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Hydraulic coupling of fuel assemblies under axial flow, confinement effect

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ABSTRACT

Nuclear industry needs tools to design reactor cores in case of earthquake. Simulation using a fluid-structure model of the core subjected to a seismic excitation is used to make a parametric study in order to observe the effect of confinement on the coupling between fuel assemblies. The modelling used to make the parametric study showed good agreement with experimental results. Simulations showed that the added stiffness should be negligible in a real core configuration, and that damping identified on one fuel assembly gives underestimated values.

1. Introduction

Safety measures are required to insure the drop of control rods and that the core is cooled when the fuel assembly spacer grids strike each other during seismic excitation of a Pressurized Water Reactors (PWR). A way to insure these two criteria is to prevent the spacer grids from buckling. Engineers need special tools for designing and maintaining reactor cores. The tools usually used involve structural modelling accounting for mass and damping coefficients induced by fluid-structure interactions. These coefficients are identified on full scale test results.

The reactor core made of fuel assemblies is subjected to an axial water flow to cool the reactor. The flow strongly modifies the dynamical behaviour of the fuel assemblies (Tanaka et al., 1988; Hotta et al., 1990; Collard et al., 2004), therefore the identification of the fluid forces is important to provide a relevant modelling of the fuel assemblies behaviour. The first approximation of the fluid forces is to consider them as added mass and damping (Rigaudeau, 1997; Viallet et al., 2003). A more complex expression of these fluid forces is given by Païdoussis (2003) in which the velocity and the relative direction of the flow with respect to the fuel assembly are accounted for. Ricciardi et al. (2009a,b) proposed a porous media approach based on the Païdoussis theory.

In previous study (Ricciardi and Boccaccio, 2012), tests dedicated to the identification of the fluid forces acting on a full scale fuel assembly were performed. These tests highlighted an added stiffness effect under axial flow. This phenomenon was first observed in Ricciardi et al. (2010) and then discussed in Ricciardi and Boccaccio (2012). Identified coefficients of stiffness, damping and mass showed a strong dependency on the lateral by-passes. These by-passes are necessary to allow the displacement of the fuel assembly. Further tests involving fluid measurements in the by-passes (Ricciardi and Boccaccio, 2014b) showed that fluid velocity fluctuations were induced by the fuel assembly displacement. A delay was observed between the displacement and the fluid fluctuations. A first attempt to model the added stiffness was made by Ricciardi and Boccaccio (2015) based on Bernoulli equation and an artificial delay, a second one accounting for the flow in the by-pass was proposed (Ricciardi, 2016) and showed good result. In these previous studies simulation were compared to experiment involving by-passes significantly larger than the distance between two fuel assemblies in a PWR core. Thus, one could wonder if the observations made on a fuel assembly with large by-passes are representative of core geometry. In this paper simulations accounting for one and three fuel assemblies are performed for various values of by-passes to asses this question. Moreover the confinement effect is validated by comparison with a few experimental data.

The paper is organized as follows: in the first section the model used to make the simulations will be recalled and adapted to the 3 fuel assemblies configuration, in the second section the experiment used for validation will be presented, in the third part the methodology to make the parametric study is described, then the 3 fuel assemblies configuration is compared to the 1 fuel assembly configuration, and the last section will be devoted to the coupling between fuel assemblies in the 3 fuel assemblies configuration.

2. Coupled fluid-structure modelling

In this section the model developed in Ricciardi (2016) is recalled and adapted to the 3 fuel assemblies configuration. The modelling proposed by Ricciardi et al. (2009a,b) is based on a porous medium approach. This approach gives access to an equivalent fluid model and an equivalent structure model both defined on the whole domain. Motion equations for the equivalent fluid and the equivalent structure

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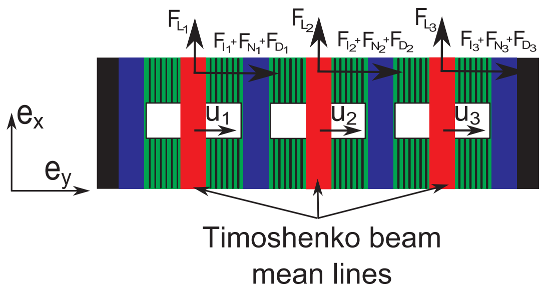

