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A MODEL FOR A FULLY-FLEXIBLE FUEL BUNDLE DYNAMICS

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ABSTRACT

A comprehensive numerical framework is proposed to model the flow-induced vibrations of a fully-flexible fuel bundle. The model includes the representations of 37 fuel rods and the two endplates. Each fuel rod is modelled by beam elements each of which has 12 degrees of freedom (2-node element and each node has 6 degrees). Each of the endplates is discretized into a number of plate elements. Contact between various systems components was modeled using the pseudo force contact method. Potential contact situation included fuel-to-pressure tube and fuel-to fuel-contact. Using this technique simulations of the dynamics of a fully flexible fuel bundle including the flexibility of the endplates were conducted.

NOMENCLATURE

$[C]$	Damping matrix
$\{F(t)\}$	External force vector
$\{F_{imp}\}$	Impact force vector
$[K]$	Stiffness matrix
$[M]$	Mass matrix
Cr	Radial Support clearance
F_c	Contact force
F_e	Nodal external force
F_{fr}	Friction force
Kc	Contact stiffness
r	Relative position vector
V_o	Limiting Velocity
$\ddot{w}, \dot{w}, w$	Acceleration, velocity and displacement
δ	Overlap
μ_k	Kinetic coefficient of friction
μ_s	Static coefficient of friction

INTRODUCTION

CANDU (CANadian Deuterium Uranium) reactors is a Canadian pressurized heavy water reactor that is being used in several countries. The reactor contains a large number of fuel channels. A string of 12 bundles is placed in the pressure tube inside each the fuel channel. The fuel bundle consists of 37 fuel rods and is held together using two endplates (see Fig. 1). In order to transport the huge amount of heat that results from the nuclear reaction, a high flow of pressurized heavy water is used. This flow through the fuel bundle can induce significant vibration levels. Such vibrations can compromise the integrity of the fuel bundles in the form cracking at endplates or fretting damage on the inside surfaces of the pressure tube.

Several research works have been reported in the literature to predict the response of fuel bundles. Païdoussis [1] proposed a mathematical model to investigate the response of vertical string of fuel bundles due to axial flow. Yetisir and Fisher [2] developed a finite element model for the fuel rod to predict the dynamic response and fretting wear due to contact with pressure tube using turbulence forces as excitation mechanism. They also performed experiments to validate the numerical work.

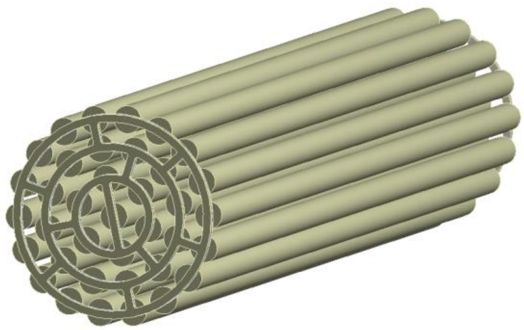


FIGURE 1: 3-D MODEL OF FUEL BUNDLE.

However, they found that the fretting wear caused by turbulence is not enough to be consistent with amount of wear observed in reality. Hassan and Rogers [3] developed a model to simulate a loosely-supported fuel elements subjected to turbulence excitation. They predicted the response and the work rate for preloaded fuel-elements but they considered only one bearing pad in contact with pressure tube. Mohany and Hassan [4] presented a numerical model to predict the response and the associated fretting wear of fuel elements due to two excitation mechanisms: turbulence excitation and seismic events. They included in their model the side spacers of the neighbouring fuel-elements and the effect of endplates by adding a rotational stiffness at the end of fuel element. Elbanhawy *et al.* [5] made further investigation by studying the effect of flow velocity and clearance on the impact force and work rate. Other efforts considered the 3-D finite modelling of the fuel elements and endplates. Example of these models such as those of Fadaee *et al.* [6] and Cho *et al.* [7] aimed at modelling the load transfer, drag loading and elements deformation.

