

Model order reduction for waterline acoustics

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Extended abstract for presentation

Introduction

Precision systems have water cooling circuits to achieve thermal conditioning. Pressure pulses propagate through these water circuits, leading to undesired forces on the isolated world of the precision system. The forces lead to a degradation of performance and accelerations of the precision system.

MI-Partners has developed a test setup [1] to measure the transfer from a pressure in the fluid circuit to the acceleration of the precision system, represented by a plate. The setup is shown in the left of Figure 1 (left), and a schematic of the setup is shown in the right of Figure 1. The goal for this test setup is to validate models that predict the propagation of pressure disturbances in the fluid circuit. In the initial models of the setup, the dynamic links were modelled as straight flexible pipes. This is clearly an oversimplification, and more detailed models are needed for accurate description. Since these fluid circuits are long, typically having lengths of more than 10m, sub-structuring and model reduction are desired.

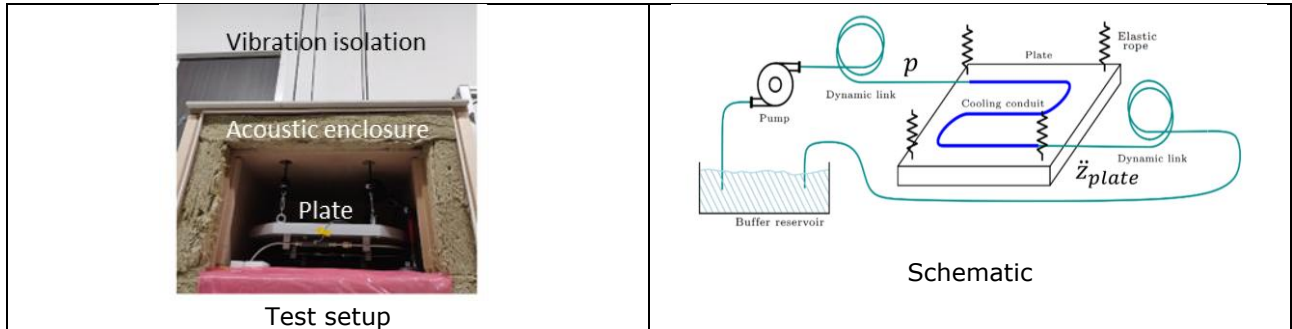


Figure 1

Test setup for characterising the transfer of pressure disturbances towards an isolated precision system (left). Schematic of the setup (right). In the setup, the pressure in the pipes p is monitored in combination with the plate's acceleration \ddot{z}_{plate} .

Force-induced vibration model

The coupling between acoustic pressure and resulting accelerations can be modelled using the finite element method, employing a combination of acoustic fluid and mechanical structural elements. The resulting equations of motion can be written as

$$\begin{bmatrix} M_s & 0 \\ \rho_0 R^T & M_f \end{bmatrix} \begin{bmatrix} \ddot{U} \\ \ddot{P} \end{bmatrix} + \begin{bmatrix} K_s & -R \\ 0 & K_f \end{bmatrix} \begin{bmatrix} U \\ P \end{bmatrix} = \begin{bmatrix} F_s \\ F_f \end{bmatrix},$$

U , P are the mechanical displacement and fluid pressure, K_s , K_f , M_s and M_f the structural and acoustic fluid stiffness and mass matrix. Furthermore, F_s and F_f represents the structural and fluid forces, ρ_0 is the fluid density and R is a bi-directional coupling matrix between both domains. Sometimes, calculations are simplified by only considering a one-way coupling from the acoustic to the structural domain. However, when flexible components are present, bi-directional coupling cannot be neglected. An example of a flexible component is a dynamic link, where the tube's flexibility plays an important role in the overall dynamics. This is illustrated by the pressure FRF for the dynamic link shown in Figure 2, where the outlet displacement is kept fixed. By taking the tube's flexibility into account, the acoustic resonance in the pipe reduces from 220 Hz to 45 Hz.

