RELATIONSHIP BETWEEN NATURAL FREQUENCIES AND PULL OUT FORCE FOR PLATES WITH TWO CLAMPED EDGES

Mauro Caresta[1] and David Wassink[2]

[1]School of Mechanical & Manufacturing Engineering, University of New South Wales (UNSW), Sydney NSW 2052 Australia

[2]Australian Nuclear Science and Technology Organisation (ANSTO), Lucas Heights, NSW 2234, Australia

Plate type nuclear fuel assemblies consist of a box like structure containing several thin rectangular fuel plates clamped to the side plates by a swage connection. In cooling these fuel assemblies during reactor operation, the coolant flow causes a longitudinal drag force that may shift the plates if they are not properly clamped. The aim of this work is to find a relationship between the natural frequencies of a plate with clamped edges and the pull out force needed to shift a plate from its designed position for various different clamped conditions. The results can be used in the future to assess the quality of the swages in plate-type nuclear fuel assemblies. An experimental rig consisting of a plate clamped along two opposite edges using bolted beams tightened to various torque settings to emulate the swage quality was built. Modal analysis was performed to relate the natural frequencies to the torque used to fasten the bolts.

INTRODUCTION

Many research nuclear reactors utilise plate-type fuel assemblies constructed as a box-type assembly. In a typical fuel assembly the plates are inserted into slots machined into the side walls of the fuel box. The clamping of the plates to the box is generally assured by a swage between adjacent plates. Fuel assembly plates are likely to be affected by structural instabilities due to the interaction with the coolant flow [1-3]. Many researchers investigated both the static and dynamic behaviour of such fuel assemblies using wide beam theory [4] or using the thin plate theory with simply-supported boundary conditions [5] or fully-clamped edges [6]. In previous work [7], the authors showed that the boundary condition of a plate with an edge fixed by a swage can be modelled assuming a perfect clamp of all the degrees of freedom (dof) except for the rotation around the axis parallel to the swage which is elastically restrained with a torsional spring, giving a theoretical and experimental justification of the model used by Kim and Davis [8]. It is evident that a poor swage will not restrain the plate from shifting axially due to the force of the coolant flow. The purpose of this work is to relate the force needed to slide a poorly clamped plate to the natural frequencies of the plate itself. In the first step, a relationship is found between the natural frequencies and the torque applied to fasten the bolts that clamp the plate. The second step is to relate the fastening torque to the pull force to shift the plate. The results from the previous experiments are then combined to find the relationship between natural frequencies and pull out force. The results can be transferred to a real fuel assembly to estimate the resistance of the swaged fuel plates to the drag force of the coolant flow.

FEM OF THE CLAMPED PLATE

A finite element model was built to predict the natural frequencies of the clamped plate. The edges are restrained fixing all degrees of freedom except for the rotation around the axis parallel to the swage which is elastically restrained with a torsional spring [7]. The plate is modelled using 4 node plate elements (QUADR element). The boundary conditions are modelled using 6 dof spring elements (BUSH element). Five dofs are fixed using a large value of stiffness while the different values for the 6th dof are used to simulate the conditions between a perfect clamp ($K_s \rightarrow \infty$) and a simple support ($K_s = 0$). Results are presented introducing a frequency parameter λ_n given by $\lambda_n = \omega_n a^2 \sqrt{\rho h/D}$, where ω_n is the natural frequency of the plate and ρ is the volume density of the material. D is the flexural rigidity given by $D = Eh^3 / 12(1-\upsilon^2)$ where E and υ are, respectively, Young's modulus and Poisson's ratio. a is the width of the plate of thickness h. Only modes with increasing wave number along the length are considered since they can be found at low frequencies and result in a large displacement. The first 4 modes for the case of a perfect clamp are shown in Fig. 1. Changing the boundary conditions from a perfect clamp towards a simply supported condition, the mode shapes appear very similar except showing a larger rotation close to the edges. The plate has a very high aspect ratio (b/a) and for this reason the lower natural frequencies are very close to each other. In the real plate, the damping increases the modal coupling making it difficult to observe the lower order natural frequencies.

The 1st, 4th, 7th and 9th frequency parameters were normalised with respect to the perfectly clamped case and are plotted against the spring stiffness in Fig. 2. They have been chosen since they are more widely spaced. Observing the slope of the curves it can be seen that the sensitivity to spring stiffness decreases with increasing modal order.