This study propose a comprehensive finite element model to model the flow-induced vibrations of a fully-flexible fuel bundle that contains 37 fuel rods and two endplates. The fuel rods are discretized using 12 degrees of freedom beam elements while the endplates is discretized using plate elements. The ultimate goal is to consider the possible 9 contact locations for each fuel rod including; the side spacers between two neighbouring fuel-elements and the bearing pads between the outer fuel elements and pressure tube. The contact forces are calculated using the pseudo force approach. The simulations consider excitation mechanisms including turbulence and pressure pulsation.

FINITE ELEMENT MODEL

To discretize such complex structure, as shown in Fig. 1, 3D block elements is usually utilized but it results in a very large number of elements and nodes. Such approach were utilized successfully in investigating the static response of the fuel system. However, for the case of simulating dynamic response for large number of time steps, such approach needs very large running time. Therefore, a reduced-order model is utilized in this work. The model represents the fuel rods by 30 beam elements each of which has 12 degrees of freedom. In addition the endplates is discretized by 1222 plate elements as shown in Fig. 2. A four-node isoparametric quadrilateral element was used to model both endplates. The 37 rods were coupled with the two endplates by utilizing common 74 nodes between the beam element and plate element.

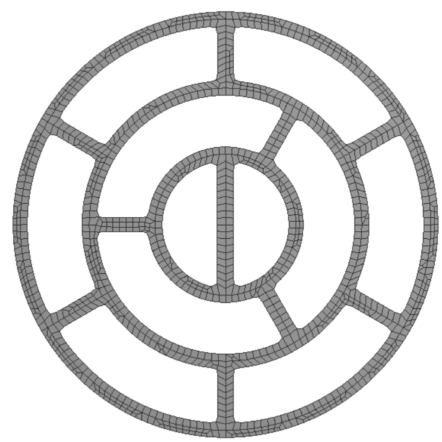


FIGURE 2: ENDPLATE DISCRETIZATION.

The governing equation to describe the dynamics in matrix form is as follow:

$$[M]\{\ddot{w}\} + [C]\{\dot{w}\} + [K]\{w\} = \{F(t)\} + \{F_{imp}(\dot{w}, w, t\} \qquad (1)$$

where M, C and K are the global mass , damping and stiffness matrices, respectively. $w, \dot{w}$ and $\ddot{w}$ are displacement, velocity and acceleration response vectors, respectively. $F(t)$ is the external fluid excitation that includes turbulence excitation, pressure pulsation, etc. F_{imp} is the impact force vector resulting from tube-to-tube and tube-to-pressure tube contacts including friction force at these sites. An in-house finite element code was used to solve the governing equation and to calculate the dynamic response. The mass, damping and stiffness matrices are calculated for both types of elements and then assembled in global matrices. The modal superposition technique was utilized to synthesis the response. The time integration scheme used in this code is NEWMARK scheme.

TURBULENCE EXCITATION

The turbulence forces are obtained by applying inverse fast Fourier transformation (FFT) to the turbulent bounding spectra of the fuel elements. This bounding spectra was obtained experimentally by Smith and Derksen [8]. An example of the turbulent forces resulting from this technique is shown in Fig. 3. Two force records were generated for each element. This required the generation of 74 force records. For simplicity, the turbulence excitation is assumed to be fully correlated along the fuel element length. This a conservative assumption. For an in-depth discussion of the efficient models of turbulence excitation with finite correlation length, the reader is referred to the excellent work of Antunes *et al.* [9].

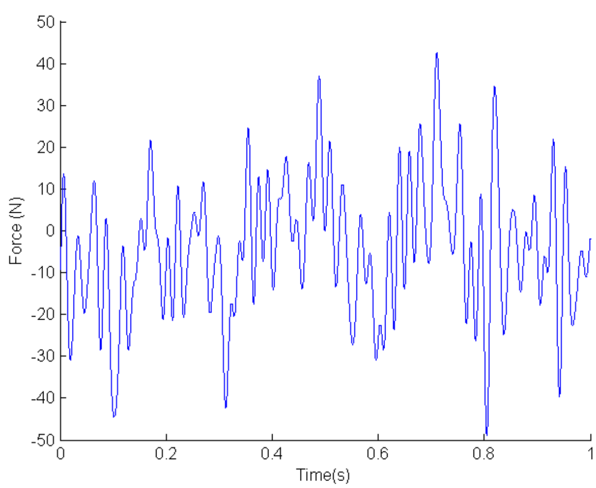


FIGURE 3: EXAMPLE OF TURBULENT FORCES.

