An Interpretation of the physics of Fluid-Structure Interaction in the frequency domain

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Summary

This paper constitutes an attempt to interpret the physics of FSI where the structure is vibrating. The structure considered is a very simple physical device, namely a flat cantilever plate. Many presumably important details are only briefly mentioned, other are left completely out of the discussion. The intention with the paper is to emphasize that there are cases where Fluid - Structure Interaction can be successfully simulated without having to utilize a Navier-Stokes Solver coupled with a (non-linear) Structural Analysis Program, a concept that requires vast amounts of computer time for linear vibrations (but is highly relevant for other applications).

The paper starts out with a brief explanation of the concept of eigenvalues and eigen-vectors (or -modes), and then the eigenforms and eigenfrequencies of the plate are described. Thereafter, the behavior of the plate in a constant flow-field is discussed with emphasis on the interaction between the plate represented by its modes and the flow-field. What is to be understood by positive and negative damping is explained, and it is discussed how disturbances in the flow field (e.g. acoustic waves) may trigger vibrations of the plate, and how the vibrations of the plate may trigger acoustic waves.

Keywords

Acoustics, Aeroelastic Stability, Eigenproblems, Fluid-Induced Vibrations,
Fluid-Structure Interaction

0. INTRODUCTION

In our daily life, we often sense vibrations that are generated through Fluid-Structure Interaction ("FSI"), although we do not always recognize them as such. Examples hereto are noise from fans, the flapping of a flag, the vibrations of a pipe.
From an engineering point of view, FSI generated vibrations are extremely important to master as they can lead to complete failure through "brute force rupture" (e.g. Tacoma Narrows Bridge) or through fatigue (e.g. feeder pipes in nuclear reactors, heat exchangers, and risers in offshore platforms).
But FSI is a difficult topic to master because it is a two-way coupled system: The fluid has an influence on the structure, and vice versa. The three disciplines involved (Computational Fluid Dynamics, Structural Mechanics, and – to some extent - Control System Theory) are considered difficult by most engineers. Some engineers are good at fluid dynamics, some at structures, and some understand control systems. Some engineers are proficient in two of these disciplines; very few master all three. And virtually nobody seems to have a firm understanding of the physics involved with FSI and vibrating structures.

In this paper, we will discuss physical phenomena where there is a strong coupling or (interaction) between fluid flow and structural response and henceforth use call it "Fluid-Structure Interaction" (or simply "FSI"). The FSI may be of static nature (e.g. the equilibrium state of a deformable body in a flow field), or dynamic (e.g. due to time varying displacements of a flexible structure interacting with the flow field).

FSI dynamics can be studied in the time-domain or in the frequency domain. Typical for the simulation of FSI in the time domain are (extremely) time consuming computer runs. In this paper, we deal with simulations in the frequency domain, or - more specific - harmonic analysis of FSI.

1. WHY DO FSI IN THE FREQUENCY DOMAIN?

There are several reasons for doing fluid-structure interaction in the frequency domain. One reason is that it is much faster than doing it in the time domain. Another reason is that it is much easier to identify the design measures needed to avoid problems (e.g. noise, fatigue). One could compare the concept with the concept of beam models and dynamics by modal superposition in a structural analysis system: The models are usually very simple, but provide a wealth of useful information.

2. VIBRATIONS OF A FLAT CANTILEVER PLATE IN VACUUM

Figs. 1 - 5 show the eigenforms of the first (lowest) 5 modes for a cantilever plate. For easier interpretation, a grid showing the undeformed structure has been added. The eigenforms correspond to free vibrations in vacuum. They will be utilized to explain the physics of fluid structure interaction interpreted in the frequency domain.
Before doing so, we will briefly recall the concept of eigenvalues in the context of structural linear dynamics. The idea is to reduce the representation of a linear structure to a set of generalized springs and masses (and dampers if applicable) in generalized (or modal) coordinates such that they are independent of each other (i.e. not coupled or "orthogonal"). This set of (physical) modes of vibration can be combined to represent any linear behavior of the structure.

