3rd International Conference on Material and Component Performance under Variable Amplitude Loading, VAL2015

Static and Dynamic Analysis of Plastic Fuel Tanks Used in Buses

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Abstract

In today’s world the conventional metal fuel tanks used in buses are being replaced with plastic fuel tanks. In this paper, a part of the project which aims to develop a computer aided methodology for developing/designing of the plastic fuel tanks based on static and dynamic analysis is presented.

Coupon tests are conducted to acquire the mechanical properties of the plastic material. In the static analysis, equivalent static loads are applied to the fuel tanks. In the dynamic analysis, the time varying loading and the inertia of the fluid and fuel tanks are taken into account using modal transient analysis.

1. Introduction

Plastic fuel tanks instead of steel tanks have been used in recent years due to their several advantageous properties such as their light weight, low material and production costs, high safety levels, greater capacity due to greater choice of shape, non-corrosiveness property and good design flexibility. In 1972, VW Passat cars with plastic fuel tanks were manufactured for the first time as a mass production. Since then the major automobile manufacturers around the world have started to show interest in and to work on the use of the plastic fuel tanks [1].

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There exist many researches on the fuel tanks in the literature. Fuel permeation mechanism was defined and multilayer fuel tanks' structural features and the development of resins were discussed by Kurihara et al. [2]. The properties, advantages and disadvantages of high capacity plastic fuel tanks of trucks were investigated in [3]. Researchers found that, because of the high levels of sulfur in the content of fuel sold in Brazil, corrosion occurred inside the steel fuel tanks and, the fuel caused some damages to the engine injection system. After examining several alternative materials in terms of certain features such as cost, safety, recycling, replacement and compatibility, they came up with the solution that the fuel tanks should be manufactured using polyethylene material.

Life cycles of steel and plastic fuel tanks are compared by Stephens et al. [4]. In order to obey the federal standards and to enhance the fuel economy, usage of light components in automotive industry became a requirement. It is also important to manufacture components regarding the environmental concerns. The results show that, while material is being manufactured, solid waste generation is high for steel compared to the polyethylene (HDPE), but the energy consumption is higher in manufacturing process of the HDPE. In the usage phase, it is desirable that the energy consumption level is lower. In fact, considering the environmental conditions, HDPE is more advantageous than steel. Considering the production stage, it is observed that the HDPE is more environment-friendly compared to the steel. However, at the end of the lifecycle, steel seems to be more sensitive to the environment because of the disadvantage of the plastic material that it is not recyclable. To be able to get rid of this disadvantageous situation, some researchers have conducted research on recyclability of these plastic materials. In one of these studies, HDPE fuel tanks that consist of 27% recycled material were manufactured. Those recycled materials were obtained from the fuel tanks of vehicles that had already completed their life cycles [5].

The effect of sloshing in a fuel tank is one of the most crucial problems for the fuel tank design, and, obviously, percentage of fullness influences sloshing effects. The sloshing in a partially full tank in the state of failure was investigated by Vytla et al [6]. In this study, there are two tanks filled up to 53% and 40%, and two breaking events, a light one and a heavy one were investigated. The sloshing problem was solved by the fluid-solid interaction technique. Water is used instead of fuel. It is found that the sloshing level increases in the case of low percentage of fullness. In other words, it takes time for a liquid to become calm and there is a high probability of fuel noise while the water percentage is lower. In another study, the fuel in a tank was modeled. After three fullness rates of 51 L, 42 L and 28 L were compared to each other, the 42 L of fuel fullness were found to cause the highest noise [7].

There are several methods to model the fluid in fuel tanks. One of them is the oscillation method which is used by Reis and Pala [8]. In this work, the effect on braking dynamics of the liquid behavior is examined. The LS-DYNA finite elements analyses program was used to solve the Fluid-Solid Interaction (FSI) problems by Vesenyak et al. [9]. Fluid sloshing problem was analyzed using the Lagrange, Euler, ALE and SPH formulations. Lagrange and ALE formulations were found as the best simulations when they were compared to the test results. Hence, it was also concluded that these two formulations could be used for automobile fuel tanks.

Modal analysis of transient liquid sloshing in partially full fuel tanks is presented by Zheng et al. [10]. In this study, the FLUENT program is employed for simulations. Cylindrical tanks are taken partially filled (40%), and longitudinal accelerations between 0.1g and 0.4g were applied to the tanks. It is found, by analyzing temporary sloshing forces and liquid surfaces, that sloshing has a periodic oscillation. Then, the Fourier Transform is used to convert the sloshing forces from time domain to frequency domain. The spectrum analysis indicates that the fundamental oscillation which has greatest amplitude is the most significant oscillation for liquid sloshing.

