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MINIMUM LEAKAGE CONDENSER TEST PROGRAM

MASTER

May 1978

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Westinghouse Electric Corporation
Power Generation Divisions
Steam Turbine Division Engineering
Lester, Pennsylvania

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REPORT

for

MINIMUM LEAKAGE CONDENSER TEST PROGRAM

Prepared for the
UNITED STATES DEPARTMENT OF ENERGY

Per Contract
EY 76-C-02-2976

by

WESTINGHOUSE ELECTRIC CORPORATION
Power Generation Divisions
Steam Turbine Division Engineering
Lester, Pennsylvania 19113

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ABSTRACT

This report presents the results and analysis of tests performed on four critical areas of large surface condensers: the tubes, tubesheets, tube/tubesheet joints and the water chambers. Significant changes in operation, service duty and reliability considerations require that certain existing design criteria be verified and that improved design features be developed. The four critical areas were treated analytically and experimentally. The ANSYS finite element computer program was the basic analytical method and strain gages were used for obtaining experimental data. The results of test and analytical data are compared and recommendations made regarding potential improvement in condenser design features and analytical techniques.

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FOREWORD

Baseload electric generating plant outage costs can reach up to \$500,000 per day, especially for large nuclear plants. The ten year equipment availability summary published by the Edison Electric Institute shows that approximately 40 hours per nuclear unit per year of forced outages are caused by condenser problems. Of the total forced outages reported by type, over 50% are due to tube failures. When these two facts are combined with the fact that a condenser's service life is normally intended to be 30 to 40 years, it is clear that there is a strong incentive to improve condenser reliability.

For the above and other reasons, the work described in this report was undertaken to study specific areas in condensers that have historically caused power outages, and to make recommendations which are in accordance with the results of this work. The areas investigated are:

- Tube vibration
- Tube pullout
- Tube plate stress distribution
- Waterbox stress distribution

The work was performed on a prototype condenser at the Lester facility of Westinghouse Electric Corporation.

With respect to waterbox design, tubesheet sizing, and tube pullout load prediction, it was found that a finite element stress analysis program was adequate to predict the stresses encountered during the experimentation. In addition, the vibration characteristics of titanium tubes were experimentally determined so that future condensers may be designed with the proper data. Finally, the impact of the experimental conditions on the rolled tube-to-tubesheet joints has been examined.

SUMMARY

Being cognizant of the costs of outages (especially in larger nuclear plants), recent regulatory changes, and design trends, the Department of Energy and Westinghouse Electric Corporation undertook the joint financing of this project. The objective of the project was to gain more information about stress levels in condenser components, titanium tubes, and double/integrally grooved tubesheets, so that more reliable condensers can be made for the power generation industry. The findings may be used to develop more advanced condenser designs with special emphasis on reducing tube leakage due to vibration and tube-to-tubesheet joint causes.

The prototype condenser used for the testing has 24,000 square feet of surface area, titanium tubes, and measures 40 feet by 12 feet by 10 feet (L-W-H). The thermal rating of this condenser is 2.5×10^8 BTU/hr. The prototype incorporates both double and integrally grooved tubesheets.

The condenser was suitably instrumented for the performance of the following tests:

- Determine the stress levels at specific locations on the condenser tubesheet under loading imposed by pressure on the cooling water side.
- Determine the stress levels at specific locations on the condenser waterbox under loading imposed by pressure on the cooling water side.
- Determine the tube stresses under loading imposed by pressure on the cooling water side.
- Determine the vibration characteristics of titanium tubes.
- Determine the magnitude of tubesheet stresses under thermal loading conditions.
- Examine for tube-to-tubesheet leakage before and after the above tests.

The results of these tests, when compared with analytic techniques, indicate that commercially available finite element stress prediction programs are adequate for understanding the stress patterns and magnitudes in the tubes, tubesheets and waterboxes. In other words, existing sophisticated design techniques are sufficient for predicting the pullout and overstress conditions in condensers. These techniques are, however, more exhaustive than those conventionally used for condenser design.

The results presented in this report also show that condensers utilizing titanium tubing can be designed in such a way that vibration failures are very unlikely to occur, and that tubesheet designs are available which will minimize the incidence of tube-to-tubesheet joint leakage.

