DYNAMICS OF A STEAM GENERATOR TUBE SUPPORTED BY AN ANTI-VIBRATION BAR

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ABSTRACT

The dynamic behaviour of U-tubes in steam generators is usually described using deterministic and statistical parameters. These U-tubes are supported by flat bars in the plane of the U-bend. Clearances between the tubes and the bars are designed to be minimal, but cumulative tolerances and manufacturing variations may lead to clearances larger than expected. Large clearances may result in ineffective support leading to in-plane and out-of-plane motion, fretting-wear and impact abrasion.

In the present work, the first modes of a simply supported single tube supported by flat bars at mid-span are investigated experimentally. The influence of clearance and preload of the flat bars on tube motion and contact force is investigated.

Preliminary results show that the flat bar redistributes the vibration energy to the higher modes. Negative clearance increases the contact forces and decreases the acceleration amplitude, and positive clearance decreases the contact forces and increases the acceleration.

1. INTRODUCTION

Steam generator tubes in the U-bend region are supported by anti-vibration bars (AVB). For assembly consideration and to allow thermal expansion, the tubes are loosely supported. Even though clearances between the tubes and the bars are designed to be minimal, cumulative tolerances and manufacturing variations may lead to clearances larger than expected. The AVB then becomes less effective, and flow-induced vibration can cause premature fretting-wear due to impacting, rubbing or both. Fretting damage that leads to tube failures in steam generators is still a serious problem in nuclear power plants. Failure of tubes results in interruption of power and costly repairs.

The wear rate is utilized to estimate the life of a steam generator tube. Knowledge of the tube dynamics and its interaction with the supports is necessary to estimate the wear rate and evaluate the support effectiveness. The complexity of the interaction makes the development of a realistic numerical contact model difficult. In addition, the system is greatly non-linear. Consequently, dynamic models of a tube supported by AVB are based on both deterministic and statistical parameters.

Continuous models are developed to obtain more realistic numerical analyses. In particular, Hassan et al. (2005) developed a new tube-to-support impact model to consider point or edge impacting. Several other analytical models have been developed to determine the dynamic response of loosely supported tubes, but few experimental results are available to compare to the numerical models. A few authors compared numerical results with experiments (e.g. Fisher et al., 1989). They found good agreement between their numerical model and experimental data.

Yetisir and Weaver (1986) carried out a theoretical study to investigate the effect of flat bar supports on the dynamic response of a heat exchanger U-tube. The simulation results showed that increasing support clearance for the same excitation level has little effects on the RMS impact force level. However, the amplitude of the tube motion increases proportionally with the clearance.

Experiments were conducted by Weaver and Schneider (1983) on U-tubes forming a triangular array subjected to air cross flow. Results were not sufficiently consistent to draw well defined stability thresholds for nonzero clearances between the tubes and the AVB.

In the present work, the first modes of a simply supported single tube supported by flat bars at mid-span are investigated experimentally. The influence of clearance and preload of the flat bars on the tube motion and contact force is investigated. The experimental set-up is described and preliminary results are shown.

2. EXPERIMENTAL PROCEDURE

2.1 Experimental Set-up

The experimental set-up consists of a simply supported two-span straight tube with flat bar supports at mid-span. The test rig (Figure 1) includes a bottom and a top support (Items 1 and 4), an electromagnetic shaker assembly (2), a flat bar displacement assembly (3), and an I-beam mounting post (5). The mounting post is also fixed to a structural concrete column.

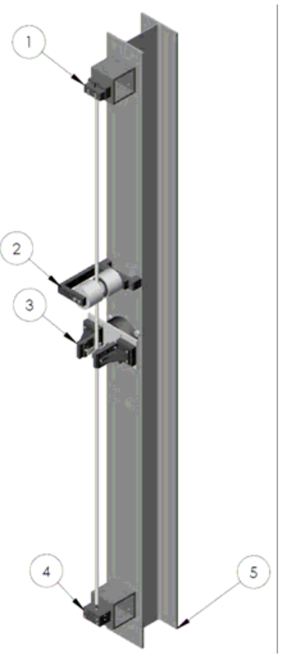

Figure 1: *Experimental set-up*

A tube of 2.5m length, 15.9mm outer diameter, and 0.965mm wall thickness, is used in this experimental set-up. The bottom and top support assemblies each include a link that simulates simple supports (Figure 2). The link is glued in the tube and the other end is fixed in the steel block. Set screws are used to fasten and prevent rotation of the link in the steel block.

