ANALYTICAL AND EXPERIMENTAL STUDIES OF TUBE/SUPPORT INTERACTIONS IN MULTI-SPAN HEAT EXCHANGER TUBES

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Steam generators and heat exchangers are essential components of nuclear power stations. Fretting damage of tubes due to tube/support or tube/tube interaction as a result of flow-induced vibrations can have costly consequences. Present analysis techniques can generally avoid vibrations excited by periodic wake shedding and fluid-elastic instability, however low levels of tube vibration will always be present due to random turbulent excitation. This is acceptable as long as data from the successful steam generators and heat exchangers can be applied to new designs. However, as new materials and new geometries for tube supports are being considered to meet new performance requirements, e.g. inhibition of corrosion, information relating to tube vibration and fretting wear is essential.

The RMS contact forces at the tube supports have been shown to play a major role in the rate of fretting-wear. This paper presents experimental and computer simulation results for a multi-span tube apparatus vibrating under a variety of conditions. For the first time, experimental measurements of tube/support contact forces are presented for a multi-span tube vibrating in orbital motion.

The computer simulations use the VIBIC code in which beam finite elements represent the tube. The modes of vibration are obtained for a tube without intermediate supports and are numerically integrated and superimposed to yield the tube response. The normal and tangential components of the contact force at each support depend upon the tube motion and are generated separately during the simulation.

The experimental apparatus consists of an open trough with a single heat exchanger tube with realistic supports. One end of the tube is clamped to simulate the tube sheet. The trough can be mounted either horizontally or vertically. Tests are performed both in air and still water. In the vertical arrangement a vibration generator is mounted on the tube; the generator consists of stepper motors turning eccentric masses. The contact forces at the four intermediate supports are measured using four force rings each consisting of a ring supported by four miniature piezoelectric force transducers. Mid-span displacements are monitored by inductive proximity probes.

The comparison of experimental and simulated results shows fairly good agreement for the support forces and mid-span displacements for the cases where the supports have typical clearance; the predicted forces are well within a factor of two of the actual forces. The predictions are also good for supports with negligible clearance. Discussion of parameters such as contact stiffness and fluid damping is also presented.

The agreement in the results indicates that the simulation technique shows promise as a tool for the prediction of heat exchanger tube motions and support contact forces. By combining this with data relating the rate of fretting-wear to contact forces and motion, estimates of wear and tube life expectancy for a heat exchanger design should be possible.

1. Introduction

Fretting damage of tubes in steam generators and heat exchangers may cause power station outage and costly repairs. Tube fretting is the result of flow-induced tube vibration which may be excited by one or a combination of three types of mechanisms, namely: periodic wake shedding, fluid-elastic instability and random excitation due to fluid turbulence.

Present analysis techniques can generally avoid vibrations excited by periodic wake shedding or fluid-elastic instability [1]. However, low levels of tube vibration will always be present due to random turbulence excitation. Some degree of tube/support interaction is therefore unavoidable. An obvious question to be answered is "What is the acceptable level of vibration?". At present, new designs may be evaluated against designs of successful steam generators or heat exchangers. However, as new materials and new geometries for tube supports are being considered to meet new performance requirements, e.g. inhibition of corrosion, information relating to tube vibration and fretting wear is essential.

The contact forces at the tube supports play a major role in the rate of fretting wear [2]. As a part of the overall vibration and fretting program [3] a computation technique [4] was developed to predict tube motions and support contact forces. By combining the force prediction with experimental fretting wear data, tube life expectancy may be estimated.

This paper presents experimental and computer simulation results for a multi-span tube apparatus vibrating under a variety of conditions.

2. Experimental Apparatus

2.1 Multispan Trough

The experimental apparatus consists of an open trough with a single heat exchanger tube and realistic supports. The trough, which is 4.86 m long x 0.18 m wide x 0.14 m deep, is made of stainless steel and is reinforced by webs at 0.4 m intervals. Tube support specimens are mounted on brackets which are then attached to the two sides of the trough. The arrangement allows easy alignment of the supports and variation of the tube spans. One end of the tube is clamped by a metal collet to simulate the tube sheet. Fig. 1 shows the trough and the multi-span tube. The tube is excited by a vibration generator which has a mass of 0.7 kg and can be positioned anywhere along the tube. The generator consists of two stepper motors which rotate two eccentric masses in opposite directions. By choosing two predetermined eccentric masses a variety of orbital motions and excitation forces can be generated. The rotational speed of the stepper motor is controlled by varying the input pulse frequency. The excitation frequency can thus be controlled accurately.

Tests can be conducted in air or in water. For the latter, the trough is divided into an upper and a lower compartment. The dividing plate where the tube passes through is sealed by a thin flexible rubber sleeve 50 µm thick. The open side of the trough is sealed by two plexiglass covers leaving an opening of approximately 0.1 m between the two compartments. The vibration generator is attached to the tube at this open location and is thus not immersed in water. The location of the vibration generator can be changed by moving the dividing plate and re-adjusting the plexiglass plates.

