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PWR FUEL ASSEMBLY MODAL TESTING AND ANALYSIS

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ABSTRACT

Pressurized Water Reactor (PWR) seismic or Lost Of Coolant Accident (LOCA) loads could result in impacts between nuclear fuel assemblies or between fuel assemblies and the core baffles. Forces generated during these shocks are often the basis for the determination of the maximum loads and of the spacer grid and fuel rod design. The knowledge of the fuel assembly kinematics is essential to compute these maximum loads, and this requires experimental tests.

Our study aims at characterizing the behavior of a full-scale fuel assembly subjected to various excitations. The effect of the assembly environment (air, still water and water under flow) is studied.

The French Nuclear Reactor Directorate experimental facility HERMES T allows hydraulic and mechanical testing of full-scale fuel assemblies. It is designed for flow rate up to 1200 m3/h and temperature up to 170°C. Specific excitation devices allow mechanical tests with amplitudes of motion up to 20 mm. Laser vibrometry, displacement transducers and tracking camera apparatus measure the fuel assembly displacement.

To identify this Multi Degree Of Freedom (MDOF) system (assembly or assembly + fluid), two dependent problems have to be addressed: the linear or non-linear model selection, and the estimation of the corresponding parameters.

Under different environments and excitation types, it is shown that the mechanical system is strongly non-linear. The damping term, essentially fluid, increases with flow rate and with motion amplitude, while the stiffness decreases with amplitude.

The main results, the measuring and identification methods and the extrapolation to the reactor thermohydraulic conditions are presented and discussed.

INTRODUCTION

A nuclear core consists of vertical fuel assemblies arranged in square pitch array. The fuel assemblies are tall and laterally flexible structures, restrained only at their ends by the upper and lower core plates. The assembly fuel rods are restrained and evenly spaced by grids.

Nuclear fuel assembly's reliability during seismic events or Lost Of Coolant Accident (LOCA) is a major concern for the Commissariat à l'Energie Atomique (CEA) and Electricité De France (EDF). The mechanical excitation of PWR internals induces large lateral motions of the fuel assemblies. This results in impact between assemblies or between assemblies and the core shroud at the grid locations. Forces generated during these shocks are often the basis for the maximum design loads of the spacer grids and fuel rods. The damping of the assemblies is generated by mechanical dissipation in the structure and, mainly, by the presence of the coolant flow. These dissipative processes decrease the energy of the system and thus reduce the impact forces on the assemblies.

Knowledge of the values of damping versus thermohydraulic conditions is necessary to prove the integrity of the assemblies. An experimental program is conducted at CEA to characterize the behavior (damping & stiffness) during large motion of the assembly, in air, in still water, and in water under flow (up to 5 m/s).

NOMENCLATURE

C modal viscous damping (N/ms)
CMS superior modes equivalent stiffness (N/m)
K modal stiffness (N/m)
M modal mass (Kg)

P modal participation factor
f natural frequency (Hz)
w circular frequency (rd/s)
w0 modal circular frequency (rd/s)
β damping ratio (%)

TEST DESIGN

Because of the complex interactions between an assembly and its environment, our hydraulic tests are designed to simulate in-core conditions as closely as possible. We perform these tests with full-scale fuel assemblies (actual or prototype). Fuel rods are loaded with natural uranium oxide or plumb pellets in order to match the mechanical characteristics of the in-core assemblies. To closely simulate in-core conditions that exist throughout the life of the fuel, grid springs may be new or released. The test facility, the HERMES T loop located in Cadarache (France), simulates the reactor geometry and thermohydraulic conditions.

