

A numerical study on air squeeze-film damping based on structure-fluid co-simulation technique

Mingzhang Ren, Miaki Tanoue, Keiichiro Tsutsui

ESTECH Corporation

Abstract: CAE applications in dealing with multiphysics problems have been drawing much attention in product development in recent years. In particular, structure-fluid interaction (FSI) problems are of major concern. In this article, a numerical simulation on air squeeze-film damping which is important in MEMS design is presented. The study employs Abaqus and STAR-CD to perform a structure-fluid co-simulation. The squeeze-film damping phenomenon of a simple plate structure is demonstrated and its mechanism investigated. An equivalent modeling method to approximately represent the squeeze-film effects is also examined. It is considered that this coupling analysis method based on Abaqus Co-simulation could be an effective approach to deal with a broad range of FSI problems.

Keywords: Squeeze-Film Damping, Fluid-Structure Interaction, Co-Simulation, MEMS, Vibration

1. Introduction

In recent years, CAE tools have been seeing increasing applications for dealing with multiphysics problems, such as fluid-structure interaction (FSI). A typical FSI example is air squeeze-film damping (SFD) phenomenon, which is characterized by the fact that when a vibrating plate moves close to a fixed surface, the air in the thin film will demonstrate resistances to the moving plate. This damping phenomenon is known from old times, but intensive studies on it are conducted only in recent years (Bao and Yang, 2007; Bicak and Rao, 2010; Pandey et al., 2007; Veijola and Lehtovuori, 2009). The background behind is the extending applications of MEMS (micro-electro-mechanical system) devices, where SFD plays an important role in improving the dynamics and functions of these devices. There is also report that SFD phenomenon is utilized to reduce noise and vibration of general mechanical systems (Ishihama and Hayashi, 2010).

It is considered that the studies on SFD can be classified into analytical (or theoretical) studies and numerical simulations. The former focuses on the basic characteristics of SFD subject to simple geometry and boundary conditions. On the other hand, the latter mainly makes use of finite element method (FEM) to solve the general Navier-Stokes equations or specialized Reynolds equations. Although the FEM approach is suitable for more complex geometry and boundary conditions and strong coupling problems, there are various difficulties in coding and solving the system equations which couple solid mechanics and fluid dynamics in terms of unified matrices. In this article, it is presented another approach, where commercial FEM code and computing fluid

dynamics (CFD) code are employed to deal with the FSI problem in a weak coupling way. That is, the structure-governing equation and the fluid-governing equation are solved independently in their own domains, and the results are exchanged at each time increment. Concretely speaking, Abaqus and STAR-CD are employed to perform a structure-fluid co-simulation to study the SFD phenomenon of a simple plate structure. It is considered that this coupling analysis method based on Abaqus Co-simulation could be an effective approach to deal with a broad range of FSI problems.

2. SFD model under study

In Figure 1 is shown the system under study in this paper. A thin rectangular steel plate, against to a rigid surface, is clamped at its two short edges. Air between the plate and the fixed surface is free to flow in and out from two long-edge sides. The air flow in the thin film is considered to be dominated by laminar flow due to the fact that the Reynold's number is usually very small. For example, assume that the gap h (film thickness) is $20\mu\text{m}$, and the plate vibrates at 1kHz , the Reynold's number can be calculated from equation $Re = \omega \rho h^2 / \mu$ to be 0.16 , where air density $\rho = 1.192(\text{kg}/\text{m}^3)$, air viscous coefficient $\mu = 1.86\text{E-}5(\text{Ns}/\text{m}^2)$. Other air characteristics such as compressibility, viscosity and temperature are incorporated into the model, and the 3D Navier-Stokes equation is solved by using STAR-CD. On the other hand, the implicit dynamic analysis of Abaqus (direct method) is employed to calculate the time response of the plate vibration under force excitation.

The data exchange between Abaqus and STAR-CD is realized by use of MpCCI (Mesh-based parallel Code Coupling Interface). The plate displacements obtained from Abaqus are passed over to STAR-CD, where the fluid domain changes its shape according to the plate deformation. This morphing process is achieved by using user subroutines in this research. In return, STAR-CD hands over the calculated air pressure to Abaqus which is then applied to the plate as distributed loads. This serial coupling scheme is shown in Figure 2.