Sloshing modes of the fluid in a rectangular tank subjected to horizontal harmonic motion are investigated by Virella et al. [11]. Both linear and nonlinear 2D finite elements are used. Natural sloshing periods and their modal pressure distribution are investigated using the ABAQUS program. Bulk response of the fluid is modeled using linear state equation and Newton’s viscous shear model. The ABAQUS program suggests that the elastic bulk module should be two or three times smaller than the real value.

In this paper, a part of the project which aims to develop a computer aided methodology for developing/designing of the plastic fuel tanks based on static and dynamic analysis is presented. First, the mechanical properties of the plastic fuel tanks are obtained from the coupon tests by universal testing machines and then these properties are used in the finite element analysis. The finite element analyses are conducted in two stages. In the first stage, equivalent static loads of real dynamic loadings are applied to the fuel tanks. Maximum/minimum “g” forces are computed using the
acceleration-time data gathered from the specific bad road runway experiments. The deflection and stress distributions in the fuel tanks are obtained, and these results are compared with the test data. In the second stage, dynamic analyses are performed to include the effect of dynamic loads.

2. Formulation

In acoustic fluid-structural interaction (FSI) problems, the structural dynamics equation must be considered along with the Navier-Stokes (momentum) equation and the continuity equation. The discretized structural dynamics equation can be formulated using the structural elements. The fluid momentum (Navier-Stokes) equations and continuity equations are simplified to get the acoustic wave equation as follows

$$\nabla \left( \frac{1}{\rho_0} \nabla p \right) - \frac{1}{\rho_0 c^2} \frac{\partial^2 p}{\partial t^2} + \nabla \left( \frac{4 \mu}{3 \rho_0} \nabla \left( \frac{1}{\rho_0 c^2} \frac{\partial p}{\partial t} \right) \right) = - \frac{\partial}{\partial t} \left( \frac{Q}{\rho_0} \right) + \nabla \left( \frac{4 \mu}{3 \rho_0} \nabla \left( \frac{Q}{\rho_0} \right) \right) \quad (1)$$

Where, $c =$ speed of sound in fluid medium $p =$ acoustic pressure ($= p(x,y,z,t)$)

$\rho_0 =$ mean fluid density $Q =$ mass source in the continuity equation

$K =$ bulk moduls of fluid $t =$ time

$\mu =$ dynamic viscosity

In which the following assumptions are made:

- The fluid is incompressible
- There is no mean flow of the fluid.

The acoustic wave equation can be written in matrix notation. The discretized wave equation is given as follows

$$[M_F]\{\ddot{p}_e\} + [C_F]\{\dot{p}_e\} + [K_F]\{p_e\} + \bar{p}_0[R]T\{\ddot{u}_{F,e}\} = \{f_F\} \quad (2)$$

Where,

$[M_F] = \bar{p}_0 \iiint_{\Omega_F} \frac{1}{\rho_0 c^2} (N)(N)^T dV =$ acoustic fluid mass matrix

$[C_F] = \bar{p}_0 \iiint_{\Omega_F} \left( \frac{4 \mu}{3 \rho_0 c^2} \right) [\nabla N]^T [\nabla N] dV =$ acoustic fluid damping matrix

$[K_F] = \bar{p}_0 \iiint_{\Omega_F} \left( \frac{1}{\rho_0} \right) [\nabla N]^T [\nabla N] dV =$ acoustic fluid stiffness matrix

$[R]^T = \partial_{\Omega_F} (N)(n)(N)^T dS =$ acoustic fluid boundary matrix

$\{f_F\} = \bar{p}_0 \iint_{\Omega_F} \frac{1}{\rho_0} (N)(N)^T dV(q) + \bar{p}_0 \iint_{\Omega_F} \left( \frac{4 \mu}{3 \rho_0 c^2} \right) [\nabla N]^T [\nabla N] dV(q) =$ acoustic fluid load vector

$\bar{p}_0 =$ acoustic fluid mass density constant \[12\].

2.1. Modal Analyses

Modal analyses are used to model the acoustic modes of an acoustic cavity (standing waves) or to compute the mode shapes of vibro-acoustic systems. For pure acoustics modal analysis the acoustic modes are computed using the following equation

$$(-\omega^2 [M_F] + j \omega [C_F] + [K_F])\{p\} = 0 \quad (3)$$

For fluid structure interaction problems the acoustic and the structural coupled modes are computed using the following equation
2.2. Sloshing

Assuming that the actual surface is at an elevation $\eta$ relative to the mean surface in $z$-direction, the pressure for a sloshing (free) surface is given by

$$p = \rho_f g \mu$$

$$\left(-\omega^2 \begin{bmatrix} M_s & 0 \\ p_b R^T & M_f \end{bmatrix} + j \omega \begin{bmatrix} C_s & 0 \\ 0 & C_f \end{bmatrix} + \begin{bmatrix} K_s & -R \\ 0 & K_f \end{bmatrix}\right) \begin{bmatrix} u \\ \eta \end{bmatrix} = 0$$

(4)

Where,

$$S_f = \frac{\rho_o}{\rho} \int_{\text{solid}} \frac{1}{\rho_f} \{N\}^T \{N\} dS$$

is the acoustic sloshing mass matrix [13].