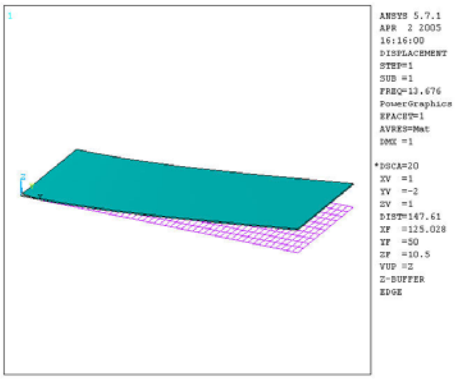

Fig. 1: Mode 1

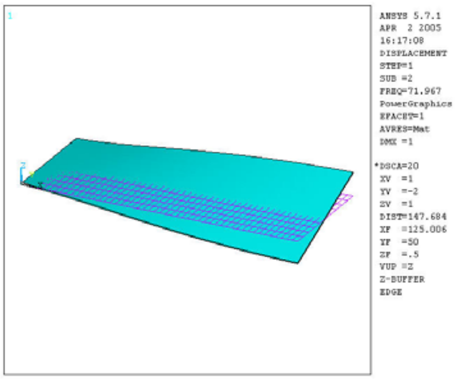

Fig. 2: Mode 2

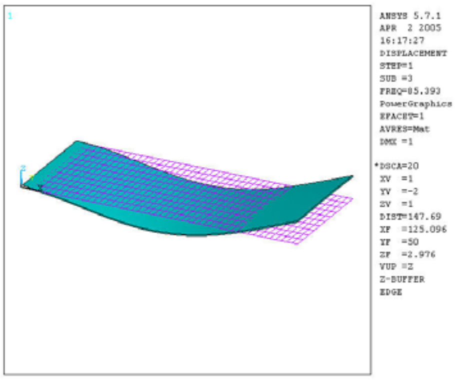

Fig. 3: Mode 3

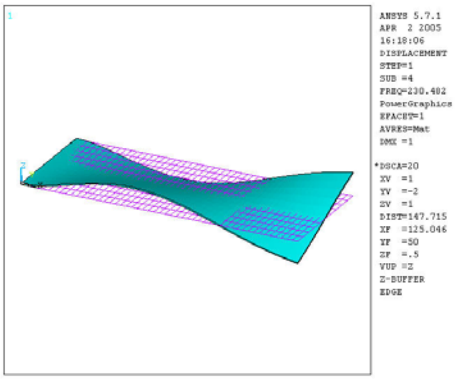

Fig. 4: Mode 4

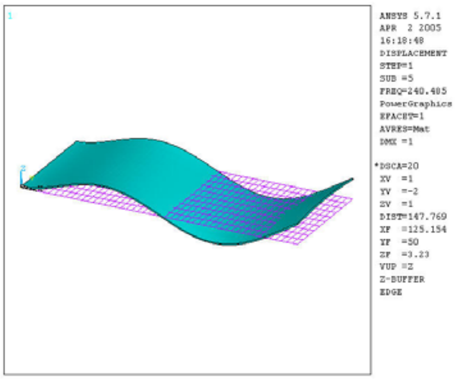

Fig. 5: Mode 5

3. VIBRATIONS OF A FLAT CANTILEVER PLATE IN A MEDIUM AT REST

When a flat plate vibrates in a medium at rest, it will move the medium and several effects come into play. Typically, the eigenfrequencies will be lower than in vacuum and the medium's viscosity will introduce damping into the system. Later in this paper, we will see that “damping” may occur even for inviscid media. When studying vibrations of a real structure in a real medium the so-called "hydrodynamic added mass" - that represents a mass to be added to the structural mass and corresponding to the reduction of the frequencies - should always be considered. It shall also be mentioned that the concept of hydrodynamic added mass is based on the assumption that the medium is inviscid. An example showing the effect can be found in [1]. The concept of hydrodynamic added mass can be considered a simple example of fluid-structure interaction. The physics gets much more complicated if the medium is moving, this is the main subject of this paper and will be discussed next.

