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# MODAL ANALYSIS OF LARGE STRUCTURES SUBMERGED IN A FLUID

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**Abstract**

Vibration of ship appendages that operate underwater during their service life can be excited by either fluid flow over their surfaces or by internal ship-generated structural loads. In order to assess the effect of these excitation loads the modal frequency response of the appendage needs to be determined in water.

Assessing the modal response of structures in a vacuum or in air via experimental and numerical means has matured, whereas the modal response of a structure in a dense fluid can still be a challenge. This paper will describe a number of techniques for handling the fluid loading and will use as an example a large wing-like hydrofoil structure to show the effect and importance of the fluid loading.

## 1. INTRODUCTION

Ship appendages such as stabilisers, rudders, and control surfaces will experience excitation loads on their structure resulting from either inboard transmitted loads (e.g. machinery) or outboard applied fluid loads (e.g. fluid flow or hull slamming). The applied loads will depend on the operational conditions, such as the sea state in which the vessel operates, and could excite vibrational modes of the structure and onboard equipment. Vibrational analysis of a structure underwater using numerical modal techniques requires an additional consideration compared to an in-vacuum analysis, namely the contribution of the hydrodynamic mass effect of the water on the submerged part. In this case, the coupling between the structure and the dense fluid leads to significant changes of the modal frequencies of vibration when compared with in-vacuum modes.

This paper describes methods for determining the modal frequencies (and vibrational response) of submerged structures using finite element and boundary element modelling techniques. The advantages and disadvantages of both techniques will be discussed. The calculation techniques are demonstrated using the example of a large hydrofoil structure.

# 2. SOLUTION TECHNIQUES

A number of techniques are possible to obtain the modal frequencies for a structure vibrating in a fluid. Some that will be discussed in this paper are:

(a) eigenvalue extraction,
(b) frequency domain method,
(c) time domain method,
(d) experimental method.

The three numerical techniques may be done with commercially available software. The Finite Element (FE) method and the Boundary Element (BE) method are two techniques that can be used alone or in combination with each other.

In the Finite Element (FE) method a structure is discretized (meshed) into nodal points, and the connectivity between the nodes then forms elements in 1D, 2D or 3D space. The properties of the materials are accounted for through element stiffness and mass matrices, which are attributed to the element nodal points. These element matrices are assembled to form global stiffness and mass matrices for the entire structure. The addition of fluid to the structural problem requires a volume of fluid to be modelled in a simular fashion to the structure except pressure degrees of freedom exist at the fluid element nodes as opposed to displacement degrees of freedom for the structure. The size of the fluid model is a function of the frequencies of interest and is generally a few times the major characteristic length of the structure. Additionally, the size of the fluid and structural elements needs to be sufficiently small to capture the highest frequency of interest. An absorption layer is also required on the outer extremities of the fluid volume in order to prevent the pressure wave reflection back onto the structure. For fluid-structure interaction problems a fluid-structure interaction layer between the structure and fluid models also needs to be defined and the nodes associated with these elements must have both displacement and pressure degrees of freedom. For practical problems the number of elements and nodes of FE models with a structure and fluid can be large. Development and run time of the model can therefore be both time-consuming and numerically intensive.

In the Boundary Element (BE) method only surfaces of the 3D problem are meshed. The BE method therefore offers an efficient way of representing the surrounding fluid, as only the surface in contact with the fluid needs to be meshed. Typically, the interaction between the fluid and structural meshes is solved using the Helmholtz Integral equation. The resulting system of equations therefore connects the nodes on one part of the mesh with nodes on essentially any other part of the mesh. The matrices that result from the BE method are dense, and may be non-symmetric, rather than sparse and symmetric as in the FE method.

FE modelling, where the varying material properties in the structure can be accurately represented, best represents complex multi-material 3D structures. Furthermore, BE modelling in solids is difficult. It therefore follows that the best approach to solving the problem of a structure in a fluid is a mixed FE/BE formulation with FE used for the structure and BE used for the fluid.

