Proceedings of the ASME 2010 3rd Joint US-European Fluids Engineering Summer Meeting and 8th International Conference on Nanochannels, Microchannels, and Minichannels
FEDSM-ICNMM2010
August 1-5, 2010, Montreal, Canada

FEDSM-ICNMM2010-31103

EXPERIMENTAL AND NUMERICAL CHARACTERIZATION OF FLOW-INDUCED VIBRATION OF MULTI-SPAN U-TUBES

Atef Mohany[1] **Victor P. Janzen** **Paul Feenstra** **Shari King**

Atomic Energy of Canada Limited (AECL)
Inspection, Monitoring & Dynamics Branch
Chalk River Laboratories, Ontario, Canada

ABSTRACT

This paper describes a test program that was developed to measure the dynamic response of a bundle of steam generator U-tubes with Anti-Vibration Bar (AVB) supports, subjected to Freon two-phase cross-flow. The tube bundle geometry is similar to the geometry used in preliminary designs for future CANDU® steam generators. This test program is one of the initiatives that Atomic Energy of Canada Limited (AECL) is undertaking to demonstrate that the tube support design for future CANDU steam generators meets the stringent requirements associated with a 60-year lifetime. In particular, the tests will address issues related to in- and out-of-plane fluidelastic instability and random turbulent excitation of a U-tube bundle with Anti-Vibration Bar (AVB) supports. Therefore, the measurements include tube vibration amplitudes and frequencies, work-rate due to impacting and sliding motion of the tubes against their supports, bulk process conditions and local two-phase flow properties. Details of the test rig set-up and the measurement techniques are described in the paper. Moreover, a numerical prediction of the U-tube vibration response to flow was performed with AECL's PIPO-FE code. A summary of the numerical results is presented.

INTRODUCTION

Flow-induced vibration is an important factor to consider in the design of future CANDU steam generators that are expected to operate for 60 years. These steam generators will be larger than previous CANDU steam generators, nearly twice the heat transfer area, with significant changes in process conditions in the U-bend region, such as increased steam quality and a broader range of flow velocities. Figure 1 shows the calculated velocity distribution in the U-bend region of a future CANDU steam generator at 100% power. This velocity distribution is a result of flow stability and fouling calculations that were performed at AECL with the THIRST (Thermal-Hydraulics In Recirculating STeam generator) code (Heppner et al., 2006). These and similar calculations predict that tubes in the U-bend region are subjected to two-phase cross-flows with gap velocities ranging from less than 1 m/s to as high as 6 m/s.

Previous experimental investigations were carried out to study the vibration response of tube bundles and the associated fretting wear under different flow, tube-bundle geometry and tube support conditions, e.g., Pettigrew et al. (1989), Pettigrew et al. (1995), Taylor and Pettigrew (2000), Guérout and Fisher (1999), and Pettigrew et al. (2002). Most of these investigations were conducted on straight single-span or cantilevered tubes in cross-flow. In those cases where the propensity for fretting-wear was investigated experimentally, the usual procedure was to measure the so-called "work-rate" (Guérout and Fisher, 1999).

The dynamic response of steam generator multi-span U-tubes with clearance supports is expected to be different, partly because the vibration modes and frequencies will be different from those of straight tubes, and partly because the dynamic response of the tube will depend on whether or not the supports are effective. Janzen et al. (2005) experimentally investigated the flow-induced vibration of a simplified U-tube bundle with a set of flat-bar supports at the apex. The mid-span region of the U-bend was subjected to two-phase air-water cross-flow. They observed, for the first time, that fluidelastic instability occurred both in the out-of-plane and in the in-plane direction.

® CANDU is a registered trade-mark of Atomic Energy of Canada Limited (AECL).

[1] Corresponding Author: mohanya@aecl.ca

This finding raised the possibility of higher-than-expected tube-to-support work-rates for U-tubes restrained by flat bars. To investigate in particular the onset of fluidelastic instability under different geometric and stiffness conditions, bundles of straight tubes preferentially flexible in the flow direction (analogous to in-plane instability of curved tubes) were subsequently used, e.g., Mureithi et al. (2005). While these tests were useful to describe the ideal behavior of tubes in cross-flow, they could not be used to verify the dynamic performance of U-tube bundles having a realistic bundle geometry and realistic (clearance) tube supports, in steam/water flow. More recently, Chu et al. (2009) investigated experimentally the fluidelastic instability characteristics of a U-tube bundle in two-phase flow. The U-tubes were arranged in a rotated square array with a pitch ratio of 1.633. Apart from the difference in array geometry compared to CANDU SG tube bundles, Chu et al. (2009) performed the experiments without anti-vibration bar supports and used air/water mixtures, which have fluid properties that differ significantly from those of steam/water.

Therefore, experiments in realistic two-phase cross-flow with a multi-span U-tube bundle are needed to clarify the relative influence of in-plane and out-of-plane fluidelastic instability, and investigate the nature and importance of random turbulence excitation in steam generators. It should be noted that the effects of random turbulence are becoming relatively more important for the next generation of steam generators, for two main reasons. Firstly, manufacturers are being asked to significantly increase the design life, from 30-to-40 years to 60 years. Secondly, the recent observations of quasi-periodic excitation forces in tube bundles (Ricciardi et al., 2010) need to be considered.

An important point is that tests in either steam-water or, alternatively, two-phase Freon would result in much more realistic hydraulic conditions than tests in air-water. Testing in Freon is much simpler than in steam-water since equivalent pressures and temperatures are much lower. The energy requirements are much less because the latent heat of evaporation is much lower. Compared to the other commonly used alternative to steam-water, namely air-water, Freon characteristics are generally much closer to those of steam-water than air-water, as described by Mohany et al. (2009).

This paper describes a set of experiments, referred to hereafter as the MSUB (Multi-Span U-Bend) test program, designed specifically to address the issues of in- and out-of-plane fluidelastic instability and random turbulent excitation of a U-tube bundle with Anti-Vibration Bar (AVB) supports. This test program is one of the initiatives that AECL is undertaking to demonstrate that the tube-support design for future CANDU steam generators meets the stringent requirements associated with a 60-year lifetime. The measurements include tube vibration amplitudes and frequencies, the work-rate due to impacting and sliding motion of the tubes against their supports, bulk process conditions and local two-phase flow properties. Moreover, numerical predictions of the U-tube vibration response to two-phase Freon flow at different void fractions and support conditions were performed with AECL's PIPO-FE code (Han and Janzen, 2009). The numerical results were used to predict the test conditions required to excite fluidelastic instability and also to help select measurement locations on the U-tubes. The results of the test program will be incorporated in AECL's design guidelines for flow-induced vibration and fretting-wear in steam generators and large heat exchangers (Janzen et al., 2009).

