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EXPERIMENTAL AND NUMERICAL ANALYSIS OF MODAL PROPERTIES FOR AUTOMOTIVE FUEL TANK

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ABSTRACT

A fuel tank is the one of the important parts in automotive fuel system that stores liquid fuel and provides space for integrating fuel pump, filter, regulator and gauge inside the tank. The dynamic characteristics of fuel tank are affected by the mass of liquid and other fluid - structure interactions. Therefore, knowledge of the effects of different parameters on modal properties is very important to design the fuel tank and integrate fuel system components inside the fuel tank. This work addresses the vibration characteristics of an automotive fuel tank under empty and fluid-structure coupling conditions. The finite element model for vibration analysis of the fuel tank is developed using ANSYS software and the modal parameters, namely the natural frequencies and mode shapes are predicted. Experimental setup is developed using accelerometer interfaced with data acquisition card. The frequency responses of the fuel tank are analyzed using LabVIEW software. The experiments are carried out for different fill levels of the tank. The findings show that the level of liquid influences greatly the dynamic characteristics of the tank. The results obtained from the numerical simulations are compared with experimental data and good agreements are observed.

Keywords: fuel tank, finite element method, modal properties, damping factor, natural frequency.

1. INTRODUCTION

The main function of the fuel tank is to provide a means for storing the fuel to be used by the engine and prevent gasoline vapors from escaping into the atmosphere. Under actual working condition, fuel tanks are subjected to continuous vibrations of internal pressures and external loads. The tank must pass stringent testing for hydrocarbon leakage and crash integrity prior to its use in a vehicle.

In automotive fuel system, each joint may be the source for fuel vapour leakage. In order to reduce the vapour leakage, the fuel pump, filters and regulator are placed inside the fuel tank. To ensure the proper functioning of these fuel system components, it is necessary to study the dynamic characteristic of the fuel tank.

Modal analysis is the process of determining the inherent dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes. The modal characteristics of liquid-containing structures are affected by the presence of liquid. Liquid changes the natural frequencies or mode shapes of the structure either by adding mass or through other fluid-structure interaction (FSI) mechanisms. The effects of Fluid-Structure Interaction on the modal properties of structures have been studied by many researchers in the past, and a number of papers have been published on this subject and some of the studies are summarized in this paper.

Hassan Jalali and Fardin Parvizi [1] developed the finite element model to study the modal properties of a pipe and cylindrical storage tank. The effects of liquid on the modal properties of liquid-containing structures are investigated experimentally and numerically. O. Curadelli *et al*. [2] proposed a numerical model taking into account the coupling between fluid and structure for analyzing the dynamic response of elevated spherical tanks subjected to horizontal base motion and numerical results are validated against the experimental results.

M. Amabili [3] experimentally investigated the large-amplitude response of perfect and imperfect, simply supported circular cylindrical shells subjected to harmonic excitation. Several interesting non-linear phenomena such as softening-type non-linearity, different types of travelling wave response in the proximity of resonances, interaction among modes with different numbers of circumferential waves and amplitude-modulated response are numerically reproduced using Donnell's non-linear shallow-shell theory.

Nan Liu [4] developed the finite element model to study the vibration characteristics of a rectangle liquid container under both empty and fluid-structure coupling conditions. The change in natural frequency of liquid containers which have same external physical dimension and different quantity and arrangement of rib are analyzed.

E. Askari *et al*. [5] developed an analytical model to investigate the effects of baffles on bulging and sloshing frequencies, and modes of a cylindrical container partially filled with a fluid.

S. Chantasiriwan [6] used method of fundamental solutions to determine the natural frequencies and mode shapes of cylindrical container, cylindrical quadrant container, cylindrical equilateral triangle container, hemispherical container, and cylindrical container with baffle.

Wang Wei *et al*. [7] established a finite element method for modal analysis of the liquid small amplitude sloshing with three kinds of contact line boundary conditions. Sloshing damping caused by energy dissipation at the wall, in the interior fluid and at the contact line is calculated and numerical results are compared with analytical and experimental results.