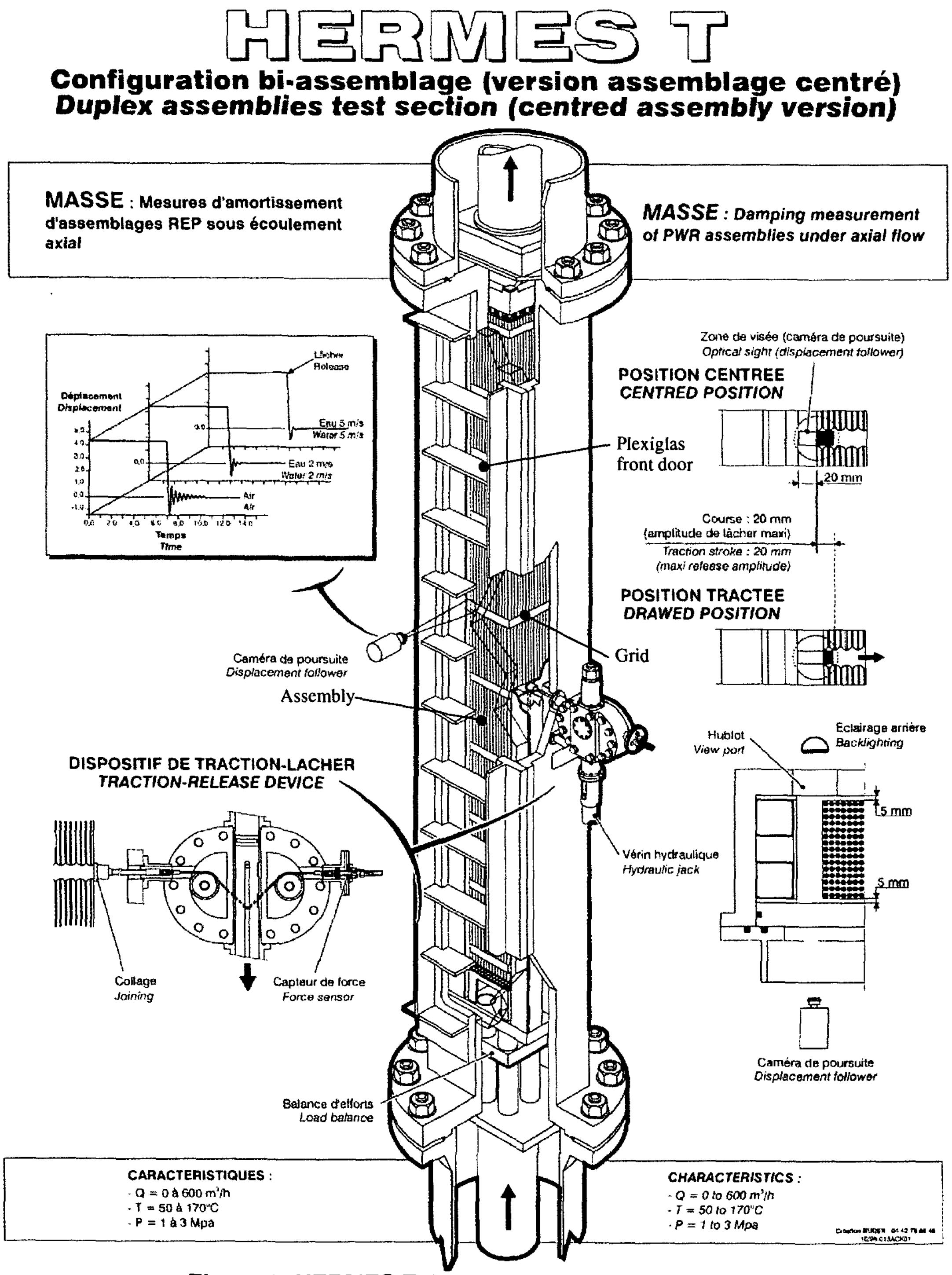

Figure 1: HERMES T duplex assembly test section

Test Facility

Tests on full-scale fuel assemblies are performed on our HERMES T hydraulic loop. Its test section is shown in Figure 1.

We usually use this loop to reproduce, measure and understand the mechanical and hydraulic behavior of the reactor core components (fuel rods, fuel assemblies, control rods, control rod guides, rod travel housings, rod drive mechanisms,…). The loop is designed for flow rate up to 1200 m^3/h, temperature up to 170 °C, and pressure up to 35 b. This loop has two test sections. The main one can accommodate 2 full-scale fuel assemblies or a single centered fuel assembly (mono-assembly configuration). The Plexiglas (50°C maximum temperature) front door provides us complete visual access to the fuel assembly. The mechanical excitation is provided by a bending and release system for the transient excitation, and by an hydraulic jack for sine or random excitation. The principal pump, that can be operated from 350 to 1000 rpm, provides heat. Temperature regulation is carried out using two heat exchangers.

