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# Acoustic Structure Interaction with Damping Interface

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***Abstract*— This paper deals with the finite element method to compute the vibrations of a fluid - structure coupled system. The fluid structure system consists of an acoustic fluid and a plate structure. The Finite element analysis is done with and without the interphase damping and the results are compared. The damping material separates both the media, the fluid and the structure. The plate structure is described by a displacement field which contains an inviscid, compressible and barotropic fluid. The barotropic fluid is described by a pressure field. The gravity effects are neglected in the study. The aim of this study is to develop new noise reduction techniques in fluid structure interaction. The originality of the work lies in introducing a damping interphase in fluid structure interaction using ANSYS software and also intends to develop a simulated experiment of impedance tube.**

***Keywords— Fluid Structure Interaction, Acoustics, Damping Interface, Acoustic Admittance, Acoustic Impedance.***

## I. INTRODUCTION

This study describes the basic finite element formulations of a coupled fluid structure system and finite element analysis of a cavity and plate problem using ANSYS. While performing finite element analysis of cavity and plate problem the modal parameters which includes the mode shape and modal frequencies of fluid alone system, plate alone system, undamped plate and cavity coupled system and damped plate and cavity coupled system where determined. For the case of fluid system the fluid elements are defined in terms of pressure degree of freedom. So the mode shapes obtained were in terms of pressure gradient. The plate mode shapes were shown in terms of displacement. Coupled system mode shapes show both pressure gradient and displacement.

The damping levels of coupled- damped FSI structure (Admittance values) were varied to observe the effects on modal parameters..

## II. FLUID ELEMENT DESCRIPTION

### *A. FLUID30 ELEMENT*

In this work FLUID30 is used for modeling the fluid medium and the interface in fluid/structure interaction problems. Typical applications include sound wave propagation and submerged structure dynamics. The governing equation for acoustics, namely the 3-D wave equation, has been discretized taking into account the coupling of acoustic pressure and structural motion at the interface. The element has eight corner nodes with four degrees of freedom per node: translations in the nodal x, y and z directions and pressure. The translations, however, are applicable only at nodes that are on the interface. Acceleration effects, such as in sloshing problems, may be included.

The element has the capability to include damping of sound absorbing material at the interface. The element can be used with other 3-D structural elements to perform unsymmetric or damped modal, full harmonic response and full transient method. When there is no structural motion, the element is also applicable to static, modal and reduced harmonic response analyses. The Element shape function is defined by:

$$N_a = \frac{1}{8}(1+\xi_a\xi)(1+\eta_a\eta)(1+\psi_a\psi) \qquad (1)$$

which is identical with the linear lagrangian element. Fluid stiffness and mass matrix (2x2x2 integration points, only for pressure). Coupling stiffness and mass matrix (2x2 integration points, only at the FS interface, of displacements and pressure)

### *B. Solution technique: The Unsymmetric Method*

Unsymmetric method, which also uses the full [K] and [M] matrices, is meant for problems where the stiffness and mass matrices are unsymmetric (for example, acoustic fluid-structure interaction problems). It uses the Lanczos algorithm which calculates complex eigenvalues and eigenvectors if the system is non-conservative (for example, a shaft mounted on bearings). The real part of the eigenvalue represents the natural frequency and the imaginary part is a measure of the stability of the system - a negative value means the system is stable, whereas a positive value means the system is unstable. Sturm sequence checking is not available for this method. Therefore, missed modes are a possibility at the higher end of the frequencies extracted. The unsymmetric eigen solver is applicable whenever the system matrices are unsymmetric. For example, an acoustic fluid-structure interaction problem using FLUID30 elements results in unsymmetric matrices. A generalized eigen value problem given by the following equation

$$[K]\{\phi_i\} = \lambda_i [M]\{\phi_i\} \quad (2)$$

can be setup and solved using the mode-frequency analysis. The matrices [K] and [M] are the system stiffness and mass matrices, respectively. Either or both [K] and [M] can be unsymmetric. $\{\phi_i\}$ is the eigenvector.

