Validation of Fuel Tank Slosh Noise and Vibration Predictions

Arnaud CHARPENTIER1; Ho Geon KIM2; Yi HWANG3; Jinchae NOH4; Stephane CARO5;

1,5 wave six LLC.
2 CD adapco, South Korea
3,4 Donghee Industrial Co., Ltd, South Korea

ABSTRACT

Noise resulting from fuel tank slosh is becoming an important issue for hybrid vehicles design. Noise issues are typically revealed at prototype stage, requiring expensive redesign and qualification tests. Prediction of noise and vibration from fuel tank slosh has been challenging due to 1) broadband characteristic of slosh noise making time domain coupled fluid-structure solutions inadequate and 2) important influence of fluid loading on structural response making traditional one-way coupled or decoupled vibro-acoustic approaches inaccurate.

A novel approach for the efficient and accurate prediction of tank slosh noise and vibration over a broad frequency range is introduced. The method relies on prediction of time varying tank wall pressure using CFD simulation and computation of fully coupled vibro-acoustic response of partly filled tank under slosh load. Validation results are shown for a simplified tank structure, including free surface, tank internal wall pressure, tank vibration and radiated noise results.

Keywords: Tank slosh noise, FEM, BEM

1. INTRODUCTION

Reduction in noise emissions from powertrain in hybrid vehicles has led previously masked noise sources to become more noticeable by vehicle occupants hence negatively affecting perceived quality. Interior noise caused by fuel tank slosh during acceleration and sudden stop has thus become an important aspect of tank design and integration for OEMs and suppliers. Presently, noise and vibration issues are mainly revealed late in design stage, once physical prototype is tested, requiring expensive product redesign and qualification tests. There is thus significant interest in being able to predict tank slosh noise early during design phase of vehicle.

While simulation method for prediction of fuel slosh has been well established (1), prediction of noise and vibration from fuel tank slosh has been challenging. Part of the challenge resides in the broad frequency content of typical slosh noise events (2) which makes traditional time domain coupled fluid-structure solutions inadequate. Furthermore, while vibro-acoustic predictions of tank slosh have in the past been carried out, the vibration calculations are typically performed in isolation, neglecting the influence of fuel fill. The presence of fluid in the tank actually has a large (mass loading) effect on the vibro-acoustic response of the tank. A fully coupled vibro-acoustic approach is thus mandatory to tackle slosh noise simulation.

In this paper, a novel approach for the efficient and accurate prediction of tank slosh noise and vibration over a broad frequency range is introduced. The method relies on prediction of time varying tank wall pressure using CFD simulation and computation of coupled vibro-acoustic response of partially filled tank as a function of time and frequency.

Validation results are shown for a simplified tank structure, including free surface, tank internal wall pressure, tank vibration and radiated noise results.

1 arnaud.charpentier@wavesix.com
2 hogeon.kim@cd-adapco.com
3 yhwang@donghee.co.kr
4 jcnoh@donghee.co.kr
5 stephane.caro@wavesix.com
2. Review of the modeling approach

2.1 Review of tank slosh noise sources and transmission mechanisms

Noise generation mechanism due to tank slosh has previously been characterized and is well documented in the literature. For instance, tank slosh noise sources are typically divided into three main categories of events (2). The mechanical excitation of tank wall due to inertial forces as a result of fluid motion is commonly referred as “hit” noise and is the main focus of the present work. In some instances, a pocket of air gets trapped between the fluid and tank wall, generating a mainly low frequency “clonk” type noise that is a result of the sudden compression and expansion of entrapped air as fluid sloshes around. Finally, noise sources are generated within the tank as a result of water-water collision, bubbles expansion (cavitation) and water droplets hitting the boundaries of the tank or fluid surface. This last category is generally referred as “splash” noise and remains a challenge for both CFD and vibro-acoustic simulations due to its complex nature and high-frequency content (2).

