EXPERIMENTAL STUDIES OF A SIMULATION CANDU FUEL BUNDLE STRUCTURE IN CONFINED AXIAL FLOW

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ABSTRACT

This paper presents an experimental study of vibration of a 43-element simulation CANDU fuel bundle in an axial pipe flow. The simulation bundle is observed to rock with different amplitudes in the coolant-flowing pipe for different volume flow rates. The frequency of this motion is far lower than the fundamental frequency of bending of each fuel element, and independent of the flow velocity, which is very different from the typical case of parallel flow-induced vibration in a cluster of cylinders.

1. INTRODUCTION

Abnormal wear pressure tube has been found in the CANDU reactors. Investigations of 37-element fuel bundles removed from reactors show that the worst fretting wear happens at the burnish mark of the pressure tube around 5 o'clock and 7 o'clock positions where the first row of the bearing pads of the inlet bundle locates. Significant fretting is also found near the bearing pads at the mid-plane of the inlet bundle around 4 and 8 o'clock positions (Norsworthy et al., 1994). Studies of the wear scar patterns reveal that erosion, corrosion and mechanical interaction may contribute to the development of the scars; however impact and sliding between the bearing pads and the pressure tube induced by the gross motion of the fuel elements on the inlet bundle are the exclusive cause of pressure tube wear in CANDU reactors (Ko et al., 2003). The motion consists of the rigid body motion of the bundle and the flexible motion of the complicated structure.

Large amount of work (Chen and Wambsganss, 1972; Lin, and Jendrzejczyk, 1984; Paidoussis and Curling, 1985) were conducted on the bending motion of a single cylinder or a cluster of cylinders in turbulent flows. Each cylinder was considered as a flexible beam supported by translational and rotational springs, which represent the endplate constraint. Under a random excitation due to the fluctuating pressure field, the maximum deflection happens around the middle point of the beam, which is a natural approach to the generation of the mid-plane fretting mark; however, this model is inadequate in explaining the more severe wear at the first row of the bearing pads near the upstream end. Even at the mid-plane, turbulence is insufficient to produce the observed wear-rate of 1 μm/day by itself (Norsworthy et al., 1994; Yetisir, 1997). Some researchers examined the acoustic characteristics of the reactor components, and found that high level acoustic pressure pulsation created by the pump and amplified by pipe resonance played a role (Misra et al., 1994).

An investigation on a 37-element fuel bundle by Smith and Derken (1998) indicates that the vibration mode is a mixture of rolling and bending motion, but the most concerned frequencies are those less than 30 Hz and characterized by almost rigid-body motion, including a low frequency motion as the bundle rocks from side to side in the pressure tube. Although the bearing pads on the fuel elements at the bottom are designed to prevent the bundle from rolling, the fuel elements and the endplates are flexible and deform under the large weight of the bundle. Besides, a small manufacture or assembly error may lead to a situation that equilibrium can be easily lost with a small disturbance, and the bundle is able to roll for a small angle until extra bearing pads contact the pressure tube wall, which causes the bundle to bounce back. If the turbulent flow continuously provides sufficient disturbance in terms of wall pressure fluctuation, this motion is able to maintain. This situation does not necessarily happen on all bundles, but only on a small number. However, as observed by Norsworthy et al. (1994), all channels bear fretting marks due to the many bundles which pass through each channel. When rocking is considered, it is easier to explain the higher fretting level at the inlet end of the bundle rather than the mid-plane.

The motions of the fuel elements in a bundle are coupled through the endplates. The vibration mode of this complicated structure cannot be approached simply by the synthesis of the bending motions of individual fuel element. Therefore, a more comprehensive rocking-bending model, which includes the rigid body motion, the flexible body motion and impact between the bundle and the pressure tube, is needed for studying the flow-induced vibration of a fuel bundle. There is clearly a lack of such models in the literature.

The present work focuses on the experimental investigation of the dynamic behavior of a 43-element simulation fuel bundle in a confined turbulent parallel flow. The response of the fuel bundle to the turbulent pipe flow is measured using accelerometers. To determine the effect of tube vibration due to structure-borne waves, vibration of the pressure tube is also measured and correlated to the fuel bundle vibration. Finally, a vibro-impact bundle vibration model is proposed based on the analysis of the experimental data.