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1 INTRODUCTION

Condenser reliability has been receiving an increasing amount of attention in recent years, and there are several reasons for this. One reason is that reports are available which clearly define some areas of condenser design that are in need of refinement. One report, the Edison Electric Institute (EEI) ten year equipment availability summary, indicates that condensers in nuclear plants cause approximately 40 forced outage hours per year. Since forced outages can cost up to 500,000 dollars per day in replacement power costs for large nuclear units, this outage-hour information is very important. Another report by the Electric Power Research Institute [1]*, concentrated on the identification of tube leak problems in a field survey analysis.

A second reason for the emphasis on condenser reliability stems from the Environmental Protection Agency's 1974 Effluent Guideline and Standards (40 CFR 423). These standards have caused many power plant designers to use cooling towers and ponds instead of once-through cooling. This has generally caused increases in the amount of dissolved solids and in the pressure of the cooling water. These effects combine to make the need of minimum leakage condensers more critical because of the incremental damage that a leak of a given size could cause due to the increased pressure and solids contamination.

In addition, in the case of PWR nuclear steam supply systems, where only volatile chemicals are recommended for water treatment in the secondary or turbine cycle, condenser leaks will result in scale formation, and perhaps accelerated corrosion, in steam generators.

Finally, the matter of condenser in-leakage is complicated by the general industry trend to increasing size, and the fact that tubes must perform a structural function as well as a heat transfer function.

It is for these reasons that the efforts described in this report were undertaken. The work has been primarily aimed at investigating existing design practices through experimentation, thus providing information on the applicability of the design practices to more advanced design features.

The apparatus used in this effort was a prototype condenser with an area of 24,000 square feet located at the Westinghouse Steam Turbine Division in Lester, Pa. It measures 40 feet by 12 feet by 10 feet high, and has a heat load rating of 2.5×10^8 BTU/hr. It is a two-pass heat exchanger with two tube bundles located in the same shell in a horizontal plane. This condenser has 1 inch OD titanium tubes and a double tubesheet at one end, with an integrally grooved tubesheet at the other end. The condenser was equipped with extensive instrumentation to measure actual stresses, temperatures, pressures, etc., under a variety of loading conditions.

*Bracketed numbers refer to listed references on pages 77 and 78.

The experiments consisted of the following:

- Filling the cooling water side with water and pressurizing to 60 psi/Measure strain in the tubes, tubesheets and waterboxes.
- Vibrating the titanium tubes/Measure amplitude and frequency.
- Thermally stressing the tubesheets/Measure strain in the tubes and tubesheets.
- Pressurizing the space between the two parts of the tubesheets/Measure pressure decay rate.

Some of these experimental results were reduced to a form which could be used for comparison with conventional prediction methods. These comparisons are made, most often, in tabular and/or graphical form. Sections 3, 4, and 5 deal with this type of analysis for the tube, tube plates and water chambers. Section 7 deals with the thermal testing of the tubesheets.

The remainder of the results were used to make recommendations regarding the advisability of incorporating some of the advanced design features into future power plant condensers. See Section 6 for titanium tubing vibration characteristics and Section 8 for results of a pressurization test on rolled tube-to-tubesheet joints.

2 EQUIPMENT DESCRIPTION

The Westinghouse Heat Transfer Products Section of the Steam Turbine Division constructed a prototype condenser to be used in the Steam Turbine Development Laboratory for turbine testing. In realizing the need for condenser design improvements to achieve greater reliability, it was decided that the condenser itself could be used as a test vehicle. The condenser was, therefore, equipped with design features that are now thought to be significant steps towards providing more reliable condenser service; the added design features are titanium tubes and special tube plates. An overall picture of the condenser is shown in Figure 1.

The titanium tubes were used because recent data (see reference 1, for example) has shown that they are considerably more resistant to the normal causes of failure in environmental and service conditions. This is very important because of the magnitude of the financial impact of unscheduled downtime and the potential damage to other power plant equipment due to leakage of contaminated cooling water into the system; thus, designs using titanium tubes should be validated so that they may be confidently selected for future condensers. The specific area of concern was their vibration characteristics since several titanium tubes have failed in service due to vibration induced damage.