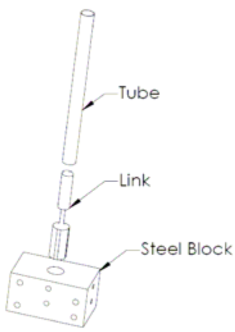

Figure 2: *Bottom tube assembly*

The link was modeled using ABAQUS. It is made of stainless steel and has a bone shape that allows for significant bending. It constitutes an extension to the tube and takes most of the bending moment away from the tube. The design criterion was to obtain the first natural frequency and mode shape of the tube and links assembly as close as possible as those of a two span simply supported tube. The radius and the neck length were varied to obtain the required properties.

The AVB displacement system (Figure 3) is composed of the AVB mounted on a translation stage and a rotation stage through a 90^{o} bracket. Figure 3 shows the tube with one AVB, a second AVB can be located on the other side of the tube, parallel to the first AVB.

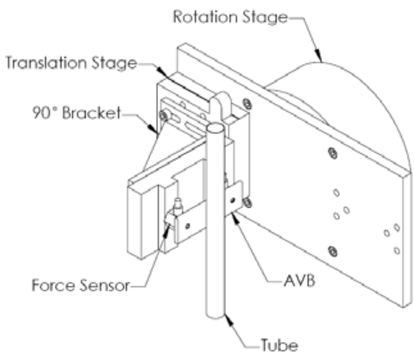

Figure 3: *Displacement system.*

The precision of the translation stage is one micrometer, and that of the rotation stage is two arc seconds. Two force sensors are located between the AVB and the rigid steel bracket. The displacement system allows for the rotation of the AVB (Figure 4) which causes the line of contact between the tube and the AVB to become a point of contact. However, the two AVB are constrained to remain parallel.

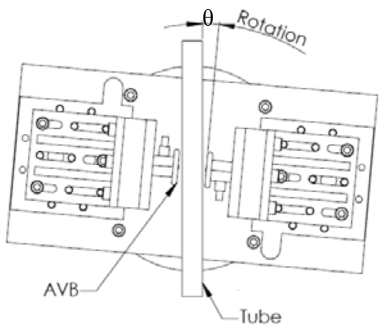

Figure 4: Rotation of the displacement system

The displacement system also allows for the translation of each AVB (Figure 5), which causes a positive or negative clearance between the tube and the AVB, as well as an eccentricity of the tube with respect to AVB.

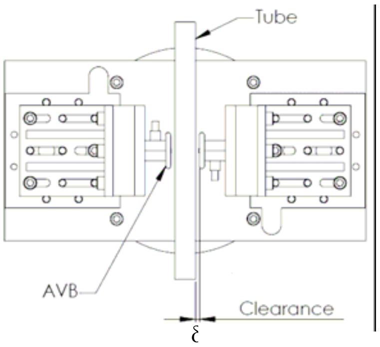

Figure 5 *: Translation of the displacement system*

Finally the flow-induced vibration excitation forces are simulated by electromagnets.

2.2 Instrumentation

The test rig is instrumented with force sensors, optical lasers and accelerometers. The force sensors are placed under the flat bars to measure the contact forces that will later be used to estimate the friction forces. The two accelerometers are mounted directly on the tube, 320mm apart and each at 160mm from the middle of the tube. They are used to measure the tube vibration parallel and perpendicular to the flat bars. A pair of laser displacement sensors is placed at the center of the tube to observe the tube displacement at the contact point with the AVB. Another pair is located at the same height as the accelerometers, above the displacement system, to correlate the acceleration and displacement measurements. Displacements are also measured parallel to and perpendicular to the flat bars.

2.3 Methodology

The controlled parameters are the frequency, amplitude and angle of the excitation forces, and the clearance, tilt and preload between the tube and the flat bars.

Prior to each test, the zero clearance position between the AVB and the tube is set. The rotation is achieved prior to setting the zero position. The gap is then achieved using the translation stage. The AVB configurations reported in the present paper are listed in Table 1.

Clearance (δ)	Rotation (θ)
0mm	0°
0.045mm	0°
-0.01mm	0.25°
-0.065mm	0.5°

Table 1: *AVB configurations*

A negative clearance means that there is a preload of the AVB on the tube.