2. EXPERIMENTS

2.1 Setup

A test rig shown in Figure 1 was set up to simulate the flow condition in the CANDU fuel channel. The test rig consists of a feeder pipe, a return pipe, a clear 4″ PVC pipe and a 10 horsepower 7-vane centrifugal pump. The pump provides a horizontal flow up to a mass flow rate of 34 kg/s. A simulation bundle, as shown in Figure 1 (a), is placed inside the PVC pipe. In the simulation bundle, the fuel elements are made of solid stainless steel rods. The dimensions and material parameters are listed in Table 1 for reference. The fuel elements are assembled via two stainless endplates. Figure 1 (b) shows the arrangement of the test rig and the data acquisition system.

The fuel bundle is connected to a stainless steel supporting rod, which is designed in such a way that the fundamental bending and rotational frequencies under fix-free boundary conditions of the bundle-rod system are close to those of a string of 12 bundles. The supporting rod is clamped to a shield plug that is fixed on the tube. Water is used as the coolant at room temperature. Pressurization of 20psi is applied to avoid bubbles.

An underwater accelerometer, VibraMetrics 1801AB, is installed in the tangential direction at 1/5 diameter from the top on the outlet endplate of the bundle to pick up the rolling motion. The accelerometer has a sensitivity of 97.5mv/g in the frequency ranges of 3 Hz - 7 kHz with a maximum error of ±5%. Another accelerometer, Dytran 3035B, is mounted on the side of the pressure tube to pick up the tube vibration. Its sensitivity is 100.8mv/g and the frequency range is 0.5 Hz - 10 kHz with a maximum error of ±5%. The acceleration signals are collected by an IMC multi-channel data acquisition system at a sampling rate of 5000 Hz, and processed using an Acer Travelmate 6591 laptop computer.

2.2 Free vibration of bundle without flow

The responses of the bundle-tube system to an impact excitation were measured to obtain the natural frequencies of the bundle in air and in stationary water. Figure 2 and Figure 3 show the acceleration time histories of 3.2 seconds in length and their energy spectral density (ESD) functions in air and stationary water, respectively.

Table 1 Simulation fuel element parameters

	Outer & Intermediate Rings	Inner Ring and Centre
Diameter (mm)	13	11
Length (mm)	500	500
Modulus (GPa)	210	210
Density (kg/m^3)	7800	7800

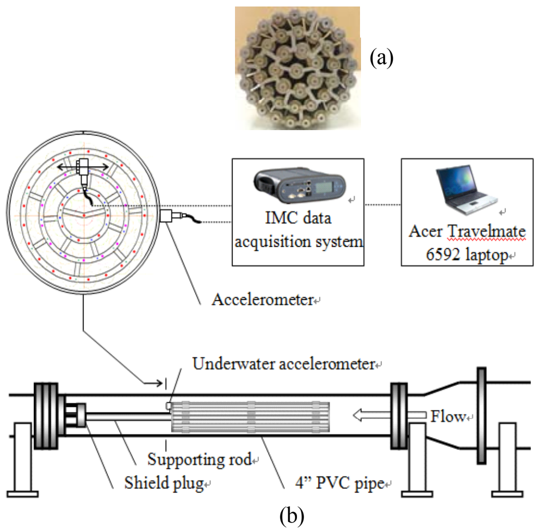


Figure 1 Test rig: (a) a simulation bundle, (b) flow channel and data acquisition system

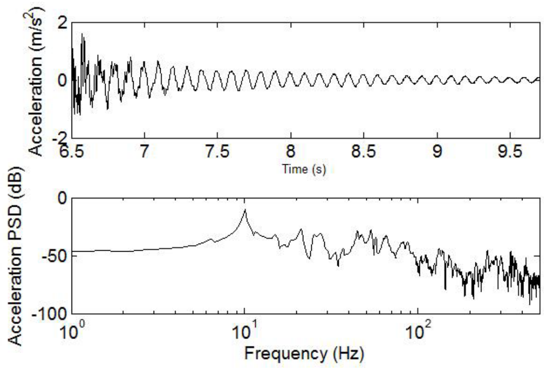


Figure 2 Time history and ESD of the response of bundle in air to an impulse

For free vibration without water, the fundamental frequency is 10.6 Hz. The motion of this mode can be seen with the naked eyes as the bundle rocks from side to side. When the flow channel is filled with water, the fundamental natural frequency reduces to 10 Hz due to the added mass of the liquid. The frequency of damped vibration is 9.1 Hz