Fig. 1. Notation of beam model displacements with fluid forces on the 3 fuel assemblies configuration.

are first established separately. For the fluid part, global fluid flow equations through the rod bundle are obtained by local spatially averaging the Navier Stokes equations written with an Arbitrary Lagrangian Eulerian approach. The resulting equivalent fluid is characterized by an equivalent velocity and an equivalent pressure both defined in the whole domain. Structure related effects on fluid are accounted for by a body force also defined in the whole domain. The structure equations are also space averaged, each fuel assembly is modelled as an equivalent structure satisfying a Timoshenko beam model with a nonlinear behaviour. For each fuel assembly, the unknowns are reduced to the displacement of the mean line u_i and the rotation of the cross section θ_i of the ith fuel assembly (Fig. 1).

Fluid related effects on structure are accounted for by a body force which is defined in the whole domain. Finally, fluid related effects on structure and structure related effects on fluid are of opposite sign and are built from expression of fluid forces acting on a rod subjected to an axial flow proposed by Païdoussis (2003).

The equations of the structure motion are:

$$\forall\ i \in [1,2,3],\quad m_{fa}\frac{\partial^2 u_i}{\partial t^2} = \frac{\partial T_i}{\partial x} + N_0\frac{\partial^2 u_i}{\partial x^2} + \frac{S_{eq}}{d_g^2}\left(F_{I_i} + F_{N_i} + F_{D_i} - xF_{L_i}\frac{\partial^2 u_i}{\partial x^2}\right) - \phi_s w_{fa}(P_{i+1} - P_i), \tag{1}$$

$$I_{fa}\frac{\partial^2\theta_i}{\partial t^2} = \frac{\partial M_i}{\partial x} + T_i, \tag{2}$$

where m_{fa} and I_{fa} are the mass and inertial moment per unit length of a fuel assembly, N_0 is the tension force at the bottom of the fuel assembly, d_g is the distance between two fuel rods centre, S_{eq} is cross-section area of the equivalent beam, w_{fa} is the fuel assembly's width, ϕ_s is the ratio of the structure area to the frontier surface between the fuel assembly domain and the by-pass domain, T_i is the shear force and M_i is the bending moment, P_i is the mean pressure in the ith by-pass F_{I_i}, F_{N_i}, F_{L_i} and F_{D_i} are fluid forces and will be defined latter in the text.

The constitutive laws of the fuel assemblies are given by:

$$T_i = GS_{eq}\left(\frac{\partial u_i}{\partial x} - \theta_i\right) + \mu_G S_{eq}\frac{\partial}{\partial t}\left(\frac{\partial u_i}{\partial x} - \theta_i\right), \tag{3}$$

$$M_i = EI_{eq}\frac{\partial\theta_i}{\partial x} + \mu_E I_{eq}\frac{\partial^2\theta_i}{\partial t\partial x} + M_{fret_i}, \tag{4}$$

where G is the shear modulus, E is the Young's modulus, I_{eq} is the quadratic moment of a fuel assembly and μ_G and μ_E are structural damping coefficients.

M_{fret_i} accounts for a progressive friction, it depends on the relative rotation $\theta_i - \theta_{0_i}$ where θ_{0_i} was the last value of θ_i when its derivative over time changed of sign, this value is updated at each change of sign. When the relative rotation is small, the structure has a linear behaviour. When the relative rotation is higher than the friction angle θ_f, the fuel rods at the extremities start to slide into the grids. As the relative rotation increases the number of rods sliding increases. This phenomenon is accounted for by the following equations:

$$M_{fret_i} = M_{0_i} + \begin{cases} \frac{N_p^4 K_c d_g^2}{12}(\theta_i - \theta_{0_i}) & \text{if } (\theta_i - \theta_{0_i}) < \theta_f \\ \frac{N_p^3 F_{ri} d_g}{4} - \frac{2N_p F_{ri}^3}{3d_g K_c^2}\frac{1}{(\theta_i - \theta_{0_i})^2} & \text{if } (\theta_i - \theta_{0_i}) > \theta_f \end{cases}, \tag{5}$$

$$\theta_f = \frac{2F_{ri}}{N_p d_g K_c}, \tag{6}$$

where M_{0_i} is the moment value when the ith fuel assembly changes its displacement direction, N_p is the number of fuel rods in one direction, K_c is the axial stiffness of a fuel rod and F_{ri} is the friction force between one fuel rod and one grid.