4. VIBRATIONS OF A FLAT CANTILEVER PLATE IN A MOVING MEDIUM

For the following, we will assume that the (vibrating) plate is in a moving medium, with the flow in the direction of the positive y-axis (the plate is in the x-y-plane with the short side (at y-axis) clamped. We will now discuss the modes in turn, with emphasis on modes 1 and 2 (first bending mode and first twisting mode); we will not discuss how the modes are exited but simply assume they somehow are.

Let us start with mode 1, and for the beginning assume that the fluid is at rest. We will further assume that there is no force from the fluid interacting with the plate, except those forces (pressures) needed to move the fluid normal to the plane of the plate. This implies that there are no forces stemming from the fluid acting in the x-y plane of the plate. This is symbolically illustrated in Fig. 6; in reality there will be a positive pressure at the side in the direction of the movement and a negative pressure at the opposite side. At this point, we will leave the discussion of mode 1 and discuss mode 2; we will revert to mode 1 after having discussed mode 2.

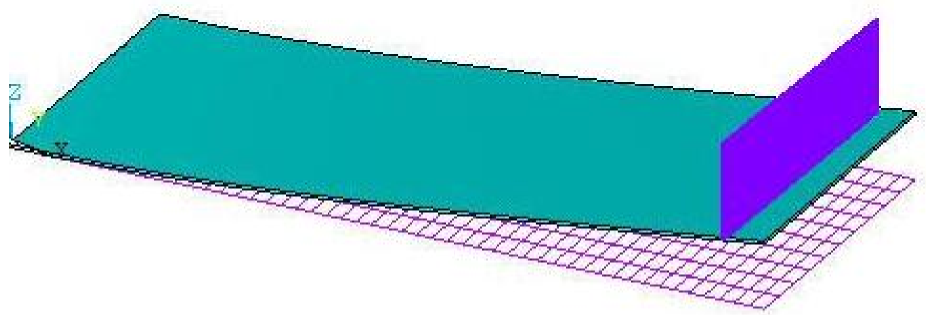

Fig. 6: Pressure over vibrating plate (Symbolic); medium at rest

Mode 2 is a twisting mode (ref. Fig. 2). Again, we will start the discussion assuming that the fluid is not moving, and again that the only interaction between fluid and structure is that of moving the fluid normal to the plane of the plate. Again, there will be a pressure distribution of (direction and magnitude) that corresponds to the movement of the fluid .

Now let us assume that the fluid is moving and (still) discuss mode 2. Mode(-shape) 2 may, on a section by section basis, be considered a plate in a flow field at an (assumed small) angle of attack. The lifting force L can approximately be computed as ([2]):

L = (Cl * A * rho * V^2)/2

with Cl = lift Coefficient
A = (total) Area
rho = mass density
V = Velocity
Cl = Clo / (1 + Clo/(pi * Ar))
with Clo = 2 * pi * alpha
and alph = angle of attack [radians]
Ar = Aspect ratio = span^2 / A

Assuming air at a velocity of 10 [m/sec] over the plate our case and an angle of attack for all of the plate of 5 [degr.], the lifting force would be

L = 20.5 [N] (corresponding to a pressure of 683 [N/m^2] and a displacement of approx. 18 [mm] of mode 1)

This pressure could also be viewed as sound pressure level ("SPL") given in Decibels.

L_sub_p = 20*log(P/P_sub_0)

With

P_sub_0 = 2*10^-5 [N/m^2]

we find

L_sub_p = 20log(683*10^5/2) [dB] = 151.85 [dB]

The level would be lower if the gradually increasing twisting of the plate would be accounted for, but still we see that this would be quite an impressive noise level!