CONTACT FORCES

Modlelling the contact forces was proposed by Hassan *et al*. [10]. They modelled the support by introducing a massless bar connected to spring-damper system at the contact site. In the current study this model is utilized to account for contact between pairs of tubes as presented in Fig. 4. Each tube pair is defined by the tangential unit vector ($\hat{e}_t$), relative position unit vector ($\hat{r}_{i,j}$) and clearance (Cr). The contact force can be written in the form:

$$F_{c,ij} = \left(-\left(Kc_{i,j}\,\delta_{i,j}\right) + sign\left(\dot{\delta}_{i,j}\right)\left(1.5\,\alpha\,\left|Kc_{i,j}\,\delta_{i,j}\right|\right)\right)\hat{r}_{i,j} + \left(sign\left(\dot{\delta}_{i,j}\right).F_{fr}\right)\hat{e}_t \quad (2)$$

where the overlap is defined as:

$$\delta_{i,j} = w_i + w_j - Cr_{i,j} \quad (3)$$

where w_i and w_j are the displacement projection of tube i and j along the relative position unit vector $\hat{r}_{i,j}$. The damping effect is considered in the coefficient α which is related to the coefficient of restitution [3]. For friction force in case of contact with pressure tube, the force-balance friction model that proposed by Hassan and Rogers is used [3]. In this model the sticking friction is checked when the absolute tangential velocity is less than a small limiting velocity (V_o). In addition to assure the occurrence of sticking, F_{fr} must satisfy the inequality ($F_{fr} \leq \mu_s F_n$). So in case of sticking the friction force will be:

$$F_{fr} = Kc\,(\vec{w}.\hat{e}_t) - F_e.\hat{e}_t \quad (4)$$

And in case of sliding:

$$F_{fr} = -\mu_k|F_n|.\hat{e}_t \quad (5)$$

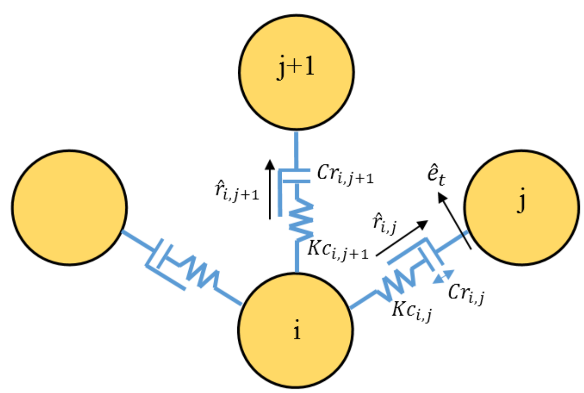


FIGURE 4: CONTACT MODEL.

PRESSURE PULSATIONS

Pressure pulsations generated by pumps can be very sever and cause damage to piping system or any device connected to the piping system such as the fuel bundle in CANDU reactor. For a centrifugal pump the pressure pulses forms when the vane passes the cutwater. So the frequency of the pulsations will be equal to the vane passing frequency which is the impeller frequency multiplied by the number of vanes [11]. For a pump running at 1800 RPM and an impeller may have five or seven vanes, the calculated pulsations frequency would be 150 and 210 Hz. Koehn *et al*. [11] conducted an experimental work on a typical Darlington fuel channel (Darlington power station, Ontario, Canada) to measure the pressure pulsations. They found that the pressure range at 150 Hz is from 40 to 100 kPa. In this study the pressure pulsations is added as a sinusoidal pressure wave acting in the axial direction of the bundle.