The method employed to solve the unsymmetric eigenvalue problem is a subspace approach based on a method designated as Frequency Derivative Method. The FD method uses an orthogonal set of Krylov sequence of vectors:

$$[Q] = [\{q_1\}\{q_2\}\{q_3\}\{q_4\}.....\{q_m\}] \quad (3)$$

To obtain the expression for the sequence of vectors, the generalized eigen value is differentiated with respect to $\lambda_i$ to get:

$$-[M]\{\phi_i\} = \{0\}$$

Substituting (3) and (2) and rearranging after applying a shift s, the starting expression for generating the sequence of vectors is given by:

$$[[K]-s[M]]\{q_1\} = \{q_0\}$$

*where* (5)

$$\{q_0\} = -[M]\{\phi_i\}$$

The general expression used for generating the sequence of vectors is given by:

$$[[K]-s[M]]\{q_{j+1}\} = \{\bar{q}_j\} \quad (6)$$

This matrix equation is solved by a sparse matrix solver.

A subspace transformation of (5) is performed using the sequence of orthogonal vectors which leads to the reduced eigen problem:

$$[K^*]\{y_i\} = \mu_i [M^*]\{y_i\} \quad (7)$$

where:

$[K^*] = [Q^T] [K] [Q]$

$[M^*] = [Q^T] [M] [Q]$

The eigen values of the reduced eigen problem are extracted using a direct eigen value solution procedure.

The eigen values $\mu_i$ are the approximate eigen values of the original eigen problem and they converge to $\lambda_i$ with increasing subspace size m. The converged eigenvectors are then computed using the subspace transformation equation:

$$\{\phi_i\} = [Q]\{y_i\} \quad (8)$$

For the unsymmetric modal analysis, the real part ($\omega_i$) of the complex frequency is used to compute the element kinetic energy.

This method does not perform a Sturm Sequence check for possible missing modes. At the lower end of the spectrum close to the shift, the frequencies usually converge without missing modes.

## III. PLATE- FLUID CAVITY ANALYSIS

In this section, finite element results obtained with the previous formulations for the analysis of interior damped structural–acoustic systems were presented. Firstly, a 3D free vibration elastoacoustic problem with damping interface is analyzed. The acoustic analysis in ANSYS was done in the following order. Fluid mentioned in these problems is air at standard atmospheric conditions.

### *A. . Input Data for Analysis*

| | |
|---|---|
| Length of the cavity, X | = 0.6 m |
| Width of the cavity, Y | = 0.5 m |
| Height of the cavity, Z | = 0.4 m |
| Thickness of the plate, H | = 6e-3 m |
| Young's Modulus, E | = 1.44e11 N/m$^2$ |
| Poisson's Ratio | = 0.35 |
| Density plate | = 7700 kg/m$^3$ |
| Sonic velocity | = 340 m/s |
| Density fluid | =1.2041kg/m$^3$ |

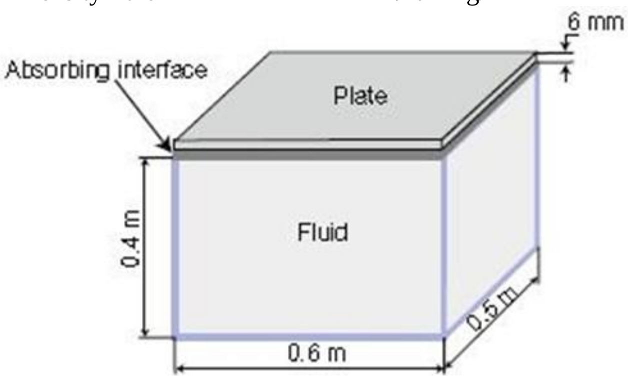


Fig 1 Acoustic cavity modeled.

## B. *Modal Analysis*

- Modal Analysis of Acoustic Fluid Cavity in ANSYS.