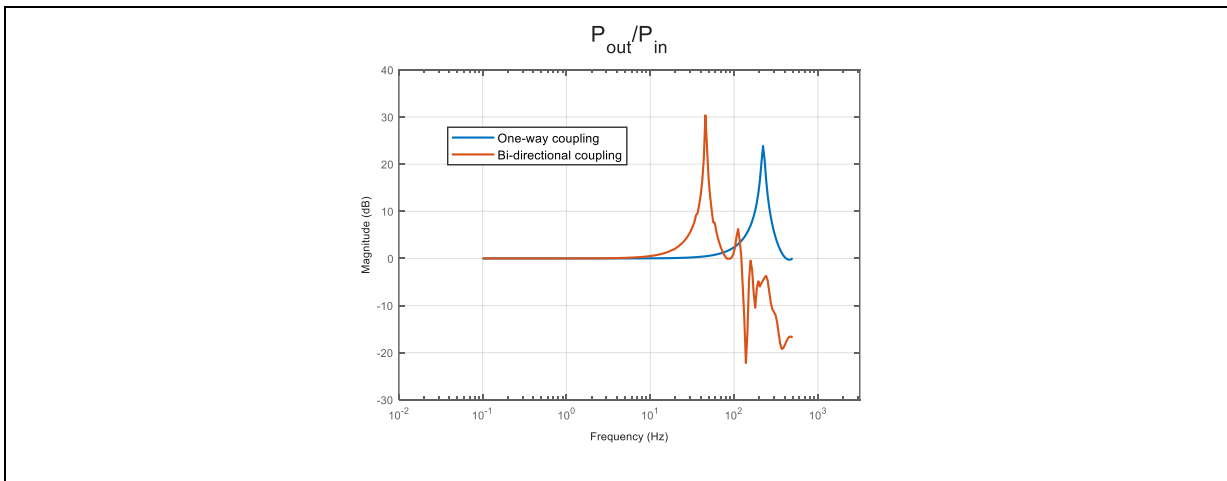


Figure 2

Transfer function from pressure at inlet to pressure at the outlet for the flexible tube considering both one-way (acoustic \rightarrow structural) and bi-directional (acoustic \leftrightarrow structural) coupling. The displacement degrees of freedom at the end of the tube are constrained.

Substructuring and model reduction

To efficiently model coupled structural-acoustic system, the model of the setup is split up in three main components as indicated in Figure 3: an inlet dynamic link (tube 1), an isolated precision machine part (measurement plate), and an outlet dynamic link (tube 2). For all three sub-models, bi-directional coupling is considered. The equations of motion are transformed into a symmetric formulation as given in [2] and [3].