RESULTS

Simulations performed in this study utilize the geometric and material properties that was used by Mohany and Hassan [4]. The fuel element is 0.0131 m in diameter and 0.495 m in length. The material properties are 80 GPa Young's modulus and 0.35 Poisson ratio. Preliminary simulations were performed to investigate the impact force, the tube response and the stresses on the end plate. Five different cases will be simulated. Case (0) simulates the full fuel bundle without taking into consideration any impact. The response under turbulence excitation for outer ring fuel element as a part of the fully flexible bundle was investigated. The bundle was subjected to axial coolant flow of 10 m/s. Fig. 5 depicts the tube trajectory in z-y plane respectively (x- axis is the axial direction). The response was random with maximum amplitude of $\pm$0.6 mm.

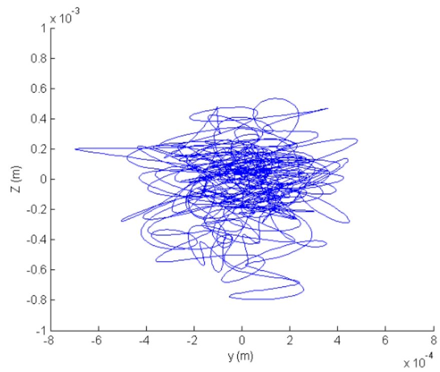
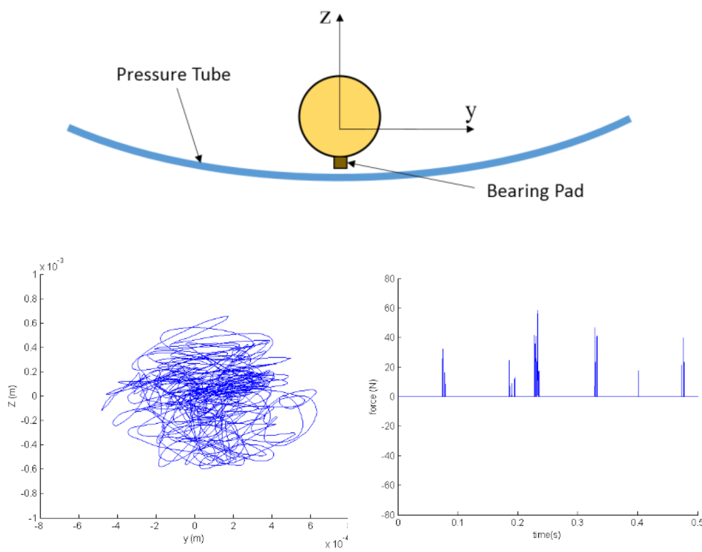


FIGURE 5: TUBE TRAJECTORY IN Z-Y PLANE AT MID SPAN FOR CASE (0).

