MASTER

VIBRATION TESTS OF A
FULL SCALE WATER MODEL OF A
CLINCH RIVER STEAM GENERATOR MODULE

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(189a, 13887) LMFBR Steam Generator

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ABSTRACT

Cross and parallel flow induced vibrations were measured in a shortened full scale model of the CRBRP steam generators using water as the working fluid. Measurements were made on five selected tubes, some of which fit through oversize tube support holes. Power spectral density plots were made of vibration parameters at selected locations. Maximum displacement was about 3 mils and occurred at the lower frequencies. The maximum vibrational stress was calculated to be 2000 psi. Measurements of vortex shedding frequencies showed that vortices are not a dominant source of tube excitation. In general, no adverse effects on the CRPRP steam generators were identified which affect service life or structural integrity.

The data were examined only for verification of the prototype design. Additional data are available for more fundamental analysis or correlation of the test results with mathematical models.

INTRODUCTION

From about June 1974 to June 1976, Atomics International under contract to General Electric, designed, fabricated, and tested a full size, 757 tube Hydraulic Test Model (HTM) of AI's hockey stick design of a Clinch River steam generator. The model is identical to the prototype steam generator in internal configuration except that the active length was shortened from about 46 to 25 feet. Figure 1 shows the model layout.

The HTM was conceived as a test vehicle to verify or provide design data for the prototype steam generator. The model was fabricated from carbon steel instead of the more expensive 2-1/4 Cr-1 Mo steel but otherwise incorporates all of the internal design features of the steam generators. The purpose of the program was (1) to measure the amplitude and frequency of both shell and tube side flow induced vibrations and (2) to describe the shell side flow characteristics. This paper reports the results of the vibration tests performed to verify the plant designs. The results of the flow tests will be reported in a later paper.

The fluid parameters affecting tube vibration and shell side flow characteristics (velocity, viscosity, and density) are generally well known, but geometrical effects are less established. Thus, it is satisfactory to use water rather than sodium for hydraulic tests, but full size geometry is needed. Natural frequency and dampening of the model is the same for both fluids; however, Reynold's number is approximately three times greater for water than for sodium. This is most important in determining the effect of Reynold's Number on Strouhal Number. In order to simulate sodium Reynold's Number, five of the six entry windows were blocked. This allowed high prototype Reynold's Number measurements of vortex shedding frequencies. The effect on cross flow vibration was less meaningful because the excitation force was about three times that with sodium.

The specific objectives of the test program were:

- (1) To determine the natural frequency of the tubes in air and water.
- (2) To measure the damping rate of the tubes in air and water.
- (3) To measure the response of the tubes, support plates, shroud, liners, baffles, and shell to prototypic flow induced forces.
- (4) Use the above data to make preliminary verification of the CRBRP steam generator design.

The CRBRP steam generators consist of shell and tube, fixed tubesheet evaporator and superheater modules. The evaporator and superheater modules are identical in design and are, therefore, interchangeable. They are of the hockey stick configuration, i.e., a 90° bend and a short, straight section in the shell and upper tube bundle provide tube flexibility to accommodate differential thermal expansion between the shell and tubes. Sodium flow is on the shell side, counter current to the water/steam flow upward through the tubes. Each secondary loop of the CRBRP contains two of these units operating as evaporators, and one operating as a superheater. The evaporators are piped in parallel; therefore, sodium flow on the shell side is 50% of superheater sodium flow.

Significant design parameters of the CRBRP Steam Generator modules are tabulated in Table 1. Table 2 is a comparison between the prototype Steam Generator and the Hydraulic Model with respect to vibration characteristics. The entries in the table are calculated numbers, generated prior to any test results.