The effect of the thin rubber sleeve on the tube response has been investigated and was found to be negligible. The effect of exposing a 0.1 m long section of the tube to air during tests in water was also found to be insignificant.

2.2 Instrumentation

Reaction forces at each tube support are monitored by four miniature force transducers of the piezo-electric type, each measuring 6 mm in diameter x 6 mm in length, located at 90° intervals behind the annular tube support specimen. The four transducers and the support specimen together form a unit, Fig. 2, which is attached to the adjustable support plate. The transducers are slightly pre-loaded to keep the support specimen in position.

The signals from these transducers are fed to an electronic summing device in which signals from two opposite transducers are combined to give the instantaneous reaction force along one diameter. Similarly, a second reaction force is obtained by summing signals from the other set of opposite transducers. Although only the root-mean-square values of these reaction forces are used for comparison in the present studies, the peak values can also be obtained both experimentally and analytically.

Two inductive type displacement transducers are mounted orthogonally at each mid-span to monitor the displacements.

3. Finite Element Model

The finite element model of the four span tube consists of 26 beam elements each having two nodes, Fig. 3. At each node the axial and torsional degrees of freedom are constrained. The node at the clamped end has its remaining degrees of freedom rigidly constrained. The material properties correspond to those of a nickel-iron alloy tube, 16 mm in outside diameter, from a steam generator design.

The intermediate supports are 6.35 mm thick and have a diametral clearance of 0.38 mm. Each support is centred at a node of the finite element on the tube. The mass of the vibration generator is added to the mass of the node at which it is located. The added mass and damping of the fluid between the supports is also included.

Using the VIBIC code [4] for vibration simulation, the modes (obtained from a modal analysis of the tube constrained only at the end) are numerically integrated and superimposed to provide the tube response. When contact at a support occurs, the normal force and tangential friction force are generated and applied to the tube. For the no-clearance case this procedure has been shown to be equivalent to superimposing the constrained modes (where the tube is assumed hinged at the intermediate supports) and is valid for the clearance case as well [5].

4. Results and Discussions

Tests were first conducted in air and then in water. The vibration generator was attached at the 3rd span between the 2nd and 3rd intermediate supports. Sinusoidal excitation forces with peak force levels of 5.4 N and 13.4 N at 17.5 Hz and 27.5 Hz respectively were applied at two different locations: (1) near the mid-span at node #16, and (2) near the 3rd support at node #18. These excitation frequencies are well below the constrained fundamental natural frequency of 54 Hz.

4.1 Annular Supports with Clearance

Results in Air Table 1 shows the results of four tests with different excitation frequencies and force levels. The experimental RMS values of the forces along two orthogonal axes at each of the four intermediate supports, and the vector sum of these orthogonal forces are listed.

In the VIBIC computation ten driving cycles were simulated for each test. A mean value was then calculated over the 3rd to the 10th cycles. In general, the simulations reached steady state after the 2nd cycle. The percentage difference between the experimental and simulated force levels was calculated with respect to the experimental values.

The results show that the VIBIC predictions are quite good considering that there are four supports with clearance and it is not always possible to make all these supports perfectly concentric with the tube. A comparison of the larger reaction forces, i.e., supports #2 and #3, which are of more concern from the fretting point of view, shows that the differences between the experimental and simulated results for the four tests are within ± 40%. The difference is generally larger for supports farther away from the point of excitation. It is also noted that the difference is greater for the two tests with the lower excitation force level. At low excitation levels concentricity between the tube and its support hole becomes an important factor. For example, for tests No. 1 and No. 3 the tube displacement at support No. 1 might have been less than the radial clearance and theoretically the tube would not make contact with the support if they were concentric with each other. However, experimentally, a small amount of eccentricity could have upset this balance and resulted in making contacts.

Table 2 shows a comparison of the mid-span displacements for two tests. Again the VIBIC predictions are in fairly good agreement with the experimental results.

Results in Water Tests were also conducted in still water and the results are given in Table 3.

The experimental support reaction forces are smaller in water than in air. This is possible due to the fluid damping effect in the tube/support clearance spaces. This squeeze film effect has not yet been included in the VIBIC simulation and therefore is not reflected in the analytical results.

Hence the difference between the experimental and analytical results is greater than those tested in air.

4.2 General Discussion

While the majority of physical parameters for the experimental model can be accurately determined, several other parameters such as support stiffness, structural damping, and to some extent the eccentricity of tube/support, can only be estimated at this stage. Rogers and Pick [4] have found from their single-span tube simulation studies that the choice of value for the support stiffness and damping factor is not critical. In the present studies the effects of support stiffness and tube/support eccentricity have been further investigated with the multi-span tube model.