Considering now the transmission mechanisms from the source to the receiver through automotive vehicle, we find that multiple transfer paths co-exist (and may be of equal interest depending on the vehicle configuration and frequency of interest) as illustrated in Figure 1 below. In particular, one of the transmission mechanisms, referred to as “airborne path”, corresponds to sound radiation from the tank surface that propagates around the vehicle and transmits through the chassis, open windows etc… The other transmission mechanism, referred to as “structure-borne path”, consists in the transmission of tank vibration through mounts and decoupling pads to the vehicle chassis and subsequent radiation into the cabin through carpet etc… Depending on the vehicle configuration (tank location, interior trim, …) and sloshing mechanism, both airborne and structure borne paths may be equally contributing to passenger’s ear response and are thus both of interest.

![Figure 1 – Tank slosh noise sources and transmission mechanisms](image)

The ultimate goal for tank designers and integrators is to be able to predict tank slosh noise inside the passenger vehicle early during design stage and thus avoid expensive redesign costs in case of noise issue (including expensive noise characterization and safety tests, parts re-tooling in case of structural redesign and other band-aid approaches used to reduce slosh noise once vehicle development freeze stage has been reached).

Practically speaking, it is often useful to consider the source and transmission paths independently. As illustrated in Figure 2 below, tank slosh noise source can be characterized individually through quantification of radiated sound power (air-borne path) and blocked forces at pads / mounts location (structure-borne path). The main frequency range of interest for structure-borne path is typically below 800Hz while airborne noise may cover a broad frequency range. For the present study, main interest was in airborne and structure borne noise source characterization below 1kHz frequency range.
2.2 Challenges with existing simulation methods

Pure CFD approach:
Computational Flow Dynamics (CFD) is widely used for the simulation of sloshing fluid in tanks (1). Using CFD method, one can simulate the motion of fluid and gas phases inside the tank when subject to an acceleration profile, typically assuming rigid boundaries at the surface of the tank. CFD analyst is then able to visualize free surface and monitor various engineering quantities such as instantaneous forces or pressures at the boundaries of the domain.

Unfortunately, relying solely on CFD analysis of fluid motion does not provide quantitative information about the actual airborne and structure borne noise source characteristics. For instance, while “hit” and “clonk” type events are the result of fluid (and entrapped gas) impacting the wall of the tank, the radiated noise and vibration from the tank typically exhibits very different spatial and spectral characteristics than the internal wall pressure resulting from fluid motion as illustrated in Figure 3 below. Consequently, it is practically not possible to evaluate the noise that will be radiated from a tank surface or the vibration at the mount locations from a pure CFD simulation.

Coupled fluid-structure approach:
In some instances, coupled fluid-structure numerical solutions are employed as a mean to predict time varying vibration of tank structure subject to slosh load as well as account for tank wall deformation in the CFD simulations. Such numerical simulations including Fluid Structure Interactions (FSI) involves transient CFD calculation of fluid motion tightly coupled to a structural Finite Element (typically non-linear) model of the tank structure. Such approach is mandatory for non-linear / large displacement type motion of structures where main output is a prediction of instantaneous displacement and stresses in the structure where the quasi-static response of the tank is critical. However, as discussed above, tank slosh noise can span a wide frequency range depending on the excitation type. As such, time domain Structural FE and especially FSI approaches are typically too computationally expensive and/or too limited in terms of frequency resolution / extension to be applicable for slosh noise simulation.

Time domain noise radiation approach:
In experimental evaluations of tank slosh noise, one typically pays attention to instantaneous external pressure response in the nearfield of the tank and possibly tank vibration. The interest is not only in the absolute noise levels but also relative amplitude of successive events during a slosh cycle (typically 3-5 seconds following sudden acceleration and braking or hard stop etc…). As a result, designers are often tempted to solve the tank slosh noise problem in the time domain directly using FSI approach described above and time domain radiation model from to the tank surface to the outside. Unfortunately, time domain acoustic simulation methods are often times very sensitive to mesh quality and may suffer from numerical instability issues making them inaccessible to most designers unless they happen to have extensive experience in numerical vibro-acoustic methods. Furthermore, time domain vibration model typically does not account for mass loading effect from partly filled tank which happens to have very significant influence on tank vibration response as discussed above.
2.3 Proposed modeling approach