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D. Tran and J. He [8] studied the fluid-structure interaction, in particular the change in mode shapes and corresponding natural frequencies of an open cylindrical tank containing liquids with the various level of liquid by FEM and experimental modal analysis (EMA). Adam Wisniewski and Robert Kucharski [9] investigated numerically the Eigen behavior of large liquid and gas storage rectangular tanks.

From the literature review, it is noted that several studies have been carried out on the modal analysis of liquid container. The various analytical, numerical and experimental techniques have applied for modal analysis of rectangular, cylindrical and spherical containers. Studying the vibration characteristics of complex shape fuel tank is helpful for integrating fuel system components within the fuel tank. This paper presents the effect of fill levels on the model properties such as natural frequency and mode shapes of an automotive fuel tank that is actually used.

Nomenclature

F	excitation vector
F_s	excitation on the structure
F_f	excitation on the fluid
K_s	solid stiffness matrix
K_f	fluid stiffness matrix
M_s	solid mass matrix
M_f	fluid mass matrix
P	fluid stress
R	coupling matrix
c	damping matrix
k	stiffness matrices
m	mass matrix
n	integer
t	time period
t_d	time period of damped vibration.
x	displacement
ϕ	phase angle
ρ	fluid density
ω	angular frequency
ω_d	frequency of damped vibration
ζ	damping factor
δ	logarithmic decrement

The contents of the paper are organized as follows. In Section 2 the equations governing the free damped vibrations and fluid - structure interactions are stated. Section 3 outlines the numerical model of fuel tank. Section 4 presents the experimental setup used for vibration analysis. The results of the simulations and experiments are given Section 5 with a discussion of important observations. Conclusions are contained in Section 6.

2. GOVERNING EQUATIONS

2.1. Equation of motion

The equation of motion of a vibrating system can be derived using principle of conservation of energy, which states that the total energy of the vibrating system is constant, if no work is done on a conservative system by external forces. i.e., sum of potential energies and kinetic energies remains constant.
The equation of motion can be written as:

$$[m]\ddot{x}(t)+[c]\dot{x}(t)+[k]x(t)=F(t) \quad (1)$$

where [m], [c] and [k] are mass, damping and stiffness matrices, respectively.

If no external force act on the fluid-structure system, then F (t) = 0 in equation (1), the equation of motion describing damped free vibration is:

$$[m]\ddot{x}(t)+[c]\dot{x}(t)+[k]x(t)=0 \quad (2)$$

The frequencies of the vibrating system are called eigen value frequencies.

The solution of the equation (2) independently for the solid or the fluid, assumes the following form of

$$x(t)=Xe^{-i\omega t} \quad (3)$$

Substituting equation (3) and its time derivatives into the equation (2), a free vibration matrix system for fluid-structure problem leads to

$$\left(-[m]\omega^2-i\omega[c]+[k]\right)x(t)=0 \quad (4)$$

For undamped free vibration the damping matrix equals zero and the equation (4) becomes

$$\left([k]-[m]\omega^2\right)x(t)=0 \quad (5)$$

The solution is defined by x = X sin (ωt +ϕ), where X is a vector of eigenvalue, ω^2= eigen value of systems and ϕ= phase angle.

Equation (5) possesses a nontrivial solution if and only if the determinant of coefficient x (t) vanishes, that is,

$$\left|[k]-[m]\omega^2\right|=0 \quad (6)$$

Equation (5) is known as eigen value problem and equation (6) is called characteristic equation and ω is called natural frequency of the system. The solution of equation (6) is to be found numerically using standard procedures developed in the ANSYS software. The natural frequency of the system and corresponding modal is fully determined by matrixes [M] and [K], which are the natural characteristics of system.

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2.2. Logarithmic decrement

The logarithmic decrement represents the rate at which the amplitude of a free-damped vibration decreases. It is defined as the natural logarithm of the ratio of two successive amplitudes.

