 Hz – Lateral mode

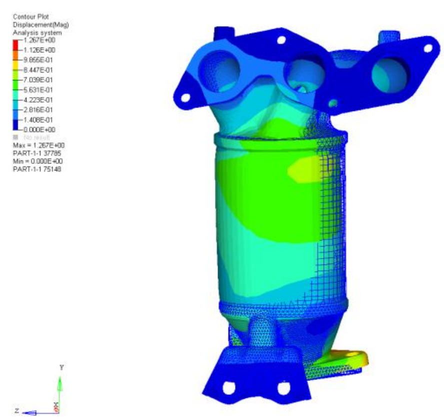

Figure 17 - Mode 2- 1216.5 Hz – torsion mode in Y axis

5.3.3. OBSERVATION

- ❖ Thermal stress generated from temperature will have impact on structural bending and torsional rigidness, which lead to decline in structural natural frequency.

6. CONCLUSION

Non-linear change in material properties caused by tail gas heating and mechanical loading from pretension leads to decline in natural frequency of the hot end exhaust system.

The result show the effects of torsional rigidity is larger than bending rigidity with temperature rising. So it is imperative to take the influence of the temperature pre-stress on exhaust manifold/ hot end vibration characteristics into account for design evaluation and validation.

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