On the rational design of fuel assemblies for reactor facilities

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Abstract. This report deals with a computational and experimental justification for the search of the optimal spacing for the axial fuel assembly (FA) spacer grid (SG) arrangement in reactor facilities subject to vibration resistance requirements. Practically, as is known, the spacing selection is postulated at the present time based on the earlier selected FA dimensions for both effective and decommissioned reactor facilities. However, in our opinion, this selection can be based on results of a computational and experimental justification. The major guidelines for the development of general rules for the rational FA design to ensure reliable operation and vibration resistance of FAs in conditions of impacts from induced hydroelastic vibrations in the axial coolant flow have been formulated with regard for the physical laws of interactions between the flow and elastic structures. A finite element FA model has been built, natural frequencies and modes of bending vibrations have been determined, mathematical relations have been defined to estimate variations in the amplitudes of bending vibrations, and major criteria have been formulated for ensuring the FA vibration resistance. The paper also presents experimentally obtained spectra of the standard multipin FA pressure fluctuations and vibrations in a hydraulic test bench in which water is used as the test environment. Specific recommendations have been developed for the axial FA spacer grid spacing selection.

Key words: Nuclear fuel assemblies (FA), optimal spacing for the axial FA spacer grid, finite element FA model, natural vibration analyses.

1. Introduction

For reactor facilities, known to be specific in having some of their components and equipment, including the FAs, not accessible for inspection and repair (replacement) and being exposed to irradiation with associated radiation creep and stress relaxation, the role of justifications with respect to vibration, vibration wear and fretting corrosion resistance is of the utmost importance [1-3]. The purpose of this paper is to look for and to develop recommendations for the selection of the optimal fuel pin FA spacer grid (SG) spacing so that no allowable levels of induced hydroelastic vibrations caused by the axial coolant flow are exceeded.

The major recommendations for the selection of general rules for the rational FA design, specifically for the selection of the optimal SG spacing to ensure reliable operation and vibration resistance of FAs in conditions of impacts from induced hydroelastic vibrations in the coolant flow, are based on the physical laws of the fluid flow interaction with elastic structures. As a precondition, this requires available data on:

– the predominant mechanism for the excitation of hydroelastic FA vibrations in conditions of the axial coolant flow inside the reactor circuit;

– the threshold value of the critical flow velocity at which the dynamic stability is lost;

– the predominant mechanism for the breakdown of fuel pins and the FA as the whole.

One of the breakdown mechanisms is the fatigue mechanism caused by a combined action of low-frequency heat-up – steady state – shutdown operating loading cycles and high-frequency vibration loading from the coolant flow. Practical operation of reactor facilities however shows that most of the failures result from the fuel cladding mechanical vibration wear and fretting corrosion processes in the fuel – spacer grid interface regions and at the FA-FA contact points. Such breakdown mechanism stems from the structural features of FAs having the form of SG-integrated close-packed fuel pin bundles with small gaps between the pins, which prevents intense transverse vibrations with a high level of stresses.

The justification of the optimal SG spacing from the point of view of the FA vibration resistance was based on a comparative analysis of results of commonly known and proven procedures used to estimate the dynamic characteristics of FAs and the practical experience of design. Keeping in mind the physical notions of the coolant flow hydroelastic interaction with an FA and its components, one may state that the spatial nature of the force excitation induced by the flow does not remain axially invariable throughout the FA and this circumstance needs to be taken into account [4-6].

Each design organization has its own practical experience in the development of FAs which is based on tested SG spacing options for effective reactor facilities and, as often happens, with no link to the coolant flow velocity, the FA geometrical dimensions and the coolant type and parameters. At the present time, strictly speaking, the spacing selection is postulated practically based on the designer insight and the existing positive experience of reactor operation.

The SG spacing is optimized so that to ensure the vibration resistance which requires the natural fuel pin and FA vibration frequencies to be detuned from deterministic excitation frequencies, with the pin transverse vibrations to be limited to avoid mutual collisions [1]. Such optimization allows minimizing the metal consumption and improving the reactor's hydraulic and neutronic performance.

2. Initial data for analysis

2.1 Design features of a standard FA

Considered as an example is a mockup of an FA with a length of L ~ 2000 mm composed of N = 160 smooth fuel pin simulators of the diameter d = 9.7 mm with a spacing of s = 13.4 mm. The SG-integrated fuel pin simulators form a fuel assembly mockup. In the initial condition, there are nine spacer grids secured in nine axial FA sections between the end pieces (the FA tail and head). An overall view of the full-scale FA mockup is shown in Fig. 1.

