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FINITE ELEMENT MODELLING OF THE DAMPING OF CYLINDRICAL SHELLS VIBRATING IN CONTACT WITH AN ANNULAR FLUID REGION

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1. INTRODUCTION

Recently published studies on natural vibration of vertical thin-walled cylindrical tanks containing liquid [1] or cylindrical shells in contact with a liquid layer in an inner coaxial annular region [2] dealt with the finite element (FE) modelling of spectral and modal characteristics of such systems. The developed FE models, where only the inviscid fluid was considered, were experimentally verified by means of modal analysis and holographic techniques. The agreement between the computed and measured natural frequencies and mode shapes of vibration was very good in both the cases studied [1, 2], however, no damping properties were modelled by the FE method.

Here is shown how the structural damping and fluid viscosity effects can be included in the FE analysis. It enables one to model the damping characteristics of such hydroelastic systems and to study, for example, the influence of the fluid viscosity on the damping ratio of the system. The damping characteristics are studied here for the identical experimental configuration of the system which was investigated in [2], i.e., for the steel thin-walled cylindrical shell containing a highly viscous oil in an adjacent coaxial (annular) region. The results of the FE analysis are compared with the previously measured data (see Figure 8 in [1]) where in particular the effect of increasing of the oil level in the shell and the effect of decreasing the thickness of the oil layer in the annular region between the outer vibrating shell and an inner rigid coaxial cylinder were investigated.

2. INVESTIGATED CYLINDRICAL SHELL PARTIALLY FILLED WITH OIL AND THE FE MODEL

The steel cylindrical shell had the following parameters: mean radius $R = 0{\cdot}07725$ m, length $L = 0{\cdot}231$ m, wall thickness $h = 1{\cdot}5 \times 10^{-3}$ m, Young modulus $E_s = 2{\cdot}05 \times 10^{11}$ N/m^2, density $\rho_s = 7800$ kg/m^3 and Poisson ratio $\nu = 0{\cdot}3$. The inner coaxial annular region was filled using oil with the following properties: bulk modulus $E_t = 1{\cdot}6 \times 10^9$ N/m^2, density $\rho_t = 933$ kg/m^3 and dynamic viscosity $\mu_0 = 3{\cdot}381$ kg/ms. For creating the fluid annulus inside the shell the solid rigid removable coaxial cylinders of various diameters $2R_d = 80$, 102, 139 or 146 mm, made of a heavy wood, were rigidly bolted to the base plate in the middle of the shell (see Figure 1). This implied that the annular gap width could assume the values $g = 36{\cdot}5$, $25{\cdot}5$, 7 and $3{\cdot}5$ mm. The height of the oil level in the annulus denoted by H was varied from $H = 0$ for the empty shell to $H = L$ for the full shell. The measurement setup and the experimental methods used for evaluation of natural

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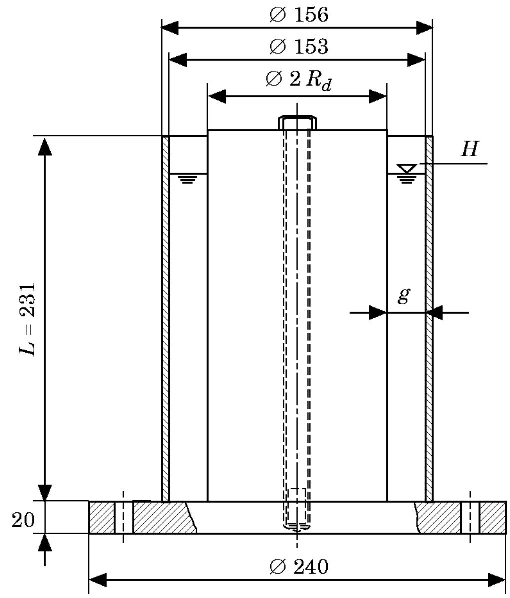

Figure 1. Experimentally studied cylindrical shell with oil in the inner annular gap of width g.

frequencies, damping ratios and modes of vibration were described in detail in references [1, 2].