TEST DESCRIPTION

The flow test facility used for testing of the HTM is located at the Rocketdyne Division of Rockwell International. This facility (Figure 2)

TABLE 1
DESIGN PARAMETERS FOR THE CRBRP STEAM GENERATOR MODULE

	<u>Evaporator</u>	<u>Superheater</u>
Thermal Power (MWt)	117	91
Sodium Flowrate (lbm/hr)	13.36×10^6	13.49×10^6
Operating Sodium Inlet Temp ($^{\circ}\text{F}$)	884	921
Operating Steam Outlet Temp ($^{\circ}\text{F}$)	626	905
Operating Steam Outlet Pressure (psia)	1866	1540
Shell Diameter (in.)	53.5	53.5
Tube Diameter x Wall (in.)	5/8 x 0.190	5/8 x 0.109
Tube Pitch, in.	1.22	1.22
Number of Tubes	757	757
Tube Length, Tubesheet to Tubesheet (ft)	62.7	62.7
Material of Construction	2-1/4 Cr-1 Mo	2-1/4 Cr-1 Mo

TABLE 2
COMPARISON OF VIBRATION CHARACTERISTICS
OF PROTOTYPE STEAM GENERATOR AND HYDRAULIC MODEL

	Prototype	Hydraulic Model
Natural Frequencies***		
Elbow region	37 - 232	34 - 237
Inlet (cross-flow) span	307	315
Adjacent span	135	.36
Mid-section span	37	34
Lower end spans	33	30
Estimated Damping	10%	10%
Analysis		
Cross-Flow Excitation		
Frequency	all	
Magnitude	2.5 lb	3 lb
Max amplitude (cross-flow span)	2-5 mil**	Small
(other spans)	1 mil**	
Parallel Flow Amplitudes		
Bergreen (Ref. 2.1)	3.2 mil	1.3 mil
ANL (Ref. 2.2)	.1 mil	.1 mil
Pardousis(Ref. 2.3)	6.7 mil	4.3 mil
Test Conditions (at an axial velocity of 12 ft/s		
Reynolds Number	6.4×10^5	1.9×10^5
Strouhal Number	0.32*	0.32
Shedding frequency (100% flow)	70 - 150	70 - ?
Pump input - frequency	97.5 - 195 Hz	300 Hz ⁺
P	± 4 psi	± 1 psi ⁺

⁺With the four pumps in the hydraulic loop, characteristics will be changed.

*Assumes S vs Re remains flat out to this range.

**When excited at all natural frequencies.

***Calculations based upon single span analysis.

was designed for water flow testing of components at flows up to 32,000 gal/min. Four 2,000 hp synchronous electric motor driven pumps take suction from a 200,000 gallon reservoir of water maintained at swimming pool type cleanliness located beneath the facility. The output of the four pumps feed a header which supplies flow to the model. The HTM was installed between the supply and a discharge system which drained into the reservoir. Flow loop operation was remotely controlled from the facility's control and instrumentation room. Facility flow measurements were made with a magnetic flowmeter that had a calibrated accuracy of ± 250 gal/min. For the HTM test, the maximum flowrate achieved during vibration testing was 29,400 gpm, or 96% of the prototype design volumetric flowrate. The model incorporates all the internal elements (baffles, tube supports, shrouds, flow distributors, etc.) required to properly simulate the CRBRP Steam Generator as show in Figure 1. The active length of the model was 25 feet less than that of the prototype (21 versus 46 feet long). The model was sufficiently long to establish steady flow; the shorter length made possible cost savings in the fabrication of the HTM and test stand without affecting the test results.

As shown in Figure 1, flow enters the shell side of the unit through one inlet nozzle, flows into and around the inlet annulus, turns and flows up the annulus; the flow then turns inward and flows across the tubes into the tube bundle, through the six windows in the shroud. The flow then turns downward and flows parallel to the tube bundle down the active length. At the outlet end, the flow exits the tube bundle through six windows in the shroud, turns and flows up the annulus, and out through two outlet nozzles (only one for the evaporator configuration).

The bent tube portion of the unit (hockey stick region) is isolated from major flow currents by the disc-donut baffle assembly. A small flow does exist in this region due to the bleed vent flow out of the top of the unit, but this flow is so low that it cannot produce tube vibration. The close coupled support of the disc donut tends to isolate the hockey stick region from the excitation of the tubes in the active portion of the unit. Additionally, three vibration suppressors restrict vibration of tubes in the hockey stick region. The vibration suppressor is a comb-like structure restricting tube motion to one plane. During the fabrication of the test model, it was noted that the tubes rested against the vibration suppressors and limited in-plane motion such that they would probably act as a support point for vibratory motion.