The solution of the equation (1) for under damped vibrating system can be expressed as:

$$x(t) = Xe^{-\zeta\omega_n t}\cos(\omega_d t - \phi) \quad (7)$$

where $\omega_d = \sqrt{1-\zeta^2}\,\omega_n$ is the frequency of damped vibration and ζ = damping factor.

Let t_n and t_{n+1} denote the times corresponding to two consecutive amplitudes x_n and x_{n+1} measured one cycle apart for an under damped system, as shown in Figure-1.

The ratio of two successive amplitude of vibration can be written as:

$$\frac{x_n}{x_{n+1}} = \frac{Xe^{-\zeta\omega_n t_n}\cos(\omega_d t_n - \phi)}{Xe^{-\zeta\omega_n t_{n+1}}\cos(\omega_d t_{n+1} - \phi)} \quad (8)$$

$$\frac{x_n}{x_{n+1}} = Xe^{\zeta\omega_n t_d} \quad (9)$$

where, t_d is the time period of damped vibration.

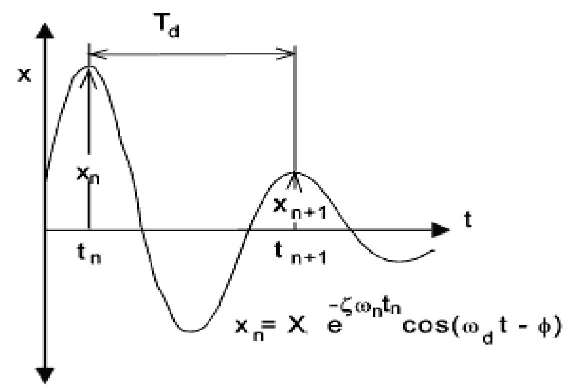

Figure-1. Response of an under damped vibrating system.

The logarithmic decrement 'δ' can be obtained from equation (9);

$$\delta = \ln\frac{x_n}{x_{n+1}} = \zeta\omega_n t_d \quad (10)$$

Substituting $t_d = \frac{2\pi}{\omega_d}$ in equation (10), δ becomes

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \quad (11)$$

For small damping, equation (11) can be approximated;

$$\delta \approx 2\pi\zeta \quad \text{if } \zeta << 1 \quad (12)$$

The damping ratio can be determined experimentally by measuring two displacements separately by any number of complete cycles. If x_1 *and* x_{n+1} denote the amplitudes corresponding to times t_1 and $t_{n+1} = t_1 + nt_d$, where *n* is integer, then

$$\frac{x_1}{x_{n+1}} = Xe^{n\zeta\omega_n t_d} \quad (13)$$

Therefore $\delta = \frac{1}{n}\ln\frac{x_1}{x_{n+1}} = \zeta\omega_n t_d$ (14)

This equation can be used to determine the damping ratio 'ζ' of the system.

2.3. Fluid-structure interaction

When the container filled with fluid vibrates, structure vibration will affect fluid, the influenced fluid will generate a reaction to structure, and therefore, wave equation in fluid and dynamic equation in solid should be calculated at the same time.

The governing finite element matrix equations then become

$$[m_s]\ddot{x} + [K_S]x = F_s - [R]P \quad (15)$$

$$[M_f]\ddot{P} + [K_f]P = F_f + \rho_o[R]\ddot{x} \quad (16)$$

where, [R] is a "coupling" matrix that represents the effective surface area associated with each node on the fluid-structure interface (FSI), M_s is solid mass matrix, M_f is fluid mass matrix, K_s is solid stiffness matrix, K_f is fluid stiffness matrix, P is the fluid stress and ρ is the fluid density.

The fluid-structure coupling equation in matrix form can be obtained combining the two equations (15) and (16) into a single equation.

$$\begin{bmatrix} M_s & 0 \\ -\rho_o R & M_f \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{P} \end{Bmatrix} + \begin{bmatrix} K_s & R \\ 0 & K_f \end{bmatrix}\begin{Bmatrix} x \\ P \end{Bmatrix} = \begin{Bmatrix} F_s \\ F_f \end{Bmatrix} \quad (17)$$

For free vibration fluid-structure coupling equation becomes

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$$\begin{bmatrix} M_s & 0 \\ -\rho_o R & M_f \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{P} \end{Bmatrix} + \begin{bmatrix} K_s & R \\ 0 & K_f \end{bmatrix} \begin{Bmatrix} x \\ P \end{Bmatrix} = 0 \qquad (18)$$

Solving the characteristic equation (18), the vibration characteristic of liquid containers under fluid-structure coupling can be obtained.