NOMENCLATURE

D	diameter of the tube
FEI	FluidElastic Instability
HEM	Homogenous Equilibrium Model
dS_X	displacement in the x-direction
dS_Z	displacement in the z-direction
δ	logarithmic damping of the tube
f	frequency
F_Y	impact force in the normal (y) direction
K	Connors constant
m	mass per unit length of the tube
P	center-to-center distance between the tubes
ρ	average density of the fluid
t	time
T	time period
$V_{p,cr}$	critical pitch flow velocity
W'	work rate.

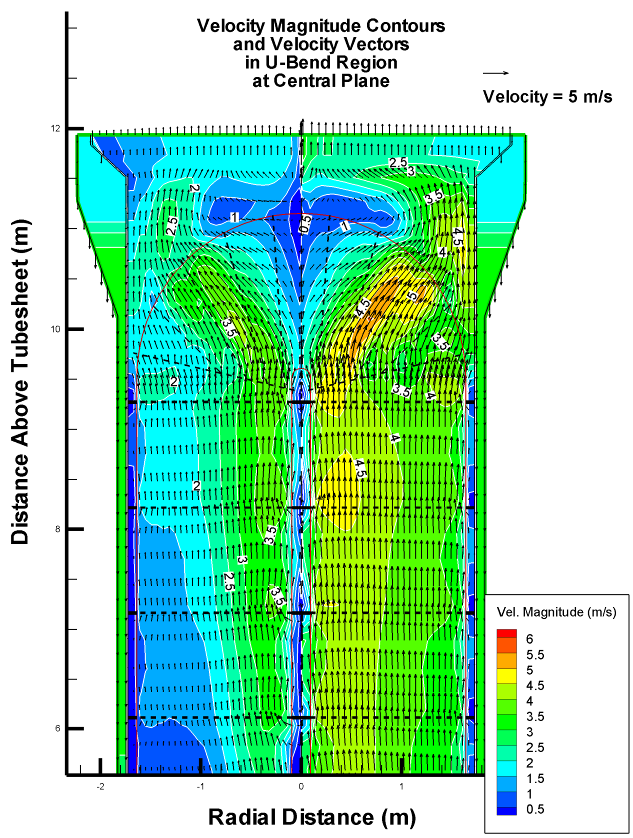


Figure 1: Velocity distribution in U-bend region for central plane of CANDU steam generator at 100% power.

EXPERIMENTAL SET-UP

The experimental set-up is shown schematically in Figure 2. The MSUB test section is attached to the Chalk River MR-3 thermalhydraulic loop, with Freon-134a as the working fluid. The test series will be carried out between 0.74 and 1.2 MPa pressure and approximately 36^{o}C and 23°C saturation temperature at the test section. The MR-3 loop can provide up to 38 kg/s of liquid flow. Freon vapour is produced by a 1.2 MW electrical heater upstream of the test section. The two-phase mixture is condensed downstream of the test section and returned to two circulating pumps. The mass flow is measured with an accuracy better than ±1% using turbine flow meters. The bulk vapour quality and void fraction are estimated from a thermodynamic balance, obtained from process conditions measured upstream and downstream of the test section. This technique is more accurate than two-phase flow-meters at higher void fractions. The accuracy is estimated to be better than ±2% of full scale at void fractions above 50%.

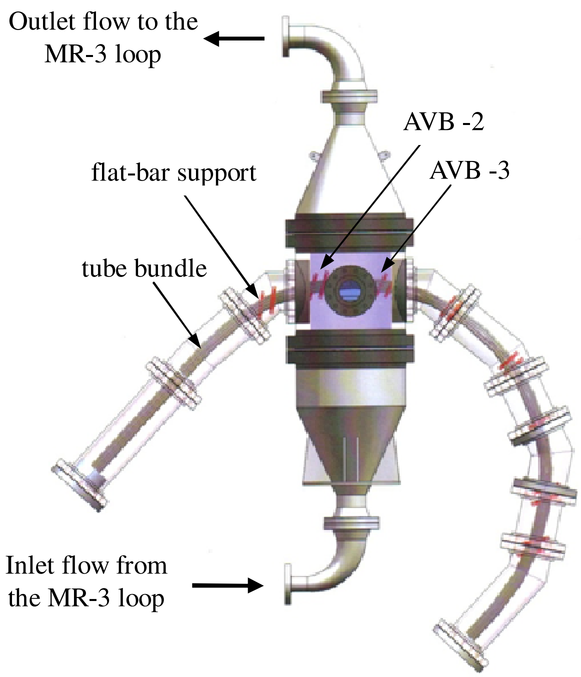


Figure 2: Side view of the test section showing the tube bundle and the flat bar supports locations.

Two-phase Freon flow from the MR-3 heater enters the test section from the left through a 152.4 mm (6-inch) diameter pipe at the bottom. The flow is then directed vertically through a rectangular flow channel containing a flow mixer and flow straightener. The static mixer consists of parallel, corrugated steel sheets in a criss-cross pattern. The honeycomb flow straightener sits on the top of the static mixer and directly below the U-bend tube bundle. The flow mixer and straightener ensure that a homogenous two-phase mixture with a well-defined flow direction is maintained until the flow enters the U-bend tube bundle from below. The flow exiting the flow channel above the U-tube bundle is directed out of the test rig into a 152.4 mm (6-inch) diameter pipe that connects back to the MR-3 loop vapour drum.

The MSUB test section has been designed to accommodate tube-bundle and tube-support geometries representative of the upper U-bend sections of typical CANDU-type steam generators. In its present form, the interior rectangular flow channel is 520 mm wide by 99 mm deep, resulting in cross-flow over somewhat more than a single span of the present MSUB tube bundle. The volume of the test-section pressure vessel surrounding the remainder of the tube bundle is open to the volume into which the flow is directed. The tube bundle consists of five columns of tubes with either four or five tubes in each column, in the rotated-triangular geometry used in CANDU steam generators, as shown in Figure 3. In its present form, each tube in the MSUB tube bundle is 12.7 mm in diameter and has a wall thickness of 0.9 mm. The tube bundle pitch-to-diameter ratio $P/D = 1.5$. These dimensions are the same as those of tube bundles used in previous AECL tests of straight, cantilevered tubes in Freon (e.g., Pettigrew et al., 2002) and two-span U-tubes in air-water (Janzen et al., 2005). Tests are also planned with tube-bundle and support geometries representative of preliminary designs for new CANDU steam generators, e.g., 17.5 mm (11/16-inch) tubes and $P/D = 1.42$ (Klarner et al., 2006).