Increasing the support stiffness by a factor of 4 from 1.75×10^6 N/m to 7×10^6 N/m only increases the force prediction of supports #2 and #3 by 1 - 2% although the changes in the two supports farther away from the excitation are larger with an increase of around 20%. Changing the relative tube/support position from nearly concentric to having an eccentricity equal to the radial clearance increases the #2 and #3 support reaction forces by 2-3%. Again, supports farther away from the excitation are affected more by this change; the increase in reaction force at these supports is around 25%.

The above findings indicate that supports No. 1 and 4, which are farther away from the excitation force, are more sensitive to deviation from the idealistic experimental model used in the simulation. Consequently the differences between the simulated and experimental results are generally higher for these two supports.

5. Conclusion

For the first time, support reaction forces at the intermediate supports and mid-span displacements of a multi-span tube in orbital motion were measured simultaneously. Comparisons between the experimental and simulated results clearly show that the computation technique is able to predict fairly accurately the tube/support interaction and dynamic response of a multi-span tube with intermittent contacts at the supports. By combining the simulated results from this computation technique with experimental data relating the rate of fretting wear to contact forces and motions, estimates of wear and tube life expectancy for a heat exchanger design should be possible.

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TABLE 1

COMPARISON OF EXPERIMENTAL AND ANALYTICAL SUPPORT REACTION FORCES (IN AIR)

	Excitation Force, N		Support Reaction Forces, N (rms)											
			Support #1			Support #2			Support #3			Support #4		
			F_{Z_1}	F_{Y_1}	F_1 $[F_{Z_1}^2 + F_{Y_1}^2]^{1/2}$	F_{Z_2}	F_{Y_2}	F_2 $[F_{Z_2}^2 + F_{Y_2}^2]^{1/2}$	F_{Z_3}	F_{Y_3}	F_3 $[F_{Z_3}^2 + F_{Y_3}^2]^{1/2}$	F_{Z_4}	F_{Y_4}	F_4 $[F_{Z_4}^2 + F_{Y_4}^2]^{1/2}$
At 1.638m from fixed end	5.4, 1.8; 17.5 Hz	Exp	0.8	0.3	0.9	3.9	2.4	4.57	4.6	3.0	5.49	1.1	1.0	1.49
		VIBIC	-	-	-	4.1	2.7	4.91	3.6	2.3	4.27	0.7	0.5	0.86
		% DIFF			-100.0			7.4			-22.2			-42.3
	13.4, 4.5; 27.5 Hz	VIBIC	3.8	1.8	4.21	15.1	7.7	16.95	14.4	7.5	16.24	3.1	1.8	3.58
		Exp	4.4	2.0	4.83	9.2	8.2	12.32	12.2	10.6	16.16	4.1	3.4	5.33
		% DIFF			-12.8			37.6			0.5			-32.8
At 1.902m from fixed end	5.4, 1.8; 17.5 Hz	Exp	0.3	0.1	0.32	2.3	1.4	2.69	6.7	4.1	7.85	1.2	0.9	1.5
		VIBIC	-	-	-	1.4	0.9	1.66	5.1	3.2	6.02	0.5	0.5	0.71
		% DIFF			-100.0			-38.3			-23.3			-52.7
	13.4, 4.5; 27.5 Hz	VIBIC	0.9	0.4	0.98	3.6	2.9	4.62	13.7	7.9	15.81	1.6	1.0	1.89
		Exp	0.8	0.5	0.94	3.7	3.3	4.96	13.8	9.3	16.64	2.2	1.8	2.84
		% DIFF			4.2			-6.9			-5.0			-33.5

TABLE 2

COMPARISON OF MID-SPAN DISPLACEMENTS
(IN AIR)

Excitation Force, N			Mid-Span Displacement, mm (RMS)			
			Span #1 D_1	#2 D_2	#3 D_3	#4 D_4
At a point 1.438m from the fixed end	13.4, 4.5 at 27.5 Hz	Exp	0.042	0.097	0.328	0.131
		VIBIC	0.073	0.126	0.431	0.108
		% DIFF	74.0	30.	31.0	-18.0
	5.4, 1.8 at 17.5 Hz	Exp	0.019	0.089	0.205	0.105
		VIBIC	0.036	0.095	0.278	0.132
		% DIFF	89.0	6.7	35.6	25.7

TABLE 3

COMPARISON OF SUPPORT REACTION FORCES
(IN WATER)