3. NUMERICAL MODEL

The point clouds of a fuel tank are generated using MicroScribe 3D Digitizer. The point clouds and CAD model of the fuel tank are shown in Figure-2. Then surfaces are constructed from point clouds using RhinoCeros software. The CAD Model of the actual fuel tank is imported into PRO-E software as IGES file.

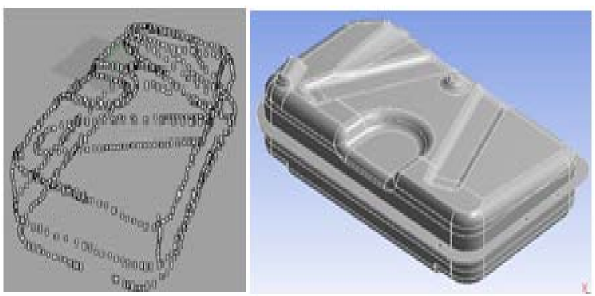

Figure-2. Point cloud and CAD model of a fuel tank.

The small features and fillet radius at the corners are approximated to reduce the complexity of the CAD model. The model is converted into a shell of thickness 2 mm by using sheet metal option and different liquid levels are specified. Then the IGES file is imported into finite element package ANSYS for modal analysis.

The SHELL181 element and HSFLD242 - 3D Hydrostatic Fluid elements are used to mesh the tank structure and fluid medium in the tank. SHELL181 can be used for analyzing thin to moderately-thick shell structures. It is a four-node element and each node has 6 DOF namely translations in the x, y, and z directions, and rotations about the x, y, and z-axes. The geometry of SHELL181 element is shown Figure-3 a). In this analysis the force is transmitted from tank wall to fluid and again from fluid to tank wall. SHELL181 accounts for load stiffness effects of distributed pressures; therefore, this element is used for meshing the tank structure.

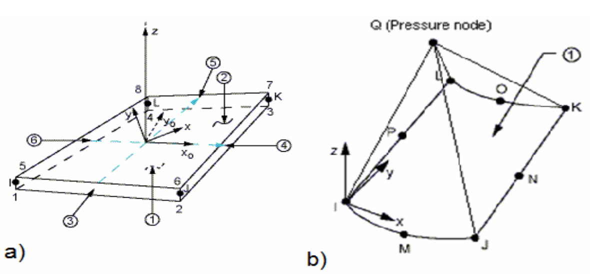

Figure-3. Element geometry.

HSFLD242 is used to model fluids that are fully enclosed by solids. The hydrostatic fluid element is well suited for calculating fluid volume and pressure for coupled problems involving fluid-solid interaction. This element is pyramid shaped with the base overlaying a 3-D solid or shell element face and the vertex at a pressure node. The element nodes shared with base element have only translation degrees of freedom. The Geometry of HSFLD242 element is shown Figure-3 b).

The number of elements and nodes generated for empty tank are 63354 and 123986 respectively. The meshed model of the tank is shown in Figure-4.

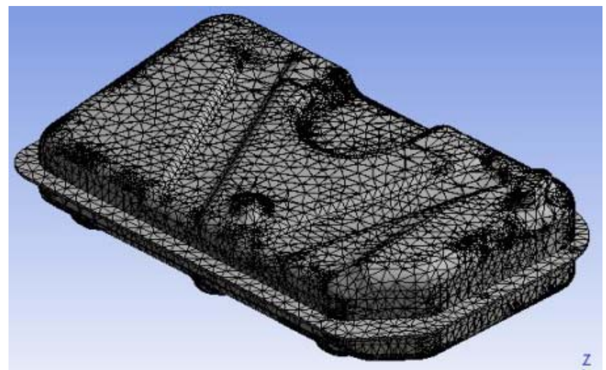

Figure-4. Meshed model of fuel tank.

