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NUMERICAL SIMULATION OF OSCILLATIONS OF A PLATE IN RESTING FLUID

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Abstract: This paper presents a numerical solution of the problem of oscillations of a flexible plate placed in a viscous incompressible fluid done with ANSYS. In constructing a finite element model of the plate, both volumetric and shell finite elements, which are more commonly used in engineering applications, are considered. By way of example it is shown that it is possible to transition from a volumetric mesh to a shell mesh without loss of accuracy in solving this problem and similar problems. The considered coupled approach to solving the problem for the shell plate mesh has an important practical application in solving real-world problems of aeroelasticity, because in engineering practice it is much more convenient to represent structures and constructions in the form of beam and shell models. The solution of this problem is of particular importance for the verification of techniques of numerical modelling of coupled aeroelasticity problems.

Keywords: numerical simulation, coupled problem, FSI, contact, numerical damping

ЧИСЛЕННОЕ МОДЕЛИРОВАНИЕ ЗАДАЧИ О СВОБОДНЫХ КОЛЕБАНИЯХ ГИБКОЙ ПЛАСТИНЫ, ПОМЕЩЕННОЙ В ВЯЗКУЮ НЕСЖИМАЕМУЮ ЖИДКОСТЬ

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Аннотация: В настоящей работе представлено численное решение задачи о свободных колебаниях гибкой пластины, помещенной в вязкую несжимаемую жидкость, с использованием программного комплекса ANSYS. При построении конечноэлементной модели пластины рассматривались как объемные, так и чаще применяемые в инженерных приложениях оболочечные типы конечных элементов. На примере показана возможность перехода от объемной модели к оболочечной без потери точности при решении подобных задач. Рассмотренный подход при решении задачи в связанной постановке для оболочечной модели пластины имеет практически важное применение к решению реальных задач аэроупругости, так как в инженерной практике конструкции и сооружения существенно удобнее представлять в виде оболочечно-стержневых моделей. Решение данной задачи имеет отдельную значимость для верификации методик численного решения связанных задач аэроупругости.

Ключевые слова: численное моделирование, связанная задача, FSI, контактное взаимодействие, численное демпфирование

1. INTRODUCTION

The interaction of structures with fluid or gas plays an important role in many construction problems. Besides suspension bridges, aeroelasticity has to be considered in wind impact analysis for structures of high-rise buildings, towers, oil platforms, and lightweight membrane coverings used in large-span structures.

The fluid-structure interaction analysis has been implemented in ANSYS since version 10.0. The CFD part can be solved with either ANSYS CFX or ANSYS Fluent, and ANSYS

Mechanical is used for stress-strain analysis. The coupling between codes is configured in ANSYS Workbench using System Coupling module. This tool is convenient for solving coupled aeroelasticity problems, but there are some limitations. At the contact between fluid and structure an interface is specified to transfer data from one module to another. In order to set such an interface, both the geometry must match and a shared surface between fluid and structure must exist. Therefore, Solid type volumetric finite elements are commonly used for aeroelasticity problems in ANSYS. However, in practical applications, most building structures are calculated using beam and shell models. Such models have smaller dimensionality and are considerably easier to create than volumetric models. For these reasons, a technique needs to be developed and verified that will allow solving the aeroelasticity problems with any type of finite elements in ANSYS.

The problem of oscillations of a plate referenced in [1] was chosen for the verification. In that paper, the problem was solved using the authors' own code. The problem is also presented in [2-4]. In [2-3] it is solved with authors' own code, too, and in [4] it is solved with ANSYS using volumetric finite elements.

In this paper, the problem is solved with ANSYS Workbench (Mechanical + Fluent).

2. PROBLEM STATEMENT

A thin flexible plate of length $\boldsymbol{L}$ = 1.0 m and width $\boldsymbol{W}$ = 0.4 m with the fixed bottom edge is considered [1]. A load in form of evenly distributed pressure $\boldsymbol{p}$ = 75 Pa is applied on the plate for 0.5 s, then removed in 0.01 s so that the plate starts to vibrate (Fig. 1).

The thickness of flexible plate $\boldsymbol{d_s}$ = 0.06 m, modulus of elasticity $\boldsymbol{E}$ = 2.5 MPa, Poisson's ratio $\boldsymbol{\nu}$ = 0.35, material density $\boldsymbol{\rho_s}$ = 2550 kg/m^3. The density of the fluid is assumed to be ρ $\boldsymbol{\rho_f}$ = 1 kg/m^3. In [1] 3 values of dynamic viscosity were considered: $\boldsymbol{\mu_{f,1}}$ = 0.2 Pa·s, $\boldsymbol{\mu_{f,2}}$ = 1.0 Pa·s, $\boldsymbol{\mu_{f,3}}$ = 5.0 Pa·s. In this paper, only one value of the dynamic viscosity $\boldsymbol{\mu_{f,1}}$ will be considered.