Excitation Force, N			Support Reaction Forces, N (rms)											
			Support #1			Support #2			Support #3			Support #4		
			F_{Z_1}	F_{Y_1}	F_1 $[F_{Z_1}^2 + F_{Y_1}^2]^{1/2}$	F_{Z_2}	F_{Y_2}	F_2 $[F_{Z_2}^2 + F_{Y_2}^2]^{1/2}$	F_{Z_3}	F_{Y_3}	F_3 $[F_{Z_3}^2 + F_{Y_3}^2]^{1/2}$	F_{Z_4}	F_{Y_4}	F_4 $[F_{Z_4}^2 + F_{Y_4}^2]^{1/2}$
At 1.638m from fixed end	5.4, 1.8; 17.5 Hz	Exp	0.4	0.3	0.5	3.1	2.4	3.92	3.5	2.3	4.19	0.8	0.5	0.94
		VIBIC	-	-	-	4.5	3.0	5.40	4.0	2.4	4.66	1.2	0.7	1.39
		% DIFF			-100.0			37.8			11.2			47.9
	13.4, 4.5; 27.5 Hz	Exp	3.4	1.7	3.80	9.9	6.7	11.95	12.4	8.1	14.81	2.9	1.8	3.41
		VIBIC	5.0	2.4	5.55	16.7	8.4	18.69	15.1	8.1	17.14	3.7	2.0	4.21
		% DIFF			46.1			56.4			15.7			23.5
At 1.902m from fixed end	5.4, 1.8; 17.5 Hz	Exp	0.2	0.2	0.28	1.4	1.5	2.05	4.5	4.7	6.51	0.6	0.5	0.78
		VIBIC	-	-	-	1.6	1.3	2.06	5.3	3.0	6.09	0.3	0.4	0.50
		% DIFF			-100.0			0.5			-6.5			-35.9
	13.4, 4.5; 27.5 Hz	Exp	1.4	1.4	1.98	2.4	2.4	3.39	9.8	9.9	13.93	1.0	1.0	1.41
		VIBIC	0.7	0.7	0.99	4.3	3.4	5.48	14.5	9.0	17.07	2.4	1.8	3.00
		% DIFF			-50.0			61.7			22.5			112.8

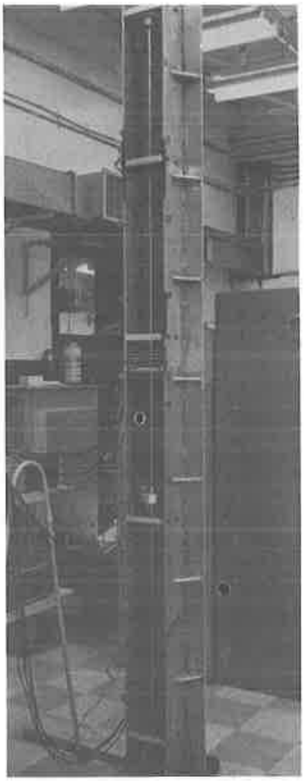

Fig. 1(a) Multi-Span Tube Trough for tests in air.

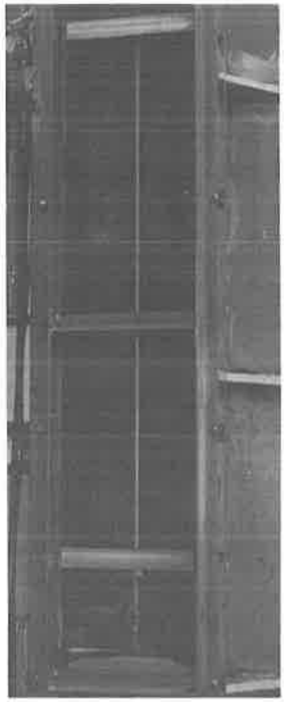

Fig. 1(b) Close-up View of Upper Water Compartment

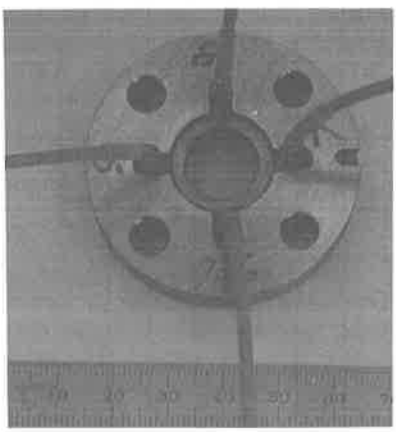

Fig. 2 Close-up View of Force Transducers and Tube Support Rings

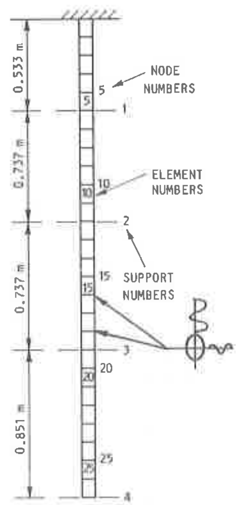

Fig. 3 Model of Multi-Span Tube using 26 Finite Elements