2.2 Experimental data on the FA pressure fluctuations and vibrations

The operating conditions of the full-scale FA mockup with nine SGs (the initial design) were simulated on a hydraulic bench in which water was used as the test environment. Inside the bench's test portion, the FA mockup was secured identically to the fastening conditions in the reactor core. The coolant flow was upward. To study the FA mockup structure vibrations, the bench was equipped with an information and measurement system to measure the parameters of the pressure fluctuations and vibrations in structural components. There were accelerometers installed in the three fuel pin simulators at different axial positions (at the bottom, in the middle and at the top), with pressure fluctuation sensors installed on the outside at different points axially along the hydraulic bench's test portion (in the region where the flow impinged the FA tail, at the center and at the FA outlets), and strain gages installed on the support tubes at the bottom near the tail. Fig. 2 shows spectra of the pressure fluctuations into the coolant flow for the three FA sections. The spectra were obtained for the rated flow velocity of 5.7 m/s corresponding to the Reynolds number of $\sim 5 \cdot 10^4$. The fluctuation spectra and the most intense components thereof are the greatest contributors to the FA vibration activity. The FA response spectra in the form of vibrations are presented in Fig. 3. In Figs. 2 and 3, the pressure fluctuation and vibration spectra corresponding to the maximum flow velocity show sharply distinct peaks matching the frequency 32.6 Hz and at the frequencies ~

65.0 and 97.5, which, in turn, match the first, the second and the third harmonics of the pumping set rotational frequency.