There were two types of tube sheets installed on the condenser: one is a double tubesheet which is made from two pieces of plate and then pinned together with a 1/8 in. gap between them; the other is an integrally grooved tubesheet which is made from a single plate and then machined so that an interconnected annular space (1/8 in. wide) at the tubesheet center is formed.

These two types of tubesheets were installed to demonstrate the manufacturability and operational characteristics of the two different methods of added leak protection. In both cases, the concept is to use the gap or annular space by pressurizing it with condensate water. Then if a leak exists, contamination of the turbine loop water with circulating water does not take place because the pressure in the annular space may be kept higher than the circulating water pressure.

To illustrate this, there are figures which show:

- The tube hole configuration for integrally grooved tubesheet (Figure 2).
- The tube hole configuration for the double tubesheet (Figure 3).
- An isometric/cut-away view of the integrally grooved tubesheet (Figure 4).

Table 1 summarizes the major characteristics of the prototype condenser, and Appendix I contains sketches of the condenser tube pattern and water chambers.

TABLE 1
PROTOTYPE CONDENSER CHARACTERISTICS

Surface Area	24,000 square feet
Thermal Rating	2.5×10^8 BTU/hour
Configuration	Single Shell, two pass, twin bundle
Condenser Overall Length	40 feet
Condenser Overall Width	12 feet
Condenser Overall Height	10 feet
Shell Material	Carbon Steel (ASTM A283 Gr. C)
Waterbox Material	Carbon Steel (ASTM A283 Gr. C)
Number of Tubes	2778
Active Tube Length	33 feet 4 inches
Tube Material	Titanium, ASTM B338 Gr. 2
Tube Diameter	1 in. OD
Tube Gage	22 BWG
Inlet/Outlet End Tubesheet	1 in. thick 90:10 Copper Nickel (ASTM B171 Alloy 705) on water side pinned to a 1 in. thick carbon steel (ASTM A283 Gr. C)
Reversing End Tubesheet	1-3/4 in. thick 90:10 Copper Nickel (ASTM B171 Alloy 706) with deep 1/8 in. grooves to form an interconnected annular space at the center line.
Tube Support Spacing	33 inch spacing
Partial Tube Support	Only 15% of the tubes have partial tube supports. They are located midway between the full supports.
Tube Support Material	Carbon Steel (ASTM A283 Gr. C)

FIGURE 1

PROTOTYPE CONDENSER

FIGURE 2
SECTION THROUGH TUBE HOLE OF INTEGRALLY GROOVED TUBESHEET

FIGURE 3
SECTION THROUGH TUBE HOLE OF DOUBLE TUBESHEET

FIGURE 4
VIEW OF INTEGRALLY GROOVED TUBESHEET
SHOWING INTERCONNECTING GROOVES

3 TUBE SHEET STRUCTURAL TEST

3.1 Objective

The objective of this test is to develop an analytical method based on the finite element method of structural analysis for predicting tubesheet stresses. The analytical results are verified by comparison with the test results.

3.2 Background

The tubesheet maximum allowable stresses are generally specified to be in accordance with the Heat Exchanger Institute Construction Standard [2] and the ASME Codes. These codes, however, do not set the procedure for determining compliance. After the allowable material stresses for the tubesheet have been determined from these codes, the engineer uses several remaining parameters to design the stresses to be less than the allowable stresses. These remaining parameters include: tubesheet thickness, tubesheet edge distance, and tube properties (OD, length, wall thickness and material properties).

Tubesheet edge distance is defined as the distance from the water chamber flange to the outermost tube of the tube array. In general, the edge distance will vary in length at different locations around the outside tube array and, of course, of various tube arrays.

Tube properties influence the tubesheet stresses by a complex interaction of the perforated tubesheet on an elastic foundation (the tubes). Estimates of this complex interaction can be made based on approximate beam-strip models [3]. These models consist of approximating the tubesheet to be a series of beams of narrow width on an elastic foundation. It is the responsibility of the designer to determine the location of most severely loaded beam strip. Further assumptions are required by the designer to estimate the edge restraint of the beam strip against rotation provided by the water chamber flange and shell structures because variations in edge fixity affect the tubesheet stresses in bending.