## 2.1 Eigenvalue Extraction

Modal frequencies in a vacuum can be obtained by solving for the eigenvalues of the matrix equation of a FE model. In a fluid, modal frequencies can also be obtained by solving for the eigenvalues using either FE or a combined FE/BE technique. The method involves the determination of the virtual mass of the structure, which is the sum of the actual and hydrodynamic masses. The latter mass results from the influence of the fluid on the vibrating structure. No external load needs to be applied as the modal frequencies and shapes are computed directly. However, when applying a fully-FE method to a fluid-loaded structure, the method is generally limited to applications that do not require more than a few thousand 'wetted' surface elements. If applied to larger models the computational time can be unrealistic. Also, the building of the fluid model for a complex structure can also be a time consuming task, especially for 3D models. For the BE method, no fluid volume model is required, and a combined FE/BE approach, as previously discussed is best.

The advantage of the eigenvalue extraction method is that it can make use of modal superposition to compute the response of the structure, or the pressure in the fluid, due to an arbitrary load, rather than by applying the direct frequency domain method discussed below. The former method is quicker in execution time, as it first calculates the frequencies and modes shapes, and then applies the excitation loading over the frequency range of interest to calculate, for example, the nodal displacements of the structure as linear combinations of the mode shapes. The direct method is more general but typically has longer run times if results are required at many frequencies.

### 2.2 Frequency Domain Method

This direct method can be applied to either FE or BE models for both the structure and fluid. The approach involves applying a unit force of single frequency on a structure to excite it. The load is generally applied at a location where all structural modes have significant displacement. The unit load is swept through the range of frequency values in which the modes of interest for the structure are expected to reside. The frequency increments will depend upon accuracy being sought. The magnitude of nodal displacements versus frequencies at the location of the applied load can be obtained for both the vacuum and water loading scenarios. Although the size of a BE model is generally smaller than a FE model the run times depend on actual model size.

### 2.3 Time Domain Method

This method can be applied to either FE or BE models for both the structure and fluid. It involves applying a time-dependent load to the structural model, thus exciting it, and then determining its time domain response at any specified location. With the time history response, an FFT is performed to obtain the modal frequencies of the structure. Solutions in a vacuum and a fluid are possible, but the authors' experience [1] with available software [2], [3] has not produced reliable results in water, with structural modal frequencies consistently lower than experimental measurements. This method is essentially a simulation of an actual testing of the structure. The total simulated run time and minimum time step required will depend upon the minimum frequency resolution and the highest frequency of interest being sought, respectively. This method is generally slower than the frequency domain method and is not recommended as it can have large model sizes (FE only), large run times and produce questionable results.

### 2.4 Experimental Approach

This approach is possible for both air (approximating a vacuum) and fluid loading, and involves the excitation of the structure with a load, often applied by a hand-held hammer. The recording of the subsequent flexural response (e.g. displacement time history) at a number of locations on the structure is undertaken, in order to obtain the deformation response shapes. These displacement time histories are then processed by FFT or by Modal analysis software to obtain the modal frequencies for the structure [4].

This is a reliable method under controlled laboratory conditions but can be a challenge when applied to large submerged structures that need to be done in the open environment under the prevailing weather conditions. The structure, of course, also needs to exist and the numerical approaches discussed previously have the advantage of being able to predict the response of alternative designs of a structure before it is built.

## 3. THE LARGE HYDROFOIL

To demonstrate the approach the modal frequencies of a large hydrofoil were calculated in a vacuum and water. The large hydrofoil is shown in Figure 1. Two boundary conditions on the shaft were investigated because it was difficult to exactly determine the correct boundary conditions for the problem. The two conditions chosen represent the extreme cases and were investigated to determine the sensitivity of the predicted mode shapes and frequencies of the structure to those conditions. In the first case, the inner shaft that controls the hydrofoil orientation is free to rotate and move axially inside the outer shaft, with the outer shaft rigidly fixed in all translational and rotational degrees of freedom. In the second case, that portion of the inner shaft exposed in Figure 1, is fixed in all degrees of freedom. There is no explicit material damping of the structure.