During fabrication of the Steam Generator Hydraulic Model single Endevco Model 2226, accelerometers were bonded to the five selected tubes at elbow bend locations. These accelerometers are sufficiently small (3/8-in. hex times 0.33 high, maximum envelope) and light (2.8 grams) that they did not affect the response of the tube or result in interference with adjacent tubes during vibration. Each unit was waterproofed during installation ^{p/1)} the leads ~~were~~ passed through the shell and sealed with epoxy. Figure 3 shows the location of all accelerometers. At other tube locations, two Model 2226 sensors were oriented 90^0 to each other and bonded to a common block which, in turn, was glued to the steam tube in order to measure biaxial movement. A movable support "dolly", accommodating two Endevco Model 2226 accelerometers, was used to measure biaxial response normal to the steam tubes. The dolly was inserted in and held tightly against the wall of the steam tube. The portability of the assembly permitted a rapid relocation to any axial position. The IITM tests confirmed the excellent correlation with bonded accelerometers which had been observed during the AI/MSG tests. Micro-miniature Endevco Model 22 accelerometers were mounted at the approximate center of the tube spacers No. 7 and 8. Endevco Model 2221C and 2217 accelerometers were mounted at various locations on the shell, shroud, and thermal liners.

Accelerometer output voltages were initially amplified by Model 7301/MQ Dynamics DC amplifiers for transmission over the 200 ft of cable from the test stand to the instrumentation control room. This signal was amplified a second time by Model 6050/6102 DC Dynamics amplifiers with the resultant signal being recorded directly on a Model VR-3300 CEC tape recorder.

Hot wire anemometers were employed to determine the shedding frequency occurring on outer row steam tubes at the shroud inlet windows. The probe used in these tests was a Model 1212-60w, Thermo Systems, Inc. unit. The diameter of the sensing element was 0.006 in. with a length of 0.08 in. This particular unit was selected because it permitted convenient entry into the model's existing penetrations and because the sensing element could be positioned behind the steam tube where vortexing measurements were desired. Data obtained from the anemometer system were recorded directly on magnetic tape using a CEC Model 330 recorder. Playback from the tape was made on both oscillographs and on X-Y plotters (via real-time analyzers).

Excitation of steam tubes for the dry exploratory tube vibration surveys was accomplished by using a Goodman 25 lb force output electrodynamic shaker driven by a 200 watt power amplifier. A Ling Model CO-10-A cycling oscillator provided the frequency signal and maintained a desired amplitude (force control). Shaker force was monitored by a Wilcoxon Model Z602 load cell in series with the shaker drive bar. The shaker

force was recorded during the entire frequency sweep on magnetic tape along with the outputs of strategically located tube accelerometers. Load from the shaker was transmitted to the steam tubes by a bar assembly and load cell bolted to the shaker armature.

TEST DATA

Five types of tests were conducted. The data collected from these tests are discussed below:

Dry Vibration Tests - Instrumented tubes on the outer diameter of the tube bundle were mechanically shaken with the Goodman Shaker with air on both the shell and tube sides. The shaker was slowly swept from 20-500 hz after being set to the desired load level at 80 hz. Three load levels were used for this test: 1.5, 2.5, and 3.5 lbs (peak to peak). At 1.5 lbs the tube was relatively quiet, at 2.5 lbs audible noise of the tube impacting in its supports started to become noticeable, and at 3.5 lbs the noise of the impacting was very pronounced. The amplitude of tube vibration did not always increase with load. It was observed that between 2.5 and 3.5 lbs, where impacting noise was very pronounced, the levels of tube displacement remained constant or decreased slightly. It is believed that this is due to the non-linear effects of the tube-to-spacer impacting coming into effect.

The tubes did not respond uniformly to input loads for all frequencies. When the shaker was driven at a constant load, the load measured between the shaker and the tube would fluctuate as a function of driving frequency. In one tube the measured load dropped two orders of magnitude over a narrow frequency band, but no marked changes were observed in the displacement levels of the tube. In general, the response characteristics of the measured load were the same for all five tubes. Resonances ranged from 20 to 500 hz for all tubes.

Flow Vibration Tests - Flow vibration testing was performed by flowing water at various flow rates through the shell side of the unit (dry water side) up to prototypic velocities. Although normally desirable, over testing of the unit was not provided due to facility limitations. The vibrational response was obtained for four tubes along the active length, in the hockey stick region, and in the lower stagnant region with both internally and externally mounted accelerometers. Measurements were made of the tube support, liner, shroud, and shell vibration levels at the locations shown in Figure 3.