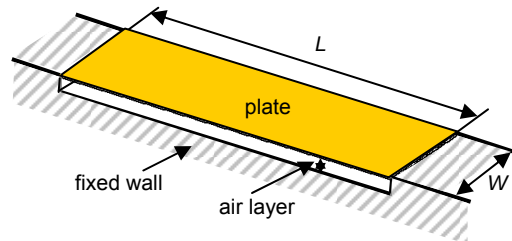


Figure 1. SFD model under study

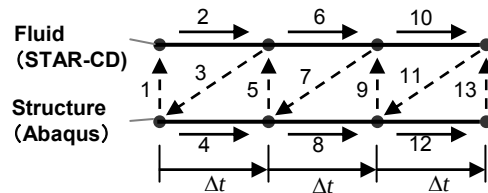


Figure 2. Serial coupling scheme

3. Implicit dynamic analysis

Before proceeding to co-simulation, let's first examine the damping definition in dynamic analysis. In Abaqus, material damping is defined in the form of proportional damping as follows,

$$[C] = \alpha[M] + \beta[K] \quad (1)$$

where, α , β are material damping coefficients, $[M]$, $[K]$, $[C]$ are mass matrix, stiffness matrix and damping matrix respectively. Unlike modal damping which can be estimated empirically, it is not a straightforward matter to estimate the values of α , β for a given structure. In the case of the plate under study in this article, it is assumed that the modal damping is given as listed in Table 1. With these damping values the response of the plate under force excitation can be calculated by using modal superposition method. A calculation example is shown by the thin line in Figure 3. On the other hand, if we calculate the response by using direct integration method, assuming a material damping $\beta = 0.005$, we get the result as shown in bold line. The two results are clearly different. Therefore, in order to use direct method to calculate the response correctly, it is first required to estimate α , β from the given modal damping.

Table 1. Modal damping of the plate

mode No.	mode (Hz)	damping ratio
1	360.08	0.006
2	925.34	0.005
3	991.64	0.005
4	1946.6	0.005
5	1952.3	0.006
6	3160.2	0.007
7	3222.3	0.008
8	4600.6	0.0085
9	4817.6	0.009
10	5462.9	0.0095

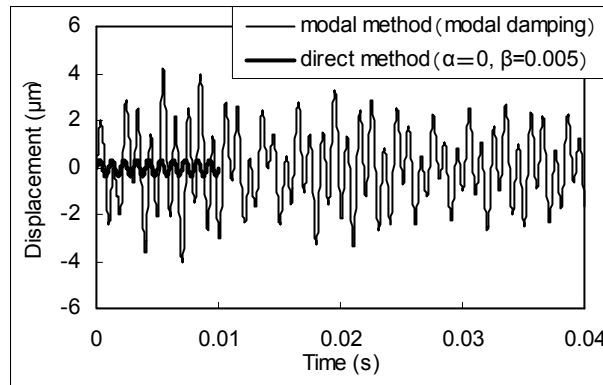


Figure 3. Displacement of the plate under force excitation

Transform Equation 1 into modal coordinates, one can obtain the following equation

$$\zeta_n = \frac{\alpha}{2\omega_n} + \frac{\beta\omega_n}{2} \quad (2)$$

which relates modal damping ratio ζ_n with the material damping coefficients α , β . From Equation 2 the material damping coefficients can be determined in two ways. The first one is to calculate α , β in terms of data at two specific modal frequencies, as follows

$$\begin{aligned}\alpha &= 2\omega_1\omega_2(\omega_1\zeta_2 - \omega_2\zeta_1)/(\omega_1^2 - \omega_2^2) \\ \beta &= 2(\omega_1\zeta_1 - \omega_2\zeta_2)/(\omega_1^2 - \omega_2^2)\end{aligned}\quad (3)$$

The second approach is to estimate α , β by using least mean square (LMS) method on the basis of all of the modal frequency and modal damping. The results are given in Equation 4.

$$\begin{Bmatrix} \alpha \\ \beta \end{Bmatrix} = \begin{bmatrix} \sum_{n=1}^N \frac{1}{2\omega_n^2} & \frac{N}{2} \\ \frac{N}{2} & \sum_{n=1}^N \frac{\omega_n^2}{2} \end{bmatrix} \begin{Bmatrix} \sum_{n=1}^N \frac{\zeta_n}{\omega_n} \\ \sum_{n=1}^N \omega_n \zeta_n \end{Bmatrix} \quad (4)$$

As an example, from the data in Table 1 and Equation 4, the material damping coefficients of the plate are estimated to be $\alpha = 28.5$, $\beta = 5.9\text{E} - 7$. In Figure 4 is given the estimated damping in the form of modal damping ratios, where thin solid line is the original data given in Table 1, bold solid line is the result synthesized from estimated α , β . For reference, the individual contributions of α term and β term are also shown. Based on these estimated material damping coefficients, the response calculated by using direct method (implicit dynamic analysis) is in good agreement with that obtained by using modal method, as compared in Figure 5.