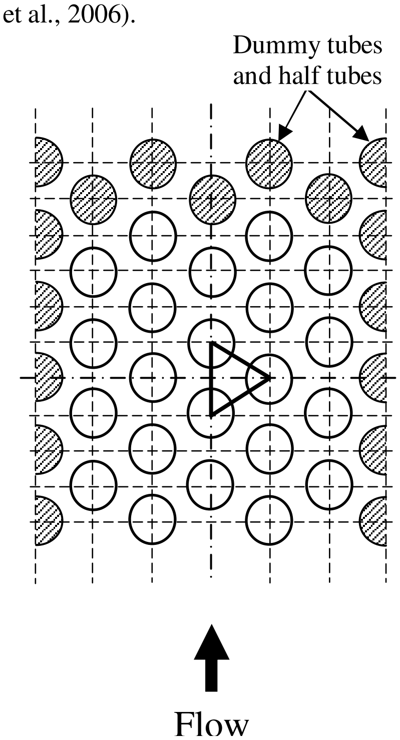


Figure 3: Cross-section of test tube-bundle with the dummy tubes and half tubes, P/D = 1.5.

Edge effects on either side of the bundle are minimized by the use of additional half-tubes on the flow-channel walls. Moreover, two rows of fixed dummy tubes are placed inside the flow channel above (downstream of) the tube bundle. The dummy tubes are fixed inside the flow channel and they are used to mount two-channel fibre optic probes for void fraction measurements of the flow, and proximity sensors for vibration measurements of two of the outer most U-tubes. They will also be used to house force transducers to measure dynamic forces acting on the essentially rigid dummy tubes due to turbulent forces. These last measurements are aimed at investigating the role of quasi-periodic forces in determining tube bundle motion.

Since the dummy tubes are fixed relative to the tube bundle, it was important to calculate the static deflection of the tube bundle due to cross-flow in order to maintain the same gap between the dummy tubes and the outer most U-tubes as that inside the bundle. The maximum static deflection of the outer most U-tubes was calculated with ANSYS, for the conditions shown in Figure 4. The radius of the outermost U-tube is 1384.3 mm (54.5 inch) and the tube is subjected to a single phase liquid Freon flow over a tube span of 520 mm (20.5 inch) in length at the maximum achievable flow rate with the MR-3 loop, i.e., 38 kg/s. As shown in Figure 5, the maximum static deflection of the outer-most U tube is about 2.8 mm (0.11 inch) in the flow direction. Note that this value is the maximum expected and is calculated conservatively. The static deflection is considered when the dummy tubes and attached instrumentation are installed downstream of the tube bundle.

The present MSUB tube bundle has a total of eight sets of flat-bar supports for the outer tubes in the U-bend region, as shown in Figure 2. A single set of flat bars consists of six, 25.4-mm-wide stainless steel bars approximately 180 mm long. At the left-hand end of the tube bundle, the tubes are hydraulically expanded into a 3-inch-thick steel tubesheet (a clamped condition). At the right-hand end of the bundle a pinned support condition is implemented, since it was not possible to add a straight section to simulate the dynamic interface between the U-bend and straight section of a full-length steam generator tube. The tube is thus free to respond in the torsional sense to U-bend vibration but is relatively stiff in the axial and translational sense.

The test section "arms" on either side are modular, so that each section is capable of being removed and re-installed to allow for adjustment of inner tubes and supports. The support structure allows sections of the "arms" to be supported to enable removal and re-installation, and with the top of the test section removed will enable access to the central "barrel" to allow for adjustment of inner tubes, supports and instrumentation. The mass of the test section is approximately 2300 kg (5000 lb_m).

Measurements of flow-induced vibration amplitudes and damping values of two tubes in the U-bend bundle will be performed using an accelerometer carriage that was designed specifically for this purpose. Moreover, the impact force of two of the outer most U-tubes against their supports will be measured using a work-rate measurement device. This device is a modified version of the work-rate instrumentation described by Janzen et al. (2005), and contains a tri-axial accelerometer and a force transducer. The fretting-wear performance in the U-bend will be determined from the measured work-rate due to impact/sliding between the selected tubes and their supports. The vibration amplitudes of the same tubes will be measured using two sets of proximity sensors. Each set consists of three proximity sensors oriented in the x, y, and z directions. Measurements will also include bulk process conditions and local two-phase flow properties.

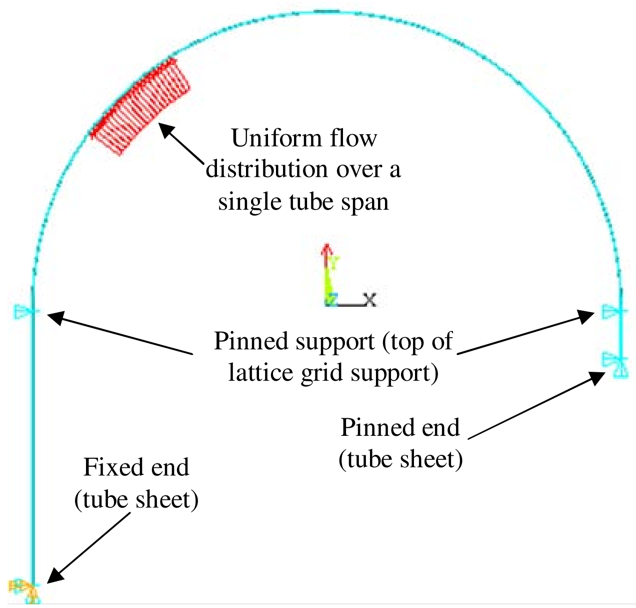


Figure 4: Boundary conditions of the outer most U-tube used in ANSYS to calculate static deflection due to cross-flow.

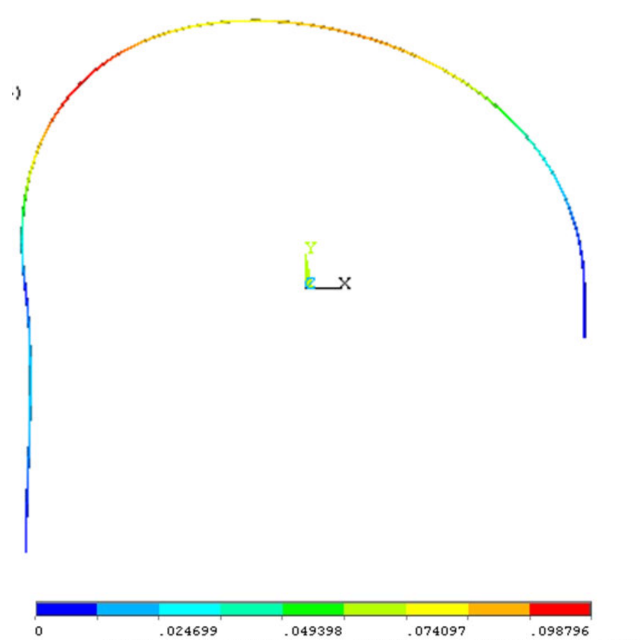


Figure 5: Static deflection of the outer most U-tube due to cross-flow. The amount of deflection is exaggerated to show the effect (scale is in inches).