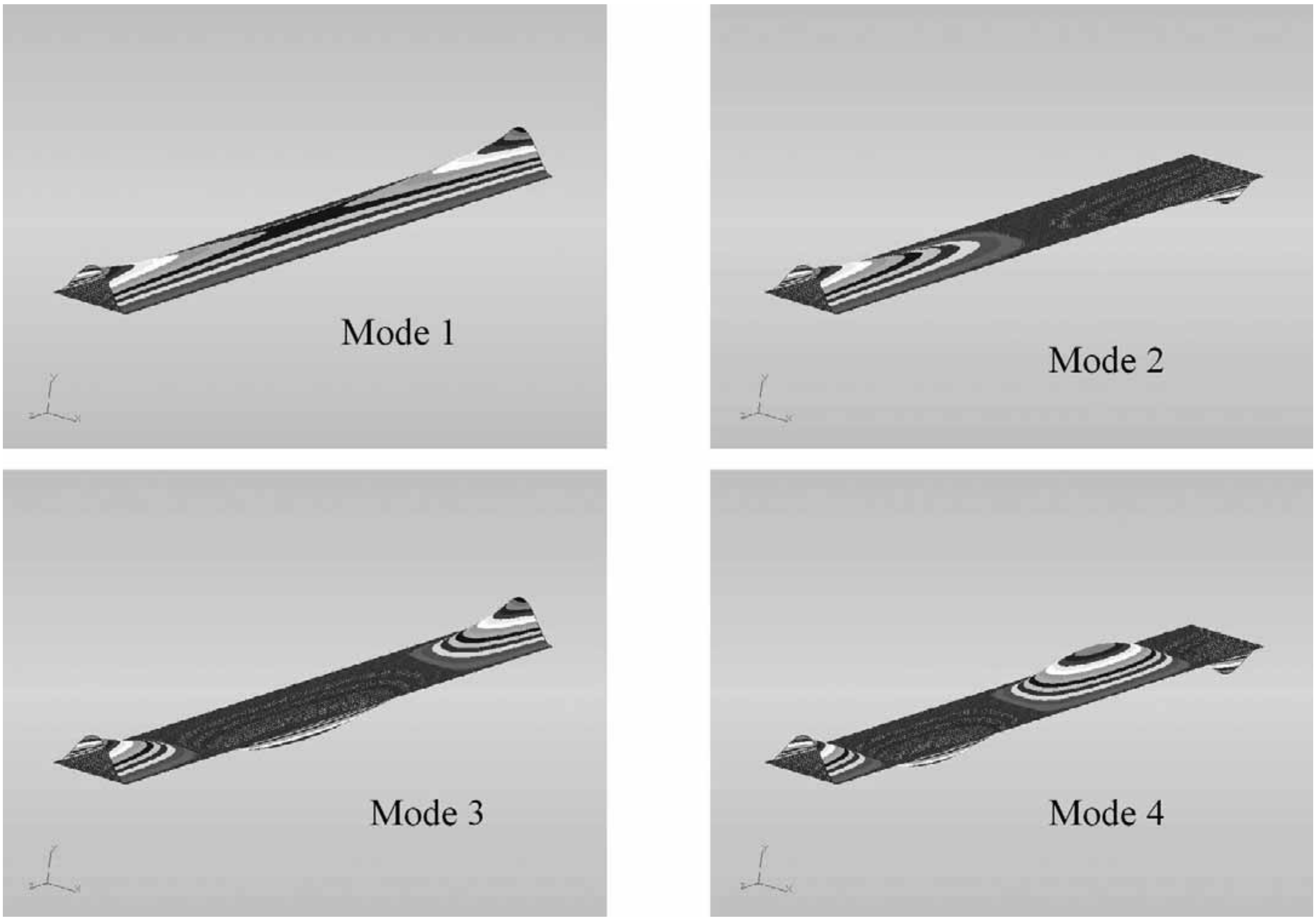

Figure 1. Modeshapes for a perfectly clamped plate (finite element results).