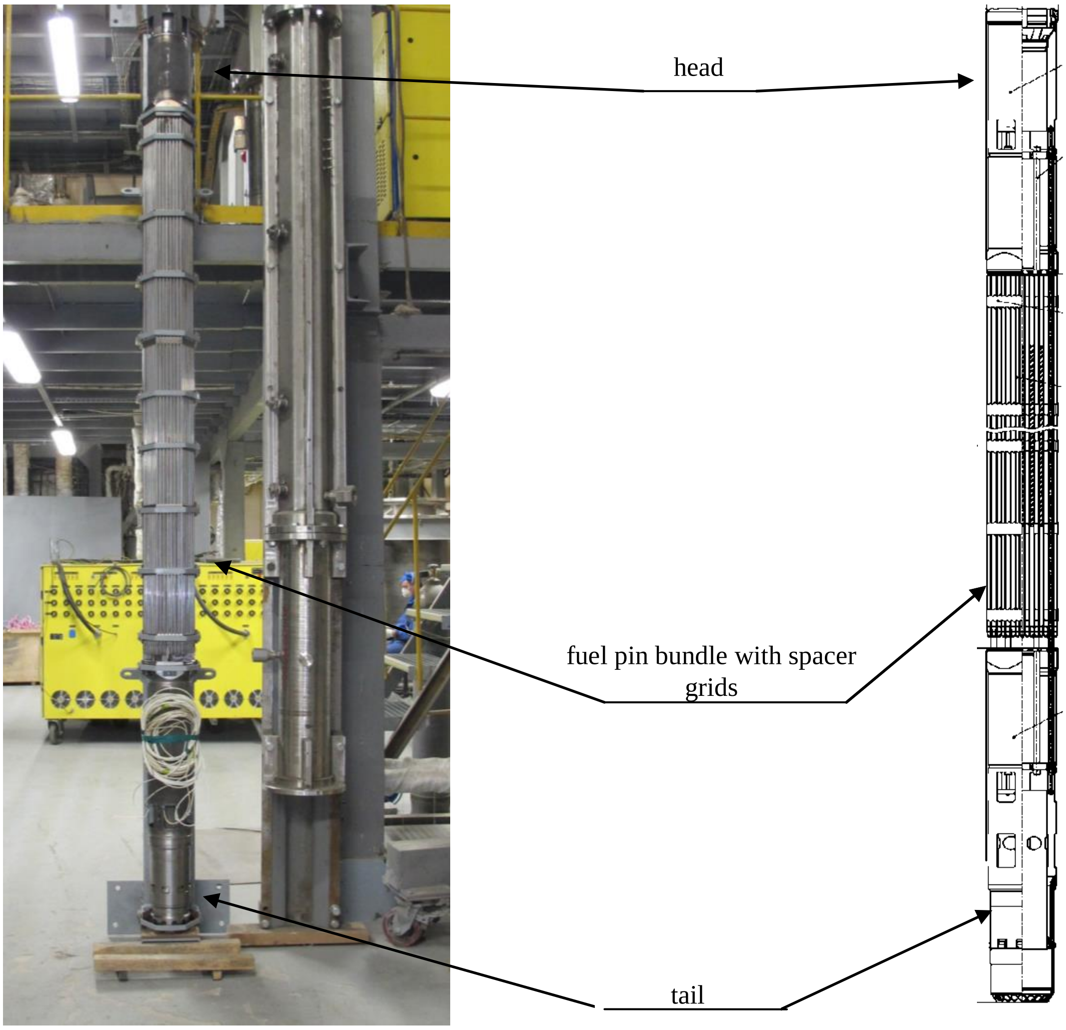


FIG 1. Overall view of the full-scale FA mockup: a photograph (left) and a schematic design (right)

Representative regions with the greatest possible intensity can be identified in the fluctuation spectra in Fig. 2. Most of the pressure fluctuation energy is liberated in the low-frequency spectrum in a range of 1 to ~ 30 Hz. The same can be observed for the vibration spectra (Fig. 3). The intensity of fluctuations and vibrations in the flow motion inside the FA central part decreases synchronously. On the contrary, there was a minor growth in the fluctuation and vibration levels recorded at the test portion outlet, explained by the fact that the vibration displacements of this FA part are higher due to the cantilever bracing used in this section. So, with information available on the loading spectrum generated by the axial coolant flow, one can say that it is the FAs that excite hydroelastic vibrations and that these vibrations have a frequency structure.
Having analyzed the pressure fluctuation and vibration spectra, one can state that the FA vibrations are forced by nature, and the maximum intensity of vibrations is concentrated in the low-frequency region.

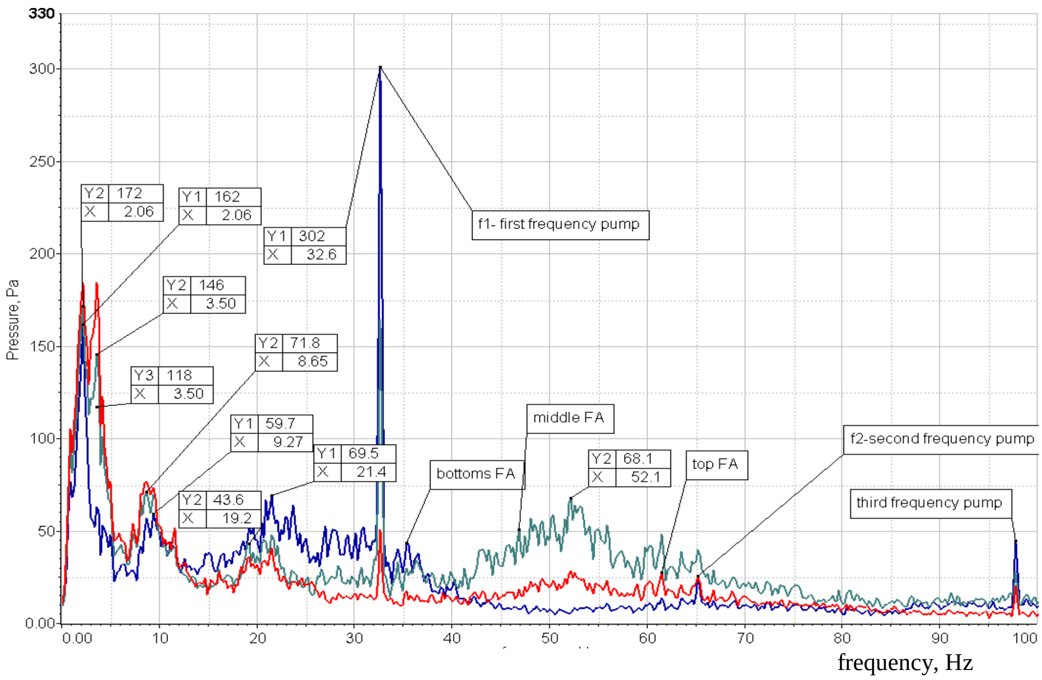


FIG 2. Spectra of the axial coolant flow pressure fluctuations in the hydraulic bench's FA test portion at the maximum flow velocity of 5.7 m/s (in the region where the flow impinges the FA tail – blue, in the region of SG 1 – green, at the FA inlet – red)

