Fluid-induced vibrations in the fuel bundle of a PbBi reactor

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ABSTRACT

In the paper, the results of fluid-induced vibration analyses on fuel pins of a lead-bismuth cooled nuclear reactor are shown.

The analyses, aimed at selecting the best number, position and elastic characteristics of fuel elements spacer grids, were performed using both empirical methods (Chen-Weber method) and FEM numerical calculations.

INTRODUCTION

The two main concerns related to the pacific use of nuclear energy are the safety of operating power plants and the safe management of nuclear wastes, with particular attention to long-living wastes.

The attention paid to the first issue lead, starting from '80s, to the development of new designs of reactors with enhanced safety features, mainly based on extensive use of passive systems.

As far as the second issue is concerned, since the birth of nuclear industry, safe disposal of long-living nuclear wastes has been deeply studied; nevertheless, no country has adopted a definitive solution due both to technical difficulties to find suitable sites and to high estimated costs to design, construct and manage such a site. On the other hand the geological disposal of long-living (thousands years) waste has to face the psychological opposition of population, afraid of potential contamination.

In last years, an increasing attention has been addressed to systems able to "burn" long-living wastes, transforming them in short or medium-living wastes that may be easier managed and stored and accepted by population.

In this framework, the European Community is promoting the construction of a demonstrations plant called ADS (Accelerator Driven System).

The solution being studied starts from the EA (Energy Amplifier) idea proposed by Prof. Carlo Rubbia in 1993. The "transmutation system" (see fig. 1) is a sub-critical fast fission reactor; the neutron source, necessary to sustain the chain reaction, is provided by the spallation of high-energy protons (supplied by an accelerator) on the nuclei of a heavy metal target located at the core centreline.

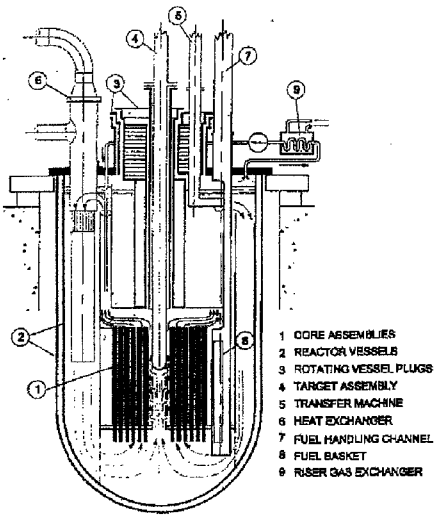


Fig. 1 – ADS Reactor Scheme

The heavy metal selected for the target is a lead-bismuth alloy, which is used also as core coolant. The core coolant flows in parallel through an equilateral triangular lattice of fuel pins, enclosed in hexagonal shrouds.

The fuel element design is based on the SuperPhenix-1 (SPX-1) fast breeder reactor fuel assembly. The different characteristics of the coolant (density and viscosity) forced to adopt a higher pitch/diameter ratio with respect to the SPX-1 solution, so the use of the elicoidal spacer wires, as in the SPX-1, is impossible and spacer grids must be introduced in the fuel assembly.

The selection of the number, of the position and of the elastic characteristics of the spacer grids requires the analysis of vibration induced by the coolant flow in the fuel pin and the parametrization of the vibration amplitude versus grids position, number and stiffness.

FUEL ELEMENT CHARACTERISTICS

Fig 2 shows a section of the fuel assembly, orthogonal to the fuel axis. The fuel assembly is formed by 90 fuel pins arranged in a hexagonal shroud with an equilateral triangular lattice 13.41 mm spaced; the external diameter of the fuel pin is equal to 8.5 mm.

Fig 3 shows a scheme of the fuel pin which is made of a foot constrained to the element lower grid, welded to the clad containing the lower plenum, the lower plug, the fuel pellets of the active zone, the upper plug, and the upper plenum; at the top the clad is closed by a head.

Foot, plugs, clad and head are made of stainless steel which, at the operating temperature (400°C) has a density equal to 7800 kg/m^3 and a Young module equal to $1.9 \cdot 10^5$ MPa. The fuel pellets are made of highly-enriched ceramic mixed U-Pu oxide, with a density equal to 10400 kg/m^3.