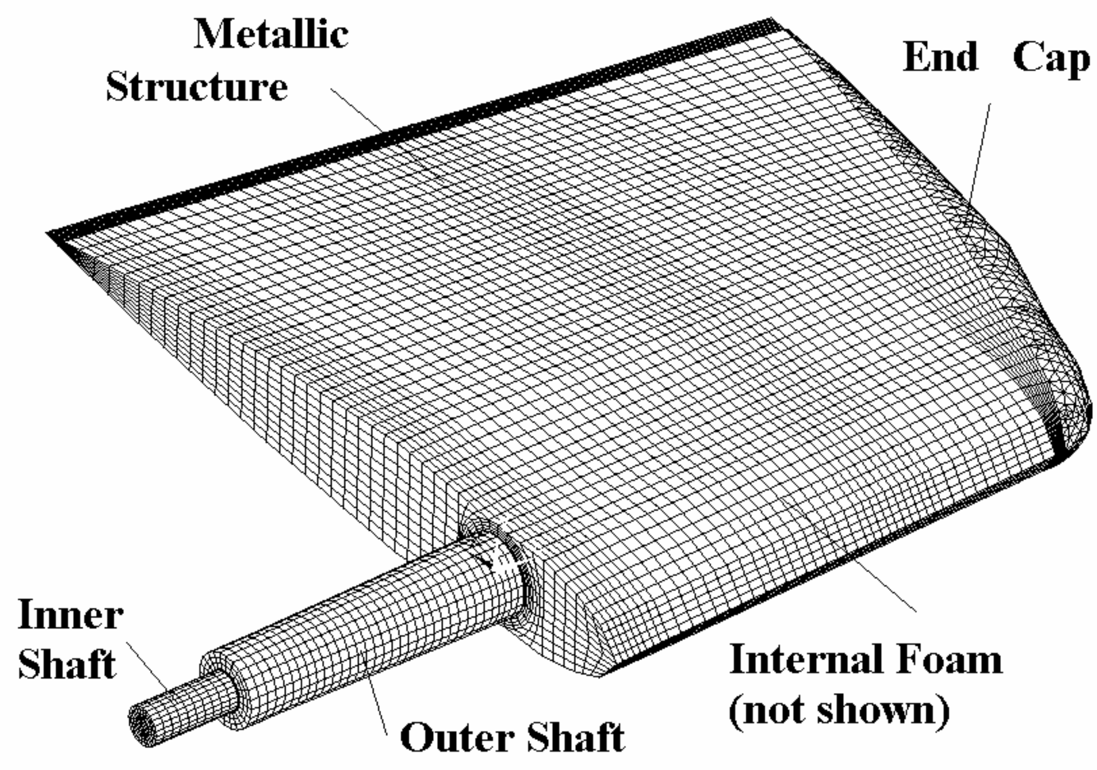


Figure 1. Geometry of the hydrofoil.

## 4. NUMERICAL MODELING

In this paper, the approach adopted for the hydrofoil was to construct a 3D FE mesh of the structure, as shown in Figure 1, using the FE modelling software *ANSYS* [2]. Scenarios in water and vacuum were analysed only in the frequency domain using both FE and BE techniques.

*ANSYS* (using the eigenvalue extraction method) was used to model and assess the modal response of the structure in a vacuum. The *ANSYS* database produced was then imported into the vibro-acoustic modelling software *SYSNOISE* [5] to determine both the modal and vibrational response of the structure (using the modal superposition method) in a fluid.

### 4.1 Model Prediction

A selection of in-vacuum modes is shown in Figures 2 and 3, with the arrows indicating either an in-plane or out-of-plane motion for the structure at that location. The boundary conditions 'free' and 'fixed' refer to the constraint on the motion of the inner shaft, as described previously. Table 2 shows the change in modal frequency under water loading.