The ANSYS 5.0 FE code implemented on a SUN-SPARC workstation was used for the calculations. The cylindrical shell was modelled by 8-node shell elements (SHELL 93) and the fluid by 3-D elements (FLUID 80) as shown in Figure 2. The fluid was assumed incompressible and the gravity effects were included in the FE model. However, the sloshing modes of the liquid vibration were not found to be important in the cases studied. Six fluid elements were normally used in the radial direction; however, for the gap widths less than 25·5 mm, only four elements were used. 8 and 24 elements were used in the axial

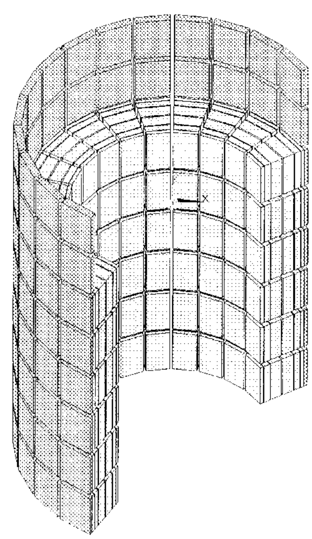

Figure 2. FE model of the shell with the coaxial fluid layer inside.

and circumferential directions, respectively. On the interface surface between the structure and fluid the shell nodal degrees of freedom were coupled to the corresponding fluid ones. At the tank bottom and at the surface of the inner rigid cylinder, all degrees of freedom were fixed.

The eigenvalue problem, given by

$$[\mathbf{K} + \lambda\mathbf{B} + \lambda\mathbf{M}][\mathbf{x}] = [0], \tag{1}$$

where $\mathbf{K}$, $\mathbf{B}$ and $\mathbf{M}$ are the global stiffness, damping and mass matrices, respectively, was solved using the ANSYS asymmetric solver based on the Lanczos algorithm.

The damping of the shell/liquid system is caused by the material damping, by a leakage of the vibration energy through the fixed tank bottom, by the existence of viscous fluid forces and acoustic radiation from the vibrating surface of the shell. However, ANSYS offers only limited possibilities for damping modelling as follows

$$\mathbf{B} = \alpha\mathbf{M} + \beta\mathbf{K} + \beta_{80}\mathbf{K}_{80} + \beta_{93}\mathbf{K}_{93} + \sum_{k=1}^{NEL} \mathbf{B}_k(\mu), \tag{2}$$

where α, β are coefficients of global proportional damping; β_{80}, β_{93} are constant multipliers based on material properties of oil and steel for the fluid ($\mathbf{K}_{80}$) and shell ($\mathbf{K}_{93}$) portions of the overall stiffness matrix, respectively; $\mathbf{B}_k$ are the fluid element damping matrices developed by specifying fluid viscosity μ, and NEL is the number of fluid elements.

The objective of the modelling of the damping characteristics of the shell/liquid system was to find a set of the parameters: α, β, β_{80}, β_{93}, μ, and NEL for fitting the experimental data which were obtained for a particular mode of vibration for all investigated filling ratios (H/L) and gaps (g), including the case of the empty shell. The modelling was predominantly focused on the following damping models:

1) *mass* model of the proportional damping:

$$\mathbf{B} = \alpha\mathbf{M}, \tag{3}$$

where $\mathbf{M}$ is the global matrix of the complete shell/liquid system;

2) *physical* model of the damping:

$$\mathbf{B} = \beta_{93}\mathbf{K}_{93} + \sum_{k=1}^{NEL} \mathbf{B}_k(\mu) \tag{4}$$

where the damping of the empty shell and the fluid viscosity are taken into account;

3) and *full damping* model:

$$\mathbf{B} = \beta_{80}\mathbf{K}_{80} + \beta_{93}\mathbf{K}_{93} + \sum_{k=1}^{NEL} \mathbf{B}_k(\mu), \tag{5}$$

where an additional damping of the fluid elements is included in the model. The values of β_{93} and β_{80} for particular mode of vibration were deduced from the measured damping ratios of the empty ($H/L = 0$) and full ($H/L = 1$) shell, respectively.