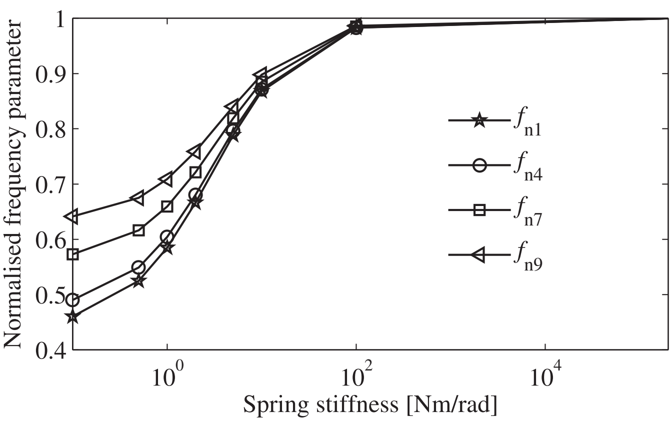

Figure 2. Variation of the natural frequencies with the spring stiffness (finite element results).

EXPERIMENTAL MODEL OF THE CLAMPED PLATE

A scale model of a fuel plate as found in a fuel assembly for the ANSTO research reactor 'OPAL' was manufactured keeping the same length for the edges but using a different thickness corresponding to commercially available sizes. Similar theory [9] allows us to easily transfer the results using materials of different thickness. The aluminium plate is restrained in the jig on the long edges by means of bolts as shown in the sketch of Fig. 3.

a and *b* are the width and length of the plate of thickness *h*. The plate is held between two thick beams detailed to recreate a situation similar to the one found in the fuel assemblies. The top beam has a fixed swage detail and the bottom beam has a small rebate where the plate is located (Fig. 4). The pressure and the clamping strength is adjusted by the fastening torque of the bolts. A wire is also used as a pivot to allow a certain contact of the top beam swage detail to the plate when the bolts are fastened.

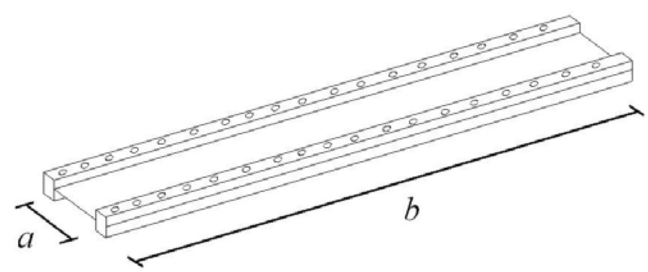

Figure 3. Model of the clamped plate

Modal analysis was performed by exciting the plate with an impact hammer and measuring the response on 21 equally spaced points on the plate using a laser vibrometer. Frequency response functions were calculated using a Brüel & Kjær Pulse signal analyser and imported into MEScope to calculate the mode shapes and determine the natural frequencies of the plate. It was difficult, when viewing the FRF, to distinguish the lower natural frequencies due to the high modal coupling mentioned before. A loose connection results in a higher damped structure since part of the vibration energy sinks at the boundaries. Focus was then shifted to the 8^{th} and higher modes. Table 1 reports the experimental and finite element method (FEM) frequency parameter, showing good agreements between the results. Figure 5 shows the 9^{th} mode shape obtained experimentally compared to the 9^{th} mode shape obtained using finite element modelling.

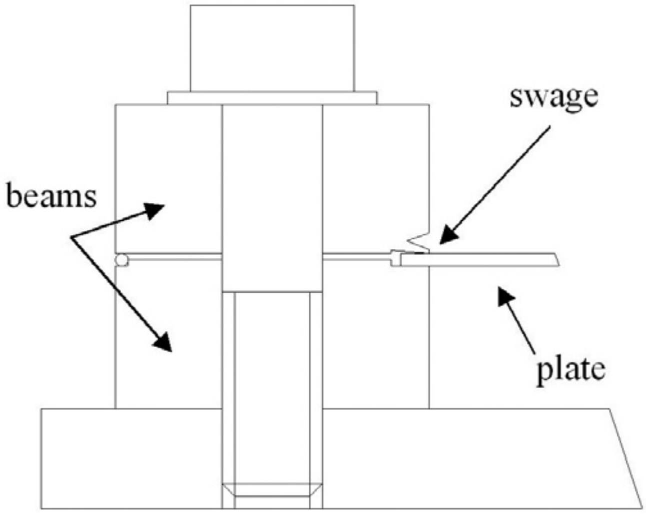

Figure 4. Particulars of the clamped edge.