The Pb-Bi reactor coolant, flowing upstream parallel to the fuel pins with a velocity of 1 m/s, has a density equal to 10300 kg/m^3.

PERFORMED ANALYSES

Two different analyses were performed.

The first one, neglecting the inertial coupling effects among fuel pins and called Parametric Uncoupled Analysis, is a parametric analysis which uses the number, the position and the stiffness of spacer grids as variables to identify solution with the lowest vibration amplitude.

In table 1, the parametric analysis grid is shown.

Table 1 – Parametric analyses grid

Parameter	Values
Constraint of the pin to the lower support grid	Pinned
	Fixed end
Number and position of spacer grids	1 – 1000 mm el.
	2 – 200 mm and 1050 mm el.
	3 – 100 mm, 625 and 1050 mm el.
	3 – 428 mm, 828 and 1228 mm el.
Spacer grid stiffness	From 1 to 4000 N/mm

The second analysis, performed with reference to the best solution pointed out in the parametric analysis, takes into consideration the inertial coupling.

Parametric Uncoupled Analysis

The parametric uncoupled analysis was carried using the Chen-Weber model, which is based on the introduction of the critical velocity (v_{cp}) of the external flow:

$$v_{cp} = f_n L \sqrt{\frac{w_e}{w_s\left[(0.002L/D)+1\right]}} \qquad 1)$$

The maximum vibration amplitude is:

$$x_p = d_h \cdot \frac{2\left(v_p/v_{cp}\right)^2}{\left[1-\left(v_p/v_{cp}\right)^2\right]} \qquad 2)$$

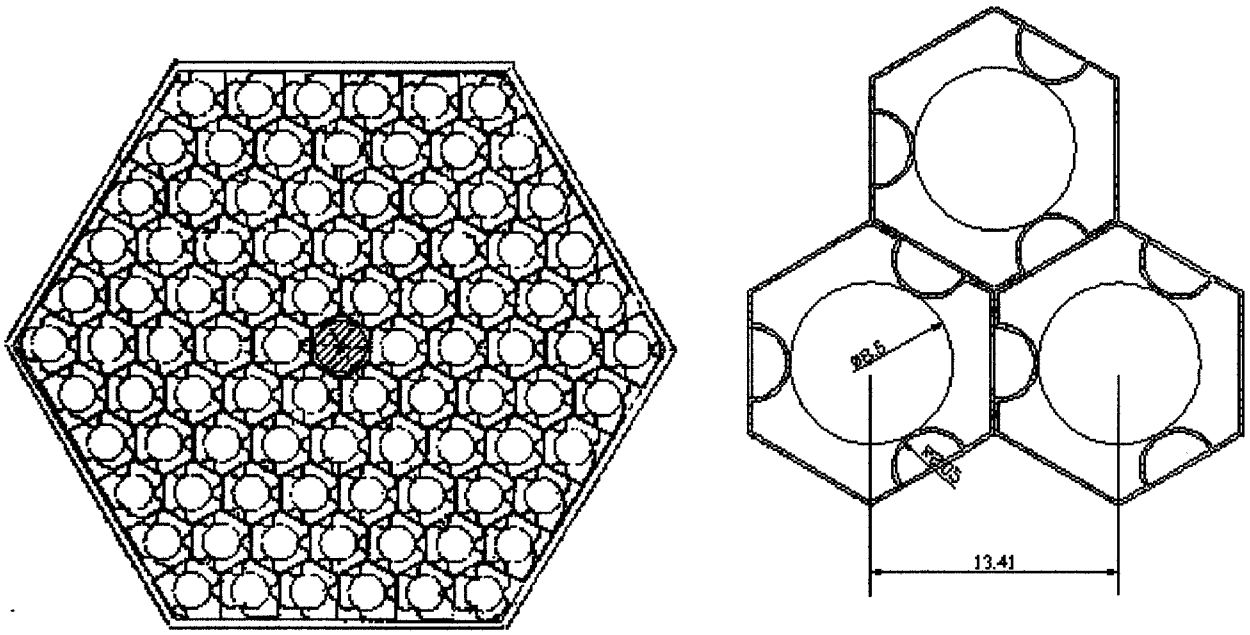
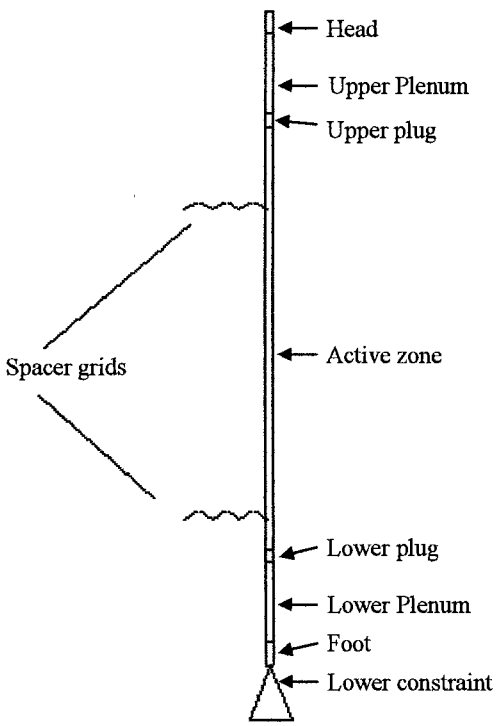