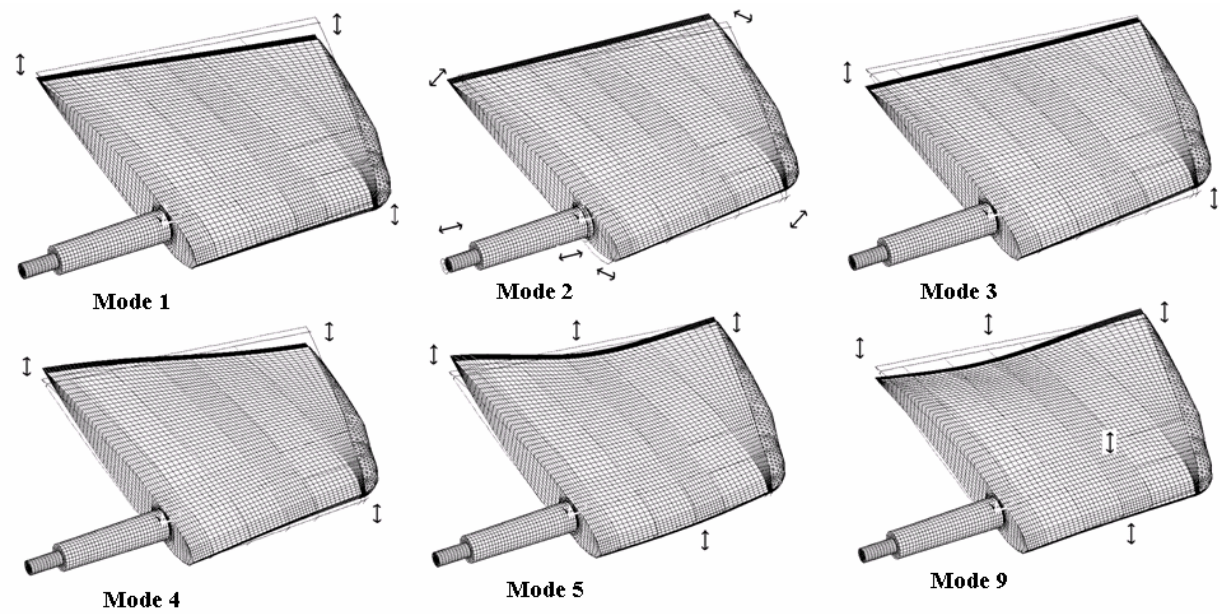


Figure 2. Vacuum mode shapes of the hydrofoil – Free boundary conditions.

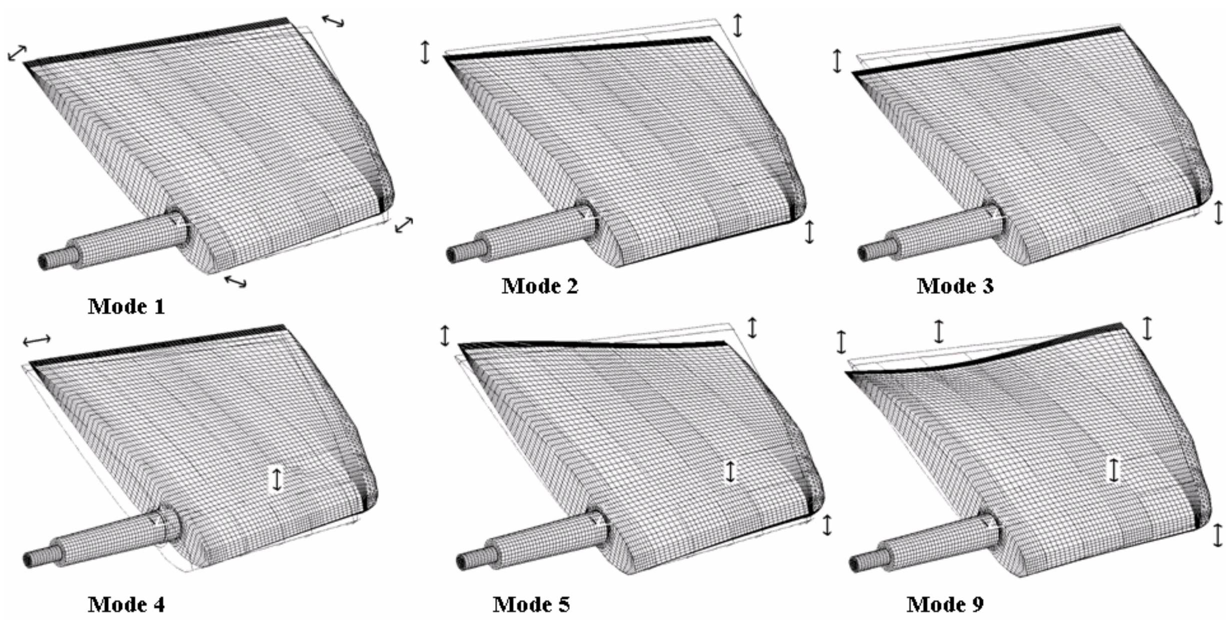


Figure 3. Vacuum mode shapes of the hydrofoil – Fixed boundary conditions.