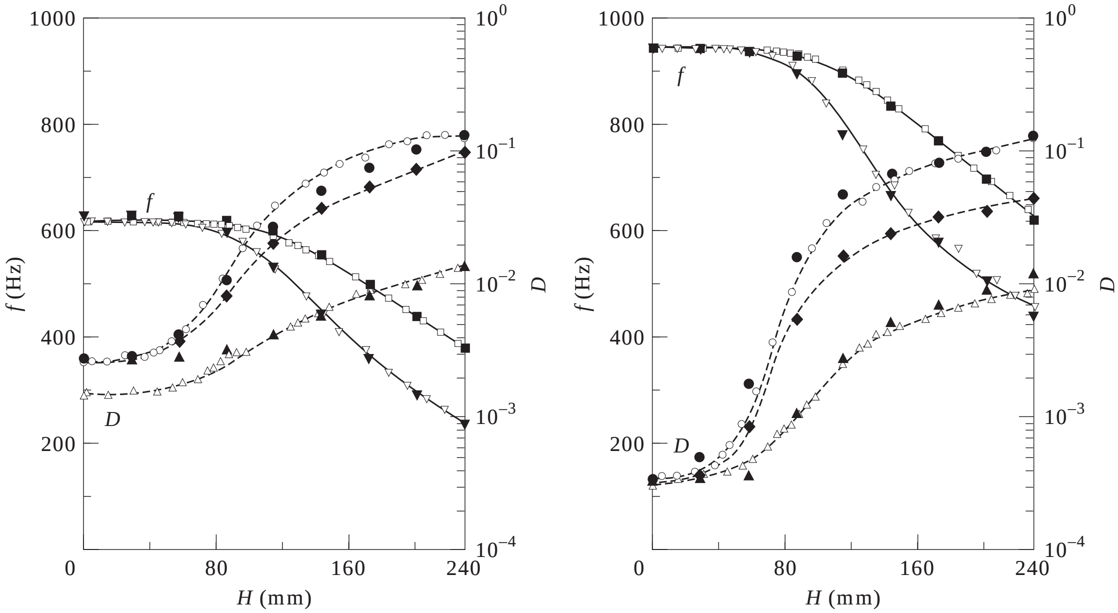

Figure 3. Natural frequencies ——, f; and the damping ratios – – – –, D; as functions of the oil level H in the adjacent annular gap of width: $g = 7$ mm and $g = 25{\cdot}5$ mm. (a) For the mode of vibration; $n = 3$, $m = 1$; (b) for the mode: $n = 4$, $m = 1$. Values of g (mm) for: measured frequencies; □, 25·5; ▽, 7; measured damping ratios: △, 25·5; ○, 7; calculated frequencies: ■, 25·5; ▼, 7; calculated damping ratios: ▲, *full damping* model 25·5; ●, *full damping* model and ◆, *physical* model 7.

3. RESULTS OF THE MODAL ANALYSIS AND DISCUSSION

The calculated natural frequencies f and damping ratios $D = |\mathrm{Re}\,\lambda/\mathrm{Im}\,\lambda|$ for increasing oil level from $H = 0$ to $H = L$ in the annulus are compared with the experimental data for the first axial ($m = 1$) and two circumferential ($n = 3, 4$) modes of vibration in Figure 3 for two different values of the annular gap width $g = 7$ mm and $g = 25{\cdot}5$ mm. The agreement between the calculated and measured natural frequencies is very good and the error for all oil levels H is less than 2·2% for $n = 3$ and less than 3·8% for $n = 4$. It is comparable with the previous results obtained for the inviscid FE model of the fluid [2]. The corresponding errors for damping are greater, because the real part of the eigenvalue λ is much lower than the imaginary part and the accuracy of measurement and calculation is worse than for the above mentioned natural frequencies. Nevertheless, for the so called *full damping* model ($\mu = \mu_0$; $\beta_{93} = 1{\cdot}41 \times 10^{-6}$, $\beta_{80} = 0{\cdot}26$ for $n = 3$; $\beta_{93} = 1{\cdot}12 \times 10^{-7}$, $\beta_{80} = 1{\cdot}29$ for $n = 4$; $\alpha = \beta = 0$) the agreement between the theory and experiment is good for all oil levels H. If the so called *physical* model (the *full damping* model except that $\beta_{80} = 0$) or a simple *mass* model of proportional damping ($\mu = \beta = \beta_{80} = \beta_{93} = 0$; $\alpha > 0$—not shown in Figure 3) were used the results were erroneous. The apparent differences in the damping ratio D for the empty shell were probably caused by slightly different installation of the measured system when various inner rigid cylinders were mounted inside the shell, because very low damping of metal structures is highly sensitive to the quality of clamping (to the boundary conditions).

4. CONCLUSIONS

Frequency modal and particularly damping characteristics of a cylindrical shell vibrating in contact with an annular coaxial region of a viscous liquid were studied in parallel by the FE method and by experimental modal analysis. The contribution is focused on the FE modelling of the damping characteristics of the cylindrical structure vibrating in a fluid when the viscous fluid forces are important, for example, for a viscous liquid in narrow gaps.

The analysis of the changes of the damping ratio when increasing the oil level in the annular gap or diminishing the thickness of the oil layer inside the cylindrical shell showed that the best results were obtained when three sources of the damping for each mode of vibration were simultaneously introduced into the FE model as follows: 1) the damping of the shell elements which characterizes the damping of the empty shell, 2) the viscosity of the fluid, and 3) an artificial additional damping of the fluid elements. The achieved agreement between the measured and calculated dynamic characteristics of the system was very good as a result, including the investigated damping ratio for each mode of vibration. When the last mentioned item was neglected, i.e., when the so called physical model of the damping was used, the damping of the system was remarkably underestimated by the FE model.

The FE analysis proved the conclusion found experimentally, that a thin spacely confined layer of a highly viscous liquid on the vibrating surface of the cylindrical shell can be a very efficient damper of structural vibrations.

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