Figure 2 – Fuel assembly section and particular of spacing grids



Overall height	1266	mm
Foot height	40	mm
Lower plenum height	150	mm
Plugs height	15	mm
Active zone height	870	mm
Upper plenum height	162	mm
Head height	20	mm
Clad ext. diameter	8.5	mm
Clad int. diameter	7.37	mm
Pellet ext. diameter	7.14	mm
Pellet hole diameter	1.8	mm

Figure 3 – Fuel pin geometry

This model was developed for tubes in shell and tube heat exchangers with parallel flow; even if some differences exist between tubes in heat exchangers and fuel pins (mainly regarding diameter, thickness and fluid characteristics) it has been assumed applicable in the case of fuel pins for preliminary evaluations.

The fuel pin natural frequency was evaluated using the FEM program MSC/NASTRAN; the added mass (the participation of displaced fluid to the pin motion) was evaluated using the equation:

$$w_s = k_s \cdot \rho \cdot \pi \cdot D^2/4 \qquad 3)$$

where the added mass coefficient k_s (which depends on the lattice type and the pitch/diameter ratio) was evaluated through the curve in fig. 4.

Figs. 5 and 6 show the results of the modal analyses of the fuel pin, pinned and fixed to the lower support grid respectively.

It may be noted that, as the grid stiffness increases, pin natural frequency fastly reaches a maximum; this maximum corresponds to grid stiffnes about 200 N/mm for more than one grid solutions and to 10 N/mm for one grid solution. Therefore stiffness higher then these limits are usefull. This is due to the fact that, as the grids stiffnes increases, the pin constrained points become vibration nodes.

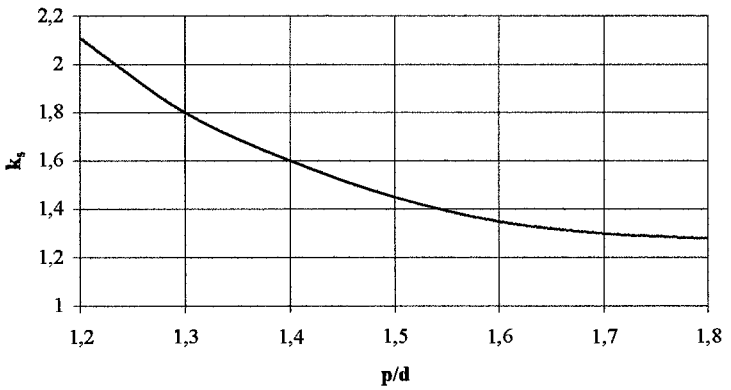


Fig. 4 – Added mass coefficient for equilater triangular lattice (Wambsganss et. Al.)