The computational domain is 20.06 × 6 × 0.4 m.

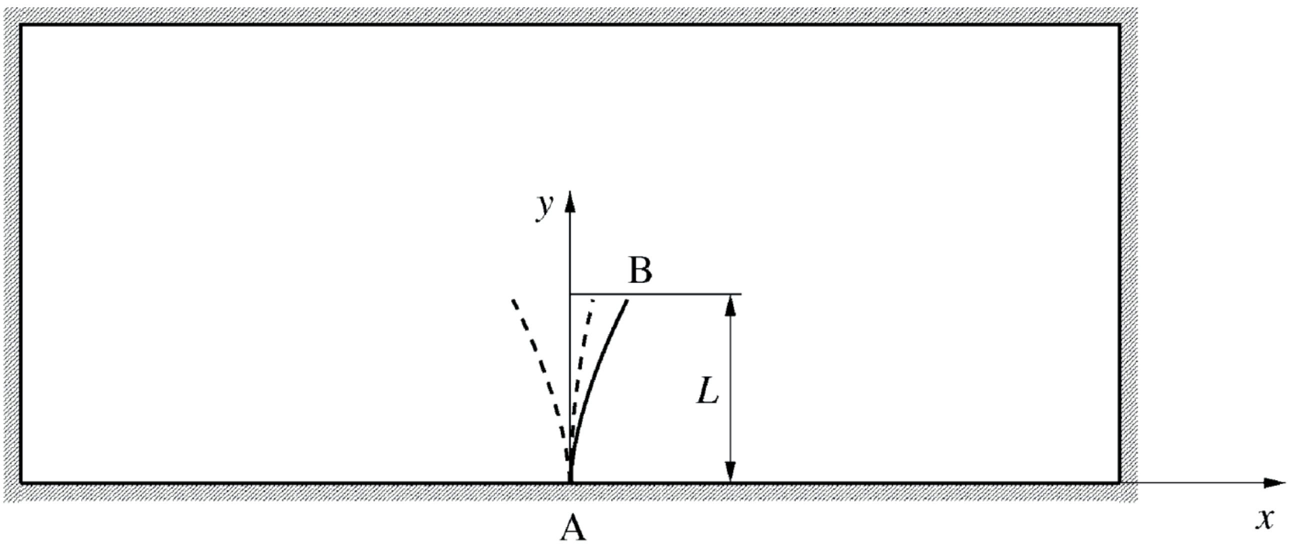


Figure 1. Oscillations of a plate problem geometry

3. COMPUTATIONAL MODELS

3.1. Computational models of the flexible plate

In this example, a volumetric and a shell type of the plate geometry will be considered (Fig. 2).

For the volumetric geometry, 3 types of grids will be considered to evaluate mesh convergence and select the optimum mesh. For the shell geometry, 1 type will be considered (Table 1, Fig. 3).

Table 1. Finite-element meshes parameters

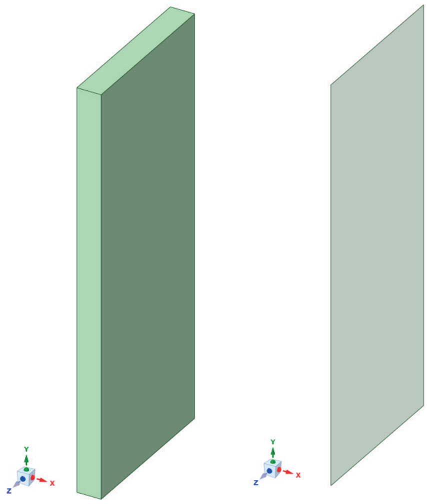
	Nodes	Elements	FE type
Model 1	15 777	3 000	SOLID186
Model 2	826 665	192 000	SOLID186
Model 3	7 353	1 000	SOLID186
Model 4	1 071	1 000	SHELL181

3.2. Computational model of viscous incompressible fluid

The computational domain has been partitioned into a structured computational mesh with inflation on the walls (Fig. 4).

Since the Reynolds number for the problem is $Re \ll 100$, the flow is laminar, and turbulence models are not used in the calculations.

Due to weakly pronounced three-dimensional effects occurring along the plate ends, the problem was solved in the quasi-dimensional formulation (one FE in the $\boldsymbol{Z}$ direction), as in [1].