VIBRATION MODES AND FREQUENCIES

A numerical prediction of the U-tube vibration response to two-phase Freon flow at different void fractions and support conditions has been performed with the PIPO-FE code (Han and Janzen, 2009), an analysis tool that AECL developed to assess the vibration response of heat exchanger tubes under different flow conditions. It predicts the tube-vibration response to vortex shedding and random turbulence, and predicts the critical velocities for fluidelastic instability. The PIPO-FE code can simulate the effects of three types of flow both inside and outside the tubes: single-phase liquid, single-phase gas and a two-phase liquid-gas mixture.

A model for use in FREMOD (Frequencies and Modes) code was developed that reproduces the geometry and physical properties of a representative U-tube as well as the locations and types of the supports. For present purposes, the fluid inside the tube was assumed to be air. Outside the tube, the fluid was assumed to be liquid Freon at 36ºC except for the tube span inside the flow channel, i.e., between flat bar supports AVB-2 and AVB-3 (see Figure 2). The fluid flowing over this span was taken to be two-phase Freon, with HEM (Homogenous Equilibrium Model) void fractions ranging from 70% to 98%.

The flow tests will be conducted with different support conditions, e.g., all supports in, one support missing and two consecutive supports missing, to investigate the effect of removing supports on the excitation mechanism of fluidelastic instability. Therefore, it is interesting to predict the effect of removing supports on the fundamental vibration properties of the U-tube. Figure 6 represents the frequencies of the out-of-plane modes of the U-tube for several different support conditions, at a void fraction of 70%. As expected, the calculated vibration frequencies decrease as the tube becomes less supported by removing Anti-Vibration Bar supports.

Mode shapes for the first four out-of-plane modes are shown in Figure 7. It should be noted that these results assume all the supports are in and effective. Figures 8 and 9 show the frequencies and mode shapes, respectively, for the in-plane vibration response of the U-tube.

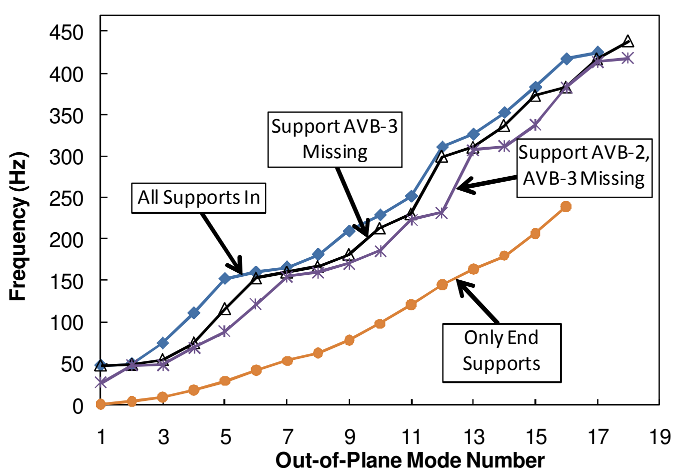


Figure 6: Frequency of the out-of-plane modes of the U-tube for different support conditions (12.7 mm tube).

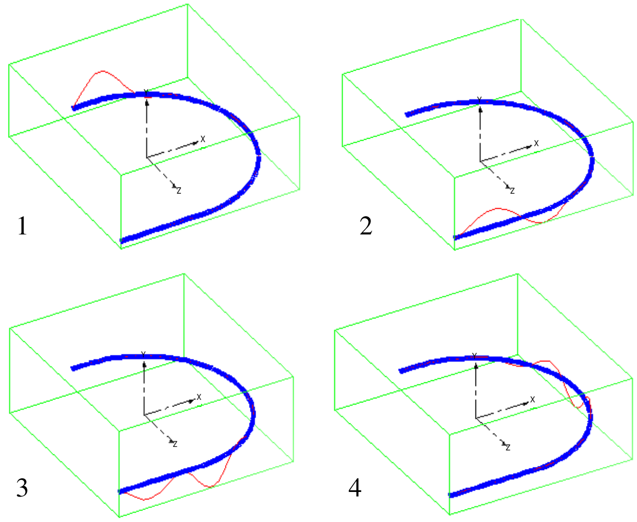


Figure 7: Out-of-plane mode shapes for the first four modes.

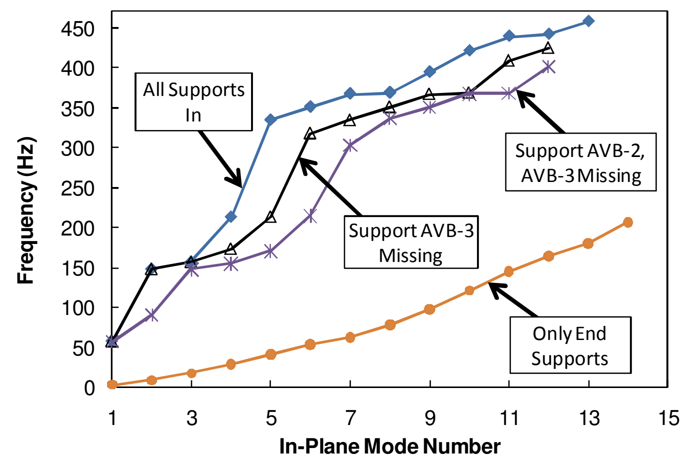


Figure 8: Frequency of the in-plane modes of the U-tube for different support conditions (12.7 mm tube).

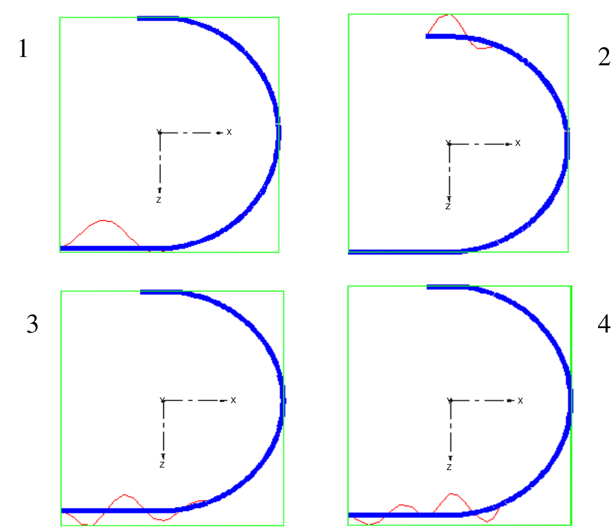


Figure 9: In-plane mode shapes for the first four modes.

FLOW CONDITIONS

For comparison purposes, data for fluidelastic instability in heat-exchanger tube arrays are typically scaled (reduced) according to two dimensionless parameters that are considered to be the most important, mass-damping parameter and reduced critical velocity.