The first mode is then excited by the electromagnets with a sine sweep spanning 10 Hertz around the first mode. Each sweep is executed over a time period of 500 seconds. The applied force is function of the electric power and frequency transmitted to the electromagnets; as a result the applied force decreases for a sweep of 10 Hertz. The amplitude of the applied force ranges from 35mN to 210mN.

The frequency spectra were obtained using the NVGate 4.22 real time frequency analyzer and plotted using Matlab 7.1.

3. PRELIMINARY RESULTS

3.1 Accelerations

The natural frequencies of the tube with no AVB are found experimentally by impact tests. The cross-spectrum of the two accelerometers signals is presented in Figure 6. The natural frequencies of the tube supported by the two links can be identified from each acceleration peak.

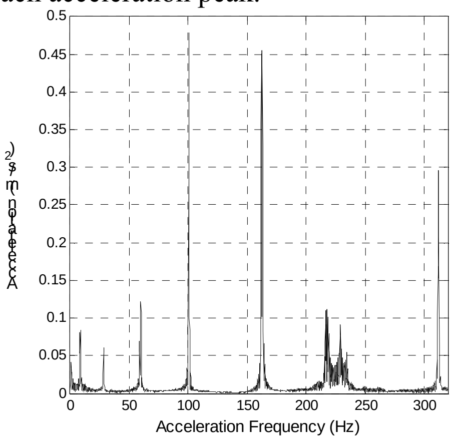

Figure 6: *Acceleration spectrum of the tube with no AVB excited by an impact.*

Sine sweeps spanning 10 Hertz around each peak revealed the precise natural frequencies of the tube with no ABV. The spacing of the two acceleration peaks at 217 and 230Hz can lead one to believe that one of the two frequencies is attributed to a frequency of the assembly.

To determine which frequency (207 or 230 Hz) is attributed to the tube, the theoretical natural frequencies of a simply supported tube were found using Fremod 7.1. As expected, the theoretical frequencies are higher than the experimental frequencies since the link does not act exactly as a simple support: the link bends at its thin section which is located outside of the tube, causing the end of the tube to move transversally. Therefore, an equivalent length was used to obtain the same first natural frequency of a simply supported tube as that experimentally determined. The difference between the experimental and theoretical frequencies proves to increase linearly with the mode order. The theoretical frequencies were adjusted using the linear error function between theoretical and experimental frequencies. The results presented in Table 2 show that the adjusted frequencies are in agreement with the experimental frequencies. Thus, the sixth mode is found to be at 230 Hz.

Mode	Experimental Frequency (Hz)	Theoretical Frequency (Hz)	Adjusted Theoretical Frequency (Hz)
1	8,1	8,4	9,2
2	28,0	33,6	28,0
3	59,5	75,7	59,3
4	99,8	134,5	103,0
5	159,6	210,2	159,3
6	230,1	302,7	228,1
7	312,1	412,0	309,4

Table 2: *Natural frequencies of the tube with no AVB*

Another impact test is performed on the tube with one AVB at zero clearance. The frequency cross-spectrum of the accelerometers is presented on Figure 7.

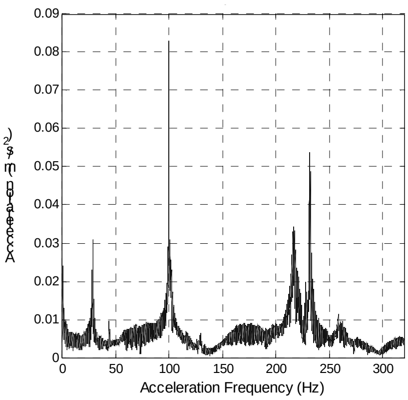

Figure 7: *Acceleration spectrum of the tube with one AVB (δ=0; θ=0) excited by an impact*

The frequency spectrum reveals the absence of the first, third, fifth and seventh modes. All the modes having antinodes at the AVB position are eliminated. For all modes present, the natural frequencies of the tube with the AVB are practically identical to the frequencies of the tube with no AVB. It seems that the AVB acts as a simple support.

The frequency spectrum of the tube with no AVB submitted to a sine sweep around its first mode by the electromagnets is presented in Figure 8. The frequencies of the sweep range from 4 to 14Hz.