The displacement constraints are specified at the outer rib of the tank as shown in Figure-5. Acceleration due to gravity of 9.81m/s^2 is applied to the entire tank in Z direction. The hydrostatic pressure is specified at the fluid-structure interface for various fill levels, namely 25%, 50% and 75% fill levels. Figure-6 shows the fluid-structure interface of tank filled with water at 25% level of fill.

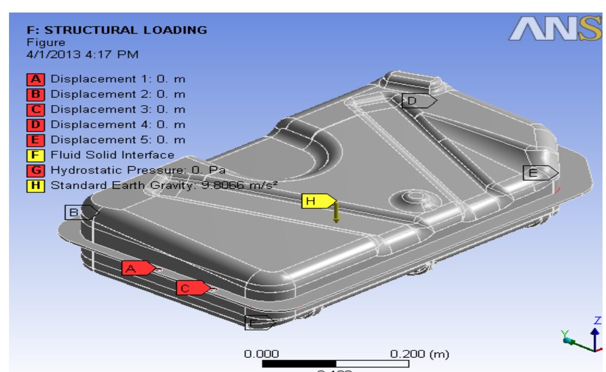

Figure-5. The numerical model with load conditions.

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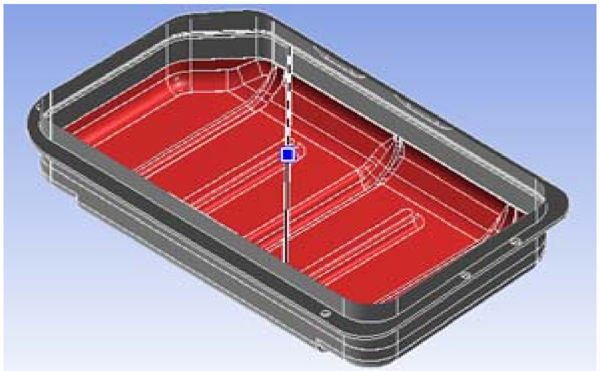

Figure-6. Fluid-structure interface of 25% fill level.

4. EXPERIMENTAL SETUP

Experimental setup was developed to measure the frequency response of the fuel tank using accelerometer (make: Instrol Devices) with National Instruments Data Acquisition Card (NI-DAQ) card (USB-6008, 8 inputs and 2 outputs). A fixture was made to support the tank as it is fixed in the vehicle. The experimental setup is shown in Figure-7.

Figure-7. Experimental setup - fuel tank with accelerometer

The accelerometer probe was mounted on the tank as shown in Figure-7 and the momentary load was applied to the tank using hammer. The frequency responses were record with the help of LabVIEW software. The experimental results are presented in the section 5.2.

5. RESULTS AND DISCUSSIONS

5.1. Finite element modal analysis

The fluid structure interaction, in particular the change in mode shapes and corresponding natural frequencies of an automotive fuel tank containing different levels of liquid is studied by numerical modal analysis. The results are delineated in this section.

5.1.1. Effects of liquid level on mode shape

The natural frequencies and mode shapes are analyzed by FEM software ANSYS, first for the empty tank then for three different fill levels, nominally 25%, 50% and 75% of fill levels. It is found that the level of liquid influences greatly the dynamic characteristics of the tank-fluid structure.

Figures 8 (a) - 8(f) show the first six mode shapes and the corresponding natural frequencies of the empty tank.

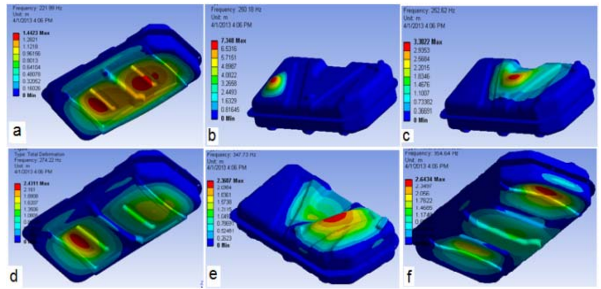

Figure-8. The mode shapes for empty tank a) first mode, b) second mode, c) third mode, d) fourth mode, e) fifth mode, and f) sixth mode.