The mass-damping parameter is given by,

$$\text{mass-damping parameter} = \quad m\delta / \rho D^2 \qquad (1)$$

where m represents the mass per unit of length of the tube including the added mass of the two-phase fluid, δ is the damping ratio of the tube in the two-phase fluid, and ρ is the average density of the fluid. Researchers have traditionally relied upon the half-power bandwidth method to determine an appropriate damping value for scaling their fluidelastic instability (FEI) data. The effect of tube-to-tube coupling poses some difficulty because it can cause the frequency peak of the modal response to be artificially broad, leading to an overly high damping value. To avoid this problem, damping measurements will be performed with all the tubes, except the monitored tube, held fixed with a thin clamping plate mounted at midspan. Connors theory predicts that the critical reduced frequency is proportional to the square root of the mass-damping parameter according to,

$$V_{p,cr} / f D \quad = \quad K\sqrt{m\delta / \rho D^2} \qquad (2)$$

The constant of proportionality, K, depends upon the pitch ratio and array geometry. Previous research has attempted to determine a representative lower-bound value for K for design purposes, to avoid the possibility of FEI.

An important aspect of the MSUB test program is to provide designers with data on the conditions at which FEI is reached in the tube array. To do this, the flow loop should have sufficient flow and heat capacity such that FEI can be reached over a desired range of void fraction. An estimate of the MR-3 loop operating envelope was made for a range of two-phase flows from 70% HEM void fraction up to 98% and is overlaid in Figure 10. Some of the input data were drawn from other sources: tube frequency was obtained from PIPO-FE-code calculations in two-phase flow, average pitch flow velocity was estimated from flow loop capacity, and average damping was estimated from previous FIV measurements in R-134a two-phase flow. This estimate indicates that, despite the relatively stiff tubes being tested, the flow conditions in the rig should be sufficient to reach the threshold of fluidelastic instability for this range of homogeneous void fraction.

Note that, within the test section flow area, the U-bend tube bundle is subjected to uniform cross flow of two-phase R-134a over one complete span, between flat-bar supports AVB-2 and AVB-3. It should be noted that the data in Figure 10 correspond to out-of-plane instability only for normal triangular and parallel triangular tube arrays.

Tests will be performed for different support conditions, e.g., the possibility of one, two or more sets of tube supports inactive. In the MSUB test rig, this involves removing set(s) of tube supports adjacent to the edge of the flow box. The effect on the U-bend tube bundle is an expected increase in the flexibility of the bundle and a corresponding reduction in the frequency of the lowest vibration modes.

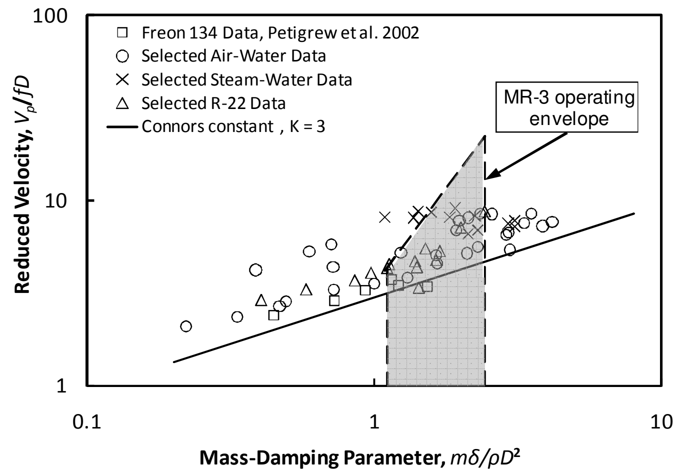


Figure 10: Plot of selected data for out-of-plane fluidelastic instability threshold in two-phase flows in various fluids with an estimate of upper-bound flow conditions attainable in the MSUB test rig. Previous data reported by Pettigrew *et al.* 2002.

An important experimental consideration is that of necessity the cross-flow in the test section extends over approximately one span of the tube bundle, not the entire U-bend as it would in a steam generator. To assess the effects of this partial admission, critical gap velocities for the onset of fluidelastic instability were predicted for various tube-bundle geometries and properties with the PIPO-FE code, taking into account detailed steam-water flow distributions predicted by the THIRST code for one of the CANDU SG designs being considered for future power reactors, with 17.5-mm (11/16-inch) outside diameter tubes, at 100% reactor power. The calculations also took into account the density difference between steam-water and two-phase Freon flow. For partial-admission flow, the calculations assumed two-phase Freon flow with 90% void fraction and a gap flow velocity of 7 m/s.

Values of the ratio of the flow velocity over the critical velocity predicted for the onset of FEI are shown for in-plane and out-of-plane motion in Figure 11 and Figure 12, respectively. A velocity ratio of one indicates the flow velocity is at the predicted threshold for FEI. The critical velocity ratios for partial-admission flow scale reasonably well with, and are not dramatically lower than, those for full admission. Given that the MR-3 loop can provide flow velocities typically 20% higher than those predicted for operational steam-generator conditions, it should be possible to observe out-of-plane FEI for tests at or close to 90% void fraction. For the more flexible

12.7-mm (1/2-inch) diameter tube bundle, it should be possible to observe the onset of FEI over a wider range of void fractions and support conditions.

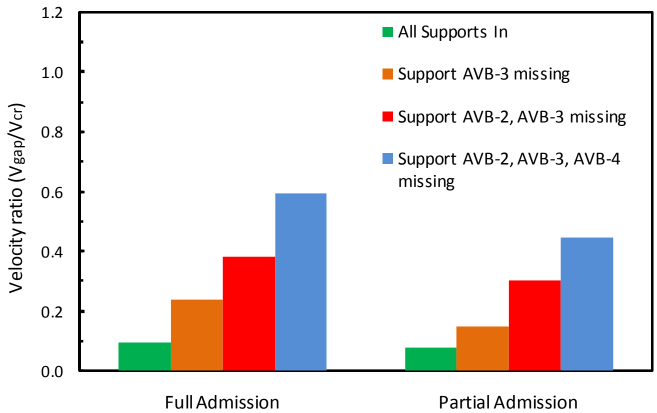


Figure 11: Gap velocity ratios for in-plane fluidelastic instability, with all tube supports and with one, two or three adjacent supports missing (17.5 mm tubes).

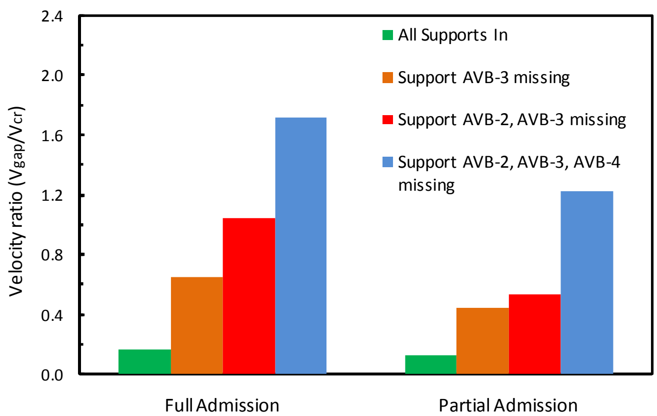


Figure 12: As in Figure 11, but for out-of-plane fluidelastic instability (17.5 mm tubes).