Volumetric type *Shell type*

Figure 2. Geometry types of the plate

Boundary conditions

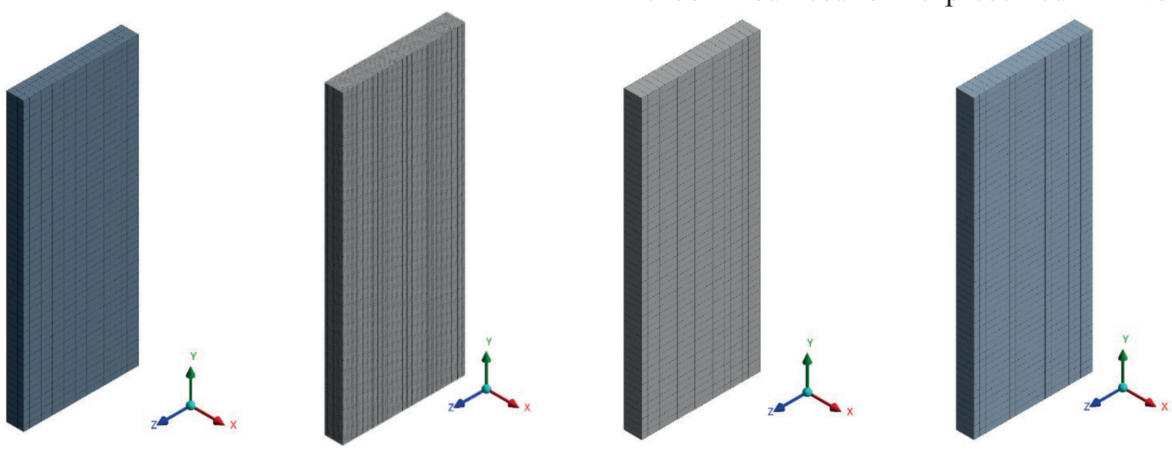
On the side faces, parallel to the plane $\boldsymbol{xy}$, the *Symmetry* conditions were set. The *No-Slip Wall* condition ($\boldsymbol{U}=\boldsymbol{V}=\boldsymbol{W}=0$ m/s) was specified for the rest of the faces.

Initial conditions

Zero velocity ($\boldsymbol{U}=\boldsymbol{V}=\boldsymbol{W}=0$ m/s) and zero relative pressure were set as initial conditions.

4. RESULTS OF MODAL ANALYSIS

The first 6 eigenfrequencies and eigenmodes were determined for all computational models. The obtained results are presented in Table 2.



Model 1 *Model 2* *Model 3* *Model 4*

Figure 3. Finite-element meshes of the flexible plate

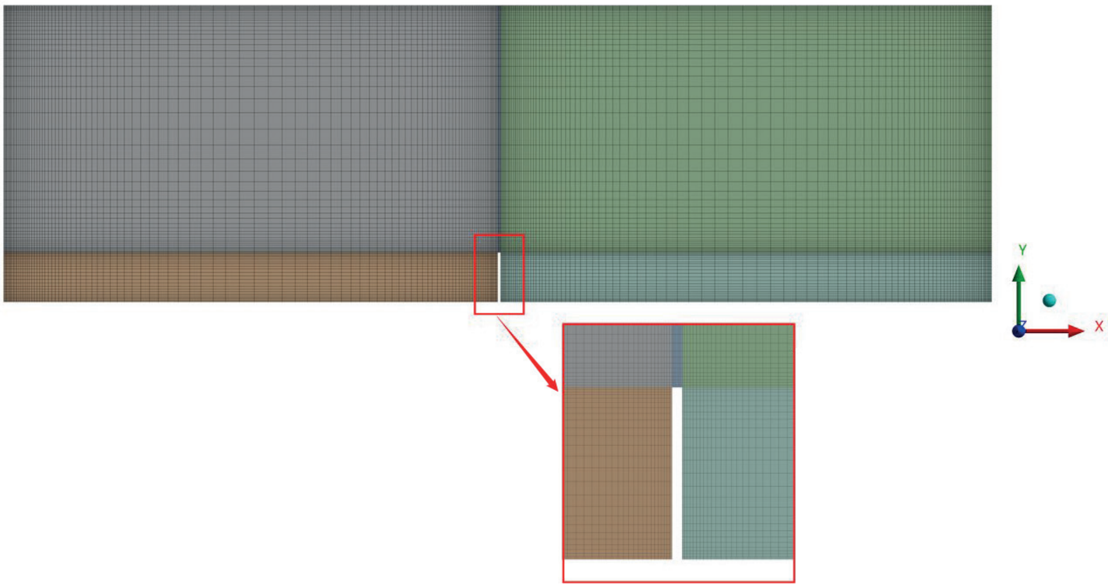


Figure 4. Finite-volume mesh of the viscous incompressible fluid (32 632 cells)

Table 2. Computed eigenfrequencies (Hz) for all models in comparison with the reference (Ref. [1])