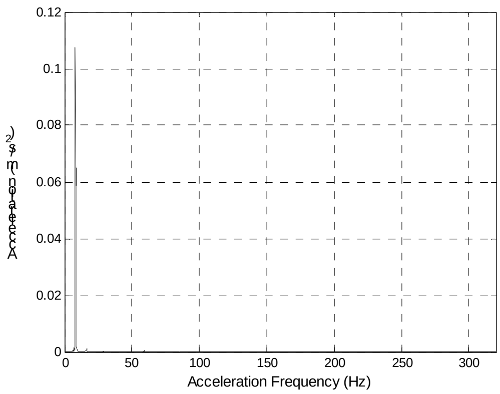

Figure 8: *Acceleration spectrum of the tube with no AVB excited at the first mode by a sine sweep*

In this case, only the first mode is excited by the electromagnets during the sweep. The acceleration peak is very narrow, indicating that there is nearly no energy dissipated. It is clearly observed that the tube has almost no damping.

The next results shown in Figure 9 are four cross-spectra of an AVB supported tube, each with a different excitation force around the first mode. The configuration of the AVB is zero clearance with the tube and no rotation.

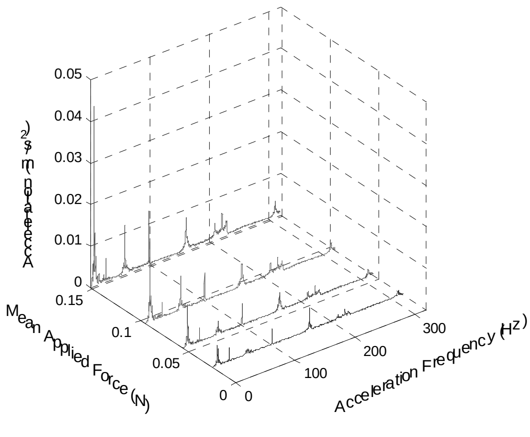

Figure 9: *Acceleration spectra of the tube with one AVB (δ=0; θ=0) excited at the first mode by a sine sweep*

In this case, all the natural modes (even and odd) are excited. Clearly, the energy provided at the first mode by the electromagnets is redistributed to the higher modes of the tube. This can be caused either by impacts or by a stick and slip interaction of the tube on the AVB. In addition, the peak of the first mode is wider than in the case with no AVB; accordingly, damping of this mode is greater. However, the first mode remains preponderant, as can be seen from the peaks of the various modes.

The four acceleration spectra in Figure 9 have a similar shape, but different amplitudes. As expected, the amplitude of the first mode increases with the excitation force.

All the acceleration spectra have the same shape for the different AVB configuration studied. The acceleration peaks are also at the natural frequency of the tube.

3.2 Contact forces

Figure 10 shows the frequency cross-spectra of the two force sensors under the AVB with zero clearance (δ=0) and no rotation (θ=0).

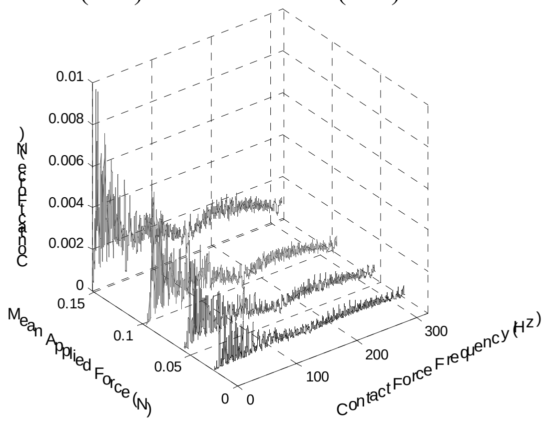

Figure 10: *Contact force spectra of the tube with one AVB (δ=0; θ=0) excited at the first mode by a sine sweep*

The force spectra cover a wide frequency band, such as the acceleration spectra. However, the shapes of the force and acceleration spectra are quite different. The force spectra are rather noisy, with a peak amplitude at the first mode (8,1 Hz), meaning that when the tube oscillates at its natural frequency, the motion is maximal and produces the largest contact forces. Also, the amplitude of the contact force increases with that of the excitation force. The amplitude reaches 10mN for an excitation force of 210mN.

For the configuration δ= -0,01mm and θ=0,25°, the contact force spectra of the AVB have similar shapes as those presented in Figure 10. The small preload and rotation cause no particular change with respect to the case with zero clearance.

On the contrary, a positive clearance, gives the contact force spectra shown in Figure 11.