Table 2: Modal frequencies of the hydrofoil in a vacuum and submerged in water. In-water frequencies are also given as a percentage of the corresponding vacuum frequencies.

| | Free Boundary Condition | | | | Fixed Boundary Condition | | | |
|---|---|---|---|---|---|---|---|---|
| Mode Number Vacuum | Frequency (Hz) Vacuum | Mode Number Water | Frequency (Hz) Water | % | Frequency (Hz) Vacuum | Mode Number Water | Frequency (Hz) Water | % |
| 1 | 15.2 | 1 | 6.4 | 42 | 14.9 | 2 | 14.1 | 93 |
| 2 | 16.3 | 2 | 15.6 | 96 | 15.2 | 1 | 6.4 | 42 |
| 3 | 31.1 | 3 | 15.7 | 50 | 31.1 | 3 | 15.7 | 50 |
| 4 | 65.2 | 4 | 33.0 | 51 | 58.9 | 5 | 54.9 | 93 |
| 5 | 105.0 | 5 | 59.8 | 57 | 65.2 | 4 | 33.0 | 51 |
| 6 | 118.4 | 12 | 118.2 | 99.8 | 105.0 | 6 | 59.8 | 57 |
| 7 | 119.1 | 7 | 70.8 | 59 | 120.3 | 13 | 120.2 | 100 |
| 8 | 133.8 | 14 | 127.8 | 96 | 120.8 | 8 | 70.8 | 59 |
| 9 | 137.7 | 8 | 77.6 | 56 | 137.7 | 9 | 77.6 | 56 |
| 10 | 148.5 | 9 | 87.0 | 59 | 140.9 | 15 | 134.7 | 96 |

## 4.2 Radiated Noise Prediction

*SYSNOISE* was used to compute the pressure, as a function of frequency, produced in the water due to a unit (1 N) excitation on the structure at a point near the trailing edge of the hydrofoil. Figure 4 shows the results at three field point locations, that is, in the three orthogonal directions from the hydrofoil, one perpendicular to (red, solid line), and two in the plane of the structure (blue, short dashes and green, larger dashes). The pressure is typically larger in the direction perpendicular to the plane of the hydrofoil due to the chosen method of excitation and the dipolar character of the radiation from the relatively flat structure. The peaks of the (radiated) pressure in the water indicate when the structure has reached a modal frequency.

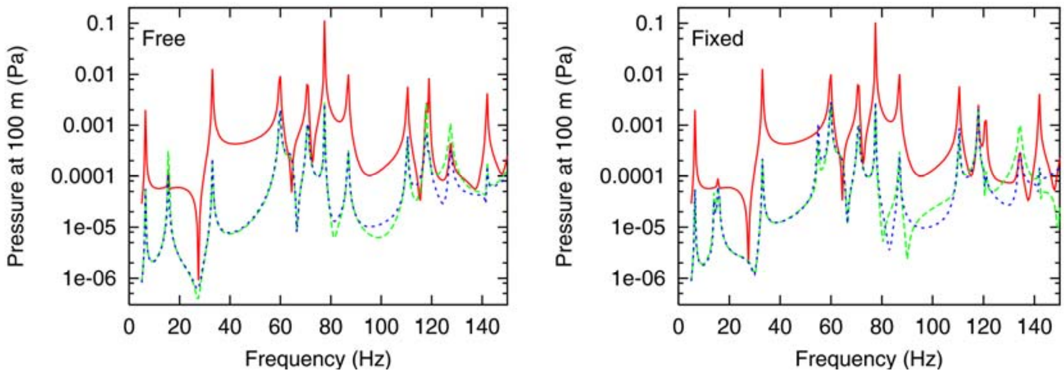


Figure 4. Pressure (Pa) as a function of load excitation frequency (Hz) for a 1N load.

## 5. DISCUSSION

There are a number of approaches possible for obtaining the modal frequencies of a large structure submerged in a fluid. Experimental methods and either or both FE and BE numerical methods are possible. Based upon the complexity of the structure a choice was made to use a combination of FE and BE methods. A fully FE approach was attempted but did not yield acceptable results.

The hydrofoil structure was built using finite elements, and then solved for the modal frequencies in a vacuum using *ANSYS*. Then the BE approach of *SYSNOISE* was used to determine the effect of water loading. The procedure requires the databases (i.e. geometry and eigenvalues) of the FE method to be passed into the BE code for the water-loading processing. This eliminates the need to build a FE model for the fluid. The need to mesh the fluid when using the FE techniques makes the model development time considerably slower than the FE/BE method, and is a claimed advantage of the BE method.