Test Configuration

We test different 17x17 full-scale fuel assemblies. We use the mono assembly configuration. The distance between the fuel assembly and the internal test section wall is 20 mm in the direction of the excitation. In the perpendicular direction, this distance is 5 mm. The axial flow velocity is 0 to 5 m/s. The fluid temperature is generally 50°C. We also performed tests at 80, 110, 140 and 170°C. The assembly is subjected to sine, random or transient (traction and release, "snap-back" or "pluck" test) excitation. The point of excitation is the middle grid of the assembly (grid 5 of the ten-grid assembly), the upper and lower core plates are fixed. The middle grid is attached to the hydraulic jack (shaker) by a pre-constrained plate (4000 N). The 3 first modes are directly excited by the hydraulic jack (excitation up to 12 Hz), which impose displacements on the structure (forced response).

Test Instrumentation

The fuel assembly motion is monitored with a displacement transducer in water (placed on the middle grid), and 8 displacement transducers (grid 2 to 9) in air to allow investigation of multiple assembly vibration modes. Three tracking cameras are also used in water to measure the displacement of 3 grids. The test section is instrumented with an accelerometer placed at the fuel assembly middle grid level to assure the absence of reference wall vibration. A load cell measures the force applied on the assembly by the excitation system.

Data Acquisition

All the instrumentation measurements are acquired via a fast 128-channel acquisition system. Each data channel is sampled at 500 or 1000 samples per second in order to provide sufficient resolution especially in the frequency domain. The test measurement are viewed and analyzed on-line (time and frequency visualization, transfer function computation, damping measurement, mono and multi-modal system identification).

Data Analysis

Using experimental modal analysis or modal testing techniques (Ewins [1]), the mode shapes, frequencies and damping of the assembly structure in water can be measured directly. The classical method peak-amplitude or circle-fit for SDOF modal analysis (Ewins [1], Barbier [2]) are inadequate for our non-linear system with relatively closely-coupled modes and heavy damping. We look to a more appropriate modal analysis. Two dependent problems have to be solved, the selection of the model (linear or non-linear, mono or multi-modal) and of the parameter identification technique.

- Mono-modal linear model

The first measurement consists of identification of the equivalent SDOF linear system, with integer terms of modal mass M, viscous damping C and stiffness K.

$$M\ddot{x} + C\dot{x} + Kx = F_{ext} \quad (1)$$

The most efficient identification is carried out by the minimization of an error criterion between the measured and theoretical mobility transfer functions. The experimental mobility transfer function H_{exp} is calculated according to the formula:

$$H_{\exp} = \frac{S\dot{x}\dot{x}}{S\dot{x}f} \quad (2)$$

where $S\dot{x}\dot{x}$ is the auto spectral density of the measured velocity, and $S\dot{x}f$ is the cross spectral density between the measured velocity and force.

The theoretical mobility transfer function H_{theo} of a mono-modal linear system is

$$H_{theo} = \frac{iPw}{1 - \frac{w^2}{w_0^2} + 2i\beta\frac{w}{w_0}} \quad (3)$$

or

$$H_{theo} = \frac{iw}{K - w^2M \ + iwC} \quad (4)$$

- Multi-modal linear model

For the important damping value, the mono-modal measurement is not accurate because of the overlap of the modes, and the multi-modal measurement is necessary. This method of measurement consists of identifying an equivalent linear system with 3 modes and an equivalent stiffness corresponding to the contribution of the higher frequency modes. The identification is carried out by the minimization of an error between the measured and the theoretical mobility transfer functions. Mobility computation (velocity / force) rather than receptance (displacement / force) makes it possible to emphasize the terms of damping of the viscous type of the form which interests us in these tests and to increase the weight of modes 2 and 3 with respect to mode 1. The experimental mobility transfer function, H_{exp}, is again calculated according to the equation 2:

This transfer function can be calculated starting from only one test sine swept over a broad band, but also starting from distinct tests carried out at different frequencies of sweeping.

The theoretical mobility transfer function H_{theo} of a linear system with three modes and contribution of the higher modes (of stiffness type) is:

$$H_{theo} = \sum_{j=1}^{3} H_i + H_{CMS} = \sum_{j=1}^{3} \frac{iP_j w}{1 - \frac{w^2}{w_{0j}^2} + 2i\beta_i \frac{w}{w_{0j}}} + \frac{iwP_{CMS}}{1 - \frac{w^2}{w_{0CMS}^2}} = \sum_{j=1}^{3} \frac{iP_j w}{1 - \frac{w^2}{w_{0j}^2} + 2i\beta_i \frac{w}{w_{0j}}} + iwCMS \quad (5)$$

that can also be written in the form :

$$Htheo = \sum_{j=1}^{3} Hi + H_{CMS} = \sum_{j=1}^{3} \frac{iw}{K_j - w^2 M_j + iwC_j} + iwCMS \quad (6)$$

- Mono-modal non-linear model

We will see that structure characteristics depend on the motion amplitude. This is to say that the system is non-linear. In this case, the linear model is efficient to measure the characteristics of the system for a fixed excitation and thermohydraulic environment but has to be improved when we have to take into account different excitation amplitudes.