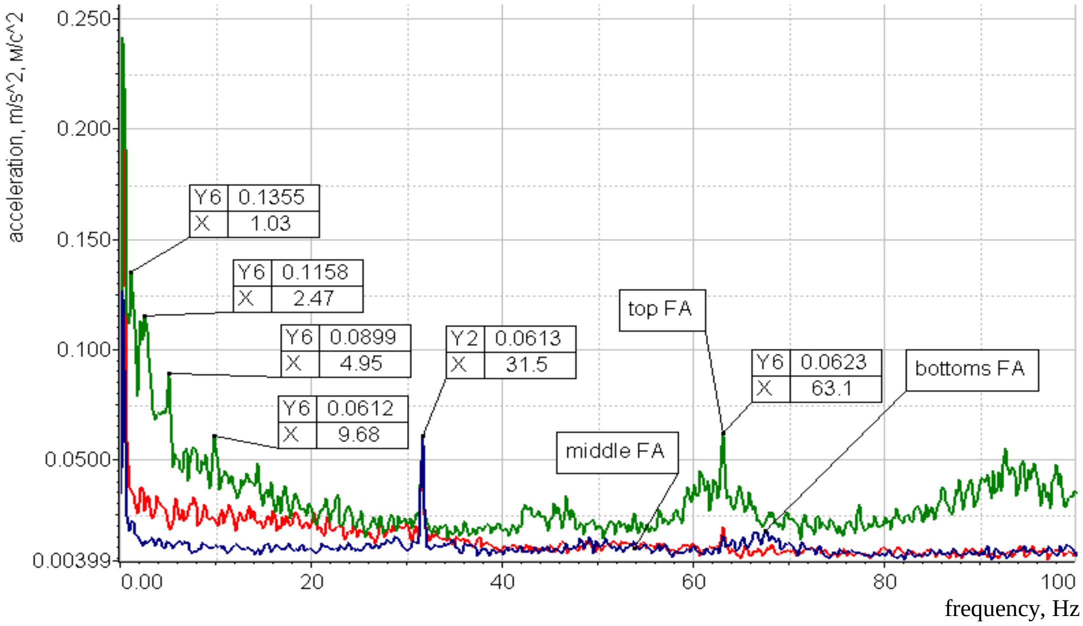


FIG 3. Spectra of the FA vibrations from the accelerometer vibration signals by sections: lower part (blue), central part (red) and upper part (green)

2.3 Mathematical FA model

A finite element model (Fig. 2) was built to calculate the natural vibrations of the FA mockup and its components. The calculation was based on the ANSYS computer system. The following equation of natural frequencies and vibration modes in a matrix form was also used for the calculation:

$$\left([K]-\omega_i^2[M]\right)\cdot\{x_i\}=0, \tag{1}$$

where $[K]$ is the symmetrical matrix of the FA stiffness formed based on the stiffness matrices of components with regard for kinematic constraints, and $\{x_i\}$ are the sought-after vectors of generalized displacements for all nodes of the ith natural vibration mode. The modal analysis was performed in an assumption that the FA is a linear system, in other words, that all structural features responsible for nonlinear behavior had been linearized for the model preparation. Therefore, relative displacements (of both the fuel pins and the support tubes in the SGs) had been forbidden at the contact points of the model components, and no nonlinear effects, such as gaps or slipping motions, were taken into account to define the boundary conditions. An 8-node SOLSH190 (Solid-Shell) finite element implementing the Timoshenko-Mindlin "shear" shell theory (more generally known as Reissner-Mindlin theory in foreign literature), based on the hypothesis of a quadratic distribution of tangential stresses through the cladding thickness, was used for the simulation. The SOLSH190 element is intended to simulate thin-wall structures with a broad range of relations of the wall thickness δ to the minimum curvature radius of the median surface ρ (approximately $\delta/\rho \leq 1/5$). The FA fastening in the support plate was modeled as if being of a hinged type and using the