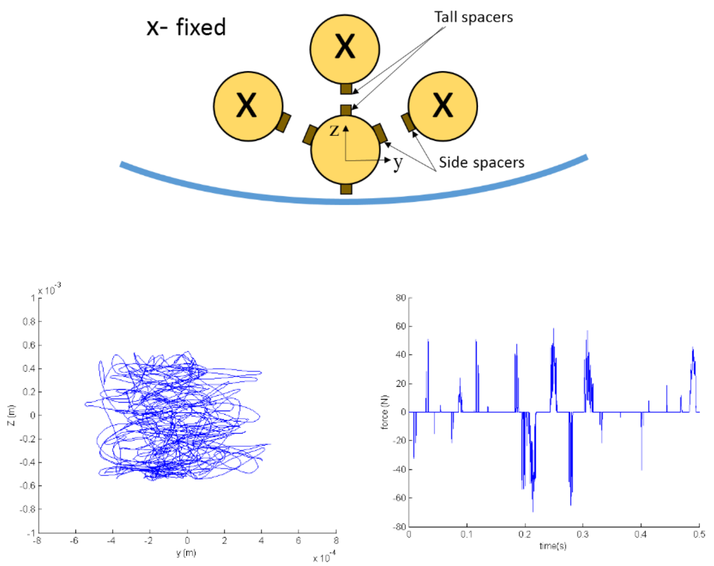
Case (1) is set to add the effect of the contact at bearing pad at mid-span with clearance of 0.5 mm. The effect of the bearing pad on the response is shown in Fig. 6 (a) and it is clear that the displacement is banded by 0.5 mm in the direction normal to the pressure tube. The resulting impact force is plotted in Fig. 6 (b). Three spacers were added in Case (2) as depicted on Fig. 4. In this case the neighbouring tubes is considered fixed with 0.5 mm clearance the same as the bearing pad. The tube trajectory and impact force of case (2) is presented in Fig. 7 (a) and (b). This a case of two-sided impact in both z and y directions. The component of y is too small and is of the order 1 N compared to the 45 N in the z direction which means for this clearance the tube barely touch the side tubes. The results of Case (1) and (2) are in good agreement with the work of Mohany and Hassan [4]. On the other hand, in Case (3) the effect of neighbouring fuel elements is considered by the interaction due to motion of the whole patch of fuel elements. The distribution of the measured clearances for side and tall spacers was reported by Mohany and Hassan [4]. For this case the clearances were choosed to be 0.3 mm for side spacers and 0.5 mm for the tall spacer and bearing pad. For case (3) the trajectory of the four tubes are plotted together in Fig.8 to show the interaction between tubes. This figure shows that the center tube is highly affected by the side tubes and the contact with pressure tube. Fig. 9 shows that contact force on the tube 3 was very small compared to other tubes. However, impact force at tube 1 is much more intense and frequent than that in Case (1) and (2).



(a) (b)

FIGURE 6: CASE (1) (a) TUBE TRAJECTORY IN Z-Y PLANE. (b) IMPACT FORCE.

The pressure pulsations is added in Case (4) as sinusoidal wave of 150 Hz frequency and 50 KPa amplitude. The stresses on the end plate are calculated for Case (3) and (4). The root mean square (RMS) of the Von Mises stress for Case (3) is shown in Fig. 10 and for Case (4) is shown in Fig. 11. The stresses in Case (3) are due to the bending moment that results from deformation of the 37 tubes but in case (4) the stresses due to the pressure pulse is added. Both cases shows that the ribs connecting the inner ring and intermediate ring have spots of maximum RMS of the stress. However in case (4) large spots in the inner ring subjected to high RMS of the stress. These results suggest that the inner ring and the connecting ribs are possible places to have cracks which is consistent with the reported damage in fuel bundles in the case of Darlington reactor [12].



(a) (b)

FIGURE 7: CASE (2) (a) TUBE TRAJECTORY IN Z-Y PLANE. (b) IMPACT FORCE

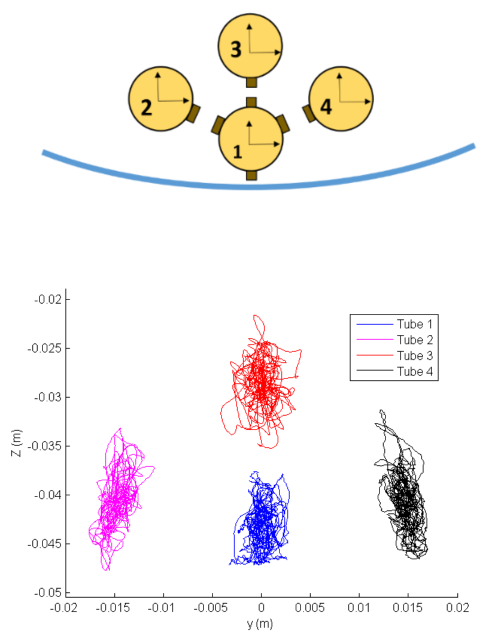


FIGURE 8: TRAJECTORY OF THE FOUR TUBES IN Z-Y PLANE FOR CASE (3).