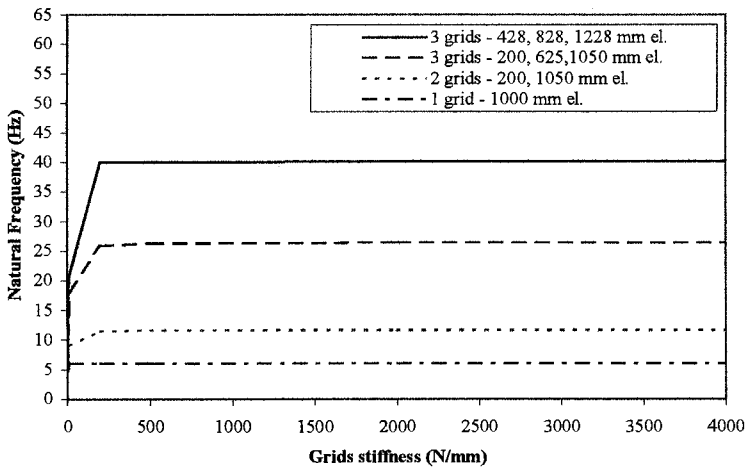


Fig. 5 – Fuel pin natural frequency (pinned solution)

The introduction of a fixed end constraint to the lower support grid, instead of pinned constraint, causes very limited benefits except than in one grid solution (where the natural frequency increase reaches 50%)

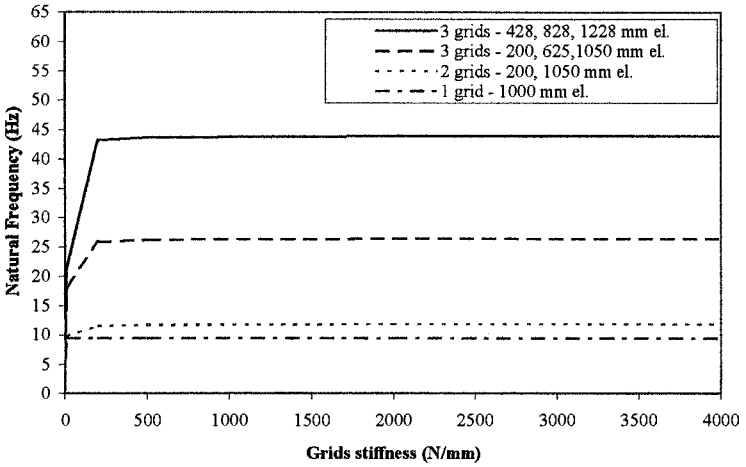


Fig. 6 – Fuel pin natural frequency (fixed end solution)

Figs. 7 and 8 show the maximum fuel pin displacements of the fuel pin, pinned and fixed to the lower support grid , rispectively (ordinates are in logaritmic scale).

Obviously lower vibration amplitudes correspond to higher natural frequency; it may be noted that only the solution with three grids located at 428, 828 and 1228 mm elevation guarantees maximum displacements lower than 0.1 mm (the obtained value is about 0.06 mm for both pinned and fixed end solutions) when grid stiffness is equal or higher than 200 N/mm.

These analyses show that almost three grids are necessary to limit pin displacements; the best grids position corresponds to an equispacing of grids along the pin length. The minimum stiffness required for spacer grids is about 200 N/mm; higher stiffness is usefull.

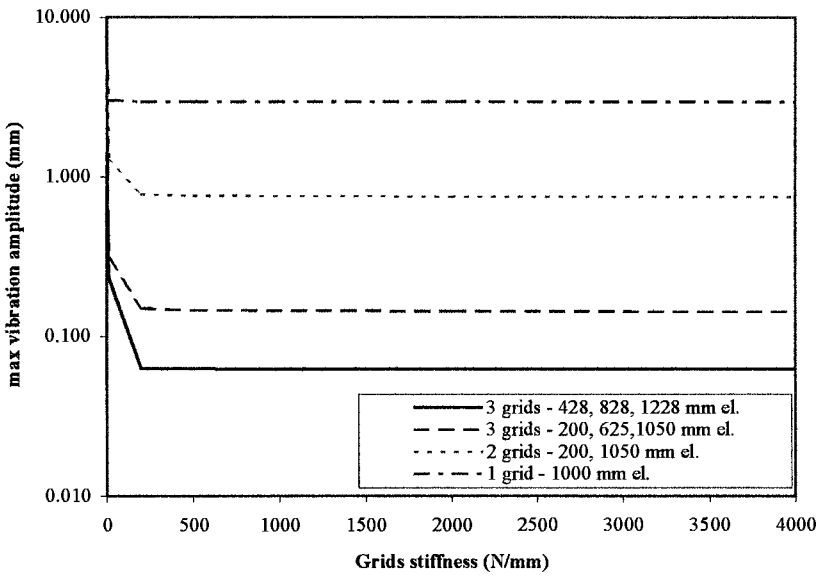


Fig. 7 – Fuel pin max vibration ampliutude (pinned solution)