Remote Displacement option: the motions of the tail piece components located in the region of the holes for the lock holders with only spherical motion of the said region permitted. The FA fastening in the support plate was modeled by defining the kinematic condition for the absence of node displacements on the tail body surface. The calculation was performed for the three mockup SG axial arrangement options: there were 7, 8 and 9 SGs installed axially along the fuel pin bundle about 2000 mm high.
The initial 30 FA natural frequencies and vibration modes were obtained by calculation, of which 24 were bending vibrations. To estimate the dependence of natural frequencies on the number of SGs in the FA, the values of natural frequencies and their respective bending vibration modes are given in the table for Fig. 4.
An analysis of the results has shown that a larger number of SGs in the FA, leading to a greater FA stiffness, causes natural vibration frequencies to grow, which is especially distinct in the event of higher vibration modes. This growth is however small in the lowest frequency region: if the first-mode lowest frequency variation is from 5.01 Hz (7 SGs) to 5.34 Hz (8 SGs) and 5.75 Hz (9 SGs), then the variation for the 10^{th} vibration mode is from 30.3 (7 SGs) to 32.4 (8 SGs) and 37.0 Hz (9 SGs). One may state that the FA lowest natural vibration frequencies depend slightly on the FA number. Calculations of the FA natural vibrations make it possible to estimate the relation between the most intense region of the pressure fluctuation spectrum generated by the coolant flow and the potential FA resonant vibrations.

3. Interpretation of calculation and experimental data

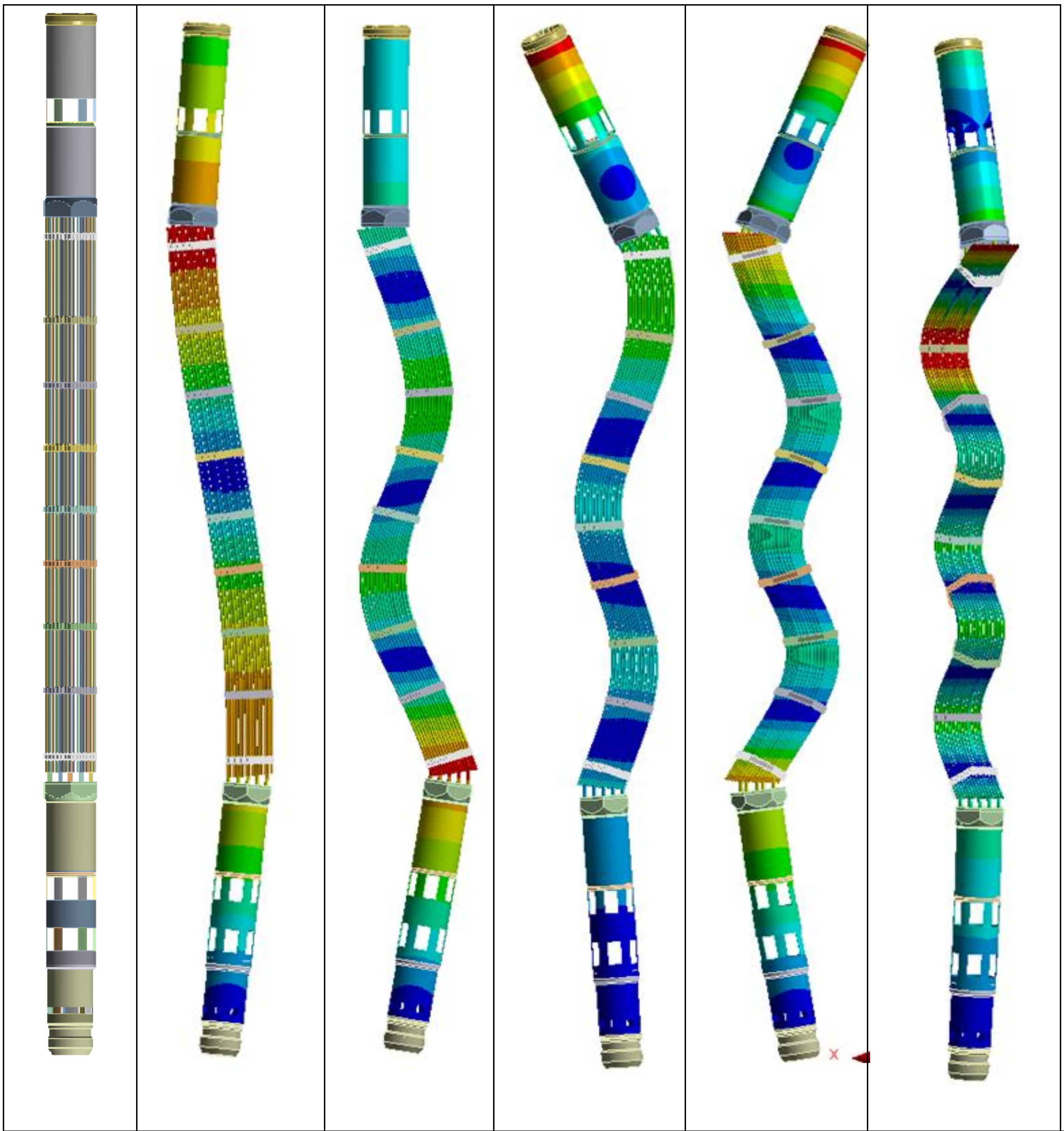
An analysis of experimental data on the FA fluctuations and vibrations in conditions of a hydraulic test bench leads to a conclusion that the assembly has the greatest vibration activity in the low-frequency region. The FA vibrations correlate well with the coolant flow pressure fluctuation behavior, which makes it possible to conclude that the mechanism of forced vibrations has the predominant role. Such excitation mechanism exhibits the coincidence of the loading and vibration spectra. It can be therefore assumed that it is reasonable to use the most intense part of the FA spectrum in a range of 1 to 30 Hz for a further FA vibration analysis. An important parameter for taking into account the FA vibrations in the coolant flow is the added liquid mass coefficient *k*. Based on results of the natural vibration measurements in two environments – air (f_0) and water $(f_в)$ – the coefficient *k* was determined from the relation