The essential thing to note here, is that a slight twisting of the plate may result in displacements of the plate that are a substantially greater than the plate thickness. Given that pressure acts as a uniform pressure over the plate, a static analysis will yield a displaced shape very similar to the first eigenmode. We also realize that, once triggered, an eigenmode may show much larger displacements than can be expected from sound pressure levels.

Now, let us revert to mode 1. Assuming that the fluid is moving, mode 1 will introduce a (small) pressure increase/disturbance at the side facing the direction of the plate's movement. If the medium is at rest, the disturbance will be symmetric (we do not consider mode 2 or higher modes at the moment). However, when the medium is moving, the disturbance will move with the medium, and hence there will be a skew load on the plate, that may (or may not) trigger mode 2; Fig. 7. To what degree the moving disturbance will trigger mode two or not, will obviously depend on the frequencies of mode 1 and mode 2, the properties of the medium, the speed of the medium, and on the other frequencies and eigenforms of the plate.

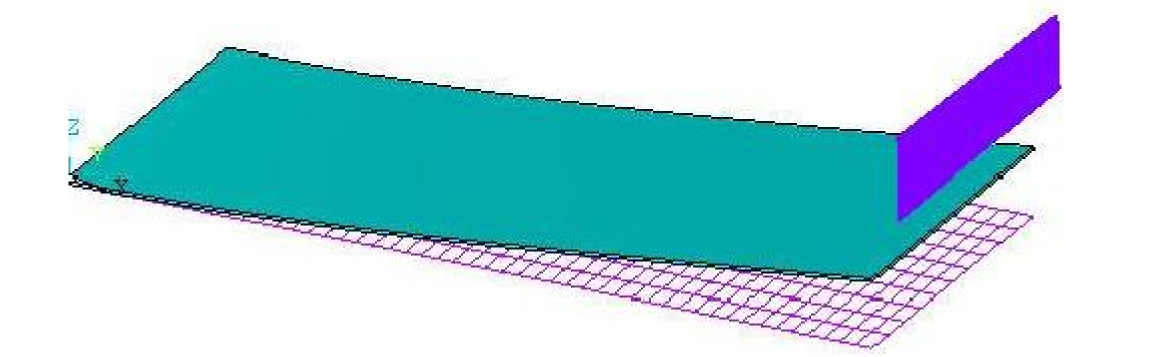

Fig. 7: Skew load on mode 1 due to pressure moving with the medium (Symbolic)

Now, let us try to imagine how the traveling pressure pulse will influence mode 2; Fig.8.

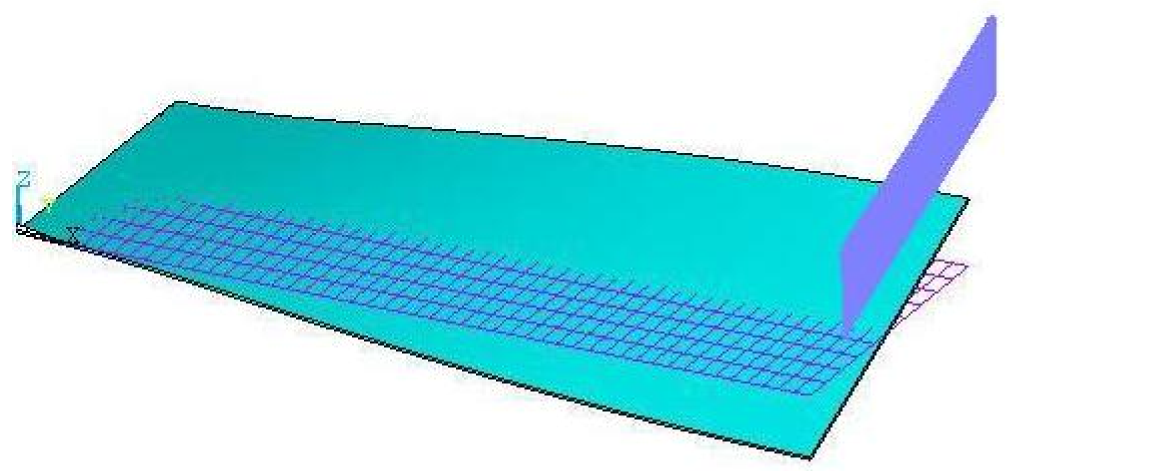

Fig. 8: Skew load on mode 2 due to pressure moving with the medium (Symbolic)