Mode of vibration	*Ref. []*	*Model 1*		*Model 2*		*Model 3*		*Model 4*	
		f_i, Hz	$\varepsilon(f_i)$, %	f_i, Hz	$\varepsilon(f_i)$, %	f_i, Hz	$\varepsilon(f_i)$, %	f_i, Hz	$\varepsilon(f_i)$, %
1. Bending (*xy*)	0.31	0.3108	0.26	0.3106	0.19	0.3112	0.39	0.31	0.01
2. Torsional	-	1.4947	-	1.4939	-	1.5035	-	1.4952	-
3. Bending (*yz*)	-	1.8243	-	1.8233	-	1.8242	-	1.8191	-
4. Bending (*xy*)	1.85	1.9071	3.09	1.9056	3.01	1.9124	3.37	1.9032	2.28
5. Torsional	-	4.7159	-	4.7131	-	4.7493	-	4.7179	-
6. Bending (*xy*)	4.93	5.2162	5.81	5.2116	5.71	5.2430	6.35	5.2101	5.68

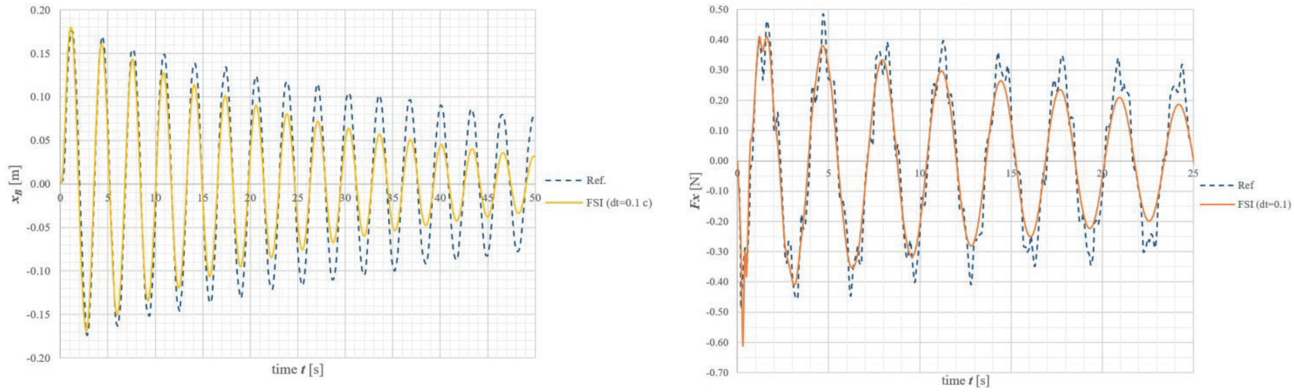
As can be seen from Table 2, the discrepancy between current results and [1] does not exceed 6.35% (*Model 3*). For further related calculations using the volumetric plate model, *Model 1* will be used since the error between this model and *Model 2* is at most 0.1%, with *Model 1* containing less nodes.

5. SOLVING A PROBLEM IN A COUPLED FORMULATION

To ensure convergence and stability of solution at each timestep, it is necessary to specify the maximum number of iterations at each timestep and the convergence criterion for loads and displacements. For this problem, 5 maximum iterations at each coupled FSI timestep were set and the convergence criterion for the loads and displacements was set to 10^{-3}. The equations of motion of the plate are solved with the Newmark method, and the total physical time is 50 s.

5.1. Coupled solution using a volumetric plate model

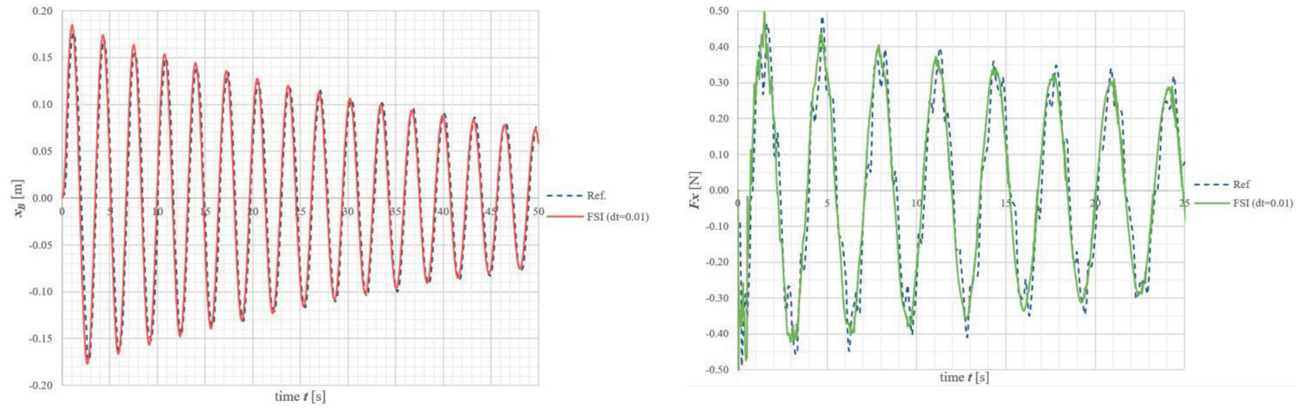
Results of the coupled analysis in comparison with the results of [1] are shown in figures 5 to 6. The timesteps are $\Delta t_1 = 0.1$s and $\Delta t_2 = 0.01$s.