WORK-RATE MEASUREMENTS

The primary experimental method for assessing fretting-wear performance is to evaluate work-rates due to impacting/sliding between components in test rigs under simulated operating conditions. When combined with a wear coefficient, the wear rate and thus the life of a generating-station component such as a steam generator U-tube can be predicted. Ideally, work-rate is the rate at which work is done by the action of a component vibrating against its support. Both work-rate and wear coefficient need to be determined experimentally, although in practice, standard values of the wear coefficient are often assumed for commonly used materials.

The work-rate that has been most often calculated is called "normal" work-rate, and can be defined as:

$$W'_{normal} = \frac{1}{T}\int_0^T \left[F_Y \sqrt{(dS_X)^2 + (dS_Z)^2} \right] \qquad (3)$$

The normal work-rate uses the force in the Y (normal) direction, F_Y, and the total displacement vector, dS_X and dS_Z, in the XZ (lateral-axial) plane. After evaluating the product of these quantities for each time interval and summing over the duration of the contact time, the work-rate is found by dividing by the total time (T). Because the displacement is perpendicular to the force, technically speaking, the normal work-rate is not actually a rate of work. It is used mainly because of the difficulties in determining the forces in the axial and lateral directions. In the simplest approach, the magnitude of the normal force is related to the force in the lateral-axial plane by the dynamic coefficient of friction for the material combination involved, with a typical value of approximately 0.5 (e.g., Janzen et al., 2005).

WORK-RATE DEVICE

The work-rate device developed for this test program consists of a miniature force transducer and a tri-axial accelerometer contained in a Teflon carriage that fits inside a 12.7-mm (0.5 inch) diameter tube, as shown in Figure 13. The conceptual design of this device is similar to that used by Janzen et al. (2005). The overall length of the Teflon carriage is 28 mm (1.102 inch) with a maximum diameter of 10.87 mm (0.428 inch). As shown in Figure 13, two small buttons, that protrude from either side of the tube and extend approximately 0.15 mm outside the tube, are used. The axial direction of the buttons is perpendicular to the surface of the flat bar supports. Therefore, upon impacting a support adjacent to the tube, the force transducer is compressed. The two buttons are used also to hold the Teflon carriage in place and to channel the impact force through the tube wall to the force transducer. However, this arrangement decreases the sensitivity of the force transducer-measuring surface. Therefore, static and dynamic calibrations were performed to determine the ratio between the measured force to that impacted on the buttons, as shown in Figures 14 and 15, respectively. This ratio is slightly different for the static and the dynamic calibration. The static calibration shows a ratio between the measured force to the input force of about 0.373, while the dynamic calibration shows a ratio of about 0.365.

A tri-axial accelerometer in the work-rate device provides the tube-vibration signals. The axial and lateral double-integrated signals provide the displacement information required to calculate the normal work-rate as given by Equation 3. A comparison between the double-integrated acceleration measured by the accelerometer and the displacement measured directly by a proximity sensor is shown in Figure 16.

For this comparison, a shaker was used to excite a straight tube at different frequencies and amplitudes. The accelerometer was attached to the tube and the displacement of the tube was directly measured using a proximity sensor that was mounted on a fixed holder outside the tube. The comparison was performed over a frequency range from 10 Hz to 150 Hz. As shown in Figure 16 the difference between the displacement measured by the accelerometer and that measured by the proximity sensor is typically about 8% except for two points where the difference is about 15%.

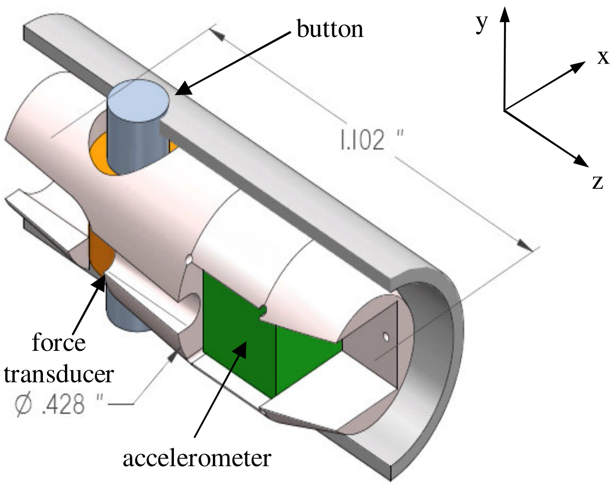


Figure 13: Isometric illustration of the work-rate device inside the tube.

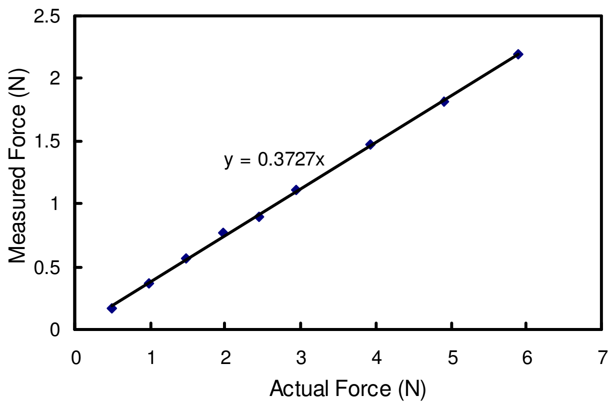


Figure 14: Static calibration of the work-rate device using hanging weights.

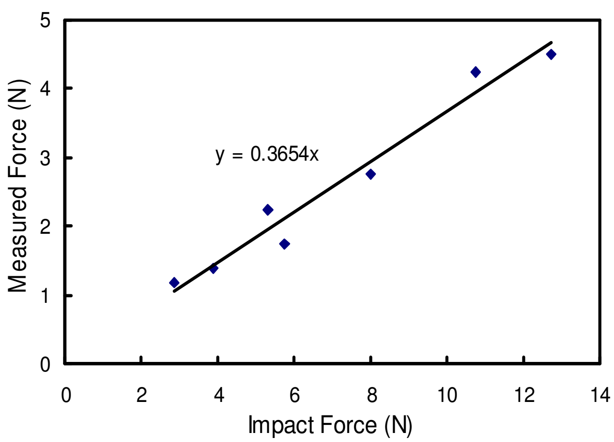


Figure 15: Dynamic calibration of the work-rate device using an impact hammer.

If the impacts occur regularly, then it will be possible to compare the impact frequency with the frequency of the out-of-plane accelerometer signal to determine if the tube is impacting one or both supports. This information can be used to infer how well the tube is centered at that tube support location.