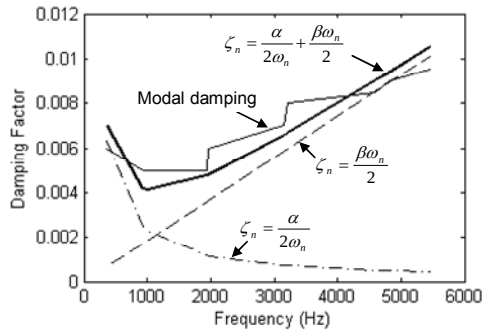


Figure 4. Results of estimated damping

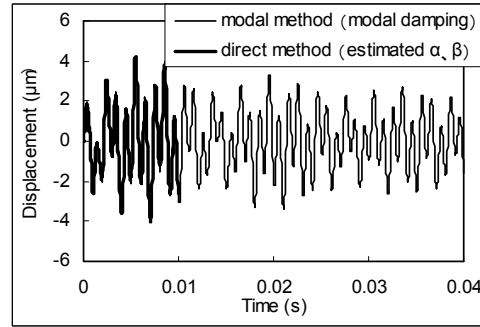


Figure 5. Response of the plate

In the following, these estimated material damping coefficients ($\alpha = 28.5$, $\beta = 5.9\text{E} - 7$) will be used in the implicit dynamic analysis employed in co-simulation process.

4. SFD study based on coupled analysis

For the system shown in Figure 1, four cases of different gap between the plate and the fixed wall, namely 1mm, 100 μ m, 50 μ m and 20 μ m, are investigated by using structure-fluid coupled analysis on the basis of Abaqus co-simulation technique. In Figure 6 are shown the calculated time responses under a 1kHz harmonic force excitation of 0.5 Newton. The thin lines are the results without considering the SFD effect, while the bold lines are the results with the SFD effect taken into account. For the case of gap=1mm, two results are almost identical, indicating that no SFD effect is noticeable when the gap is as large as 1mm. For other three cases, SFD effects are more or less observed. The smaller the gap is, the stronger the SFD effect.

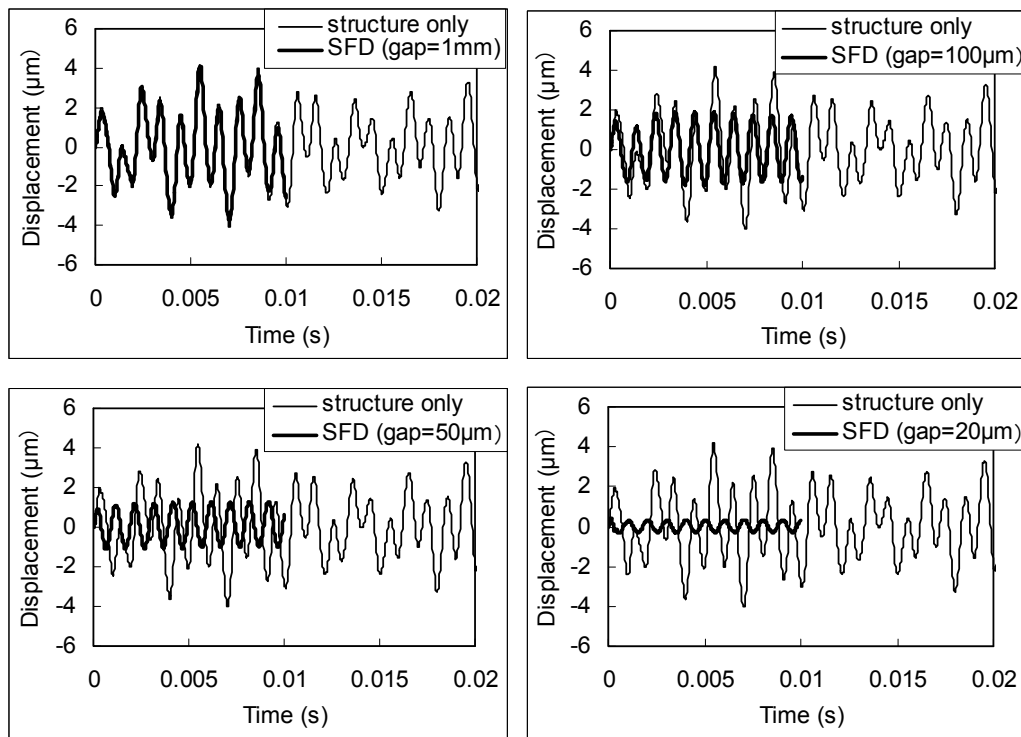


Figure 6. Responses of the plate with / without SFD effect

Furthermore, the response under impact excitation is also calculated. The impact force is defined as shown in Figure 7, and the resulting responses are given in Figure 8. Clearly, the transient response of the plate is well suppressed by the SFD effect at smaller gaps.