Figures 9(a) - 9(f) show the first six mode shapes and corresponding natural frequencies of the 25% filled tank.

Figures 10(a) - 10 (f) show the first six mode shapes and corresponding natural frequencies of the 50% filled tank.

Figures 11(a) - 11 (f) show the first six mode shapes and corresponding natural frequencies of the 75% filled tank.

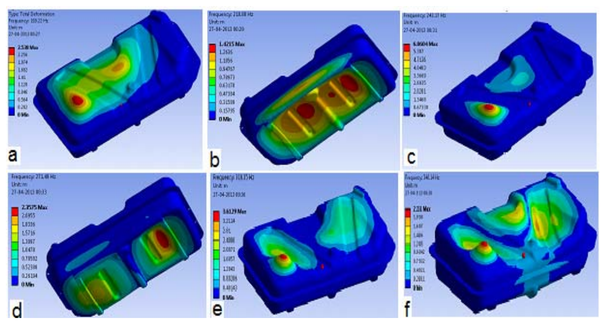

Figure-9. The mode shapes for 25% fill level a) first mode, b) second mode, c) third mode, d) fourth mode, e) fifth mode, and f) sixth mode.

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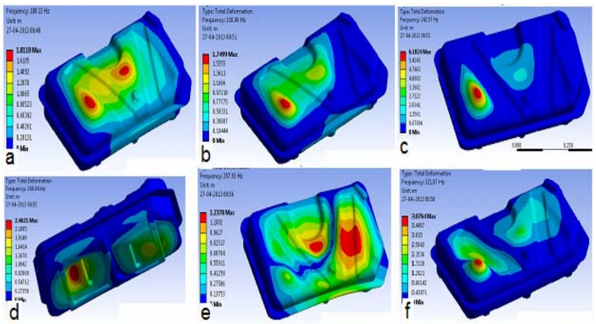

Figure-10. The mode shapes for 50% fill level a) first mode, b) second mode, c) third mode, d) fourth mode, e) fifth mode, and f) sixth mode.

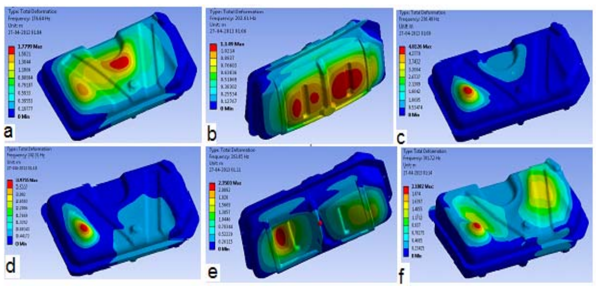

Figure-11. The mode shapes for 75% fill level a) first mode, b) second mode, c) third mode, d) fourth mode, e) fifth mode, and f) sixth mode.

The results obtained from numerical analysis show that the mode shapes are affected as the fill level increases in the tank. From the simulations the following points are observed:

Some of the modes are wiped out at certain liquid levels, for example the modes shown in Figures 8 (a), 8(c), 9 a), 9(b) 9(e) appear only in empty tank and 25% filled tank.

The mode shapes shown in Figures 8 (b), 8(d), 9(c), 9(d) 10 (c) 10(d), 11(c) and 11(e) exist at all levels of liquid.

Some of modes are no longer exist at other liquid levels, i.e. the mode shape shown in Figures 8 (e), 8 (f), 9 (f), 10 (a), 10(b), 10 (e), 10 (f), 11 (a), 11 (b), 11 (d) and 11(f) are the new modes formed at certain fill levels.

5.1.2. Effects of liquid levels on natural frequencies

The natural frequencies of a fuel tank are also affected as the fill level increased. Figure-12 shows the influence of water level on the natural frequencies at all the six modes. All natural frequencies decreases as water level is increased, this results show that the effect of fill level on the vibration characteristics of the tank is the addition of mass.