These techniques, though satisfactorily reliable in the past, may not be adequate in light of today's more demanding requirements. Improved analytical techniques, therefore, need to be developed and evaluated for potential application.

Finite element analysis has been applied to many complex systems. Condenser tubesheet stresses are influenced by a complex interaction of the tubesheet, tubes, waterchamber, flanges and diaphragm and the finite element method lends itself to this type of analysis.

3.3 Test Plan

The test plan was to perform a strain gage test on the tubesheets of the condenser and compare the results of those based on the finite

element method. The loading of the tubesheets was accomplished by hydraulically pressurizing the circulating water side of the condenser. For each of two tubesheets, waterproofed strain gages were attached to both sides.

3.4 Strain Gage Locations

The strain gage locations on the tubesheet are shown in Figures 5 and 6. Figure 5 is for the integrally grooved tubesheet and Figure 6 is for the double tubesheet with a 1/8 inch gap.

It should be added that the testing did not include an effort to actually locate the point(s) of maximum stress in the tubesheet. However, gages are located on the outer edges of the tubesheet where the bending stress is generally the highest. Gages at these locations have a second gage located 1 inch away acting as a "backup" gage, and both are located in the direction of principle stress.

Gages were located in a series along the upper right edge for the purpose of determining the stress variation with respect to edge distance. Gages were also located along the bottom right edge for the purpose of determining the stresses due to the effect of the hydrostatic head of water in the water chamber.

Several gages were located on the reverse side of each tubesheet for the purpose of checking the membrane stress. Due to the symmetry of the tubesheet geometry and loading, only one half of the tube sheet need, in theory, be analyzed. As a check on this, however, some gages were located on the left tube bundle of the integral tubesheet at locations where peak stresses were expected (near tubes located closest to the flange).

Strain gage rosettes (gages capable of measuring triaxial states of stress) were located in two corners where the stresses were not expected to be a maximum; confirmation of this expectation was obtained from two gages.

3.5 Test Procedure Summary

The test procedure can be summarized in two simple steps (for a detailed description, see Appendix II): first, the water chamber and tubes were filled with water; second, the water chamber pressure was gradually increased while the strain gauge outputs were recorded. The maximum test pressure did not exceed 60 psig.

3.6 Analytical Modeling

The integrally grooved tubesheet was structurally modeled by the

1 - NUMBERS IN CIRCLES ARE GAGES LOCATED ON STEAM SIDE OF TUBESHEET.
UNCIRCLED NUMBERS ARE ON CIRCULATING WATER SIDE

2 - GAGES P60 TO P67 AND P60 - P61 ARE USED FOR THERMAL ANALYSIS ONLY

FIGURE 5
STRAIN GAGE LOCATIONS INTEGRALLY GROOVED TUBESHEET

NOTES:

- 1 - NUMBERS IN CIRCLES ARE GAGES
LOCATED ON STEAM SIDE OF TUBESHEET
UNCIRCLED NUMBERS ARE ON CIRCULATING
WATER SIDE.
- 2 - GAGES IP9 TO IP28 ARE USED FOR THERMAL ANALYSIS ONLY

FIGURE 6
STRAIN GAGE LOCATIONS DOUBLE TUBESHEET

finite element method of structural analysis. See Appendix III for background information on the finite element computer program selected. The model consists of a tubesheet (partially perforated plate), tubes, water chamber flanges and a diaphragm type expansion provision. A cross-section view of these components is shown in Figure 7 which also shows the location of the strain gages with respect to the water chamber flanges. The diaphragm is welded to the shell wall on one end and on the other end is welded to one of the flange sets (also called condenser end ring). The purpose of the diaphragm is to accommodate the thermal expansion between the shell and tubes during normal operation. The tubesheet is bolted between the end ring and water chamber flange as shown. The tube pattern of the tubesheet is shown in Figure 6.