The porous media approach is based on space averaging of the local Navier Stokes equations considering an Arbitrary Lagrangian Eulerian formulation, in order to obtain equation of the porous media some assumptions are accounted for:

H1 The fluid is viscous, incompressible and newtonian.
H2 Gravity effects are neglected.
H3 The rod section does not deform.
H4 Distance between two rods remains constant.
H5 Turbulent kinetic energy is negligible in comparison with the turbulent diffusion.

The space averaging operation consist in integrating the Navier Stokes equations over a control volume and considering the mean values of the velocity $\mathbf{V_{eq}}$ and the pressure P_{eq}. Using the Leibniz and Gauss theorem and making some simplifications based on the assumptions leads to:

$$\rho_{eq}\frac{\partial\mathbf{V_{eq}}}{\partial t} + \rho_{eq}div\mathbf{V_{eq}}\otimes\mathbf{V_{eq}} = -\nabla P_{eq} + \mu_{Teq}\Delta\mathbf{V_{eq}} + \rho_{eq}\frac{\partial\mathbf{U}}{\partial t}\cdot\nabla\mathbf{V_{eq}} - \rho_{eq}\mathbf{V_{eq}}\cdot\nabla\frac{\partial\mathbf{U}}{\partial t} + \frac{1}{V_{\Omega_t}}\int_{A_s(x,y,z)}\boldsymbol{\sigma}\mathbf{n}dS, \tag{7}$$

$$div\mathbf{V_{eq}} = 0, \tag{8}$$

where ρ_{eq} is the equivalent fluid density defined by $\rho_{eq} = \phi\rho$ where ϕ is the porosity defined as the ratio of fluid volume over the total volume, ρ is the fluid density, μ_{Teq} is the equivalent porous turbulent viscosity and $A_s(x,y,z)$ the fluid structure frontier surface. Note that ϕ which is related to a volume ratio is different from ϕ_s in Eq. (1) which is related to a surface ratio.

The term $\frac{1}{V_{\Omega_t}}\int_{A_s(x,y,z)}\boldsymbol{\sigma}\mathbf{n}dS$ represents the structure body force acting on the fluid. An empirical expression of this force is given by Païdoussis (2003), it accounts for added mass effect F_I, flow induced damping F_N, damping in still water F_D and axial drag force F_L.

Considering a control volume accounting for the whole cross section of a fuel assembly and assuming a two dimensional problem gives:

$$\mathbf{V_{eq}} = V_{fa_i}\mathbf{e_x} + V_{cf_i}\mathbf{e_y}, \tag{9}$$

$$P_{eq} = P_{fa_i}, \tag{10}$$

where V_{fa_i} is the axial component of the mean velocity in the ith fuel assembly, V_{cf_i} is the transverse component of the mean velocity in the ith fuel assembly and P_{fa_i} is the mean pressure in the ith fuel assembly (Fig. 2).

Assuming the viscous term $\mu_{Teq}\Delta\mathbf{V_{eq}}$ is negligible compared to the fluid-structure terms F_{I_i}, F_{N_i}, F_{D_i} and F_{L_i}, the equations of the equivalent fluid in the ith fuel assembly become:

$$\frac{\partial V_{fa_i}}{\partial t} + V_{fa_i}\frac{\partial V_{fa_i}}{\partial x} = -\frac{1}{\rho_{eq}}\frac{\partial P_{fa_i}}{\partial x} - \frac{1}{d_g^2\rho_{eq}}F_{L_i}, \tag{11}$$