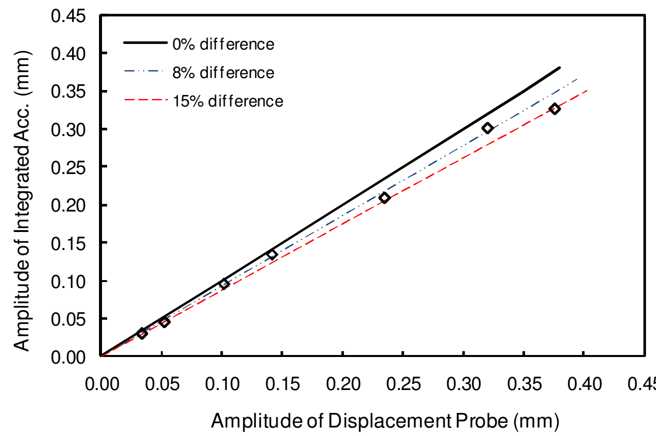


Figure 16: Comparison between the double integrated acceleration and the displacement measured directly by the proximity sensor.

ACCELEROMETER CARRIAGE

This carriage was designed to measure flow-induced vibration amplitudes and damping values for two tubes in the U-bend bundle. As shown in Figure 17, the carriage consists of a stainless steel spring, a two-sided Teflon carriage, three sets of screws and a ferrite disc. A tri-axial accelerometer, similar to that used in the work-rate device, is placed inside the carriage. The stainless steel spring is used to hold the carriage in place inside the tube. The ferrite disc, about 1 mm in diameter, is used to identify the orientation of the carriage and hence the orientation of the accelerometer. An eddy current probe is used outside the tube to detect the location of the ferrite piece. The stiffness of the stainless steel spring is sufficient to keep the carriage in contact with the tube inner surface at all times, as shown in Figure 18.

To ensure that the accelerometer response is not affected by the carriage assembly itself, a comparison was performed with another accelerometer that was rigidly mounted on the external surface of the tube at the same location and with the same orientation as the accelerometer in the carriage inside the tube. The tube was subjected to sinusoidal excitation forces applied by two mechanical shakers. The shakers were used to independently excite the in-plane and out-of-plane modes of a single U-tube mounted on a vertical plywood board with 8 flat bar supports. Several different excitation amplitudes and frequencies were tested.

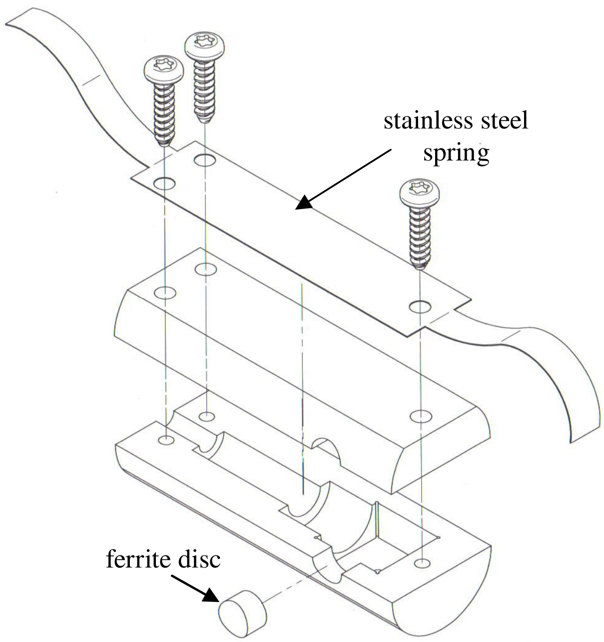


Figure 17: Isometric drawing of the accelerometer carriage.

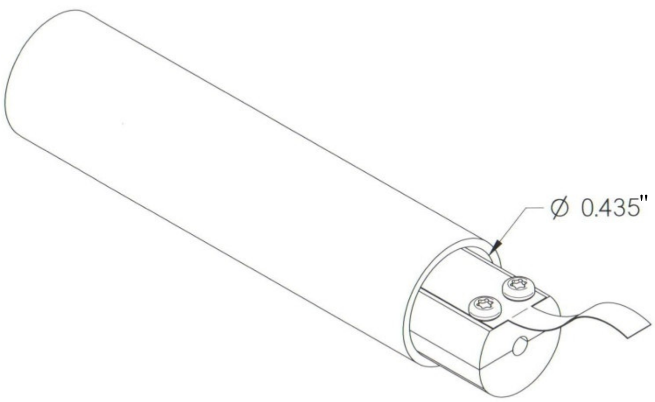


Figure 18: Isometric drawing showing the accelerometer carriage as it is being inserted inside the tube.

Figure 19 shows the response of both accelerometers for an out-of-plane excitation at 50 Hz. As can be seen from this figure, the two accelerometers are responding in almost identical fashion. The difference between the acceleration amplitude measured by the externally mounted accelerometer to that measured by the accelerometer inside the carriage was only 1.8%.

Figure 20 represents a comparison of the frequency spectra for the root mean square (r.m.s) acceleration measured by both the internal and the external accelerometer subjected to a sinusoidal excitation. As it can be seen from this figure, both spectra have a dominant peak at 50 Hz, which corresponds to the second out-of-plane vibration mode of the U-bend, as shown in Figure 7.

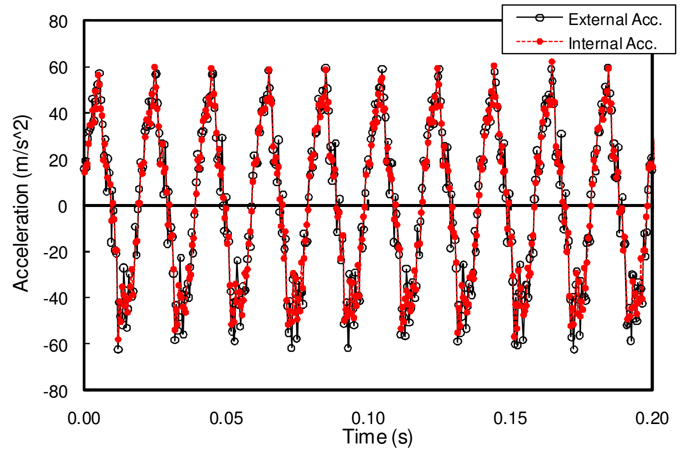


Figure 19: Comparison between the acceleration amplitudes measured by an externally and an internally mounted accelerometer.