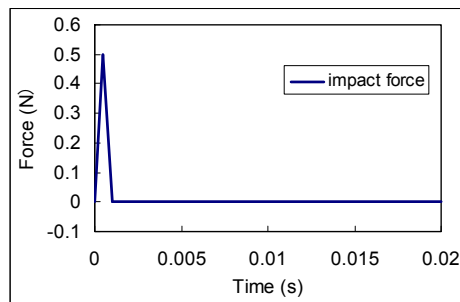


Figure 7. Impact force definition

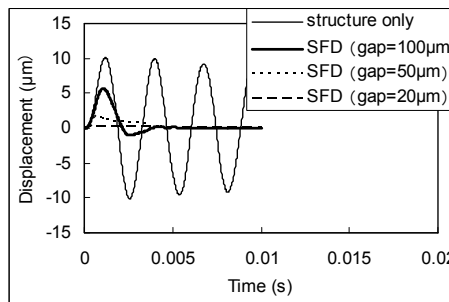


Figure 8. Responses under impact excitation

In order to examine the behavior of the thin air film, Figure 9 gives the calculated results in the case of $\text{gap}=50\mu\text{m}$. The ones on the left are the plate deformations at specific instants, and the ones in the middle are the air pressure distributions backing the plate, the ones on the right the air flow velocity maps at the same instants. When the plate deforms downwards from its initial position, positive pressure occurs in the air film and the air flows outwards. On the other hand, when the plate deforms upwards, negative pressure occurs and the air flows inwards. That is, as the plate vibrates, there will be pressure oscillation generated in the thin air film, and the pressure is always against the plate deformation. This is the mechanism of squeeze-film damping phenomenon.

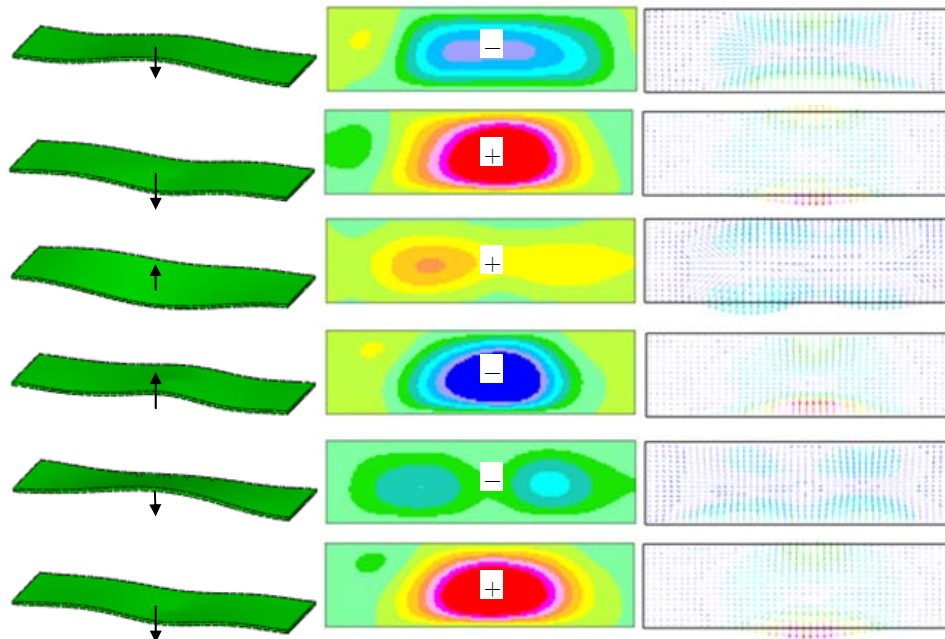


Figure 9. Plate deformation (left) and air pressure (middle) and flow velocity (right)

Here we will go a little further to examine the characteristics of the air pressure. Figure 10 shows the air reaction forces on a fluid cell and the corresponding plate displacements at the same location. For the case of gap=1mm, the air reaction force is very small and has no influence on the plate response. As the gap becomes smaller, the air reaction force becomes larger and consequently exerts significant influence on the plate deformation. As far as the phase is concerned, the case of gap=100 μ m demonstrates a nearly 90° phase difference between the air reaction force and the plate displacement. As the gap goes smaller such as the case of gap=20 μ m, the phase difference approaches 180°. As we know, when force and displacement are in 90° phase difference, the force can be considered as viscous force. While the phase difference is 180°, the force can be considered as elastic force. Therefore, the air reaction force in SFD can be regarded as possessing the characteristics of both viscous force and elastic force. As the gap changes from relatively large to small, it is considered that the feature of air reaction force change from viscous-dominated to elastic-dominated. For an easy understanding, Figure 11 depicts the characteristics of air reaction force in SFD.

