Acoustic-Structure Simulation of Passive Fluid Pulsation Dampers

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Introduction

Valve actuation and pump fluctuation in piping systems generate propagating sound waves in the fluid path [1], which in turn can lead to undesired excitation of structural components [2]. Fluid pulsation dampers are introduced to reduce the vibro-acoustic excitation of fluid- and structure-borne sound transmission, thus achieving noise reduction.

This can be done using two different principles. On the one hand, material damping leads to sound level attenuation due to dissipation of sound energy. On the other hand, the principle of destructive interference has the capacity of reducing sound transmission. The latter one can either be obtained by active or passive reflection dampers, where the active ones use sensors and actuators in combination with a control law to diminish the incident sound field.

This research focuses on passive reflection elements while neglecting additional noise reduction due to material damping. One class of them are in-line pulsation dampers, which attenuate incident waves due to impedance mismatches. Thus, sound power is transmitted back into the inlet pipe. The impedance mismatch is achieved by modifications of the pipe diameter leading to resonance chambers [1], or by reducing the sound speed locally. In the latter case, flexible hoses are built-in, which increase the effective longitudinal compressibility of the enclosed fluid, leading to reduced sound speeds [3]. On the other hand, side-branch resonators use a closed side branch to generate reflections which cause destructive interference.

If pipes convey heavy liquids such as water or oil, the acoustic field strongly interacts with the structural dynamics. Therefore, it is necessary to take a full surface coupling at the acoustic fluid-structure interface (FSI) [4] into account. In addition, local sound transmission phenomena often include acoustic fields with a non-uniform pressure distribution over the pipe cross section, so that modeling in 3D becomes necessary.

Two methods are applied to model both inline and side-branch resonators. The finite element method (FEM) using ANSYS in 3D is compared to the transfer matrix method (TMM) [2,5] in 1D. It is worth noting that the TMM does not include a detailed model of the pipe structure or attached structural components of the piping system. Only the Korteweg wave speed in the fluid can be considered to model longitudinal hydroacoustic phenomena.

Inline Resonator

The inline resonator used in this study shows a quite simple geometry, where the resonator volume is a widened pipe compared to inlet and outlet pipes. As a result, a resonance chamber similar to an exhaust muffler exists. Figure 1 shows a longitudinal cut of the inline resonator. The acoustic fluid-

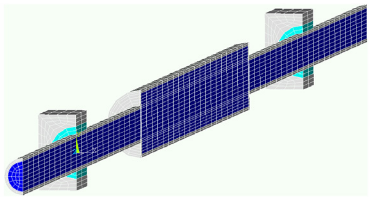

Figure 1: FE Model of a single-chamber inline resonator, where the blue elements symbolize the fluid, while the grey and cyan elements build the pipe structure and the rubber layer in the pipe mounts, respectively.

structure interface is located on the coupling interface, but it is not depicted in the figure.

In order to measure the effectiveness of the inline resonator, an acoustic source is modeled at the beginning of the inlet pipe. An absorbing boundary condition is applied at the end of the outlet pipe to avoid reflections back into the piping system. As a result, it is possible to truncate the pipe model at this point, which is necessary if extended pipelines are investigated. In the first step, an harmonic analysis of the piping system including the pulsation damper is performed. In a second simulation, the resonator volume is replaced by a pipe with the same radius as the inlet and an outlet pipes. From both results, the insertion loss, i.e. the difference of the sound pressure levels in the fluid at the outlet, is evaluated and shown over frequency. Frequency bands with positive insertion loss mark sound attenuation, whereas negative values indicate deterioration in terms of sound transmission.

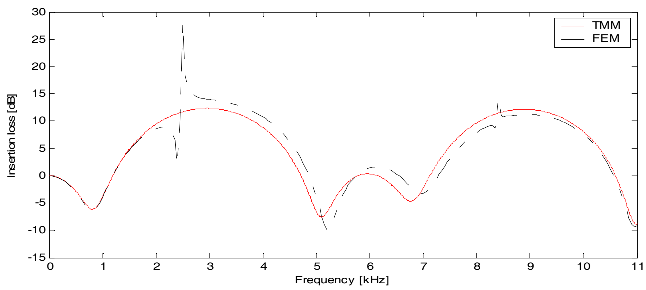

Figure 2: Insertion loss for the inline resonator in the fluid path by TMM and by FEM.