Here, one has to distinguish two cases

- Mode 2 is moving against the pressure
- Mode 2 is moving in the direction of the pressure

In the former case, the force exercised by the pressure can be considered a damping force for mode 2. Further, the pressure will decrease the angle of attack, and so decrease the (negative) lifting force.
In the latter case, the force exercised by the pressure can be considered a negative damping force that introduces more kinetic energy into the second mode of plate. Here, the angle of attack will increase, and larger amplitudes can be expected.

Summarizing modes 1 and 2, we can say that mode 1 will influence (trigger) mode 2, but it is not given that mode 2 may influence (trigger) mode 1. Also, it is apparent that the influence of mode 1 on mode 2 will in any case not be the same as the influence of mode 2 on mode 1.

So far, nothing has been said about the frequencies of the modes. For completeness, the eigenfrequencies of the 5 lowest modes in vacuum are given in Table 1. It is worth noting, that the eigenfrequencies and modes for the combined problem are complex and are established as combinations of the real eigenfrequencies of the eigenvalue problem for vacuum. These eigenfrequencies are of little significance if the fluid is moving, as a moving medium may shift the frequencies substantially, /1/.
However, the associated eigenforms are of interest, as they may help in determining what design measures should be taken if there are design changes to be made.

Mode	Frequency
1	13.7
2	72.0
3	85.4
4	147.7
5	240.5

Table 1: Eigenfrequencies

Looking at Table 1, one can see that the eigenfrequency of mode 4 is very close to twice the eigenfrequency of mode 2. Based on experiences from structural analysis, one could be tempted to believe that modes 2 and 4 can combine into a mode that will increase in amplitude. However, keep in mind that the fluid will influence the eigenfrequencies and that the influence is not necessarily linear. Further, please keep in mind that that the speed of the fluid will determine how the disturbances in and from the flow field will “hit the various modes”.

Now, we want to have a brief look at some combined modes. The combined modes we show are first displacement normalized (i.e. max displacement is unity), and then added to give the combined displacement shape.
The purpose is to show that there are combined modes where the plate cord will not be at an angle to the flow direction, and others that will. In other words: There are combined modes that are stronger candidates than other to interact with the fluid.
Figures 9 to 12 show the combined modes, and one can speculate what combined modes are candidates for interacting with the fluid.
No individual scaling of the modes has been considered, so one should not draw any definite conclusions based on the displacement shapes shown.

In the considerations above, it has been (implicitly) assumed that the upstream flow conditions are not altered by the movements of the plate. To what degree such effects are present and whether they significantly influence the behavior of the plate or not shall not be discussed here.

However, it is obvious (ref. Fig. 8) that *the speed of the fluid* will have an influence on the response of the structure since it controls when and where a pressure pulse will hit the structure, and hence what modes are triggered. That the speed of the fluid might have a significant influence on the vibrations of a structure can be seen in pipe flow, where one can observe that there is no “resonance” with low speed, then, as the speed increases, significant vibrations appear until the flow speed is further increased.