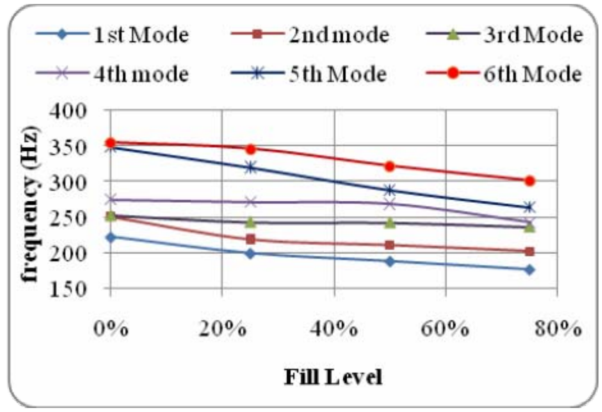

Figure-12. Effect of fill level on the natural frequencies of tank.

5.2. Experimental modal analysis

To validate the results obtained by numerical analysis and to determine the influence of fill level on the damping characteristics of the tank, experimental modal analysis are carried out. The vibration characteristics of the empty tank and various fill levels are analyzed using acceleration meter interfaced with DAQ card and LabVIEW software.

The frequency spectra of the amplitude histories are estimated by applying Welch's method. In Welch's method, the time signal is divided into a number of overlapping segments and then average value of squared magnitude DFTs of the signal segments are calculated.

The Welch's Power Spectral Density (PSD) estimate $G_w(i)$ is given by [13]

$$G_w(i) = \frac{1}{L(2S-1)} \sum_{s=0}^{2S-2} \left| Y_s(i) \right|^2 \ , 0 \le i \le \mathrm{L} \tag{19}$$

where

$$Y_s(i) = DFT\{w(b)x_s(b)\}, 0 \le \mathrm{s} < 2\mathrm{S} -1 \tag{20}$$

S = number of signal segments

$x_s(b)$ = segment number '*b*' of the measured signal

L = segment length

w (b) = window function

The PSD estimated by Welch's method for various fill level are shown in Figures 13 - 16.

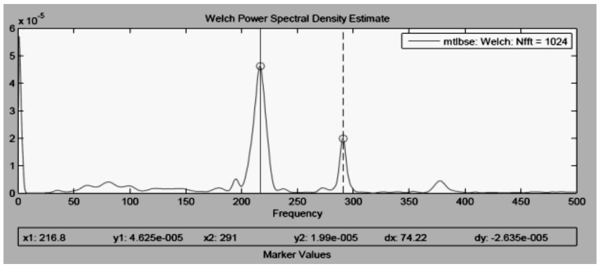

Figure-13. Frequency response for empty tank.

Figure-13 shows the PSD of empty tank. The first fundamental frequency obtained for an empty tank is 216.8 Hz.

The frequency response for 25% filled tank is shown in Figure-14. The fundamental frequency of 171.9 Hz is obtained.

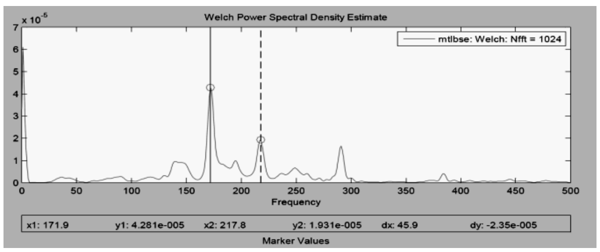

Figure-14. Frequency response for 25% filled tank.

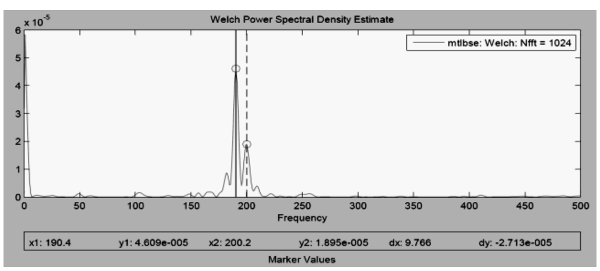

Figure-15. Frequency response for 50% filled tank.