Modeling of acoustic fluid elements in ANSYS have been studied and an acoustic fluid cavity has been modeled in ANSYS using 3D FLUID 30 element, which is an eight noded brick element. Modal analysis has been done on the modelled acoustic cavity using ANSYS. The sonic velocity selected was 340 m/s and the density of the acoustic cavity was selected as 1.2041 kg/m$^3$.

Table 1. Fluid alone results (Resonant frequencies)

| Mode No: | Frequency (Hz) |
|---|---|
| 1 | 283.625 |
| 2 | 340.35 |
| 3 | 425.437 |
| 4 | 443.036 |
| 5 | 511.312 |
| 6 | 544.825 |
| 7 | 569 |

- Modal Analysis of Plate (Using Shell 181 Element)

The plate is modelled using shell181 element in ANSYS. With density ρ=7700 kg/m3, Young's modulus of 144×1011 Pa and poisson ratio υ = 0.35. Plate thickness is selected as 6 mm. Modal analysis were done in ANSYS and modal parameters are plotted.

Table 2. Plate alone results (using shell181)

| Mode No: | Frequency (Hz) |
|---|---|
| 1 | 156.50 |
| 2 | 283.00 |
| 3 | 352.73 |
| 4 | 468.58 |
| 5 | 487.57 |
| 6 | 654.91 |
| 7 | 662.99 |

- Undamped plate- fluid Coupled System.

For undamped plate fluid coupled system analysis the plate was modeled using shell181 element and the fluid portion were modeled using FLUID30 element with rigid wall boundary condition and structure (plate) present at one of its boundaries. Shell181 element is selected because it gives better result for bending problems compared to solid shell element. The dimensions of the system were same as the previous analysis.

Table 3. Plate fluid coupled undamped system (using shell181 and fluid30 elements)

| Mode No: | Frequency (Hz) |
|---|---|
| 1 | 152.862 |
| 2 | 279.715 |
| 3 | 287.653 |
| 4 | 339.213 |
| 5 | 355.763 |
| 6 | 425.52 |
| 7 | 442.382 |

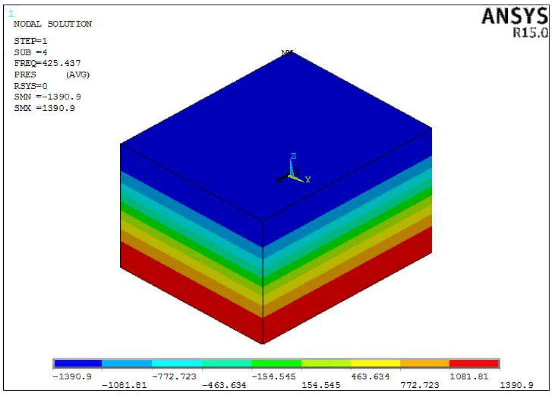


Fig 2.Pressure gradient plot of acoustic cavity at 425 Hz

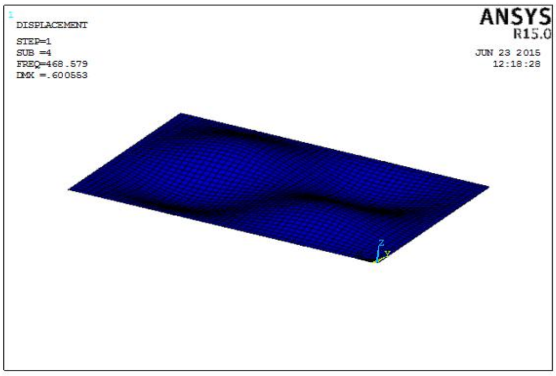


Fig 3. Mode shape of plate (shell181) at 468 Hz.