The modulus of elasticity of the 90-10 Cu-Ni unperforated tubesheet material was 18×10^6 psi and the corresponding Poisson's ratio was 0.3. The equivalent modulus of elasticity for the perforated area of the tubesheet was determined from testing to be 2.7×10^6 psi (the corresponding equivalent Poisson's ratio was 0.6 [4]). The tests consisted of loading perforated tubesheet specimens in bending and recording the deflections. The test specimens were loaded in a Baldwin Testing Machine equipped with a stress-strain recorder. The test specimen was 1 in. by 1 ft. by 44.5 inches. Multiple tests were used to establish the load-deflection curve. The testing began with a solid plate. After confirming the test procedure with this undrilled plate of known properties, the plate was drilled with the hole pattern for the test condenser and tested again to determine the weakening effect of the holes. The same test specimen was then integrally grooved and retested.

The tubesheet dimensions are 56 by 120 inches. The modulus of elasticity for the titanium tubes is 15.5×10^6 psi. The flange and diaphragm have a modulus of elasticity of 30×10^6 psi and a Poisson's ratio of 0.27.

Due to symmetry, only one-half of the condenser was analyzed. The total number of elements in the finite element model was 1,070 and the number of nodes was 780. The element size was 3 inches square with the exception of the flanges where the size was 3 by 4 inches. The element size was checked by changing the element size to 4 inches square and noting that the stresses changed by less than $\pm 2\%$. Figure 8 shows the element arrangement.

The flexible expansion diaphragm was considered fixed at the shell wall where the two pieces are welded together. The water chamber was designed to be very stiff at the flanges and to force the flanges to translate and rotate as a rigid body. This structurally uncouples the water chamber (with the exception of the dead weight of the water which was included in the analysis) from the tubesheet. The net result is that the water chamber does not need to be included in the element list, thus reducing computation time.

FIGURE 7
COMPARISON OF CALCULATED VS. MEASURED TUBESHEET BENDING STRESSES
AT A LOCATION 1.5IN FROM INSIDE OF WATERBOX SHELL

FIGURE 8
FINITE ELEMENT GRID LAYOUT

The loads on the structure consist of the pressure in the water chamber (60 psig) and the dead weight of the water contained in the water chamber. The net effect of the dead load is to produce an overturning moment on the tubesheet.

3.7 Comparison of Finite Element Analysis and Test Results

The stresses from the finite element analysis are compared with the strain gages in Figure 7. The stresses are plotted as a function of distance along the top of the tubesheet at a distance of 1.5 inches from the inside edge of the flange (see sketch in Figure 7). The stresses are plotted for a 60 psig pressure. Examining the stresses at other pressures confirmed that the condenser structure is linearly elastic (doubling the pressure doubles the stresses).

Case 1 in Figure 7 is a finite element analysis based on the tubes sliding through the tube supports. This means that the elastic foundation of the tubesheet is based on a tube length measured from the tubesheet face to the transverse centerline of the condenser.

The calculated stresses on this basis are high when compared to the test results. Several analytical methods were attempted to reconcile the large differences and finally it was hypothesized that, for geometric and physical considerations, the elastic foundation constant determined by usual methods may not be valid for this structure. Note that the tubes, which form the elastic foundation, are flexible. The tubes are in compression in areas of the tubesheet and they can deflect laterally and change to the point of contacting the support plates. When this happens, the support plates take on some degree of fixity and therefore change the elastic foundation. This binding at the first and other supports suggest that a modified elastic foundation constant could help reconcile the calculation/test differences.

Treating the tubes as locked at the first support resulted in an analysis that gave good agreement with the test results. As a result of this analysis, a modified foundation constant (K^*) has been defined which can be used in analysis. This constant, as modified, bridges the physical phenomena in a manner similar to the familiar effective elastic modulus which permits analysis of perforated plates [5 and 6]. K^* is derived mathematically by using the length of tubing in the elastic foundation as the length from the tubesheet to the first support plate.

Eight strain gages were located on the double tubesheet as shown in Figure 6; the stresses from these gages are shown in Table 2. A bimetallic stress effect is present and can be seen by noting that the stresses in the steel plate are higher than those in the 90-10 copper-nickel plate. The copper-nickel and steel plates are both 1 inch in thickness but the neutral axis is shifted towards the steel plate (higher modulus of elasticity). The flexure stress as computed by $S = Mc/I$ (S is stress, c is distance from neutral axis, I is moment of inertia) would be the stress in an equivalent homogeneous beam. This stress can be converted to the stress in the composite beam by multiplying by the ratio of the two moduli of elasticity.