The vibration levels and resonant frequencies of the tubing, shells, liners, shroud, and spacers are shown in Tables 3 and 4 must be interpreted with care because water was used instead of sodium and because of the geometrical differences between the model and the prototype. The use of water to simulate sodium introduces an error in Reynolds number which may affect the cross flow vortex shedding frequency, the cross flow lift and drag forces, and the parallel flow excitation forces. Additionally, due to the difference in density between the two fluids, the velocity head and thus the flow induced excitation forces, are changed. Structural differences exist, including overall length, shell thicknesses, and material. Also differences in temperature and internal tube pressures and fluids must be considered. The influence of these factors has been estimated, and based on data presented in the open literature (Reference 3.2) resonant frequencies should be within 5% of each other for the HTM and prototype, and the vibration levels in the HTM may be 30% lower than those in the prototype. These differences will be accounted for in the final design of the prototype unit.

Examination of power spectral densities (PSD's) for various spans along the length of the tube shows that the whole tube becomes involved in single mode shapes and thus simplified analyses assuming each span acts independently are incorrect. For example, the 14" cross flow span participates in a 42 hz mode shape, while analysis would predict that a

Table 3
SUMMARY OF VIBRATION DATA
FOR TUBE 3-099, FULL FLOW TEST

<u>Region</u>	<u>Span</u>	<u>Span Length</u> <u>(in.)</u>	<u>P-P RMS Displacement</u> <u>(mils)</u>	<u>Major</u> <u>Resonances (hz)</u>
Hockeystick Region	M	34.1 (R=63.25)*	1.3	
	N	33.1 (R=63.25)*	3.8	
	O	46.0	2.8	
Active Region	A	15.5	3.5	
	B	22.6	1.5	
	F	46.0	0.8	
	J	22.6	0.8	
	K	15.5	0.7	
Lower Stagnant Region	L	49.0	6.2	

* R is the Bend Radius

Table 4
SUMMARY OF VIBRATION DATA FOR COMPONENTS OTHER THAN TUBES,
FULL FLOW TEST

	<u>Location</u>	<u>P-P RMS Displacement</u> (mil)	<u>Major Resonances</u> (hz)
Shell	P	113	18, 31
	Q	6.3	22
	R	4.5	27, 38, 83
	S	6.3	19
Shroud	W	0.3	31, 58, 68, 78
Inlet Liner	V	0.7	42, 67, 80
Outlet Liner	X	2.0	20
Spacer	U	0.1	57
	T	0.1	57

single span of that length would have a first mode shape of 315 hz. Resonant frequencies for tube 3-099* are shown in Table 3. Note that only frequencies below 150 hz were predominant, compared to dry vibration testing, where the tube showed resonances over the total range of the test, 20-500 hz. This could be due to either dampening out of higher frequencies with water or low excitation from shell side flow at the higher frequencies. Experiments by Burgeen (Reference 1) showed that parallel flow excites the tube at its resonant frequency. This may be what occurs here. Several dry vibration tests were run where the tube was excited by a white noise signal to the shaker, a similar excitation mechanism to that postulated by Wambsganns (Reference 2) for parallel flow induced excitation; results of these white noise tests showed spectra similar to the flow test data. That is, higher frequencies did not appear as they did in the discrete frequency excitation dry vibration tests. This result supports the conclusion that excitation of the tube under prototypic flow conditions is by white noise, rather than by discrete frequencies.

Results of a comparison between analytically predicted** and experimentally measured resonances for Tube 3-099 are presented in Table 5 for the active region of the tube bundle. Direct comparison between the measured and predicted frequencies is not obvious, but the lowest resonance frequency of the tube is predicted. Analytical prediction of these types of results is difficult, since numerous support conditions can exist for the tube. For example, any tube support (there are 16 support points in the HTM) the tube may be centered in the support hole, not contacting unless excited, or resting against one edge, or wedged tightly against one side of the support.