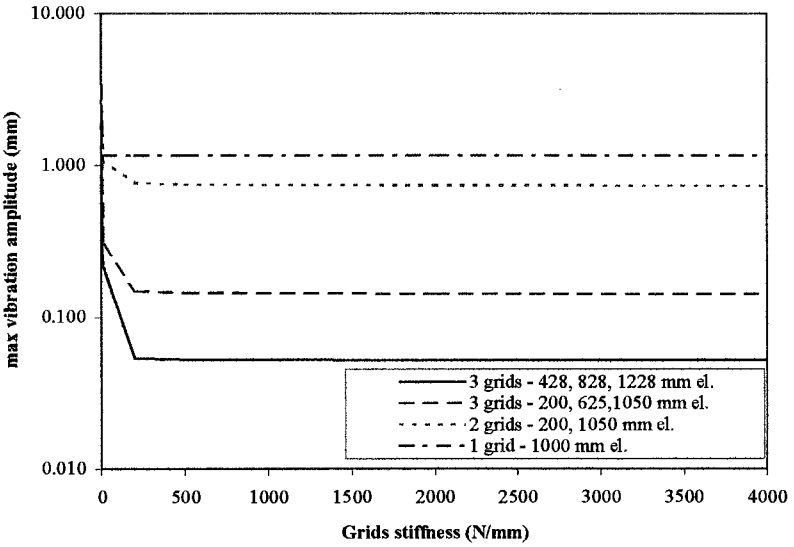


Fig. 8 – Fuel pin max vibrazion amplitude (fixed end solution)

Coupled Analysis

When a solid body vibrates in a fluid, other than the participation of a fluid mass to the body motion ("added mass effect" already introduced), another phenomenon may be present causing modifications in vibration modes due to the presence of other masses (bodies) in the same fluid ("inertial coupling effect").

To investigate the presence of inertial coupling and to evaluate its influence on fuel pin vibration amplitude, a FEM analysis was performed applied to the solution showing the best behaviour in the parametric analysis (3 grids - 428, 828 and 1228 mm el. – 200 N/mm grids stiffness).

Since this analysis is very "resource expensive" (a huge amount of memory and of storage capacity is required), a three-dimensional model of a 7 fuel pin group was made. The FEM model, which includes 2646 nodes and 4648 elements, is shown in fig. 9.

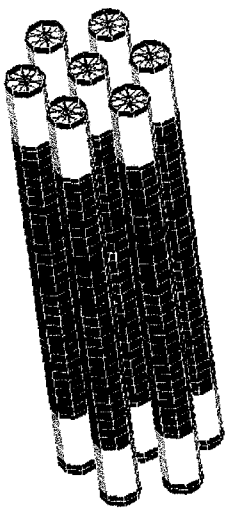


Fig. 9 – Fuel pin group FEM model (the image is compressed in the longitudinal direction)

Two modal analyses where performed using MSC/NASTRAN, the first one neglecting the presence of the external fluid, the second one taking it into consideration (hydro-elastic analysis).

Table 2 shows the results of both analyses. One can note that neglecting the presence of the fluid, the first ten modes of pins have the same frequency because they represent the motion, in two perpendicular planes, of each pin; the frequencies are higher than those previously evaluated since the added mass effect is not present.

Table 2 – Frequency of vibration modes

Mode No	Frequency	
	Fluid absence	Fluid presence
1	75.42	41.29
2	75.42	42.24
3	75.42	42.37
4	75.42	44.70
5	75.42	45.32
6	75.42	45.92
7	75.42	47.10
8	75.42	47.95
9	75.42	49.91
10	75.42	51.07

Taking into account the presence of the external fluid and allowing the program to evaluate both "added mass" and "inertial coupling" effects, the modes represent real different vibration modes of the pin group; on the other hand, the frequencies are quite lower than previous ones mainly due to the added mass effect.