The present modeling approach relies on Volume Of Fluid (VOF) method as part of CFD simulation to predict temporal and spatially varying pressures on the internal surface of the tank assuming rigid walls. All predictions discussed in this paper were performed using STAR-CCM+ software (3). While flexibility of the tank boundaries is ignored in these CFD simulations, it is assumed that the spatial and temporal discretization of CFD model are more critical to the accuracy of broadband noise predictions. In particular, frequency content of internal wall pressures at time of hits requires accurate reproduction of free surface motion which is typically not computationally practical when employing an FSI scheme.

Vibro-acoustic predictions were performed using wave6 software (4) and rely on a Finite Element Model of the tank structure (possibly including straps, mounts etc…) fully coupled to a Boundary Element Model of the internal (fuel fill) as well as external fluids (radiation into unbounded space, possibly including ground plane etc…) as illustrated in Figure 4 below. As discussed in previous section, it is important to account for mass loading effect of the fluid, hence the fully coupled vibro-acoustic approach employed here. Because the speed at which noise and vibration travels through the tank is several orders of magnitude faster than the speed at which the free surface of the fluid moves, the fluid surface can be assumed stationary for noise and vibration transmission calculations.

Vibro-acoustic predictions are performed in the frequency domain, in a block-wise manner. In particular, time varying wall surface pressures from STAR-CCM+ are split into overlapping time segments, effectively resulting in multiple load sets that are fed to the vibro-acoustic model. For past studies on real tank structures, it was found that 200ms time blocks with 50% overlap and hanning window weighting were typically adequate to distinguish between main hit events. These values were used for the present study.

The wall pressure loading is automatically converted to frequency domain and conservatively mapped onto the structural mesh as CFD mesh is typically mush finer. Results of wave6 predictions consist in time varying (blockwise) spectra of a) radiated sound pressures (SPL), b) overall radiated sound power (SWL), c) vibration or acceleration response on tank surface, d) blocked forces at pads and mount locations. Results are integrated over frequency to compute (blockwise, A-weighted) overall levels as function of time. As discussed in previous sections, predicted radiated sound power and blocked forces results can be readily used in combination with vehicle airborne and structure-borne noise transfer functions to estimate driver’s ears SPL.
3. Application to idealized tank model

The objective of the present study was to validate the modeling approach using an idealized tank structure as presented in the following sections.

3.1 CFD model review

The sloshing simulation was carried out using VOF method in STAR-CCM+ software (3). Specific model settings are illustrated in Figure 5 below. Time varying internal wall pressures were automatically stored at cells center on the boundary of the domain for the duration of slosh cycle (5 to 10 seconds duration). Solution time was ~1min per 1ms of data using 32cpus.

Figure 5 – Idealized tank CFD model content
3.2 Vibro-acoustic model review

The vibro-acoustic simulation was carried out using wave6 software (4) starting from a) STAR-CCM+ time varying wall pressure data (.trn format), b) tank structure geometry (CAD data exported from STAR-CCM+), c) water fill height information (14L and 40L configurations considered). The wave6 model content and solution times on a typical windows workstation are illustrated in Figure 6 below.

Structure and fluid meshes were automatically generated in wave6 with target upper frequency limit of 2kHz. Structural FE model is based on quad dominant shell mesh while internal and external BEM fluids are based on tria mesh.

Pinned boundary conditions were assumed at fixation points between tank structure and sloshing rig cart. 150 (in-vacuo) structural modes were computed below 2kHz.

As discussed in the previous section, the wave6 model is solved in the frequency domain. For this study, 5s duration slosh excitation from CFD was split into 50 overlapping time blocks (200ms), resulting in 50 sub-load cases used for the subsequent vibro-acoustic calculations.

The coupled FEM-BEM model was solved in 5Hz steps between 5Hz and 600Hz, representing a total of 120 frequencies. While calculations could be performed at higher frequencies, experimental validations were confined to frequencies below 600Hz to focus on hit and clonk noise rather than (presently unpredictable) splash noise. Total solution time for the wave6 calculation was under 1h.