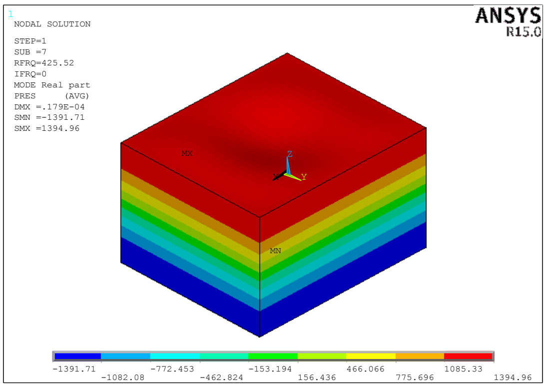


Fig 4. Modal plots of coupled plate fluid undamped system at 425 Hz

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### *C. DAMPED PLATE- FLUID COUPLED SYSTEM*

For damped plate fluid coupled system analysis the plate was modeled using shell181 element and the fluid portion were modeled using FLUID30 element with rigid wall boundary condition and structure present at one of its boundaries. Shell181 element is selected because it gives better result for bending problems compared to solid shell element. The dimensions of the system were same as the above. Acoustic boundary admittance was provided at the boundary for giving the damping effect in the coupled structure.

Acoustic boundary admittance values were varied in repeated analysis of the same system to analyse the effects of admittance values. Three different admittance values were used 0.2, 0.5 and 0.7. Mode shapes were plotted at each modal frequency, to observe the effects of damping. The mode shapes were compared with the previous undamped coupled system analysis to analyse what damping actually does and how they affect the system.

Table 4. plate fluid coupled damped system (using shell181 and fluid30 elements), Modal frequencies for applied real admittance 0.5.

| Mode No: | Frequency (Hz) |
|---|---|
| 1 | 156.476 |
| 2 | 282.67 |
| 3 | 294.008 |
| 4 | 352.217 |
| 5 | 425.268 |
| 6 | 458.778 |

Table 5. Plate fluid coupled damped system (using shell181 and fluid30 elements), Modal frequencies for applied real admittance 0.2.

| Mode No: | Frequency (Hz) |
|---|---|
| 1 | 156.48 |
| 2 | 282.594 |
| 3 | 284.919 |
| 4 | 341.281 |
| 5 | 352.652 |
| 6 | 425.202 |
| 7 | 444.906 |

Table 6. plate fluid coupled damped system (using shell181 and fluid30 elements), Modal frequencies for applied real admittance 0.7.

| **Mode No:** | **Frequency (Hz)** |
|---|---|
| 1 | 156.472 |
| 2 | 282.679 |
| 3 | 305.848 |
| 4 | 352.144 |
| 5 | 365.591 |
| 6 | 425.339 |

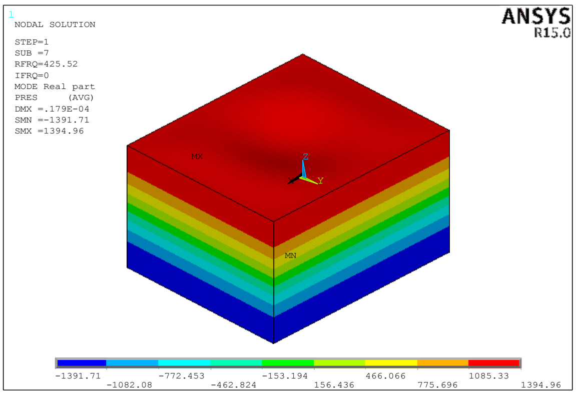


Fig 5. Sixth mode of undamped coupled system where pressure gradient is normal to the damping surface.