Figure 2 depicts the insertion losses using the inline resonator for both TMM and FEM. Both curves show a similar behavior except for two deviations at 2.4 kHz and 8.5 kHz. A succeeding modal analysis clarifies the two deviations. Figure 3 shows the coupled mode shapes at the critical frequencies. The first mode is a structurally-dominated mode, where the whole pipe body vibrates in the pipe direction within the pipe mounts. The mounts are fixed on the ground due to a displacement boundary condition. This mode shape is supported by the low stiffness of the annular rubber inlay, which is a part of the joints. The associated acoustic field at this eigenfrequency shows a continuous pressure distribution. The same statement is true for the second mode, although it is a higher harmonic longitudinal mode. Its structural deflection results from

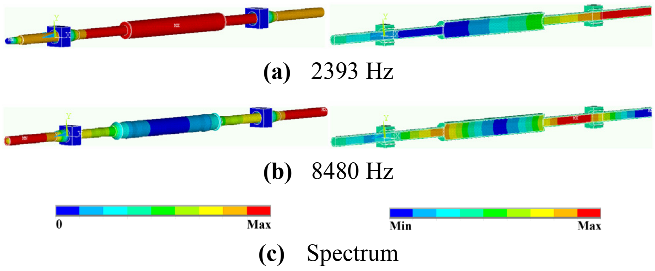

Figure 3: Coupled modes of a piping segment with inline resonator represented by the displacement vector sum (left) and the acoustic pressure distribution (right).

longitudinal extension/compression, where the inlet and outlet pipe move in opposite directions.

Helmholtz Resonator

The Helmholtz resonator is an example for a side-branch resonator and provides a large fluid volume, which is connected to the pipe by a narrow neck. This leads to a mass-spring like oscillating behavior in the fluid, which can be tuned in order to absorb fluid sound at a certain frequency. Figure 5 shows the insertion loss in the fluid (dashed lines), where a virtual sensor hydrophone is placed at the outlet to compare sound pressure levels. Strong sound attenuation is achieved around 2.3 kHz at the design point of the resonator. The FEM and TMM results again agree except for some sharp deviations. Coupled acoustic-structure modes are identified according to the abnormalities along the insertion loss curve. It becomes obvious that negative insertion losses are present around eigenfrequencies, since incident waves may excite larger vibration and noise levels.

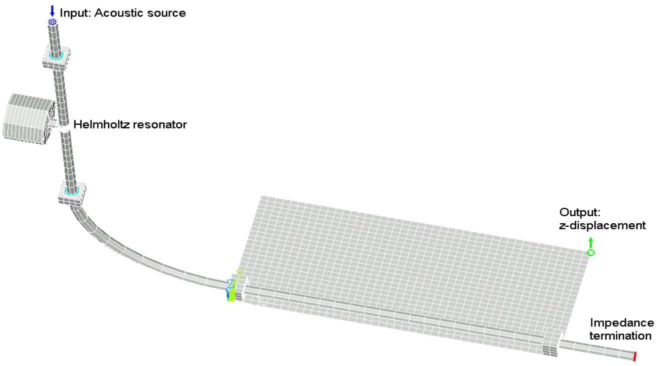

Figure 4: Piping assembly with straight and elbow pipes, joints, a target structure (plate) and the Helmholtz resonator.

So far the noise reduction in the fluid has been investigated. But the FE method with the FSI is also capable to quantify the influence of the passive reflection dampers on attached structures. In order to examine this, a larger part of a piping system including a target structure is assembled (see Figure 4). In this case, the two upper mounts are fixed in space, while the other two mounts are ideally connected to the plate. Again an acoustic source is placed at the inlet and an absorbing boundary condition is present at the outlet. By considering the out-of-plane displacement on the target plate, instead of the sound pressure level at the outlet, the insertion loss accounting for structural waves is obtained. The resulting structural insertion loss is plotted in Figure 6 (green line). The insertion loss of the plate tends to follow

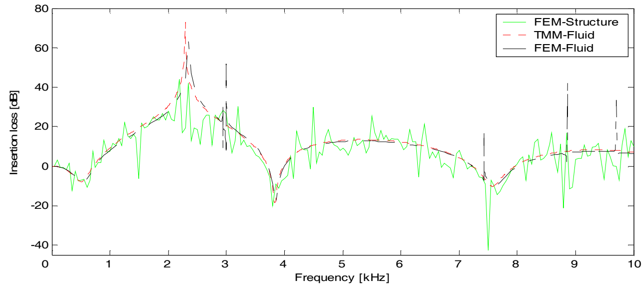

Figure 5: Insertion losses: Fluid path by FEM and by TMM, on the target plate by FEM.

the insertion loss in the fluid, where the trembling of the curve results from the high modal density of the target plate. Therefore it becomes clear that the inserted fluid damper can affect the structural dynamics of attached structures in a positive manner. The structural responses are a main reason for undesired airborne sound radiation and structural failure.

Conclusions

In order to achieve noise reduction in the fluid path of piping systems, this study investigates passive pulsation dampers by using numerical methods. The 1D transfer matrix method as well as the 3D finite element method (ANSYS) allow predictions of the insertion losses evaluated over frequency. Both methods account for influences due to the acoustic source, the pressure boundary condition and geometrical properties of the passive elements, inlet and outlet pipes. FE modeling also includes the influence of structural resonances and 3D effects, which are neglected by the TMM models. For these reasons, the FEM renders a more complete model of the piping components compared to the TMM. This is a clear advantage for tuning purposes of real components, and justifies the increased complexity and computing costs of the FEM. Furthermore, the effects of sound transmission on attached target structures can be integrated and analyzed by using FE models.

References

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