VOID-FRACTION PROBES

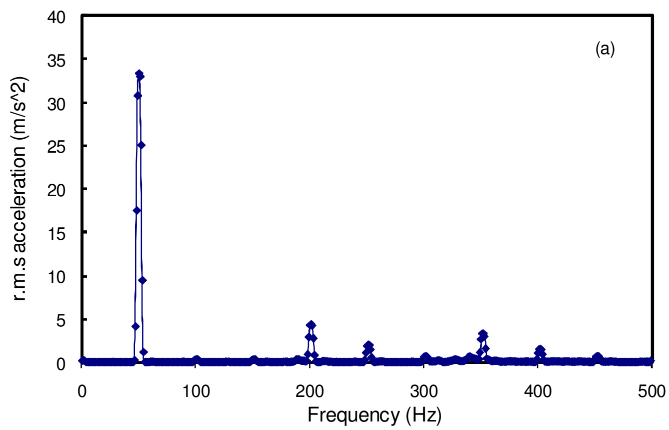
The characteristics of two-phase flow surrounding the tubes will be measured with two dual-channel fiber-optic probes. Each fiber-optic probe has a conical tip and is made of an optical fiber of either 50 or 170 µm diameter. The tip acts as a phase sensor based on the different level of light reflection between air and water. Each dual-channel probe will be capable of providing information on void fraction, bubble (void) size and bubble velocity, that will be used to assess two-phase flow properties and flow-regime effects. While the HEM is convenient model for defining input parameters for generating a two-phase flow, the void fraction measurements obtained from the VF probes will allow for more accurate estimation of these parameters

CONCLUDING REMARKS

A test program to measure the dynamic response of a bundle of steam generator U-tubes with AVB supports, subjected to Freon two-phase cross-flow, has been presented. The test program represents the most recent advance in an extensive series of vibration and work-rate measurements aimed at characterizing the flow-induced vibration and fretting-wear response of steam-generator tube bundles under increasingly realistic conditions. The main objectives of this program are to address the issue of in and out-of-plane fluidelastic instability, and to characterize random turbulent excitation of a U-tube bundle with AVB supports, including the possibility of quasi-periodic flow-related forces. The design principles of an accelerometer carriage for measuring flow-induced vibration amplitudes and a work-rate device for determining the tubes fretting wear against their supports have been outlined.

ACKNOWLEDGMENTS

The authors would like to thank Yingke Han, John Pietralik, Sherry Laroche, Andrew Kittmer, Jon Pozsgai and Mohammed Saleem for their valuable contributions to this project.



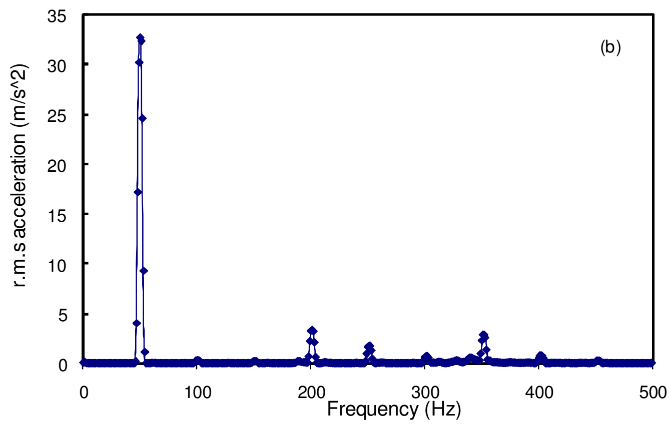


Figure 20: Frequency spectra of the r.m.s acceleration amplitude measured by (a) an externally mounted accelerometer, and (b) an accelerometer in the carriage inside the tube.

REFERENCES

[1] Heppner, K., Laroche, S. and Pietralik, J., 2006, The THIRST Chemistry Module as a Tool to Determine Optimal Steam Generator Corrosion Control Strategies, Proceedings of the 5th CNS International Steam Generator Conference, Toronto, Canada, November 26-29 (Canadian Nuclear Society, 2006).

[2] Pettigrew, M.J., Kim, B.S., Taylor, C.E. and Tromp, J.H., 1989, "Vibration of Tube Bundles in Two-Phase Cross-Flow - Part 2: Fluidelastic Instability", ASME J. Pressure Vessel Technol., 111, pp. 478 – 487.

[3] Pettigrew, M.J., Taylor, C.E., Jong, J.H. and Currie, I.G., 1995, "Vibration of a Tube Bundle in Two-Phase Freon Cross Flow", ASME J. Pressure Vessel Technol., 117, pp. 321 – 329.

[4] Taylor, C.E. and Pettigrew, M.J., 2000, "Effect of Flow Regime and Void Fraction on Tube Bundle Vibration", Flow-Induced Vibration, Ziada, S. and Staubli, T. eds., Balkema, A.A., Rotterdam, pp. 529 – 536.

[5] Guérout, F.M. and Fisher, N.J., 1999, "Steam Generator Fretting-Wear Damage: a Summary of Recent Findings", ASME PVP-Vol. 389, Flow-Induced Vibration, pp. 227 – 234.

[6] Pettigrew, M.J., Taylor, C.E., Janzen, V.P. and Whan, T., 2002, "Vibration Behaviour of Rotated Triangular Tube Bundles in Two-Phase Cross Flows", ASME J. Pressure Vessel Technol., 2, pp. 144 – 153.

[7] Janzen, V.P., Hagberg, E.G., Pettigrew, M.J. and Taylor, C.E., 2005, "Fluidelastic Instability and Work-Rate Measurements of Steam Generator U-Tubes in Air-Water Cross-Flow", ASME J. Pressure Vessel Technol., 127, pp. 84 – 91.

[8] Mureithi, N.W., Zhang, C., Ruël, M. and Pettigrew, M.J., 2005, "Fluidelastic Instability Tests on an Array of Tubes Preferentially Flexible in the Flow Direction", Journal of Fluids and Structures, 21, pp. 75-87.

[9] Chu, I.C, Chung, H.J and Lee, C.H., 2009, "Fluid-Elastic Instability of Rotated Square Array U-Tubes in Air-Water Flow", J. Pressure Vessel Technol., 131, 041301 1-8.

[10] Ricciardi, G., Pettigrew, M.J. and Mureithi, N.W., 2010, "Fluidelastic Instability in a Normal Triangular Tube Bundle Subjected to Air-Water Cross-Flow,"submitted to ASME J. Pressure Vessel Technol.

[11] Mohany, A., Feenstra, P., Janzen, V.P. and Richard, R., 2009, "Experimental Modeling of Flow-Induced Vibration of Multi-Span U-Tubes in a CANDU Steam Generator", Proceedings of the 6th International Steam Generator Conference, Toronto, Canada, November 8-11 (Canadian Nuclear Society, 2009).

[12] Han, Y. and Janzen, V.P., 2009, "PIPO-FE: An Updated Computer Code to Evaluate Heat Exchanger Flow-Induced Vibration," Proceedings of the 6th CNS International Steam Generator Conference, Toronto, Canada, November 8-11 (Canadian Nuclear Society, 2009).

[13] Janzen, V.P., Han, Y. and Pettigrew, M.J., 2009, "Design Specifications to Ensure Steam-Generator and Heat-Exchanger Flow-Induced Vibration and Fretting-Wear Performance," Proceedings of the ASME PVP2009 Conference, 26-30 July, 2009, Prague, Czech Republic, PVP2009-78078.

[14] Klarner,R., Idvorian, N. and Steinmoeller, F., 2006, "Steam Generator Design for ACR-1000," Proceedings of the 5th CNS International Steam Generator Conference, Toronto, Canada, November 26-29 (Canadian Nuclear Society, 2006).