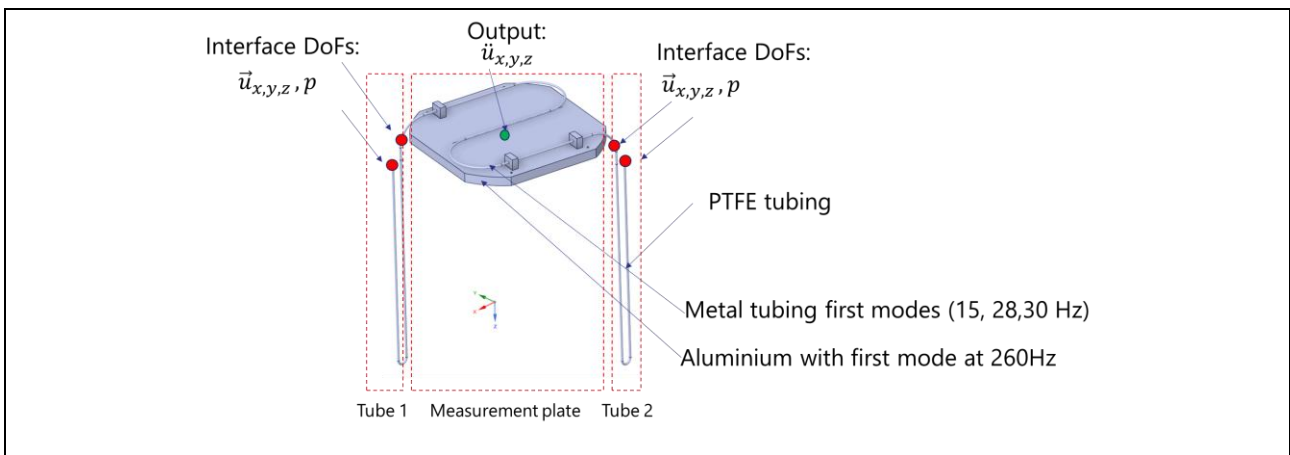


Figure 3

Model used in establishing the transfer from input pressures towards measurement plate accelerations. The system consists of two dynamic links (tube 1&2) and isolated machine component (measurement plate). The inlet displacements of tube 1 are constrained. The output of the tube 2 constrained for displacement and pressure. For modelling details see [6].

The FRFs of the assembled system from pressure at the inlet of tube 1 towards acceleration on the plate is shown in Figure 4 for three cases:

1. Full model without reduction, consisting of 294936 degrees of freedom;
2. Reduced model, resulting from the Craig-Bampton method [2,3], containing 449 degrees of freedom;
3. Reduced model, resulting from the second order MIMO reduction according to [5], in this paper referred to as block SOAR, containing 452 degrees of freedom.

For all models Rayleigh damping was assumed, resulting in 1-2% of modal damping in the frequency range of interest. As shown in Figure 4, both model reduction methods predict the combined structure effectively. The Craig-Bampton method has the advantage that physical degrees of freedom are kept, whereas the SOAR reduction seems slightly more accurate.

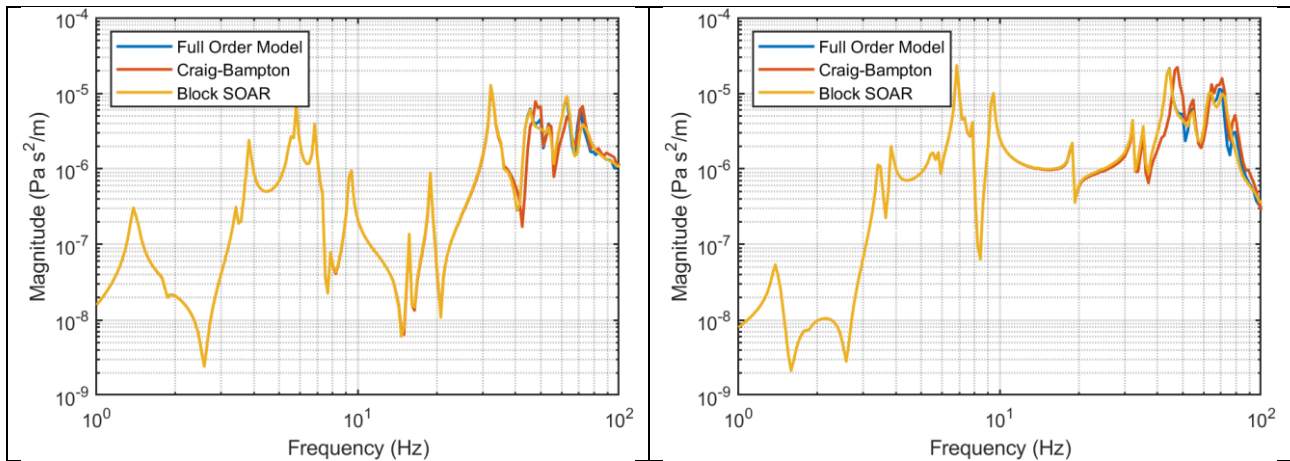


Figure 4

FRFs from pressure at the input of the tube to acceleration on the plate in the x-direction (left) and z-direction (right), based on the full-order model, the Craig-Bampton (CB) reduced model and the block-SOAR reduced model.

Outlook

This paper has shown that reduced order models show good correspondence to a full order model for acoustic-structural systems. In this paper, the effect of viscoelastic damping in dynamic links was neglected. Visco-elastic dynamic links and viscoelastic hoses add a significant amount of damping. The block SOAR method can be extended to incorporate the damping while resulting in efficient models. A next step will be the comparison to the measured transfer function.

References

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