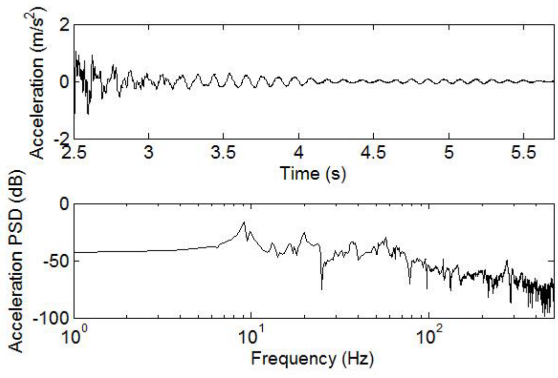


Figure 3 Time history and ESD of the response of bundle in stationary water to an impulse

2.3 Flow induced vibration

The fuel bundle and pressure tube vibrations under different flow conditions were measured. The hydraulic diameter of the bundle is calculated using

$$D_h = \frac{4A}{C} \quad (1)$$

where A is the effective cross-sectional area of the flow channel; C is the wetted perimeter. The hydraulic diameter of the flow channel that contains the bundle is 8.0 mm. The Reynolds number in the pipe flow is defined as

$$Re = \frac{U_b \cdot D_h}{\nu} \quad (2)$$

where U_b is the average axial velocity of flow over the bundle; ν is the kinematic viscosity of water, whose value is 10^{-6} at the room temperature. Table 2 shows the average flow velocity and the Reynolds numbers for nine tests. The flow is highly turbulent even with the lowest average flow velocity.

Following Paidoussis (2004), the flow velocity is nondimensionalized as

$$u = U_b L \sqrt{\frac{\rho A}{EI}} \quad (3)$$

where L, ρ, A, E, and I are the length, the density, the cross-sectional area, Yong's modulus and the cross-sectional area moment of inertia of a fuel element. The highest dimensionless velocity is u = 0.39 for type1 rod and u = 0.33 for type 2 rod, which are far lower than the critical velocity of u = 3.5 for fluid-elastic instability to occur according to Paidoussis (2004).

The power spectral density (PSD) of the bundle acceleration is shown in Figure 4 against 34 flow velocities ranging from 4.2 m/s to 11.3 m/s. Two apparent spikes appear in a range from 380 Hz to 550 Hz, decreasing with the increasing flow velocity. The amplitudes of the spikes increase and the bandwidths extend with the flow velocity. These characteristics are in agreement with those of the flow-induced vibrations of slender cylinders in parallel flow. However, the contribution of high frequency acceleration is greatly reduced when it is integrated to displacement. For small amplitude vibration, the displacement amplitude at a certain frequency f can be estimated by dividing the acceleration by $(2\pi f)^2$, and thus the influence of high frequency components vanishes. Therefore, the frequency components below 100 Hz are of particular interest.

Table 2 *Average flow velocity in the bundle and Reynolds number*

Tests	U_b(m/s)	Re(x10^5)
1	4.4	0.352
2	5.3	0.423
3	6.1	0.490
4	6.9	0.549
5	7.5	0.600
6	8.1	0.649
7	8.8	0.705
8	9.8	0.784
9	11.3	0.904

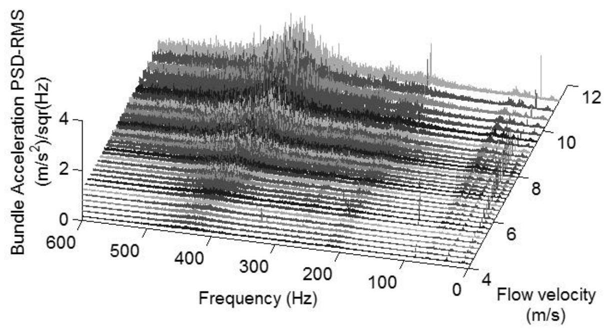
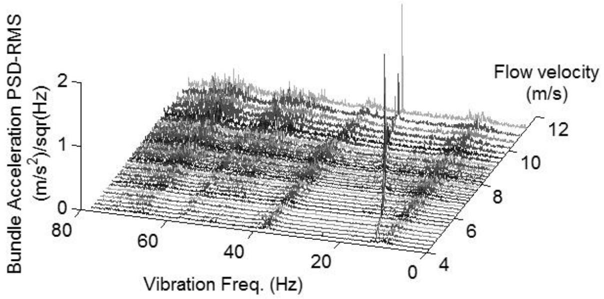
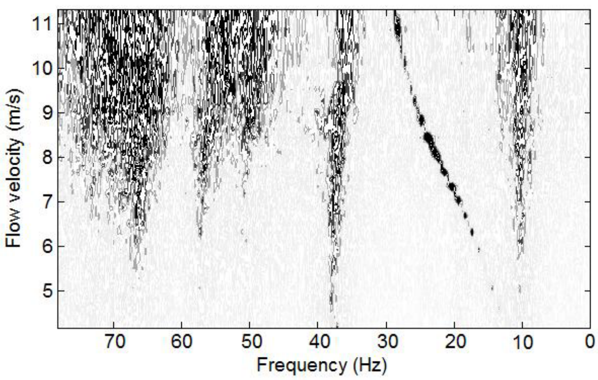