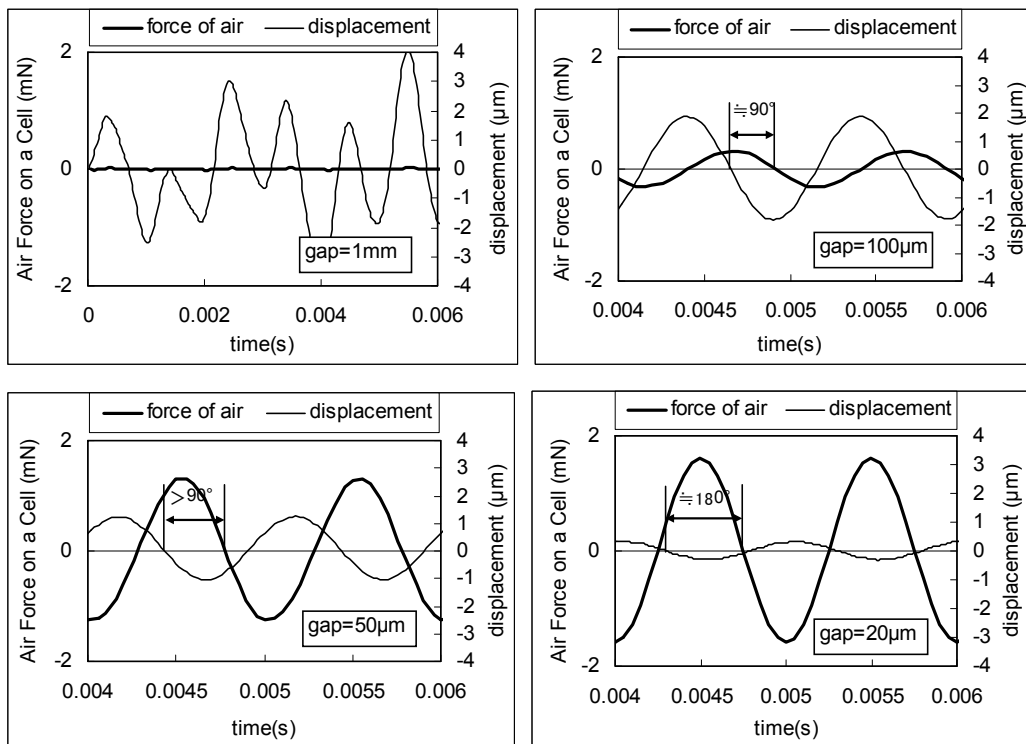


Figure 10. The relationship between air reaction force and plate deformation

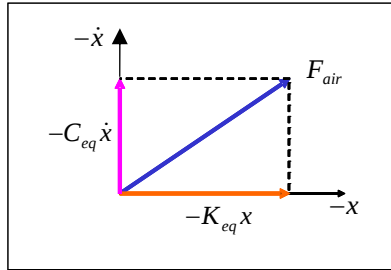


Figure 11. The characteristics of air reaction force

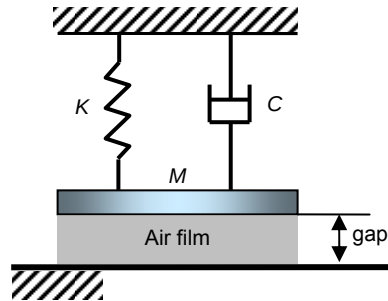


Figure 12. Rigid piston model
($f_n=1\text{kHz}$, damping ratio=0.5%)

5. Equivalent modeling of SFD

As discussed in the proceeding section, the air reaction force of SFD possesses features of viscous and elastic forces. It is therefore a straightforward idea to employ viscous and elastic elements to equivalently represent the SFD effects somehow. In order to verify this consideration, here we employ a simple single degree of freedom vibration system as shown in Figure 12 to perform structure-fluid coupled analysis.

The piston system without air film is assumed to be with a natural frequency of 1kHz and a damping ratio of 0.5%. The forced vibration of the system under a harmonic force excitation of 1kHz is calculated without or with the air SFD effect considered. The results are shown in Figure 13. Without considering air SFD, the piston resonates and the vibrating amplitude becomes larger and larger. With air SFD taken into account and the gap being small, piston vibration is apparently suppressed.

In the following, we will examine the coupled analysis results in the cases of gap=50 μm and gap=20 μm a little further, in order to investigate the equivalent modeling method of SFD.

5.1 The case of gap=50 μm

Figure 14 shows the air reaction force and piston velocity in this case (exciting frequency=1kHz). It can be seen that these two quantities are almost in opposite phase, indicating that the air SFD is dominated by viscous effect. Therefore, it is reasonable to represent the air SFD effect by employing an equivalent viscous damping element. Here we will show how to determine the equivalent viscous damping from these calculated results.