![Image of wave6 model content and solution time](image)

4. Experimental validation

4.1 CFD model validation

Validation of the STAR-CCM+ VOF model of fluid slosh was carried out by comparing measured and predicted tank internal pressure time histories. As illustrated in Figure 5, a specific test sample was built to accommodate hydrophones flush mounted to the tank internal wall surface. Pressure time histories were measured and compared against predictions as illustrated in Figure 7 below. As can be observed, overall sloshing trends are well captured. Variability in test results is observed for some of the hit events which makes comparisons against predictions challenging. Due to limitations in the acquisition system, spectral content of internal wall pressure predictions could not be compared against test, this is subject of future work.

In addition to instantaneous internal wall pressure, free surface predictions were also compared against tests as illustrated in Figure 8 below for the 14L fill, “ACC” profile configuration. For these experiments, high-speed video recording of the slosh cycle was performed using tinted water for the tank fill. Predicted sloshing cycle is following the same trends as measured video recording. When comparing video animations, some discrepancies are observed at hits such as variations in water height across the hit surface. It is strongly suspected that the peculiar geometry of the case studied (square box with parallel / perpendicular walls to the slosh direction) causes a great deal of challenge to reproduce results due to the high stochastic nature of the problem (as small change in water movement causes fluid to move toward one of the other corners of the tank). This causes issue not
only with comparison of tests and predictions but also negatively affects tests repeatability as shown in Figure 7 and in later section of this paper.

4.2 Vibro-acoustic model validation

A validation of the wave6 vibro-acoustic model was carried out by performing impact hammer tests on the candidate tank structure. Acceleration and noise measurements were performed in the test cell used for tank slosh measurements (near anechoic acoustic boundary conditions).

Mechanical properties (Young’s modulus and structural damping loss factor) assumed for the structural model were first adjusted to match measured drive point mobility at various locations.
around the tank surface for the empty tank suspended on bungee cords. Tank vibration response on the tank surface and radiated sound pressure level were then validated against (impact hammer) tests under various mounting and water fill conditions as illustrated in Figure 9 and 10 below. As can be observed, water fill has a large influence on vibration response and radiated noise which is accurately reproduced by the fully coupled structural-acoustic model.

**Figure 9** – wave6 model validation results (impact tests) – acceleration response

**Excitation:**
1N at 5 locations on the tank surface

**Response:**
accelerometer on top surface

**Figure 10** – wave6 model validation results (impact tests) – SPL response 0.5m above the tank

**Excitation:**
1N at 5 locations on the tank surface

**Response:**
Microphone at the center of top surface

Figure 10 – wave6 model validation results (impact tests) – SPL response 0.5m above the tank
4.3 Tank slosh noise validation

Tank slosh noise validation was performed on the same acrylic tank for 2 water fill levels (14L and 40L) and 2 different acceleration profiles (acceleration and creep types). 3 microphones were placed 50cm away from the tank in the middle of front/rear/top surfaces. 4 accelerometers were placed on the top/front/read surfaces of the tank.

Pressure and acceleration time history recordings were post-processed such that overall levels vs. time could be compared against predictions (e.g. split into overlapping segments with hanning window, converted to frequency domain, A-weighted, band-integrated from 100Hz to 600Hz). Note that frequencies below 100Hz were discarded because noise in this region came from non-modeled test cart resonances (the first structural mode of the tank occurs at over 100Hz, even under heavy fluid loading as illustrated in Figure 9 and Figure 10).

Sample validation results are shown for the 14L and 40L fill cases in Figure 11 and Figure 12 below. In the upper portion of the figures, predicted overall SPL and acceleration levels vs. time are compared against tests. Additional measurements with empty tank were performed to identify non-modeled rolling cart noise and vibration as well as other measurement background noises. Although validations were performed against individual sensors, only space average results are shown here for sake of brevity.