Figure 4 Acceleration PSD of the bundle vs. vibration frequency and flow velocity

Figure 5 shows the responses of the bundle and the pressure tube. The sharp spikes, which increase with the flow velocity, are recognized as the pump produced pressure pulsation, since their frequencies are exactly the pump rotating frequency and its integer multiples up to a factor of 7 (7 vanes). Figure 5 shows the root mean square (RMS) power spectral density of the bundle acceleration. Spikes can be seen at 10 Hz, 37 Hz, 50 Hz, 57Hz, and 67 Hz to 75 Hz. The central frequencies of the spikes at 37 Hz and higher frequencies drop with the increasing flow velocity, which is similar to the typical response when fluid-elastic forces are considered. On the other hand, the central frequencies of the spikes at 10 Hz remain unchanged while the flow velocity goes up. This indicates that the fundamental mode of the bundle motion is not related to fluid-elastic interaction, and thus the fluid-elastic forces can be ignored when the rocking motion is studied.



(a)

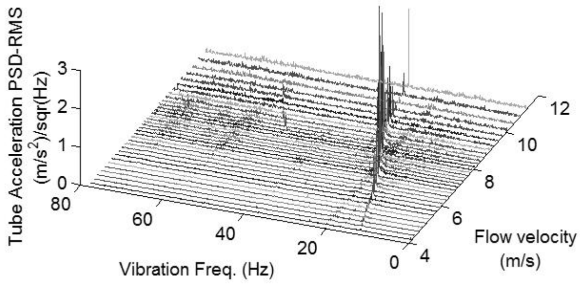


(b)

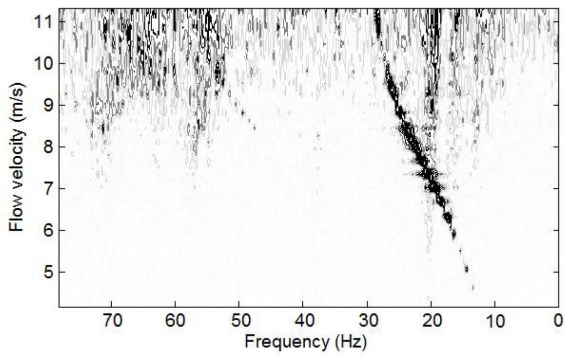
Figure 5 Bundle acceleration: (a) PSD-RMS (b) logarithmic contour after averaging

Smith and Derken (1998) have reported that the 37-element bundle is subjected to a fluctuating side force at a reduced frequency, fD/U_b = 0.11, which may cause the bundle to rock. The side force is found to have an oscillating frequency that linearly increases with the flow velocity. This force comes from the vortex-shedding of the upstream shield plug and can be reduced by changing the design of the shield plug. In the present measurement, upstream shield plug is not installed and typical vortex-shedding-like response was not found.

The PVC pressure tube has considerable flexibility. The pressure tube vibrates under the excitation of the pump and the flow. To investigate its influence on the bundle vibration, the pressure tube vibration was measured simultaneously with the bundle vibration. The root mean square power spectral density of the pressure tube acceleration is shown in Figure 6. The response is correlated to the bundle vibration and expressed in terms of the root mean square cross spectral density (CSD) in Figure 7. It is seen that most of the pressure tube vibration energy is associated with the pump-induced frequencies. The CSD shows almost no correlation between the tube and the bundle at the fundamental mode (10 Hz). This gives confidence that the 10 Hz mode is flow-induced vibration.