TABLE 2

COMPARISON OF INTEGRAL & DOUBLE TUBESHEET STRESSES

Location*		Stress (Minus indicates compression)	
		Integral	Double
P1	Water Side	4671	5328
P2		4149	5697
P12		4140	4842
P13		4239	----
P39	Shell Side	-3639	-7215
P40		-4392	-7620
P41		-4887	-8730
P42		-5247	-8835

90-10
Cu-Ni

90-10 Cu-Ni

Carbon Steel

*Location is referenced to integral tubesheet

The finite element analysis can model this by considering a model consisting of two layers of elements corresponding to 90-10 cooper-nickel and steel or by considering a single layer of elements with equivalent stiffness properties.

3.8 Conclusions and Recommendations

The experimental and analytical results were brought into agreement by modifying the elastic foundation in the model. Within the scope of this study there are two choices of how to handle the effect of the elastic foundation constant: one, to include the "effective" elastic foundation constant; and two, disregard the "effective" constant and accept a more conservative analytical solution. These two choices would also apply to conventional practice and are not limited to the finite element technique.

Choice one is not recommended for the following reasons:

- The degree of binding of locking at the first tube support is highly unpredictable due to effects of tube and tube hole tolerances.
- The effect of the tube itself, in the rigidity of the system, cannot be accurately determined since it is affected by the tube straightness.
- The actual alignment of the tube plates and tube supports has a large influence on the system spring constant.
- In many instances, the tube support spacing is variable over the length of the condenser and effects of tube binding at multiple locations cannot be treated with current technology.

For the above reasons, it is recommended that the second choice be used; the advantages are:

- Conservatism in the design of the tubesheet.
- Simplicity of analysis by using the more generally recognized elastic foundation theories.

The anticipated cost penalty for adopting this approach is expected to be near 10% of the condenser cost.

4 TUBE-TO-TUBESHEET DESIGN TEST

4.1 Objective

The objective of this test is to verify an analytical method, based on the finite element method of structural analysis, for determining tube compressive and tensile loads. The analytical results are verified by comparison with the test results.

4.2 Background

The condenser design engineer determines the loads in the tubes and compares them to the maximum allowable load which, when applied to a tube, will force the tube out of the tubesheet. The maximum load generally occurs in the outside tubes of the tube bundle. The ends of the tubes are expanded in the tubesheet holes by use of a roller/expander. The amount of expansion is determined by the reduction in tube wall thickness after the tube has been rolled in the tubesheet. Design parameters that determine the maximum allowable tube load include:

- Combination of tubesheet material and tube material
- Percent of tube wall thickness reduction
- Internal configuration of the tubesheet hole

The effect of each parameter is generally determined by testing. The parameters that determine the expected tube tensile load include:

- Tubesheet thickness
- Tubesheet material
- Tube properties (OD, length, wall thickness and material properties)
- Tubesheet edge distance
- Tubesheet deflection

Tubesheet edge distance is defined as the distance from the water chamber flange to the outermost tube of the tube array. In general, the edge distance will vary in length at different locations around the outside of the tube array.

Tube and tubesheet properties (size, thickness, material, etc.) influence the tube load by a complex interaction of the perforated tubesheet with the tubes and water chamber. Estimates of this complex interaction can be made as described in section 3.2

The general tube load distribution pattern of the tube bundle has been determined [3]. The highest stressed tubes in the

tubesheet are located in the outermost tube locations next to the flange. Many of the design parameters used to determine tube loads are also design parameters for determining the tubesheet stresses as described in section 3.3.

In summary, the finite element method offers the potential for determining tube loads in the condenser. This method relieves the design engineer of the responsibility of locating the most severely loaded beam strip when using the beam strip analysis method.

4.3 Test Plan

The test plan was to perform a strain gage test on the tubes of the test condenser and compare the results to those found with the finite element method. The loading of the tubes was accomplished by hydraulically pressurizing the circulating water side of the condenser and waterproofed strain gages were placed on the tubes.