Displacement of the free edge of the oscillating plate in x direction [m]

Drag force F_x [N]

Figure 5. Comparison of the simulation results (Δt_1 =0.1 s) with the reference [1]



Displacement of the free edge of the oscillating plate in x direction [m]

Drag force F_x [N]

Figure 6. Comparison of the simulation results (Δt_2 =0.01 c) with the reference [1]

In the ANSYS Workbench Transient Structural module a numerical damping value is utilized to stabilize the scheme of numerical integration of equations of motion by damping high-frequency modes [5]. By default, it equals 0.1.
For the Newmark's method, if the parameters α, δ satisfy the conditions:

$$\delta \geq \frac{1}{2} \tag{1}$$

$$\alpha \geq \frac{1}{4}\left(\frac{1}{2} + \delta\right)^2 \tag{2}$$

then it can be unconditionally stable [5]. If the above conditions are not satisfied, the additional amplitude damping ratio γ is introduced into the equations:

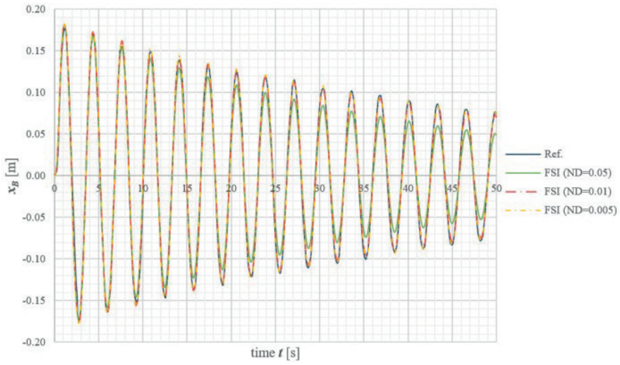
$$\delta = \frac{1}{2} + \gamma \tag{3}$$

$$\alpha = \frac{1}{4}(1 + \gamma)^2 \tag{4}$$

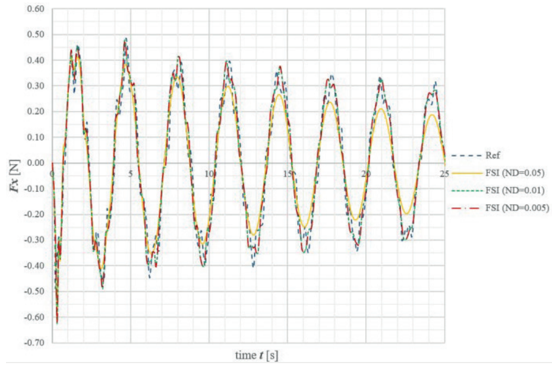
$$\gamma \geq 0 \tag{5}$$

While the amplitude damping ratio γ allows to make the integration procedure stable, it affects the resulting solution.
The results of the calculations are presented below. It can be seen that for Δt_1 =0.1 c a rapid damping of the plate vibration is obtained, but the reduction of the timestep (Δt_2 =0.01 c) gets rid of it.
Since reducing the timestep increases the computation time, it is necessary to minimize

the effect of numerical damping. In order to achieve that, calculations at Δt_1 =0.1 with different numerical damping values (*ND1* = 0.05, *ND2* = 0.01, *ND3* = 0.005) were performed. The results of the coupled analysis in comparison with the results of [1] are shown in Fig 7. As it can be seen, reducing the numerical damping value allows to approach the solution referenced in [1].



Displacement of the free edge of the oscillating plate in x direction [m]



Drag force F_x [N]

Figure 7. Comparison of the simulation results at Δt_1 =0.1 s for different numerical damping values (ND_1=0.05, ND_2=0.01, ND_3=0.005) with the reference [1]

5.2. Coupled solution using a shell plate model

The next step in solving this problem was the transition from the volumetric plate model to the shell model. However, a problem arises in the assignment of the interface between the fluid and the plate due to some limitations: as pointed out before, the geometries where contact occurs must coincide. To circumvent this, a dummy surface was created at the fluid-structure boundary (Fig. 8), which served as an interface to transfer data from one module to another.

In order for the flexible plate and the dummy surface to work together, a Bonded type contact was established between them, in which no separation or sliding between the faces was allowed. This type of contact is linear and is calculated in one iteration. Multi-point constraints were used for the interface which is a form of Lagrange multipliers method. The algorithms used did not significantly increase the number of iterations and the counting time. *Model 4* was chosen for the plate.