The frequency response for 50% filled tank is shown in Figure-15. The first fundamental frequency of 190.4 Hz is obtained.

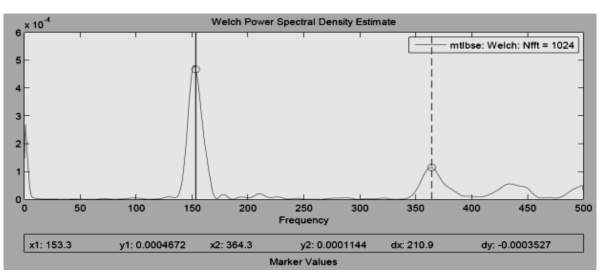

Figure-16. Frequency response for 75% filled tank.

The first fundamental frequency obtained from the experimental modal analysis for 75% fill level is 153.3 Hz and it is presented in Figure-16.

The first natural frequency values obtained from finite element modal analysis and experimental modal analysis are presented in Figure-17.

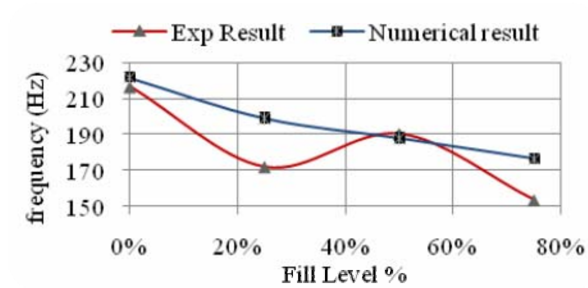

Figure-17. Comparison of first natural frequency of numerical and experimental results.

Results obtained from the numerical analysis show that the frequency decreases as water level increases for all the mode shapes. This is because increasing level of water would increase mass of the system and also increases damping.

In experimental modal analysis frequencies were not always found to decrease with increasing liquid levels. At 50% fill level experimental result has deviation from numerical result. The reason is that there are many modes of close frequencies; the result reported by experimental analysis probably includes the contribution of these neighboring modes.

The damping factor of an automotive fuel tank for different fill levels is experimentally calculated using equation (14) under free vibration and the results are shown in Figure-18.

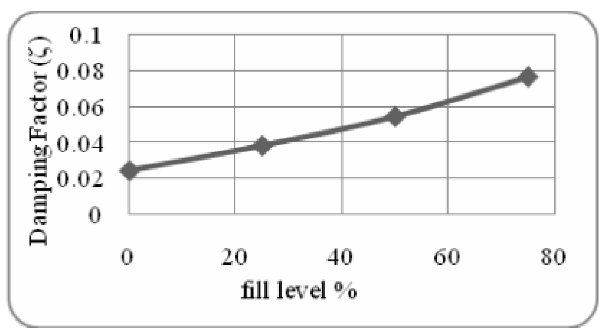

Figure-18. Damping factor versus percentage of fill levels.

It is observed that the damping factor increases when the fill level increased, this is due the increase in the mass of the system.

6. CONCLUSIONS

In this work, the effects of liquid level on the modal properties of the fuel tank are analyzed experimentally and numerically. The natural frequencies and mode shapes are predicted using ANSYS software. Experimental studies are carried out to validate the simulation results and determine the damping factor of the fuel tank for different fill levels. The natural frequency values obtained from the simulation are in good agreement with the experimental results.

The results obtained from the finite element analysis show that, some of the modes are wiped out and new modes are appeared at certain liquid levels, and the natural frequencies of the tank are decreased as the fill levels increased. This is due the increase in mass and damping characteristic of the tank. However, the natural frequencies of 50% filled tank are affected by neighboring modes of very close frequencies.

This study helps understand the effect of liquid levels on the dynamic characteristics of the fuel tank and design the fuel tank and completely avoid the occurrences of resonance at all operating conditions of the vehicle.

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