$$(f_0 / f_в)^2 = 1 + (m_{cool} / m_{FA}), \quad k = m_{cool}/m_0, \tag{2}$$

where m_{FA}, m_{cool} , m_0 are the FA weight per unit length, the added mass and the displaced liquid mass respectively.
The coefficient *k* for the FA inside the hydraulic bench's test portion has turned out to be equal to ~ 7 which agrees with the data in [4]. With this coefficient allowed for, the FA natural vibration frequencies will decrease in accordance with the relation

$$f_{cool} = f_{air} \cdot \left(\rho_{FA} / (\rho_{FA} + k\, \rho_{Cool})\right)^{0.5}, \tag{3}$$

where f_{cool}, f_{air} are the vibration frequencies in water and in air respectively; and ρ_{FA} and ρ_{COOL} are the densities of respectively the assembly material and the water.



Mode No.	1	7	10	15	22
Natural vibration frequencies, Hz					
9 SGs	5.75	28.8	37.0	57.6	113
8 SGs	5.34	25.9	32.4	54.8	94.4
7 SGs	5.01	25.5	30.3	49.8	83.1

FIG 4. FA finite element model and its natural vibration modes

It is not difficult to find by calculation that the natural vibration frequencies in water, as compared to air, will drop by a factor of about 1.4. Meanwhile, no vibration modes however vary.

With experimentally obtained fluctuation spectra at hand, it becomes easy to select the axial FA SG number. We do not reach the potential for the excitation of vibrations in the fuel rod spans between the SGs since there are no pressure fluctuations in this frequency region. The calculated natural vibration frequency values for the fuel rods, as a multispan beam, assuming that the SGs were uniformly distributed in the axial direction were 192 Hz for the initial FA with 9 SGs, 126 Hz for the modified FAs with 8 SGs, and 83 Hz for the FAs with 7 SGs. With regard for the added mass of water, these will be equal to 137, 90 and ~ 60 Hz respectively. Therefore, even for the FA with 7 SGs, the fuel pin natural vibration frequency is two times as high as the upper boundary of the excitation spectrum, which shows that natural frequencies have been safely detuned from excitation frequencies.

To avoid collisions of adjacent fuel pins in the bundle, we will estimate the variation in the amplitude of bending vibrations so that the requirement $2A < \delta$ is fulfilled, where A is the vibration displacement amplitude, and δ is the width of the gap between the pins.

The variations in the FA vibration amplitudes can be estimated comparatively using the prediction procedure in [7, 8] that can be used for considering two, the modified and the initial, FAs. The estimation in this case may be given with the following assumptions:

– the frequency spectrum of excitations in a frequency range of Ω_1 to Ω_2 is constant, where Ω_1, Ω_2 are the fundamental natural vibration frequencies for the modified design and for the initial design;

– the damping factor for the modified design differs slightly from that for the initial design;

– natural vibration modes for the modified design and for the initial design are equal;

– there are no acoustic resonances in the system.