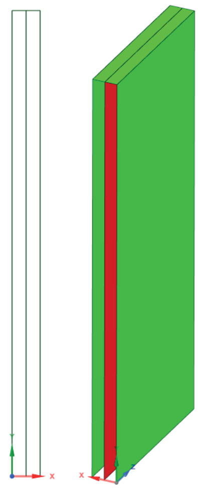


Figure 8. Geometry of the flexible plate (red) with the dummy surfaces (green) acting as interfaces to the fluid domain

The dummy surface was modelled with a *Shell181* FE. Three models of a flexible plate with the dummy surface were considered with differing thickness:

- *Model A*: 1 mm;
- *Model B*: 0.1 mm;
- *Model C*: 0.01 mm.

The material of the dummy surface corresponds to that of the flexible plate.

In order to ensure similar behaviour of the flexible plate with and without the dummy surface, a modal analysis was performed for *Models A, B, C.*

Results of the modal analysis

The results of calculations versus the results of modal analysis for *Model 4* are presented in Table 3 and Figs. 9-14.

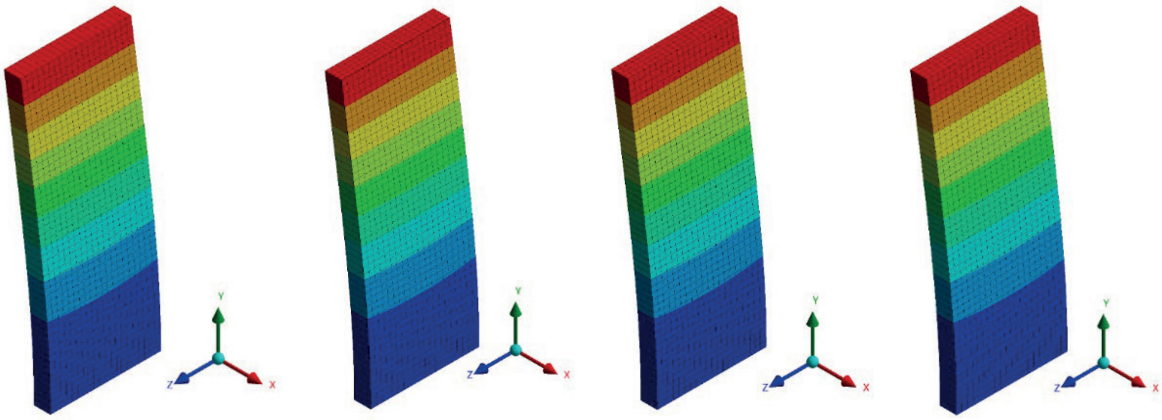
It can be seen from the results that *Model A* shows a different behaviour from *Model 4*. A significant difference is seen not only in the eigenfrequencies (up to 21.47% for mode 2), but also in the eigenmodes (2 and 3). For *Model B* and *Model C* the eigenmodes correspond to *Model 4*, and the maximum error in the eigenfrequencies is observed in the mode 2 (3.14% for *Model B*, 0.33% for *Model C*). Therefore, *Model B* and *Model C* have been chosen for the coupled calculations.

Fig. 15 shows the results of the coupled analysis compared to the results of [1] for *Model B*. The time step size is $\Delta t = 0.1$ s and the numerical damping coefficient is 0.005.

No results were obtained for *Model C* due to the strong distortion of the dummy surface grid caused by the small thickness of the shell.

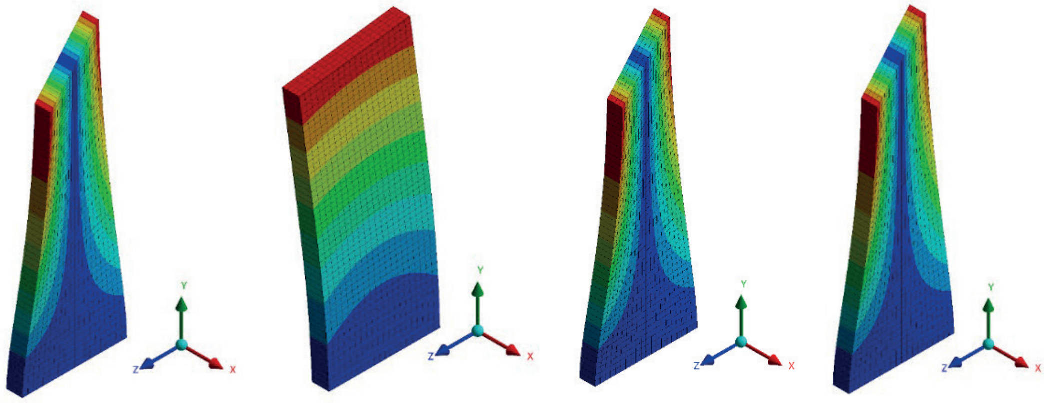
Table 3. Computed eigenfrequencies (Hz) for all models