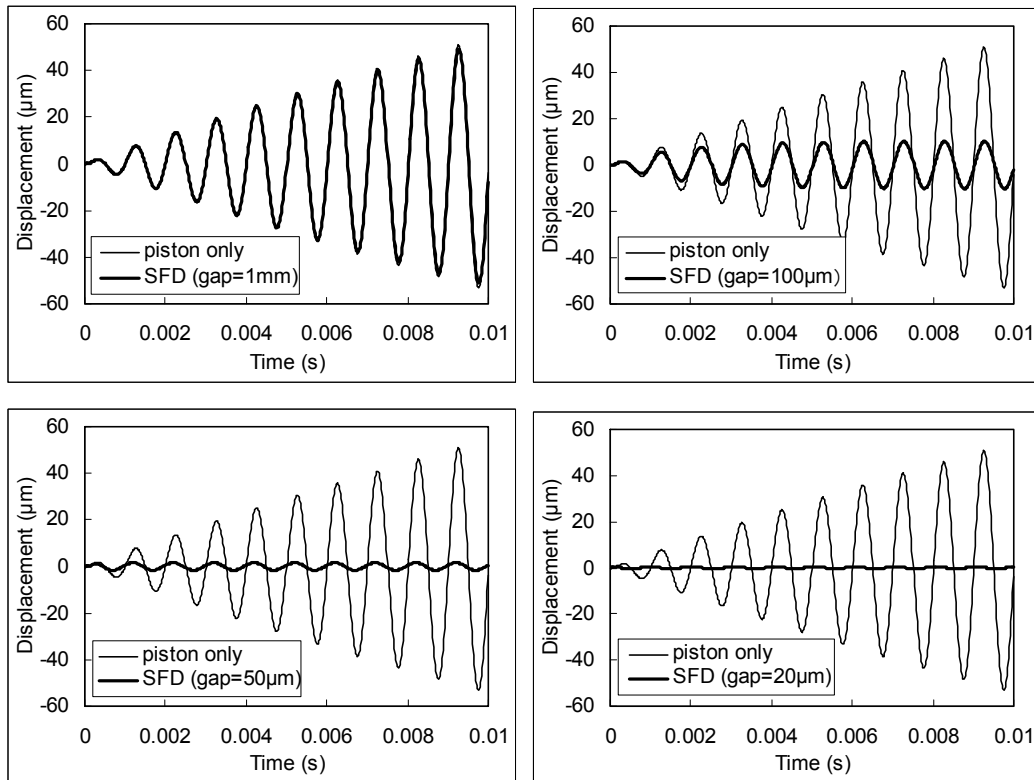


Figure 13. Responses of the piston with / without SFD effect

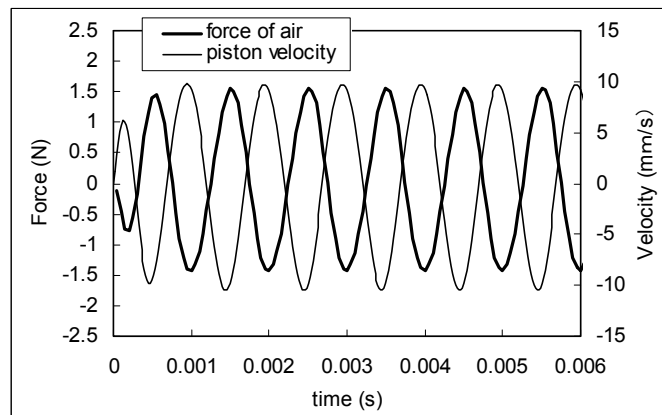


Figure 14. The results of case gap=50 μm (exciting frequency=1kHz)

The work done by air reaction force to the piston can be estimated from the following equation

$$W_{air} = \left| \int F_{air}(t) dx \right| = \left| \int F_{air}(t) \dot{x}(t) dt \right| \quad (5)$$

On the other hand, the work done by an equivalent damping can be calculated from

$$W_{eq} = \left| \int F_{eq}(t) dx \right| = C_{eq} \left| \int \dot{x}^2(t) dt \right| \quad (6)$$

Assuming $W_{air} = W_{eq}$, one can obtain the equivalent damping coefficient of air film as

$$C_{air} = C_{eq} = W_{air} / \left| \int \dot{x}^2(t) dt \right| = \left| \int F_{air}(t) \dot{x}(t) dt \right| / \left| \int \dot{x}^2(t) dt \right| \quad (7)$$

Based on the above equations and the results given in Figure 14, the equivalent damping coefficient is determined to be $C_{eq}=135$ (Ns/m), which is about 115 times of the original damping. In terms of damping ratio (ζ_n), the value goes from original 0.5% up to 57%. Replacing the air film with this equivalent damping we get an equivalent model as depicted in Figure 15. The piston response of this system can be calculated from a general structure dynamic analysis. The result is in good agreement with that obtained from coupled analysis, as compared in Figure 16. The feasibility of this equivalent modeling approach is therefore verified.