We have compared different type of polynomial representations of the system with different parameters (x or $|x|$, $\dot{x}$ or $|\dot{x}|$, $\ddot{x}$ or $|\ddot{x}|$) and degrees. The most accurate form appears to be (Pisapia [3]):

$$M\ddot{x} + C\dot{x} + Kx = M_0\ddot{x} + C_0\dot{x} + K_0 x + C_1|\dot{x}|\dot{x} + K_1|x|x = F_{ext} \quad (7)$$

This form is efficient in air, still water and under flow. The identification is the same that for the linear identification, except for the number of identified parameters. A multi-test (multiple amplitudes) identification is necessary (Pisapia [3]).

TEST RESULTS AND ANALYSIS

This study aims at determining the behavior of the fuel assembly with direct excitation of its three first modes.

Transient excitation

The measurements show that for any environment, the damping term is larger with transient excitation (bending and release, see Fig. 2) than with swept sine excitation of the first mode. Computations show that for an assembly, this difference is partly induced by the effect of the higher modes, which is more important with transient excitation and dissipates the energy of the system. With a pure sine excitation, only the first mode is excited and this induces less dissipation. Measurement of all the grids of the assembly enables us to identify the contribution of the different modes.

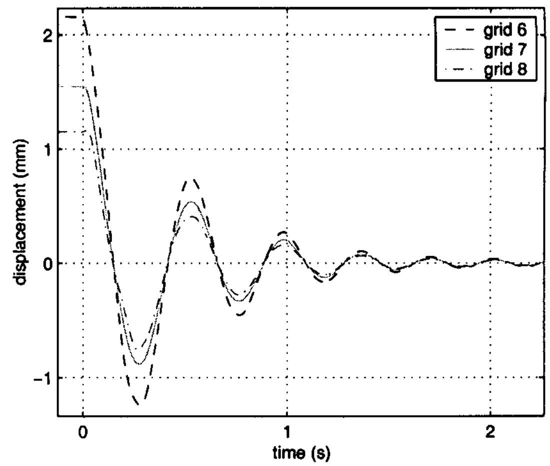

Fig 2: Bending and release test, displacement of the grids 6, 7, and 8 of the assembly

For high damping (damping ratio over 20%), bending and release excitation is not very accurate because of the very few cycles of the assembly. From the data, we are able to identify the global system but not to separate the effect of the different modes. Therefore continuous excitation (sine or random) is more efficient.

Sine excitation

The Figure 3 presents the deformation of the assembly in its first mode measured at each grid of the assembly. The black points represent the excited grid (grid 5).

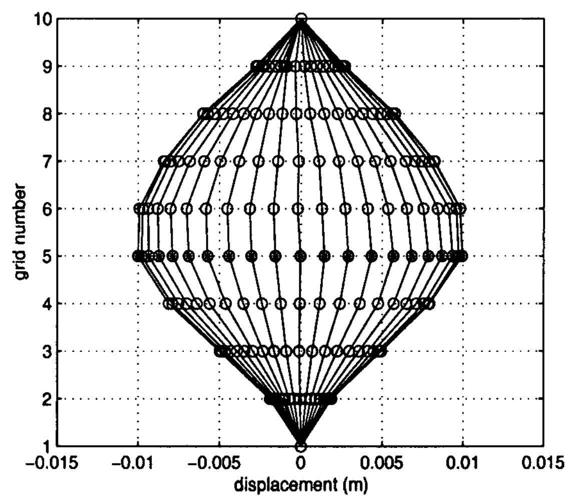

Fig 3: First mode deformation of the assembly with sine excitation

The stiffness measured with quasi-static (very low frequency) or sweep sine excitation are equal. It is very similar in air or still water. Flow velocity seems to slightly change this observation. We see (Fig. 4) that the stiffness is smaller for large displacements and, hence, that the system is non linear. The decrease of the stiffness for large displacement results of the rods support conditions (springs), more efficient for small displacements. With dynamic sine excitation around the first mode and identification of the system, we measure a very similar stiffness. The optimal expression for non-linear stiffness is $K = K_0 + K_1|x|$ (Eq. 7) where K_0 and K_1 are real values that depend of the thermohydraulic environment.