Mode of vibration	*Model 4*	*Model A*		*Model B*		*Model C*	
		f_i, Hz	$\varepsilon(f_i)$, %	f_i, Hz	$\varepsilon(f_i)$, %	f_i, Hz	$\varepsilon(f_i)$, %
1	0.31	0.3049	1.65	0.3095	0.16	0.31	0
2	1.495	1.816	21.47	1.542	3.14	1.5	0.33
3	1.819	1.8482	1.61	1.8188	0.01	1.8191	0.01
4	1.903	1.8701	1.73	1.8997	0.17	1.9029	0.01
5	4.718	4.8284	2.34	4.7298	0.25	4.7191	0.02
6	5.210	5.1205	1.72	5.2009	0.17	5.2092	0.02



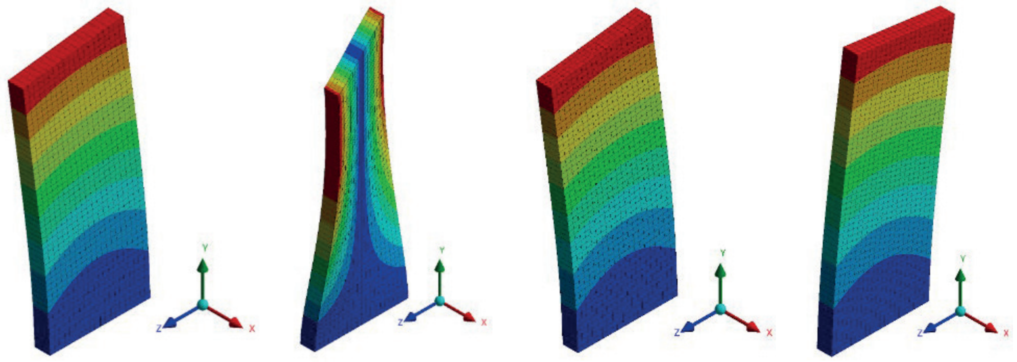
Model 4 (f_1 = 0.31 Hz) *Model A* (f_1 = 0.3049 Hz) *Model B* (f_1 = 0.3095 Hz) *Model C* (f_1 = 0.31 Hz)

Figure 9. Eigenmodes and eigenfrequencies (mode 1)



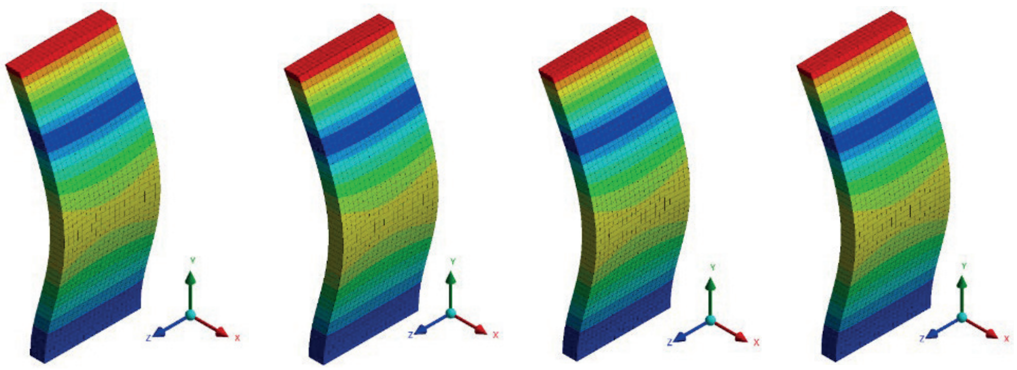
Model 4 (f_2 = 1.4952 Hz) *Model A* (f_2 = 1.816 Hz) *Model B* (f_2 = 1.542 Hz) *Model C* (f_2 = 1.5 Hz)

Figure 10. Eigenmodes and eigenfrequencies (mode 2)