(a)



(b)

Figure 6 Pressure tube acceleration (a) PSD-RMS; (b) logarithmic contour after averaging

3. VIBRO-IMPACT MODEL

3.1 Mechanism of the rocking motion

The experimental data show that the main flow-induced bundle vibration mode is a rocking-like motion. This motion is not apparently related to fluid-elasticity and vortex shedding. It is more likely caused by the flexibility of the bundle and its weight. The fuel elements in the outer ring around the 6 o'clock position bear the weight of the whole bundle. They are supported by the two bearing pads near both ends. The bearing pads on the mid-plane are not active since the beams bend towards the center of the bundle under the weight. This can be imagined as a large cylinder supported by springs on its circumference, because the beams have bending stiffness. If a disturbance is applied to the cylinder on the side, the cylinder will shake in a direction perpendicular to its axis. The bundle is more flexible than a rigid cylinder and may have a lower vibration frequency. It must be pointed out that at most of time, the bundle is supported by two fuel elements and thus four bearing pads. It is observed that not all of the bearing pads always keep contact with the pressure tube wall. It is more often that three of them are active while the other one is inactive. This produces a situation that the bundle is sitting on a three-point support with two of the supports very close (a distance of 14 mm) while the other one is at a distance 30 times of the distance between the two.

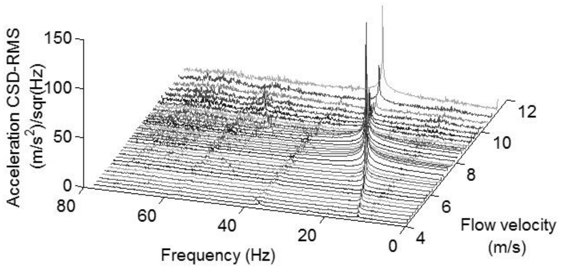


Figure 7 Bundle and pressure tube acceleration CSD-RMS

Considering that the diameter of the entire bundle is about 99 mm, equilibrium is fragile under such a situation. If the disturbance is sufficiently large, the bundle will leave its stable equilibrium position and even pass the unstable equilibrium position. In this case, the bundle will roll under the effect of gravity until another bearing pad contacts with the pressure tube. Changing the supporting bearing pads will change the boundary conditions. The accurate rocking frequency can be hardly obtained because it is accompanied by contact. In addition, friction must also be considered, because the bundle will slide if the friction forces between the bearing pads and the pressure tube are insufficient to hold the bundle. Sliding will further complicate boundary conditions. Therefore, a finite element elastic contact model of the bundle is in need.

3.2 Excitation forces

Turbulence has been considered as the exclusive excitation source of the bundle vibration. Chen and Wambsganss (1972) predicted the wall pressure power spectral density on a rod in confined axial flow. Curling and Paidoussis (1991) developed more elaborated equations to predict the wall pressure cross spectral density in a cluster of cylinders; but the approximation is valid only for a Strouhal number higher than 0.2, which excludes the case of the fundamental mode in the above experiment. For lower Strouhal number pressure fluctuations, Wilson and Jones (1983) found that the wall pressure power spectral density is flat down to a Strouhal number about 0.008, rather than sloping as predicted by Chen and Wambsganss (1972). A numerical simulation of a fuel element cluster conducted by Kim and No (2004) using the Large Eddy Simulation (LES) turbulent model in Fluent 6 shows the same flat trend of the wall pressure spectra in low Strouhal number range. There are very limited experimental data of wall pressure fluctuations in a 37-rod or a 43-rod bundle in the open literature. The small diameter of the fuel element makes measurement almost impossible without altering the flow. Therefore, numerical simulation becomes valuable.

4. CONCLUSIONS

This paper presents the measurement of the flow-induced vibration of a 43-rod simulation bundle. Acceleration responses of the bundle and the housing pressure tube have been recorded with 34 different velocities ranges from 4.4 m/s to 11.3 m/s. The bundle is found to have a predominant rocking motion at a low frequency (10 Hz), characterized by a mixture of flexible body quivering and rigid body rolling. The frequency of the rocking motion is not affected by the flow velocity, which excludes the fluid-elasticity and/or vortex shedding induced vibration. Based on the experimental results, a vibro-impact model that involves rigid body motion, flexibility, contact and low Strouhal number turbulence excitation is considered necessary to analyze the flow-induced vibration of a fuel bundle

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