* Tube 3-099 is a long radius tube ($R = 63.25$ in.) on the outside row of the tube bundle

** Analytical predictions were made on Atomics International Computer Code DRIPS. DRIPS is a finite element structural analysis code, capable of solving a variety of static and dynamic structural problems.

TABLE 5
COMPARISON OF PREDICTED AND EXPERIMENTAL RESONANCES
RESONANT FREQUENCIES (Hz)

<u>Active Region</u>	
Predicted	Measured
41	37
65	42
105	52
139	58
171	67
186	86
191	100
and higher	140

Amplitude levels for various spans of tube 3-099 observed for the full flow condition are shown in Table 3.1. Using the relation developed by Paidousis for parallel flow induced vibration (Reference 3), the center span of 46" would be expected to displace 4.3 mils* at full flow. Displacement of this span was seen to be 0.8 mils. The 46" span on three other tubes had maximum deflections of 1.3 mils.

Correlation of the shell and tube accelerometers was attempted at several axial locations. Results showed that in the hockey stick region, where the shell is coupled directly to the tube bundle by both sodium and the mechanical action of the vibration suppressor, good correlation exists in the shell and tube signals. However, in the portions of the unit where the shell does not support the tube bundle, which is over most of the unit, the measurements would not be related. This is significant, because during operation of the plant units, vibration monitoring must necessarily rely on external, shell mounted accelerometers because of the adverse internal environment (high temperature liquid sodium). Based upon the analysis of the test results to date, it appears that the shell side monitoring of the tubes is feasible for the bend region of the tubing. Due to the high attenuation introduced by the presence of liners and shrouds in the remainder of the unit, shell side monitoring elsewhere will not provide acceptable results.

Stress levels produced in the tubing of the Hydraulic Model by flow induced vibration were predicted using analytically calculated mode shapes and experimentally determined displacement. The results are shown in Table 6. The maximum stress was calculated to be 2000 psi, and the acceptability of this stress will be determined in the final design analysis of the CRBRP prototype steam generator.

* Considering the range of data compiled by Paidousis, this value could vary from 1.3 to 6.4 mils.

Table 6
VIBRATION INDUCED STRESS LEVELS IN HTM TUBES

Span	Flow Condition	Resonance Frequency	Peak-to-Peak Displacement Amplitude	Calculated** Stress
A (15.5")	Cross Flow	42 hz	4 mil	2000 psi
F (46")	Parallel Flow	52 hz	1 mil	60 psi
L (49")	None*	36 hz	6 mil	400 psi
N	None*	18 hz	4 mil	20 psi

* Span located in a stagnant region

** Stress calculated using mode shape from analytical model of the HTM tubes

Vortex Shedding Tests - Vortex shedding occurs at discrete frequencies which can produce resonant conditions and lead to severe structural damage. The literature (Reference 4) shows that for a single tube vortex shedding the Strouhal number (a correlating parameter for cross flow vortex shedding) jumps in the Reynolds number range of interest in the prototype. In order to obtain prototypic Reynolds numbers (1.5×10^5), five of the six inlet windows were blocked off to increase the velocity to compensate for the decreased density of water. At the higher velocities, problems occurred with breakage of the anemometer wires. Data obtained for less than 100% flow were incomplete at the time of publication.

The tests to date show Strouhal numbers for the HTM to be approximately those reported by Chen (Reference 5) for tube bundles with dimensions of the prototype. Chen reported a Strouhal number of 0.31 and the tests measure one of 0.3 to 0.4.* More importantly, the test shows that vortex shedding does not appear to be a predominant excitation mechanism for the prototype. There was not a strong vortex shedding frequency, and it was hard to distinguish it experimentally from the background turbulence.

Simulated Steam Testing - The possibility of the tubes being excited by droplet impingement on the tube side was investigated by flowing a variety of air-water mixtures through several tubes. Accelerometers were used to monitor the tubes in the bent region of the hockey stick. The test was performed under two conditions, one with the shell side empty and the other with flowing water on the shell side. The first allows more detailed perception of tube excitation, while the latter shows if any coupling between several excitation mechanisms might exist.

* Incomplete test results

Results of the test show that no excitation of the tubes was perceptable, and any excitation in the range from 10-500 hz is less than 0.1 mil. This result is as expected, since tube side flow is not a significant excitation force.