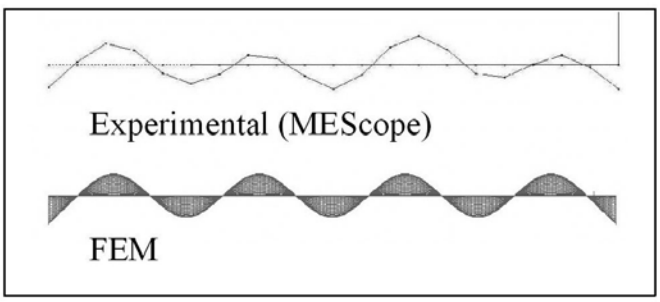

Figure 5. Mode shape of the 9^{th} natural frequency.

Table 1. Experimental and FEM results for the frequency parameter λ_n

Modal order	Experimental	FEM	% diff
8	28.2117	28.2345	-0.1
9	29.5605	29.5377	0.1
10	31.3209	31.0923	0.7
11	33.1499	32.8526	0.9

VARIATION OF FREQUENCY PARAMETER WITH FASTENING TORQUE

Modal analysis was performed using different values of the fastening torque on the bolts. A perfect clamp condition was achieved with around 8 Nm of torque applied onto the bolts. Figure 6 shows the variation of the 9^{th} frequency parameter with the fastening torque. The maximum sensitivity is found for low values of the torque where the frequency parameter increases following a steep curve. For each side there are 23 identical bolts that are fastened one after the other, the resulting preload can be assumed to be reasonably constant from bolt to bolt.

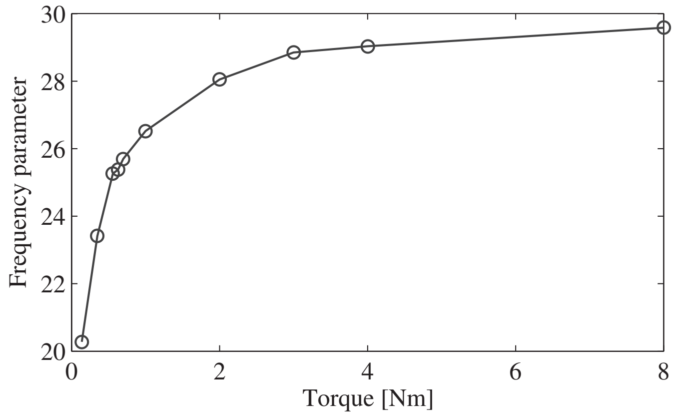

Figure 6. Experimental variation of the frequency parameter with the fastening torque.

PULL OUT EXPERIMENT

The plate clamp was mounted vertically on a solid structure and weights were applied to the bottom edge of the plate by means of a special clamp as shown in Fig. 7. Weights were slowly applied until the plate started to shift. A maximum load of up to 90 kg was able to be applied to the plate.

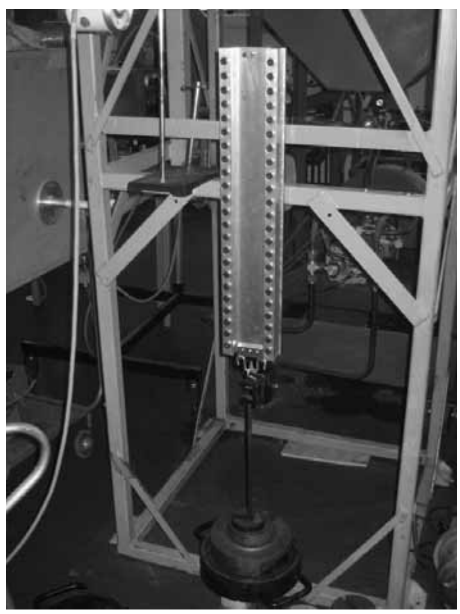

Figure 7. Photograph of the pull out experiment.

The relationship between the torque T and the pull force P needed to shift the plate is shown in Fig. 8, with a linear interpolation of the data which seems appropriate. A small torque was enough to hold a significant weight. The linear interpolation is found to be $P = c_1 T + c_2$ with $c_1 = 2300$ m^{-1} and $c_2 = 7.9$ N. Finally, using the linear approximation calculated by the previous figure, the relationship between the frequency parameter and the pull out force needed to shift the plate is shown in Figure 9. Fig. 9 can be used to estimate the resistance of a fuel assembly plate to a pull out force once the natural frequency of the plate is known.