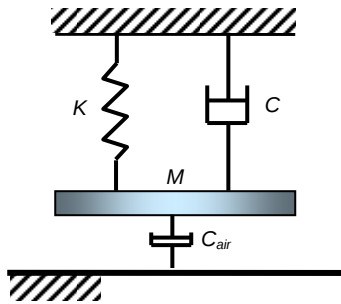


Figure 15. Equivalent damping

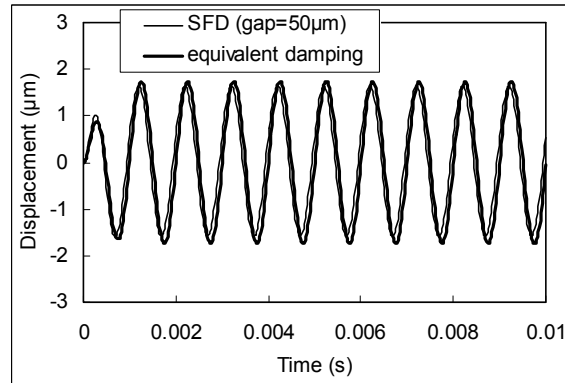
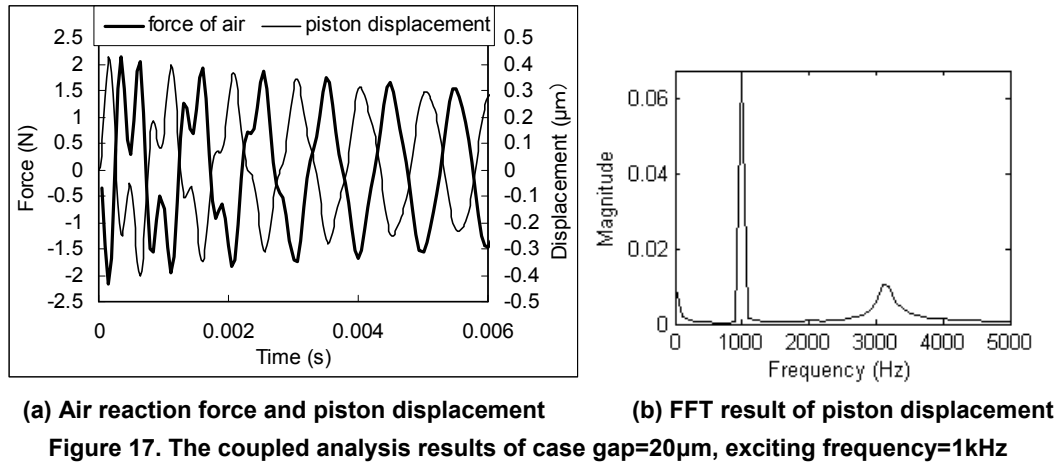


Figure 16. Coupled analysis vs equivalent damping

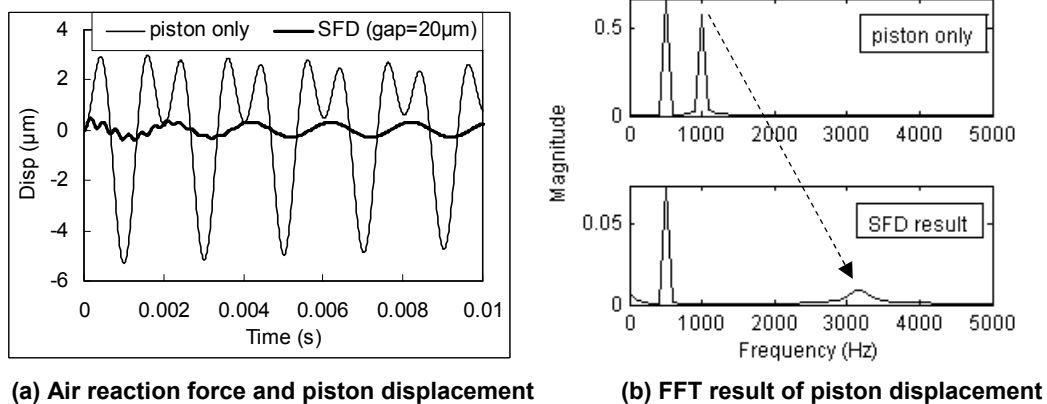
5.2 The case of gap=20 μm

For this case, the air reaction force and piston displacement under 1kHz force excitation are shown in Figure 17(a). It is interesting to note that this time the air reaction force is almost out of phase

with the displacement, and also that there is other frequency component besides the exciting frequency 1kHz. With fast Fourier transformation (FFT), it is shown in Figure 17(b) that there is a new peak around 3.1kHz.



In order to clarify the frequency of 3.1kHz, change the exciting frequency from 1kHz to 500Hz and conduct the coupled analysis again. The results are given in Figure 18. For the occasion of not considering the SFD effect (piston only), there are two peaks, at the exciting frequency 500Hz and the system natural frequency 1kHz, on the FFT result. When SFD is taken into account, the peak at the exciting frequency 500Hz remains, but the peak at 1kHz shifts to 3.1kHz. That is to say, the natural frequency of the system changes from 1kHz to 3.1kHz. In other words, the air film exerts strong influence on the stiffness of the system.