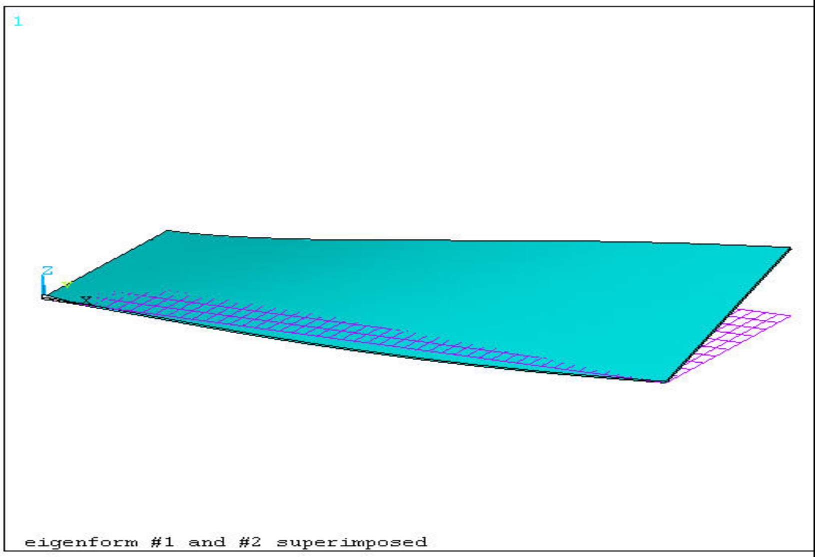

Fig. 9: Modes 1 and 2 combined (may interact with the flow ?)

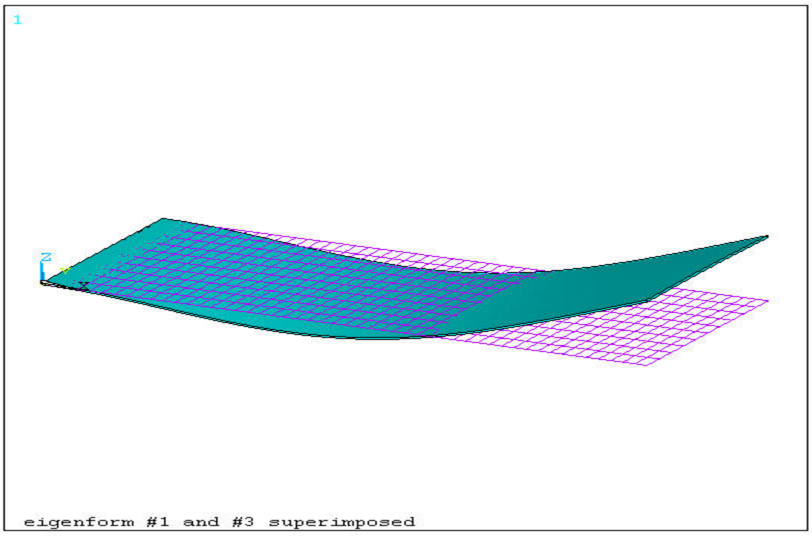

Fig. 10: Modes 1 and 3 combined (no interaction w. the flow?)

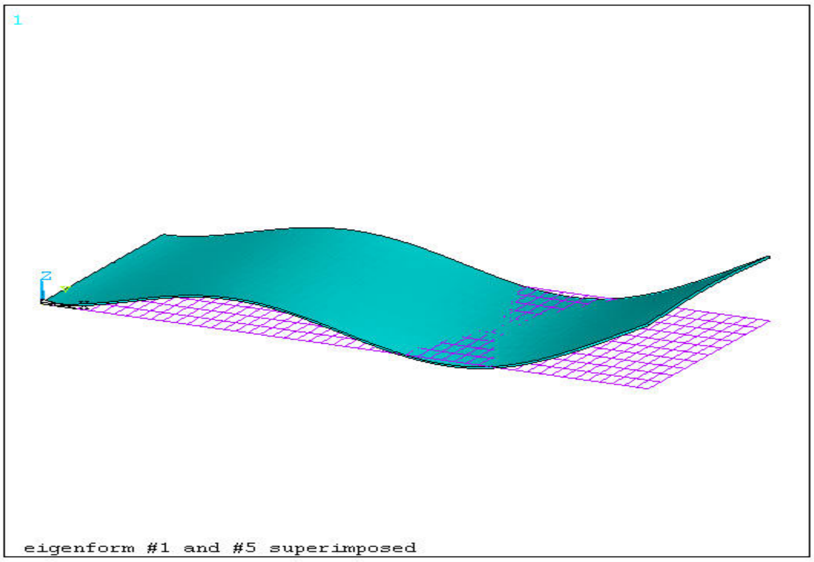

Fig. 11: Modes 1 and 5 combined (no interaction w. the flow?)