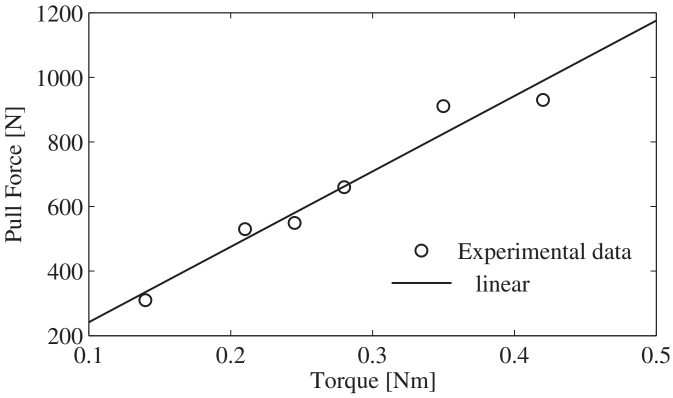

Figure 8. Experimental variation of the pull out force with the fastening torque.

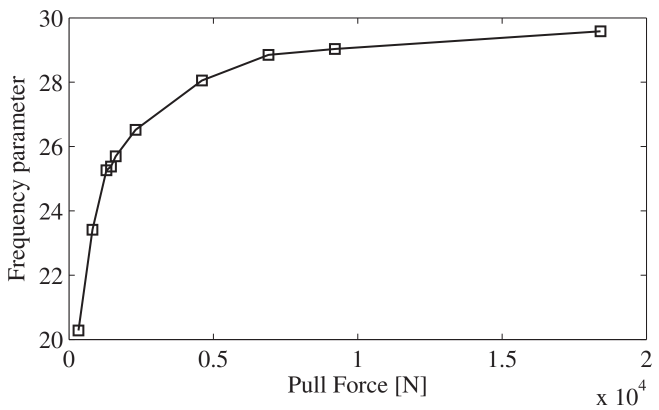

Figure 9. Experimental variation of the frequency parameter with the pull out force.

CONCLUSIONS

A method to relate the pull out force and the frequency parameter in plates with variable clamping strength was presented. The aim was achieved with two steps. In the first step a relationship between the frequency parameter and the fastening strength was found using standard modal analysis. The second step determined a relationship between the pull out force and the fastening torque by attaching weights to the bottom edge of the plate. The results were then combined in order to identify the variation of pull out force with the frequency parameter of the plate. Results can be scaled to plates with different material properties and thickness.

REFERENCES

[1] R. D. Blevins, "Flow-induced vibration in nuclear reactors: a review", *Prog. Nucl. Energ.*, **4**, 25-49 (1979)

[2] M. P. Paidoussis, in *Practical Experiences with Flow-Induced Vibrations*, Vol. 829-832 (Ed: E. E. Naudascher and D. Rockwell), Springer-Verlag, Berlin 1980

[3] M. Ho, G. Hong and A. N. F. Mack, "Experimental investigation of flow-induced vibration in a parallel plate reactor fuel assembly", *Proceedings of the 15th Australasian Fluid Mechanics Conference*, Sydney, Australia, 13-17 December 2004

[4] D. R. Miller, "Critical flow velocities for collapse of reactor parallel-plate fuel assemblies", *J. Eng. Power T ASME*, **82**, 83-95 (1960)

[5] S. J. Pavone and H. A. Scarton, "Laminar flow-induced deflections of stacked plates", *Nucl. Eng. Des.*, **74**, 79-89 (1982)

[6] D. C. Davis and H. A. Scarton, "Flow-induced plastic collapse os stacked fuel plates", *Nucl. Eng. Des.*, **85**, 193-200 (1985)

[7] M. Caresta and D. Wassink, "Vibrational characteristics of roll swage jointed plates", *Acoustics Australia*, **38**, 82-86 (2010)

[8] G. Kim and D. C. Davis, "Hydrodynamic instabilities in flat-plate-type fuel assemblies", *Nucl. Eng. Des.*, **158**, 1-17 (1995)

[9] W. Soedel, "Similitude approximations for vibrating thin shells", *Journal of the Acoustical Society of America*, **49**, 1535-1541 (1971)