The lower portions of the figures are aimed to illustrate what the first 4 main slosh events correspond to. Due to technical difficulties, it was not possible to synchronize video recordings and vibration/SPL measurements. As a result, predictions and measured time histories were manually realigned based on review of audio recordings (listening), video recordings and visualization of the mapped tank internal wall pressures in wave6 as illustrated in lower portions of Figure 11 and Figure 12.

It should be noted that, while video recordings correspond to instantaneous free surface position, the wave6 result represent the overall pressure over the 200ms time block for a given event. As such, these comparisons are only aimed to validate the event type, not the free surface position as was done in section 4.1.

Last, as shown in the figures, experiments were repeated 10 times to check for test repeatability. As discussed in previous section, ensuring good test repeatability proved to be difficult. The authors suspect this is the result of the idealized square geometry employed and acceleration profiles (both intensity and direction). In particular, authors suspect that large variations in measured levels occur depending on how the fluid “climbs” the front and rear walls of the tank. Any slight change in the motion profile / water movement causes fluid to move preferably to either corners of the tank yielding a strong hit/clonk type event or not. Further, the acceleration profiles fed to the tank resulted in very chaotic fluid motion with severe air entainment, which is a challenge to reproduce in CFD simulation unless very fine mesh and time steps are used (typically not practical due to limited computational resources).

As a result of test repeatability issues discussed above, minimum and maximum levels measured across 10 experiments are compared against predictions in Figure 11 and Figure 12 below. Events with good experimental repeatability are predicted with reasonable accuracy. On the other hand, events with large experimental variability (later part of the slosh cycle) are hardly predictable. It can observed that 40L fill results yield louder radiated noise and higher vibration levels than 14L fill results in both tests and predictions. As such, the proposed simulation approach shows good potential for comparative studies between various tank designs and / or fill levels.
Figure 11 – slosh noise validation results – 14L fill / “ACC” type acceleration profile

Figure 12 – slosh noise validation results – 40L fill / “CREEP” type acceleration profile
5. Conclusions

This study demonstrated analysis process for characterizing sloshing of fluid within fuel tank as a “noise and vibration” source. The analysis process combines CFD (STAR-CCM+) model of sloshing fluid in a rigid tank with aero-vibro-acoustic (wave6) model of noise and vibration response of an elastic tank. The modeling approach was applied to an idealized tank (square shape acrylic box) to predict radiated sound pressure level and tank wall acceleration (as a function of time and frequency).

The vibro-acoustic model of idealized tank was validated against tests using impact hammer data (P/F and A/F). Changes in response when the tank was mounted onto the test rig and partially filled with water were also validated against test.

As a second step, free surface and instantaneous internal wall pressures were validated for two different acceleration profiles and two different water fill levels. It was shown that overall trends were well captured by CFD simulation but comparison of pressure at hits was difficult due to sometimes large variability in measured results.

Finally, slosh noise predictions were compared against tests for the same sloshing conditions. Again, noise and vibration predictions at hits generally agree favorably with tests but comparisons were difficult to handle due to large variability observed in tests. The symmetric shape of the test item (sharp corners, walls parallel and perpendicular to flow direction) combined with highly energetic motion profiled are suspected to cause significant variability in tests and subsequent challenges with the correlation of measured noise and vibration levels against predictions.

Nevertheless, for slosh noise events that showed good experimental repeatability, radiated noise and tank vibration predictions were found to match measurements with good accuracy. In addition to this, the same trends between experiments and tests were observed when comparing fill levels. In particular, higher fill levels caused larger radiated noise and vibration response. As such, the proposed approach is promising for use with early stage tank design. Future work will consist in applying the same modeling and validation methodology for real tank system.

ACKNOWLEDGEMENTS

The authors wish to thank Phil Shorter and the wave six R&D for the invaluable support with methodology development. The authors also thank Samir Muzaferija of CD adapco for the support with VOF simulations. Finally, the authors wish to thank Thomas Kye of Donghee Industrial Co., Ltd for supervising this joint study, providing test facilities, equipment and staff support for the validation effort.

REFERENCES

4. wave6, version 1.3, wave six L.L.C., 2016