The equivalent stiffness in accordance with 3.1kHz is determined to be 9.6 times of that of the original system. It is found that in addition to the equivalent stiffness, equivalent damping effect should also be considered. As a result of correlation, appropriate equivalent damping is incorporated so that the resulting damping ratio is increased from original 0.5% to about 2%. Replacing the air film with the equivalent stiffness and damping elements we have an equivalent model as shown in Figure 19. The response of this equivalent model can be easily calculated from a general Abaqus dynamic analysis. The result is compared with the SFD co-simulation result in Figure 20. The two results are in reasonably good agreement, indicating that the equivalent model is correct.

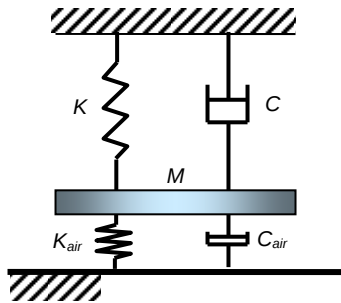


Figure 19. Equivalent model

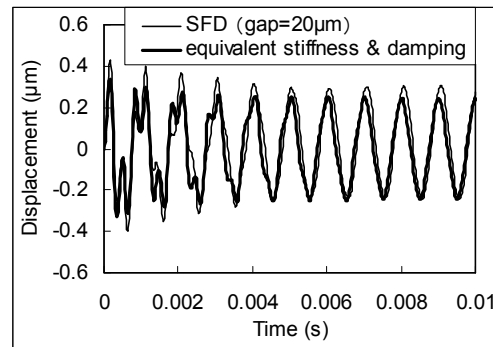


Figure 20. Coupled analysis vs equivalent model

6. A practical example

As an application of the equivalent modeling method, Figure 21 shows the frequency response of a practical mechanical system, where a component structure and the machine body have a thin air film between them, and thus constitute a SFD. When the thickness of the air film (gap between the part and the body) is relatively large, the calculated frequency response function is in fairly good agreement with the measurement result, indicating that the SFD effect can be ignored. When the gap is small, the measured frequency response changes a lot that cannot be correlated with FEM analysis. In this case, it is proved that SFD effects are significant and the influence should be taken into account in the analysis model. Although it is possible to conduct a structure-fluid coupling analysis, here we adopt the equivalent modeling method for simplicity. With equivalent stiffness and damping elements adopted into the model, the analysis result correlates reasonably well with the test result.

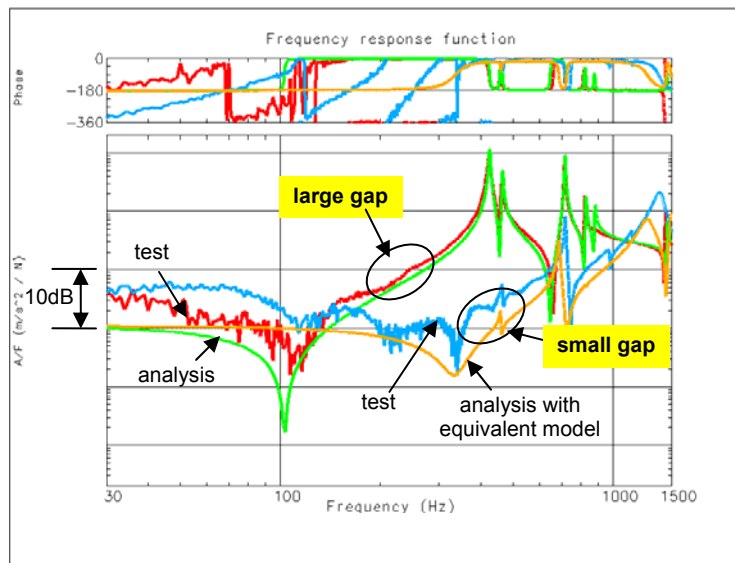


Figure 21. Frequency response functions of a real structure showing the SFD effects

7. Summary

In the article, the structure-fluid coupled analysis based on Abaqus Co-simulation option is applied in the study of air squeeze-film damping. The air SFD phenomenon is demonstrated and its mechanism investigated. It is shown that as the film thickness is decreased, the air SFD effect changes from viscous-dominated to elastic-dominated. It may be possible to represent the SFD effect with lumped stiffness and damping elements. This equivalent modeling method has been demonstrated through a practical example. However, it should be pointed out that the equivalent model may not necessarily results in satisfactory results in some complex conditions. On those occasions, it may be necessary to conduct a co-simulation analysis as introduced.

Although the object under study in this paper is simple, it is considered that this coupling analysis method based on Abaqus Co-simulation could be an effective approach to deal with a broad range of FSI problems.

Finally, it is mentioned that Direct Coupling Interface is implemented in Abaqus later than Ver.6.8, which illuminates the need for MpCCI in a co-simulation with Abaqus and STAR-CD.

8. References

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