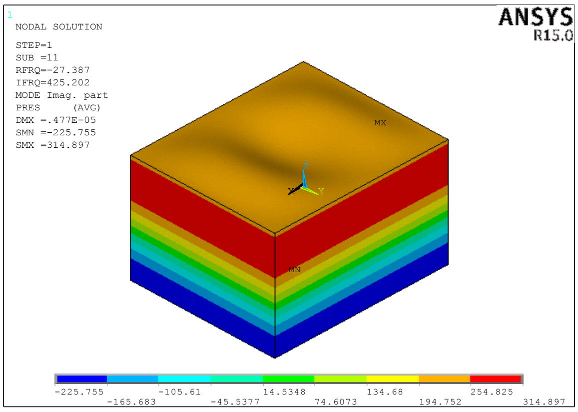


Fig 6. The sixth mode of damped system of real admittance value of 0.2

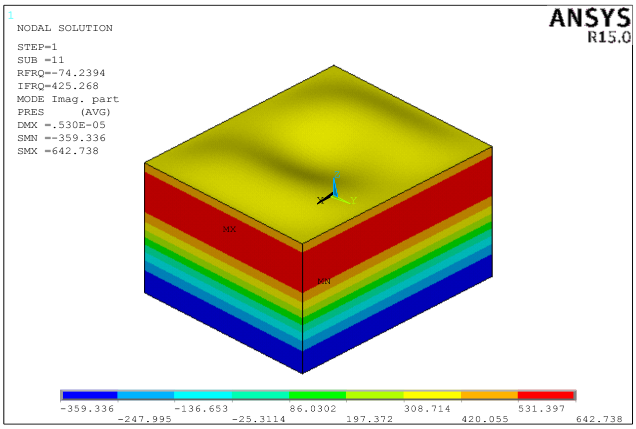


Fig 7. Fifth mode of damped system of real admittance value of 0.5

Fig 7. Fifth mode of damped system of real admittance value of 0.5

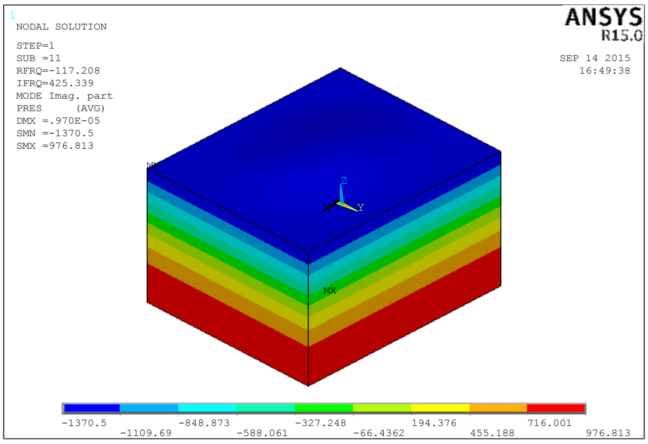


Fig 8. Sixth mode of damped system of real admittance value of 0.7

## IV. RESULTS AND DISCUSSION

The results of the modal analysis of damped coupled system shows that damping or application of real admittance in the system affects both the structure displacement and fluid pressure modes . The difference in pressure generation and displacement is shown in the following figures. But it can be observed that in certain modal plots pressure gradient is much significant than the displacement, these modes are known as fluid dominant modes. The modes where structure displacement have significant values are known as structure modes and the modes in which both the structure and fluid modes are significant are known as coupled modes.

The first mode for all the three admittance values (damped system) and undamped coupled system are coupled modes. The application of damping reduced its displacement as well as pressure gradient values. And it can be noted that only normally incident pressure gradient values shows damping effects, i.e. the pressure gradient which is in the direction normal to damping is affected, the other pressure gradient plots which is not in the direction normal to damping or dissipating surface is not affected by the admittance values (Acoustic damping).

## V. CONCLUSIONS

Damping of the coupled fluid plate system affects the pressure gradient significantly, hence it can be concluded that the damping significantly affects the fluid mode of a coupled system. When damping is introduced it should be noted that only the pressure gradient which is normal to the damping surface is reduced. This inference helps to study the effects of different types of damping material in an acoustic system by knowing its acoustic impedance value. The application of damping interface in a cabin system of automobile or aircraft reduces the normally incident pressure gradient in the system.

Damping of acoustic systems can be done in ANSYS using boundary admittance method. The application of boundary admittance in damped acoustic systems in ANSYS is similar to the application of a damping material in an acoustic system in a real situation. This can be justified using simulation of impedance tube test.

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