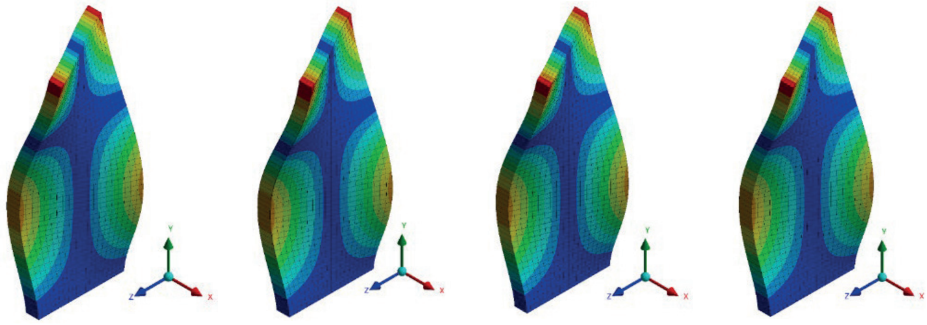
Model 4 (f_3 = 1.8191 Hz) *Model A* (f_3 = 1.8482 Hz) *Model B* (f_3 = 1.8188 Hz) *Model C* (f_3 = 1.8191 Hz)

Figure 11. Eigenmodes and eigenfrequencies (mode 3)



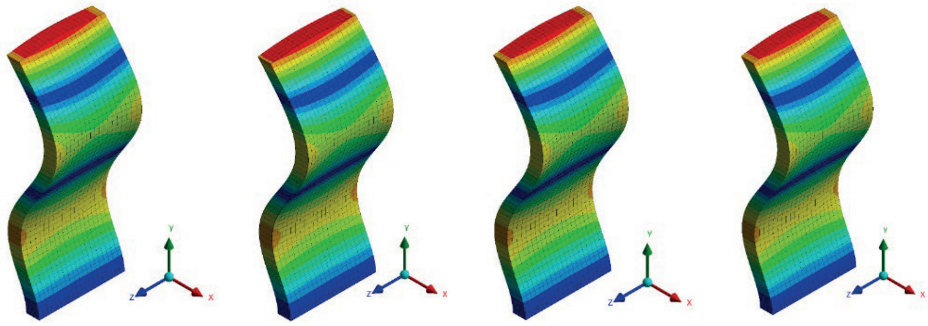
Model 4 (f_4 = 1.9032 Hz) *Model A* (f_4 = 1.8701 Hz) *Model B* (f_4 = 1.8997 Hz) *Model C* (f_4 = 1.9029 Hz)

Figure 12. Eigenmodes and eigenfrequencies (mode 4)



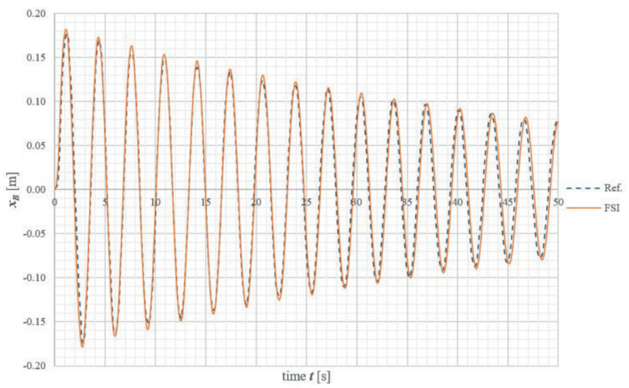
Model 4 (f_5 = 4.7179 Hz) Model A (f_5 = 4.8284 Hz) Model B (f_5 = 4.7298 Hz) Model C (f_5 = 4.7191 Hz)

Figure 13. Eigenmodes and eigenfrequencies (mode 5)

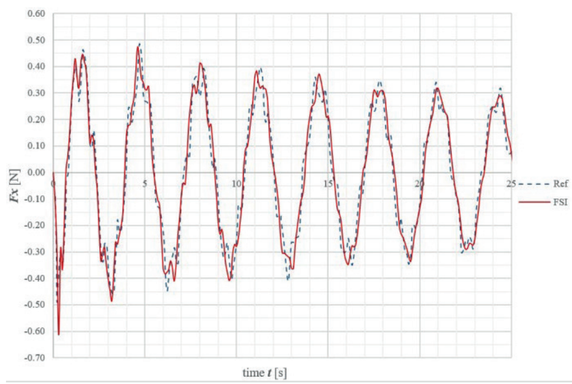


Model 4 (f_6 = 5.2101 Hz) Model A (f_6 = 5.1205 Hz) Model B (f_6 = 5.2009 Hz) Model C (f_6 = 5.2092 Hz)

Figure 14. Eigenmodes and eigenfrequencies (mode 6)



Displacement of the free edge of the oscillating plate in x direction [m]



Drag force F_x [N]

Figure 15. Model B: Comparison of the simulation results with the reference [1]

6. CONCLUSION

The considered problem is included in the development of technique for estimating the aeroelastic instability of building structures ([6-7]) as a verification problem. This example shows the possibility of transition from a volumetric model to a shell model. The results obtained for both types of the flexible plate models showed good agreement with the reference. The considered coupled approach, which utilizes a shell model of the plate, has practically important application to the real-world problems of aeroelasticity, since in engineering practice the building structures are most often modeled by beam and shell models.

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