Damping Measurements - Damping measurements were made of various tubes at select locations with and without the presence of water on the shell side of the unit. Measurements were made of damping using the logarithmic decrement, which was determined by striking the tube and observing the envelope of the decaying signal. Measurements made by this technique showed damping of 2-1/2% for both with and without water on the shell side of the unit. Although the presence of the water on the shell side of the unit did not affect the damping, it did produce a drop in the resonant frequency of the tubes. The method used here for determination of damping is conservative, since it only considers linear damping, and does not involve non-linear events, such as tube-to-spacer impacting, which add to the total damping of the system.

TEST RESULTS

The acceleration-time histories which constituted the results of the flow induced vibration tests were examined in the following ways to verify the structural integrity and service life of the CRBR steam generators:

Separation of Frequencies - In the cross flow region, the maximum expected excitation frequency (for example, due to vortex shedding) should be a factor K below the corresponding lowest structural natural frequency. This simplified rule can sometimes be applied and stress calculations obviated. It is usually considered adequate if K=3 in straight tube regions and K=4 in tube bend regions.

In the hydraulic model the fundamental tube frequency for a continuously supported tube is 42 hz in the active region. The vortex shedding frequency is 70 hz and flow induced dominant frequencies are seen as low as 18 hz in the hockey stick region. Therefore, there is insufficient separation of frequencies and the oscillating stress range of the tubes was examined.

Oscillating Stress - The oscillating stress range of the tubes should be low enough to avoid fatigue failure. Fatigue evaluation should also consider the effect of mean stress due to other sources of loads on the tubes. The following additional considerations should be taken into account:

a. The frequency of flow induced vibration is usually high. When the tubes respond in a simple harmonic oscillation, the corresponding alternating stress should be low to endure a large number of cycles (about 10^{10} to 10^{12}). In practice, it is almost impossible to obtain fatigue data at this high cycle region, but conservatively defined "endurance limit" stress levels can be specified for design purpose.

b. In many cases of turbulent excitation, the tube responses are of random amplitude at several dominant frequencies, usually the natural frequencies of the tube. The above requirement can be modified to take advantage of the varying amplitudes. It is usually a good approximation to assume that the response is stationary and is of Gaussian distribution. Thus, the amplitude distribution will be a Rayleigh distribution and can be easily related to the RMS value of the response. The rule of accumulative damage can be assumed to be valid and the total sum of partial usage factors from all participating responses should be less than the fraction of usage allocated for tube vibration. A criterion in terms of the RMS values of the response at all dominant frequencies can thus be obtained. The calculated or measured RMS values of response should fall below the allowable values.

With the amplitudes and frequencies of vibration shown in Table 3, the maximum vibratory stress is 2000 psi, occurring in tube 3-099. This stress assumes a continuously supported tube vibrating at a mode principally at the cross flow span. This stress level will be examined during the prototype design to determine its acceptability for plant service.

Impact Wear - Fretting and wear due to tube vibration at the tube support locations for the lifetime operation should be less than the amount of additional wall thickness provided. The amount of wear is generally obtained from simulation tests and extrapolated to the design life of the steam generator.

The maximum displacement must be less than half the minimum separation between tubes to avoid tube-to-tube impacting. From Table 3 it can be seen that the maximum vibration amplitude at the equivalent full sodium flow condition is +3 mils. This is much lower than half of minimum tube separation and no tube impact is anticipated.

CONCLUSIONS

It is concluded from these tests that the CRBR steam generators will, in general, experience no adverse structural or endurance problems from the extra load imposed on the tubes from vibrational forces. The maximum allowable oscillating load will be determined during prototype design.

The data were analyzed only for their applicability to the CRBR steam generators. Over 400 power spectral density plots were obtained and these are available for correlation with mathematical models or more fundamental analysis.

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Figure 1 Hydraulic Test Model Layout

- FIXED ACCELEROMETERS ON TUBES
- ▲ MOVABLE ACCELEROMETERS IN TUBES
- ▼ FIXED ACCELEROMETERS ON SHELL,
- FIXED ACCELEROMETERS ON SHROUD
- ⊕ FIXED ACCELEROMETERS ON LINERS
- ⊗ FIXED ACCELEROMETERS ON TUBE SUPPORTS

Figure 3 Accelerometer Locations