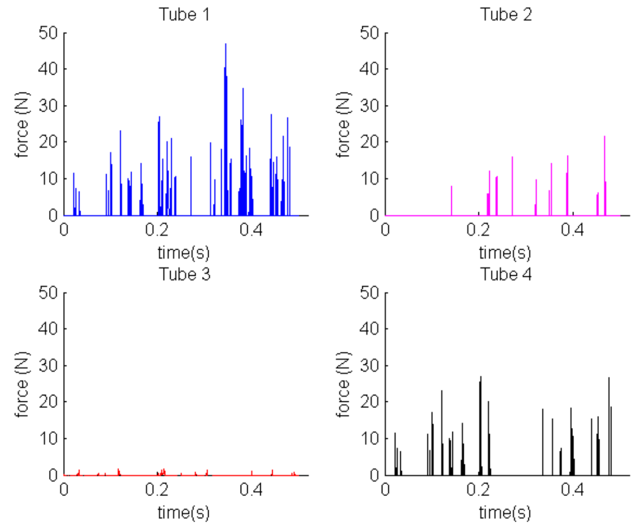


FIGURE 9: IMPACT FORCE ON TUBES FOR CASE (3).

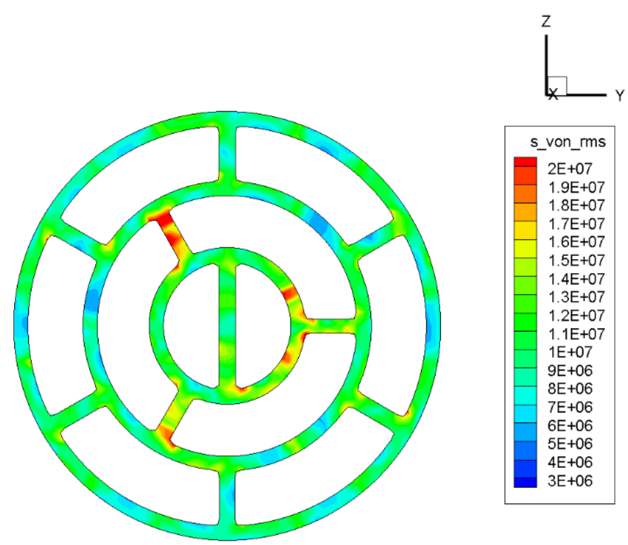


FIGURE 10: RMS OF THE VON MISES STRESS FOR CASE (3).

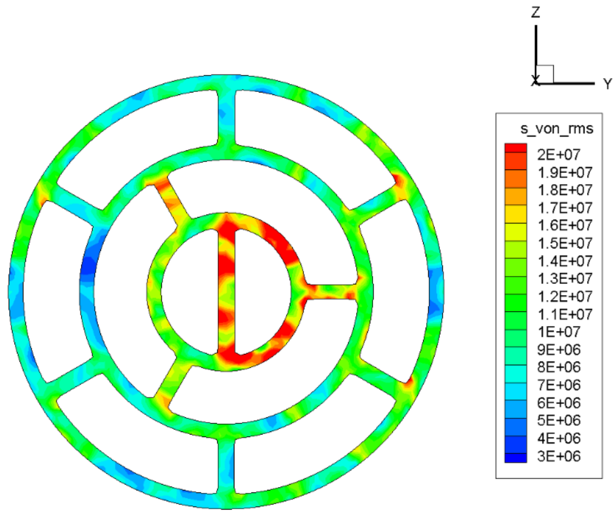


FIGURE 11: RMS OF THE VON MISES STRESS FOR CASE (4).

CONCLUSION

A comprehensive model for a fully flexible bundle dynamics is presented in this study. The model consider the 37 elements as beam elements and the endplates is divided to number of plate elements. The model includes the turbulent excitation and the effect of pressure pulsations. Using a pseudo force technique the contact forces at bearing pad and spacers are investigated. Stresses on the endplates due to the bending moment results from the elements deformation and pressure pulsations is calculated.

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