Fig. 10 shows two vibration modes of the pin group.

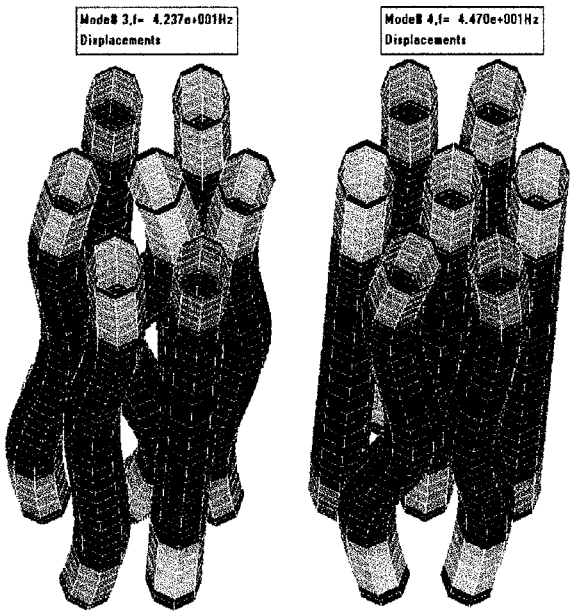


Fig. 10 – Two vibration modes of the pin group (the image is compressed in the longitudinal direction)

In addition, a frequency response analysis was performed on the pin group using the forcing frequency and amplitude obtained from the Chen-Weber model. As a matter of fact, in this model the tubes are considered as a unidimensional oscillator with:

$$f/f_n = v_p/v_{cp} \quad \text{and} \quad F = 2S\left(v_p/v_{cp}\right)^2 d_h \qquad 4)$$

So, from eq. 1), 2) and 4) and thanks to the pin flexional stiffness, evaluated by means of MSC/NASTRAN and equal to 108 mm/MPa, the forcing frequency and amplitude is equal to:

f = 1.75 Hz
F = 5.6 Pa

Fig. 11 shows results of frequency response analysis, where a maximum displacement slightly higher than 0.04 mm is obtained.

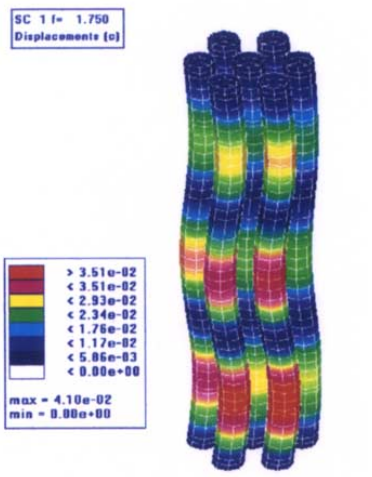


Fig. 11 – Results of frequency response analysis (image is compressed in the longitudinal direction)

CONCLUSIONS

The analyses performed allowed to identify the best number (three), position (428, 828 and 1228 mm el.) and the stiffness (200 N/mm) of spacer grids in an ADS Demo fuel assembly. They also show that the forcing frequency (1.75) is sufficiently far from the assembly natural frequency (about 40 Hz) to guarantee the absence of resonance phenomena.

The maximum displacement evaluated is about one half of millimetre tenth. Anyway, since the empirical approach used as a base for these analyses shows a confidence range of about one decade, displacements around some millimetre tenth could be expected.

Experimental campaigns should validate such results and would show if the vibration amplitude may guarantee the absence of pin damage.

NOMENCLATURE

f_n = fuel pin natural frequency (Hz)
L = fuel pin length (m)
D = fuel pin external diameter (m)
w_e = fuel pin weight per unit length (N/m)
w_s = weight of the external fluid displaced by the pin per unit lenght (N/m)
x_p = maximum fuel pin displacement (m)
d_h = hydraulic diameter of parallel flow (m)
v_p = velocity of the parallel flow (m)
ρ = external fluid density (kg/m^3)
k_s = added mass coefficient (-)
f = forcing frequency (Hz)
F = forcing amplitude (MPa)
S = pin fuel flexional stiffness (mm/MPa)

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