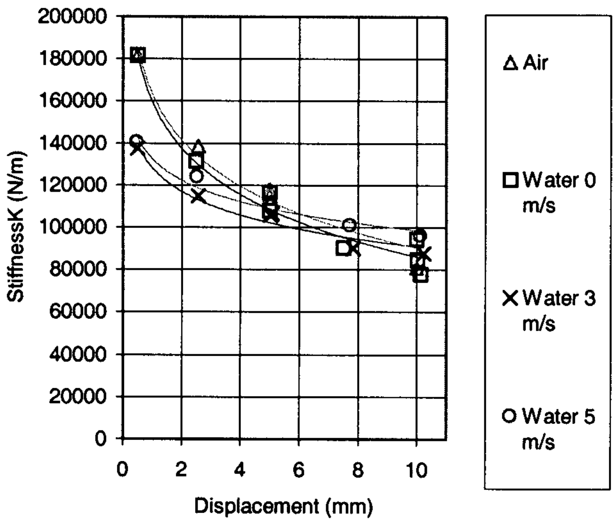

Fig 4: Assembly stiffness term K measured in air and water with sweep sine excitation

We measure the damping term C for the different environments with sine excitation around the first mode of the assembly. Damping in the fuel assembly is mainly determined by the rod support conditions (Koebke [4]). At small displacement, the support spring forces prevent any relative movement/friction between the rods and their support. For larger displacement, sliding frictions and chocks appears and increase the overall structural damping.

We see (Fig. 5) that damping in still water is greater than damping in air. This is the result of drag forces caused by the viscous dissipation of the fluid induced by the lateral motion of the assembly. We can also notice that the damping term changes with the amplitude of the motion (which corresponds to velocity and sliding friction changes), and is, therefore, non-linear. In water under flow (5 m/s), we measure much higher damping, twice or triple that in still water. This is the result of the addition of lift forces, which, according to theory, are proportional to the axial flow speed (Rigaudeau [5], Nhili [6]). Axial flow enforces circulation around the assembly and produces much larger lift forces opposing lateral motion. Our measurements seem to confirm this theoretical result.

For transient excitation, the measured damping is larger, but the overall comportment is similar.

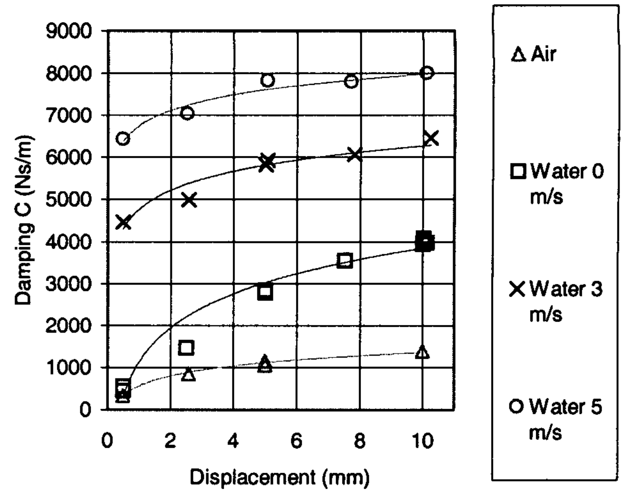

Fig 5: Assembly damping term C measured in air and water with sweep sine excitation

The most accurate positive non-linear damping term $C_1|\dot{x}|$ (see Eq. (7)) is the same as the one observed in still water in the literature for a single tube in still water (Blevins [7]).

The modal or generalized mass also varies with the thermohydraulic environment. For a dimensionless mass of 1.0 in air, we measure a mass of 1.4 in still water and 1.1 in water with a flow of 5 m/s (in our hydraulic conditions). Increase of the generalized mass in still water is induced by the inertial effects of the fluid (necessary acceleration of the fluid) which seem to be modified with flow rate. Moreover, modal deformation of the assembly is slightly different.