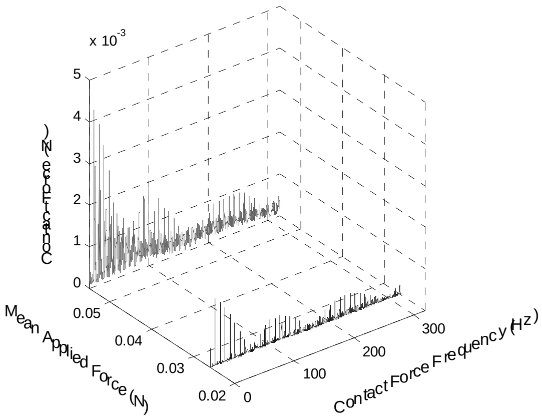

Figure 11: *Contact force spectra of the tube with one AVB (δ=0.045mm; θ=0) excited at the first mode by a sine sweep*

In Figure 11, the general shape of the spectra is similar to that in Figure 10. However, a harmonic component is clearly observed in this configuration, the spacing of the harmonics is approximately 8Hz, which is the first natural frequency of the tube.

The shape of a sinus x is also discernible, which seems to modulate the amplitudes of the various harmonics. The presence of sinus x modulated harmonics can be explained by a discontinuous time signal. This is very likely, since the contact force may exist only during part of each oscillation cycle of the tube. The force promptly rises to a maximum value and then decreases to zero; the period of these pulses is given by the frequency of the oscillations, which is 8 Hz.

It should be noted that such modulated harmonics also exist in the previous configurations, although less noticeable due to the high noise level. Also, the period of the sinus x differs in each case, indicating that the duration of contact is different.

4. ANALYSES

For preliminary analysis, only the influence of the clearance is taken into account. The angle of the flat bar will be studied in depth in forthcoming work.

Theoretical and numerical analyses by Goyder and Teh (1989) predict that the magnitude of impact forces decrease with decreasing clearance. Table 3 shows contact force amplitude ratios at the first mode for an excitation force of 80mN. To obtain the ratios, the contact forces associated to the clearances in the rows are divided by the contact forces associated to the clearances in the columns.

Since the behavior of the tube can be completely different for a negative or a positive clearance, the ratio for the -0.01mm and 0.045mm configurations is not noted.

	0mm	0,045mm
0mm	1	0,81
-0,01mm	1,47	-

Table 3: *Contact force amplitude ratios at the first mode for an applied force of 80mN*

Theoretical analysis predicts a ratio greater than one for all the cases noted in Table 3. This is true for our preloaded flat bars (negative clearance). The ratio 0.81 at positive clearance is theoretically unexpected. The angle of tilt of the AVB and the precision of the clearance measurement could be reasons why the results do not follow theoretical predictions.

Table 4 shows the acceleration amplitude ratios at the first mode for an excitation force of 80mN. The acceleration ratios are calculated in the same way as the force ratios.

	0mm	0,045mm
0mm	1	3,80
-0,01mm	0,84	-

Table 4: *Acceleration amplitude ratios for an applied force of 80mN*

The results in Table 4 indicate that the acceleration amplitude of the first mode increases with the clearance. Similarly, a preload (or negative clearance) causes the acceleration to decrease.

From the preliminary results, it is shown that negative clearance increases the contact forces and decreases the acceleration amplitude and positive clearance decreases the contact forces and increases the acceleration amplitude.

5. DISCUSSION AND CONCLUSIONS

In the present paper, the experimental set-up and procedures were described for the investigation of heat exchanger tube dynamics. The effects of clearance, preload and rotation of the AVB are studied. Preliminary results show that:

- → the AVB does not change the natural frequencies of the tube;
- → the flat bars redistribute the vibration energy of the first mode to higher modes, possibly due to impacts or due to a stick and slip interaction of the tube on the AVB;
- → a negative clearance increases the contact forces and decreases the acceleration amplitude, as expected from the literature; and
- → a positive clearance increases the acceleration, as expected from the literature, but it also decreases the contact forces, which is unexpected.

From these observations, it is clear that the positioning of the flat bar with respect to the tube plays a major role on the effectiveness of heat exchanger tube supports. Further investigations will provide valuable experimental data to be used in tube dynamics models, and will allow correlations to be made between the tube motion, the contact forces and the positioning of the AVB.

6. ACKNOWLEDGMENT

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