The collection of in-vacuum mode shapes for the hydrofoil is shown in Figures 2 and 3 for both ‘free’ and ‘fixed’ boundary conditions. Similar mode shapes occur for both cases. Mode 2 from the ‘free’ case and mode 1 from the ‘fixed’ case are similar, with mostly in-plane vibration, the bulk of the hydrofoil displaying a pivoting motion. However, there is a difference in that the inner shaft is moving axially inside the outer shaft for the ‘free’ case, whereas there is no motion of the inner shaft for the ‘fixed’ case. Out of plane motion for modes 1, 3, 4 and 9 from the ‘free’ boundary condition are similar to modes 2, 3, 5 and 9 from the ‘fixed’ boundary condition.

There are also non-similar mode shapes resulting from the differences in the boundary conditions. Mode 4, from the ‘fixed’ case does not have an equivalent in the ‘free’ case. Also, the inner-shaft motion of mode 2 in the ‘free’ case does not exist in the ‘fixed’ scenario. A summary of the frequencies for both cases is shown in Table 2. For those modes that are similar between both boundary conditions, similar mode frequencies are calculated.

The reduction in modal frequencies due to the water loading is a function of how much water is being moved by the motion of the mode. If the projected area in the direction of motion is larger then the frequency reduction is greater. Also, at higher mode numbers (not shown in Figures 2 and 3), where there are more sinusoidal cycles of displacement across the hydrofoil, the effect of the loading is less, due to lesser surfacer area of the structure moving relative to lower mode numbers.. The general trend with increasing mode number is therefore less reduction due to water loading.

As examples, in the free boundary condition case of Figure 2, mode 1, the structure is undergoing an out-of-plane bending motion, whereas for mode 3 the structure is pivoting about a diagonal line across the wing and the effect of the water loading is slightly less. In mode 2 there is basically only an in-plane rocking motion and the effect of the water loading is very small. The reduction in the modal frequencies for these modes is greatest for mode 1, at 42% of the vacuum frequency, followed by mode 3, at 50%, and mode 2 at 96%. The modes in Table 2 with small reductions due to water loading (near 100%) involve relatively small areas of the total hydrofoil area with motion into the fluid. This can be due to mostly in-plane motion of the whole hydrofoil, as with mode 2, or to modes involving predominantly small parts of the structure even if the motion is out of plane.

## 6. CONCLUSION

This paper has described some techniques for handling the fluid loading on an underwater structure and it has used as an example a large complex hydrofoil to show the effect and importance of the fluid loading with respect to the structural modal frequencies.

The shape of the structure and the modes are critical in determining the modal frequencies under fluid loading. Reductions in modal frequencies of more than a factor of two were calculated for the hydrofoil considered here. The effect of two boundary condition cases on the hydrofoil inner shaft, ‘free’ and ‘fixed’, showed only a small effect on the magnitude of the modal frequencies; although extra modes were introduced in the ‘free’ case.

The numerical technique adopted to obtain a solution is important, and is determined by a number of factors, like the structural size, the external and internal complexity of the structure, and the resources and time available to do the task. For this fluid-structural problem the frequency domain approach with modal superposition method was adopted using a combination of FE/BE methods to model the structure and fluid.

The FE method alone would generally require much more pre-processing effort than the BE method, simply because a fluid model is required in the FE approach. This has obvious consequences in terms of model size and run time for the FE method. It is the authors’ experience that the implementation and execution of the BE method is a much more attractive and versatile approach in dealing with the fluid loading aspect than the FE method for the modal solution of structures submerged in a fluid.

## REFERENCES

[1] I. R. MacGillivray and J. Marco, “Review of Numerical Techniques for Modal Analysis in Fluid-Structure Interactions”, *INTERNAL DOCUMENT*, DSTO, Australia, 2006.

[2] *ANSYS*, Version 10, ANSYS Inc, Southpointe, 275 Technology Drive, Canonsburg, PA 15317 USA, 2005.

[3] *LS-DYNA*, Version 970, LSTC, 2876 Waverley Way, Livermore, California, USA, 2005.

[4] C. Norwood, “The Measurement Of Natural Frequencies and Mode Shapes of Submerged Cylinders using PVDF Strip Excitation”, *Proceedings of INTERNOISE 95, The 1995 International Congress on Noise Control Engineering,* 10-12 July 1995, pp. 1337-1340.

[5] *SYSNOISE*, Version 5.6, LMS Engineering Innovations, 3001 Leuven, Belgium, 2003.