3. Model

The solid models (CAD) of the selected fuel tanks, which were damaged during the tests, are taken from the CATIA library at MBT and are transferred into the ANSYS finite element program to generate the finite element models. Figure 1 shows the meshed geometry of the plastic fuel tank. Liquid is modeled as a solid thru acoustic elements. Finite element analyses are executed for different fullness rates. During analysis, 52841 SHELL181 elements are used in the structural parts and 374766 FLUID30 elements are used in 100% full tanks. First, for all these analyses, fuel tanks are assumed to have no constraint. Later, fuel tanks are assumed to be fixed from the lower surfaces and all analyses are repeated. The motion of the tank in three directions in space was constrained by straps which tie the tank to the framed structure. Thus, the accuracy of the results has been improved.

Fig 1. Fuel Tank Meshed Geometry

4. Material Properties

Tensile mechanical properties of the plastic material at room temperature were obtained by the tensile tests with the guidance of the ASTM D638-10 Standard [14]. The tests are performed using the universal testing machine with the load cell to obtain load data, and $0^\circ$-$90^\circ$ rosette type strain gauges for each specimen. Six specimens were tested. The results for ultimate strength, 0.2 % offset yield strength, modulus of elasticity and Poisson’s ratio of those specimens with the average values are given in Table 1. Stress-strain histories of all tested specimens until almost the limit of the strain gauges are shown in Figure 2.

Table 1. Tensile material properties of the plastic at room temperature.

| Specimen | Ultimate Strength (MPa) | 0.2 % Offset Yield Strength (MPa) | Modulus of Elasticity (MPa) | Poisson’s Ratio |
|----------|-------------------------|----------------------------------|-----------------------------|----------------|
| T-1-RTD  | 15.46                   | 8.20                             | 890.16                      | 0.43           |
| T-2-RTD  | 16.06                   | 7.50                             | 885.18                      | 0.42           |
The density of the plastic material is determined by the liquid immersion method that is specified by the ASTM 792-13 Standard [15]. The water was chosen as the liquid. The tests for water were done using the density at 24.5 °C. Five specimens were tested, the density values of the specimens and the average density of all of them were calculated and shown in Table 2.

Table 2. Test results for density of the plastic

| Specimen | Density (kg/m³) |
|----------|-----------------|
| D-1-RTD  | 921             |
| D-2-RTD  | 917             |
| D-3-RTD  | 919             |
| D-4-RTD  | 916             |
| D-5-RTD  | 917             |
| **Average** | **918**         |

5. Numerical Solution and Discussion

5.1. Natural Modes

Normalized natural frequencies for different fullness ratio of fuel tanks are presented in Table 3. According to the analysis results, the acoustic elements reduce the natural frequencies as expected. Results of the PERMAS programs for no constraint cases are also included in Table 3.

Table 3. Normalized Natural Frequencies-According To Different Fill Ratio of Fuel Tanks

| ANSYS RESULTS | Mode #1 | Mode #2 | Mode #3 | Mode #4 | Mode #5 | Mode #6 |
|---------------|---------|---------|---------|---------|---------|---------|
| Empty Tank    | 54.93   | 56.86   | 65.37   | 72.89   | 78.83   | 90.21   |
| Empty Tank - fixed | 39.64   | 61.73   | 67.32   | 70.96   | 77.80   | 93.00   |
| 1/4 - Full Tank | 1.93    | 2.13    | 2.42    | 17.87   | 26.61   | 30.19   |
| 1/4 - Full Tank - fixed | 23.27   | 28.38   | 44.97   | 48.44   | 55.48   | 59.33   |
| Half - Full Tank | 2.51    | 2.71    | 3.00    | 15.49   | 19.14   | 25.61   |
| Half - Full Tank - fixed | 14.88   | 18.28   | 32.39   | 35.97   | 41.56   | 42.21   |
| 3/4 - Full Tank | 1.17    | 2.82    | 2.98    | 3.31    | 7.67    | 18.12   |
| 3/4 - Full Tank - fixed | 1.00    | 2.03    | 16.08   | 20.64   | 23.92   | 27.75   |
| Full Tank     | 2.35    | 2.77    | 3.32    | 5.73    | 7.40    | 14.39   |
In this model, fuel was modeled by means of the solid elements with low elastic moduli. By the natural frequency analysis, the dynamic structural behavior of the fuel tank was gathered. The first three vibration mode shapes of the full tank are shown in Figure 3. In this case, the model is formed using structural solid elements, and has no constraints at the bottom surface.