In a general case, the root-mean-square value of the vibration level <x> is the function of such parameters as bending stiffness *(EI)*, damping α, flow velocity squared U^2 and natural vibration mode $X(x)$ which is the function of the boundary conditions, that is

$$< x >= f\left((EI), \alpha, U^2, X(x)\right) \quad (4)$$

We will assume that the FA is subjected to a load that can be in a general case represented as a Fourier expansion

$$Q(x,t) = g(x) \cdot \xi(t), \quad (5)$$

where $\xi(t)$ is the broadband random process caused by the turbulent fluctuations of the flow with the intensity $D_{\xi\xi}$. By natural mode, the average square of transverse vibrations $<x^2>$ will be equal to:

$$< x^2 >= D_{\xi\xi} \cdot \chi^2 / 4 \cdot \alpha \cdot \Omega^2 , \quad (6)$$

where $\chi^2 = \nu^2 \int g(x) \cdot X(x) \cdot dx$, $\nu^2 = \int_0^l X^2(x) \cdot dx$ is the natural function norm squared.

Then the vibration level of the new FA design will be governed by the relation

$$< x_m^2 > = < x_i^2 > \left(D_{\xi\xi}^i \cdot \nu_i^2 \int_0^{li} g_i(x) \cdot X_i(x) dx \cdot \alpha_i \cdot \Omega^2{}_i \right) \cdot \left(D_{\xi\xi}^m \cdot \nu_m^2 \int_0^{l_m} g_m(x) \cdot X_m(x) \cdot dx \cdot \alpha_m \cdot \Omega^2{}_m \right)^{-1}$$

(7)

The assumption that the intensities of the dynamic impact $D_{\xi\xi}^i = D_{\xi\xi}^m$ are equal is valid in as much as the condition that the coolant flow velocities in the FAs and the media densities are equal or, at least, close is fulfilled. However, the media densities ρ being different, this circumstance shall be allowed for through force parameters, and, in the given case, through the hydrodynamic head ratio of $\frac{\rho V^2}{2}$ that determines to some extent the dynamic load applied to the FA pins. With the boundary conditions being invariable, $\nu_m^2 = \nu_i^2$. If the pattern of the load distribution by the fuel pins and the assemblies remains generally the same, it may be suggested that the ratio of integrals in relation (7) will be approximately equal to unity. The ratio of the intensities will be proportional to the relation $\left(\rho V^2\right)_i / \left(\rho V^2\right)_m$ that will be equal to unity in our case. The damping will be also approximately equal. Then the average square of the modified FA vibration displacements will be determined only by the relation of natural frequencies, that is:

$$< x_m^2 > = < x_i^2 > \cdot \Omega_i^2 / \Omega_m^2 \tag{8}$$

Therefore, for the modified FA with 8 SGs, as compared to the initial FA with 9 SGs, the variation in the root-mean-square value will be $< x_m > \approx 1.07 < x_i >$ for the lowest (most active) frequency component equal to ~ 5 Hz and $< x_m > \approx 1.1 < x_i >$ for the upper boundary of the vibrationally active range equal to 30 Hz. With the SG number reduced to 7, the maximum variation will be $< x_m > \approx 1.13 < x_i >$. Under the assumptions made, it can be therefore seen that a smaller number of SGs will lead to a minor and harmless growth in the vibration level, this allowing a conclusion that the number of SGs axially along the FA may be reduced. But, in any way, the modified FA requires to be tested in the hydraulic bench conditions to confirm the calculation results.

4. Conclusion

The paper presents the results of a computational and experimental justification for the selection of a rational FA geometry to ensure the vibration resistance in the selection of the number of the FA spacer grids in the axial direction. Experimentally obtained spectra of pressure fluctuations and vibrations are presented for the case of the axial coolant flow velocity of 5.7 m/s. It has been shown that a reduction in the SG number leads to the level of vibrations growing proportionally to the ratio of frequencies for one and the same vibration mode. The increase in the FA vibration levels in the most intense part of the fluctuation spectrum is small. In any case, however, the modified FA needs to be proof tested to confirm the computational and experimental justification results.

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