Random excitation

Random excitation in a 0-12 Hz frequency band on the fifth grid of the assembly enables us to identify the non-linear system for different deformations of the assembly and larger frequency responses. Results are slightly different to the ones observed with large frequency band sweep sine-excitation. Measurements are still in progress.

MULTI-MODAL ANALYSIS

For our heavily damped system, measurement of the second and third modes (ultimately of the first mode) is not accurate with mono-modal identification because of the modes overlapping. For this analysis, we have to excite the assembly over a large frequency band, typically 0-12 Hz. The excitation may be swept sine or random excitation. Figure 6 presents the real and imaginary part of the mobility transfer function measured on our assembly. We can identify the three first modes of the assembly and the higher modes equivalent stiffness contribution according to Eq. (5) or Eq. (6) (linear equivalent system).

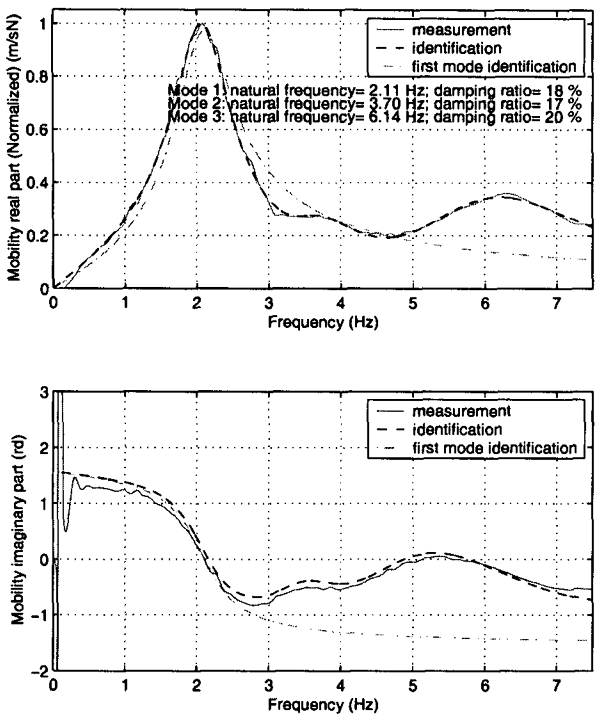

Fig 6: measured and fitted (linear model) mobility transfer function

We observe the overlap of the modes 1, 2 and 3 for this medium damping test (3 mm amplitude, still water environment, 18% damping ratio for the first mode). The result of such an analysis is a more accurate measurement of the heavily damped system, the relative damping correction on the first mode is less than 20% compared to a mono-modal identification.

DYNAMIC MODELS

The experimental measurement analysis and extrapolation to the reactor thermohydraulic conditions is a major concern. Tests show that the measurement results depend of the excitation. For example, the damping measured with sine and transient excitation is significantly different. Random excitation provides again slightly different results. The classical measurement methods or our more sophisticated identifications are efficient but limited for extrapolation exercise. Moreover, in-reactor flow conditions are slightly different. The simple mechanical models used for measurement are not accurate for reactor conditions extrapolation. We need more sophisticated representation of the system.

Works on this subject already exist (Rigaudeau [5], Broc [8], Brochard [9], Shah [10]). One of our approach aims at compute a full reactor core comportment under seismic or LOCA loads: models must be as simple as possible but meanwhile realistic.

We represent each assembly with a single beam. It features nodes at assembly top, bottom, and grid levels. The model contains shear and rotation stiffness, damping, and hysteresis (both objective and equivalent parameters are required). Impact model is a classical spring damper at each grid level. We perform specific tests for impact study (Broc [8], Collard [11]). Hydrodynamic is computed with a finite volume or finite element method, the water is considered as viscous and incompressible. Fluid and structure dynamic are totally coupled. We solve the system in the time-space domain. First results (forces, static and modal deformations) in air or still water are consistent with single assembly tests. We perform hydrodynamic coupling tests (multi-assembly tests) in still water (Broc [8]) and in water under flow in order to qualify the hydrodynamic coupling and damping models.

Complete developments and validations are in progress.

CONCLUSIONS

The experimental facilities and theoretical capabilities of our laboratory enable us to characterize the behavior of full-scale fuel assemblies in semi-realistic mechanical and hydraulic configurations. These measurements are necessary to characterize the system and give information to the theoretical models. Extrapolation to reactor conditions is then possible by a theoretical approach of the coupled fluid-structure system.

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