| PERMAS RESULTS | Mode #1 | Mode #2 | Mode #3 | Mode #4 | Mode #5 | Mode #6 |
|----------------|---------|---------|---------|---------|---------|---------|
| Empty Tank     | 49.19   | 51.87   | 61.39   | 67.58   | 72.35   | 81.87   |
| 1/4 - Full Tank| 37.66   | 40.26   | 46.86   | 47.67   | 53.17   | 56.24   |
| Half - Full Tank| 36.03  | 37.75   | 42.26   | 44.82   | 46.80   | 49.58   |
| 3/4 - Full Tank| 33.94   | 36.25   | 37.36   | 40.68   | 42.63   | 45.35   |
| Full Tank      | 33.38   | 34.56   | 37.01   | 40.70   | 43.09   | 45.78   |

Figures 4 show the first three mode shapes for fixed empty tank and fixed full tank, respectively.

5.2. Modal Transient Analysis

Figure 5 shows static analysis and transient analysis model. The PERMAS solver program is used for these analyses. Equivalent static loads are applied to the fuel tanks. Maximum/minimum “g” forces that the fuel tanks can encounter with under operational conditions were obtained using the acceleration-time data collections from the
specific bad road runway experiments of the test department of MBT, R&D Center. Under these loadings, the deflection and stress distribution in the fuel tanks are examined.

Maximums and minimums of the collected acceleration values by MBT testing department were used (Fig 6). In each direction, equivalent internal pressure distribution which simulates liquid was calculated by the formula below. Frictionless contact is considered. 24 load cases are created using the following ratios. X coefficient is determined by MBT.

Table 4. Acceleration Ratio

| P_x | P_y | P_z |
|-----|-----|-----|
| ±100% P_x | ± X% P_y | ± X% P_z |
| ± X% P_x | ±100% P_y | ± X% P_z |
| ± X% P_x | ± X% P_y | ±100% P_z |

Basic Formula to calculate internal liquid (fuel) pressure is

\[ P = \rho \times a \times h \times (\text{acceleration ratio}) \quad (\text{N/mm}^2) \]  

Where,

- \( P \): internal liquid (fuel/Ad-Blue) pressure \quad (\text{N/mm}^2)
- \( a \): accelerations measured by test \quad (\text{mm/s}^2)
- \( \rho \): density of the liquid \quad (\text{ton/mm}^3)
- \( h \): height of the fluid \quad (\text{mm})

Equivalent pressure is obtained using the formula in Eq. (7) and X coefficient in Table 4. The macro code was developed using the Microsoft Excel and pressure loads were calculated. During this process, accelerations, participation factor, fill factor, dimensions of the tank (it can be calculated automatically), density of the liquid, pretension, name of the Medina file and job name are assigned as inputs of the macro code. Analysis parameters and loads were obtained as outputs. Accelerations for the total of 24 cases are created. Then analyses are realized by the PERMAS program. The solution of the full tank is presented in Table 5.
5.3. Transient Dynamic Analysis

In this analysis, acceleration – time data history is applied to the bottom of the structure using the Large Mass Method. The RMS values of the stress – time data histories are collected and additional hotspots are searched from the static load based analysis (Fig 7). Pre-tensioning of the retaining straps is not taken into consideration.

![Stress Effected By Minimum Acceleration](image)

Fig 7. Stress RMS Values for Transient Analysis

6. Conclusion

A part of the project which consists of the design, analysis and optimization of plastic fuel tanks used in the buses is presented in this paper. Fuel tanks are subjected to varying dynamic loads due to severe road conditions. These loads may cause failures and cracks in the fuel tanks. To design desirable fuel tanks with high reliability, it is vital to have an idea about both the results of their failures and their life span. For this purpose, static and dynamic analyses are performed using finite element method. The aim of the project is defining the analysis methodology to prevent the damages. The project is ongoing until the analyses are verified by the tests. So far, an important part of the simulation method has been completed for plastic fuel tanks for buses.

Acknowledgements

This work, which is a collaboration project of Mercedes-Benz Turk A.S. and the Istanbul Technical University, is supported by the Republic of Turkey, the Ministry of Science, Industry and Technology under the project number [0488.STZ.2013-2].

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