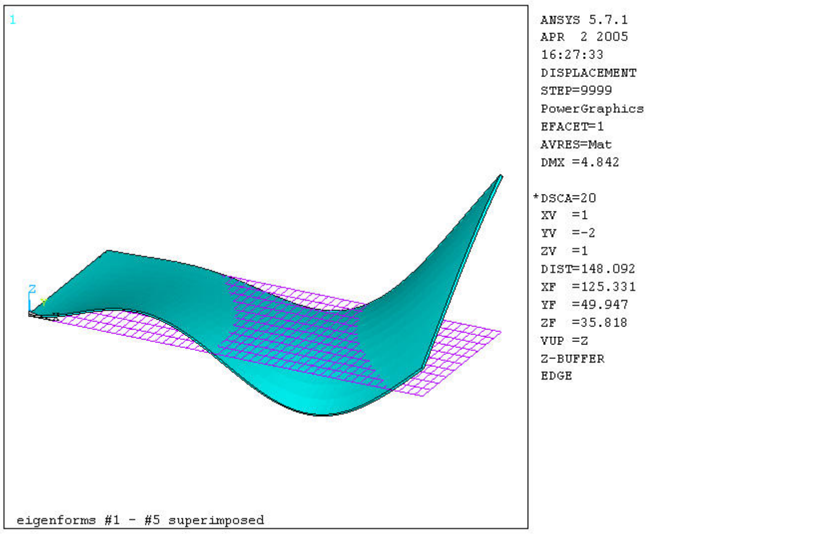

Fig. 12: Modes 1 to 5 combined (interaction w. the flow !?)

4. ACOUSTICS

It was shown above that fairly high acoustic pressures are needed to generate displacements comparable to those generated by lift. This becomes substantially more pronounced considering other structures, e.g. with pipe vibrations. It is believed, that in the majority of vibration problems where FSI plays a role, the observed acoustic pressures are generated by the displacements of the vibrating structure, not vice versa.

However, the modes must be triggered so that they can interact with the fluid and extract energy to become more pronounced until the damping of the system prevents further amplitude growth. There are two ways of triggering the modes. One is by structural vibrations (e.g. through vibrations of the support structure), the other is by disturbances in the pressure field. The disturbances in the pressure field may be due to some vibrating source external to the pipe, or they are due to vortex shedding (in which case the disturbances might be calculated in a Navier-Stokes Solver like ANSYS/CFX).

In any case the vibrating structure will interact with the fluid, and – if the amplitudes are big enough – generate significant acoustic pressures, pressures that might be significantly higher than the acoustic pressures triggering the FSI.

CONCLUDING REMARKS AND CONCLUSIONS

Understanding Fluid-Structure Interaction vibration problems is difficult. Some of the concepts emphasizing the structural side have been presented. Neither how to perform the adequate simulations, nor the functions and features required for the simulation software doing FSI in the frequency domain have been discussed. The mechanism of the coupling of structural modes through the fluid has only been hinted at.

It can be concluded that

- The speed of the fluid might influence what modes will be dominating the FSI problem
- Acoustic pressures might trigger the FSI mechanism
- Very high acoustic pressures are needed to introduce significant structural displacements
- An interpretation of the physics of FSI in the frequency domain has been given

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***Note**:*
This paper will be subject to revisions. The outcome of Numerical Experiments Utilizing LINFLOW™ will be included.