4.4 Strain Gage Locations

The strain gage locations on the tubes are shown in Figure 9. The gages were located near the center of the tube length for the purpose of eliminating the effect of tube bending that may be present near the tubesheet. A series of gages is located along the top right outer tubes for the purpose of determining tube load variation with respect to edge distance. Gages were also located along the bottom right outside tubes for the purpose of determining the tube loads due to the effect of the dead weight of water in the water chamber. A series of gages were located along the vertical centerline of the right tube bundle for the purpose of determining the variation of tube load from the outer tube to the center. For this variation, the tubes at the outer edges are expected to be in tension [3], and the center tubes are expected to be in compression. As a check on symmetry of the tubesheet, several gages were located on the left side tube bundle.

4.5 Test Procedure Summary

See Section 3.5 on page 10.

4.6 Analytical Modeling

The finite element model used was exactly the same as that used for the tubesheet stress analysis (see section 3.6). The boundary conditions and loads are also identical.

The tube loads were calculated from the node deflections of the finite element analysis computer output. The node locations did not necessarily correspond to locations of tubes; therefore, interpolation was necessary. The tubesheet deflections at tube loca-

FIGURE 9
TUBE STRAIN GAGE LOCATIONS

tions were determined by linear interpolation of the nearest four nodes enclosing the tube location. The tubesheet deflection at the location of a tube is also shared by the tube itself and therefore the tube elongation is known. The tube load is calculated as follows:

$$P = \frac{\Delta AE}{l}$$

where P = tube load (lbs)
 Δ = Axial deflection of tube
 A = Cross-section area of tube
 E = Modulus of elasticity for tube
 l = effective length

4.7 Comparison of Analytical Analysis and Test Results

Tube loads from the strain gage test and analytical analysis are given in Table 3, where it can be seen that there is considerable scatter in the test data. This is expected because some of the tubes slide in the tube supports and others bind at the first support plate. The largest variations occur at low loads. For example, along the vertical centerline of the right tube bundle at strain gage 8, the load was 36 lb and the analytically determined load was 3 pounds. Using the test result as a calculation basis, the percent difference is 92%. At the higher tube loads, the percent differences are lower. At the top of the same vertical centerline, the strain gage load is 333 pounds and the analytically determined load is 203 pounds and the percent difference is 39%. The maximum tube load is the most important load and was 344 lb at a percent difference of 41%.

The discussion of this effect is in the Tubesheet Structural Test" section. The best overall correlation between the test results and analytical results was obtained by basing the effective foundation modulus for the upper half of the tubesheet on an effective tube length of one support spacing and the effective foundation of the lower half on an effective tube length of one-half the actual tube length. The values in Table 3 are for this situation. This is in agreement with the physical situation where it is noted that partial tube supports are installed in the upper portion of the inlet side, thus making the tube clearances tighter there and changing the elastic foundation. For additional discussion see section 3.7.

4.8 Conclusions and Recommendations

The two choices of how to handle the effect of the elastic foundation are discussed in section 3.8.

TABLE 3
TUBE LOADING RESULTS

A	B	C	A	B	C
1	213	205	32	N/A	---
2	174	156	33	192	213
3	333	203	34	167	188
4	N/A	---	35	189	202
5	N/A	---	36	215	210
6	161	108	37	251	198
7	87	41	38	257	206
8	36	3	39	239	181
9	- 22	- 13	40	254	191
10	- 63	- 11	41	234	189
11	- 95	- 25	42	N/A	---
12	-114	- 63	43	187	169
13	-138	- 97	44	201	170
14	N/A	---	45	13	151
15	N/A	---	46	246	189
16	-166	-148	47	-109	-111
17	N/A	---	48	-105	-123
18	215	206	49	-103	-116
19	223	220	50	-113	-128
20	183	174	51	212	114
21	278	225	52	N/A	---
22	193	182	53	N/A	---
23	194	194	54	317	-116
24	184	150	55	317	170
25	222	195	56		
26	218	150	57	218	169
27	170	117	58	183	213
28	142	79	59	N/A	---
29	N/A	---	60	N/A	---
30	192	134	61	344	203
31	N/A	---	62	223	156

A - Tube Strain Gage Number

B - Measured Tube Load (Lbs)

C - Calculated Tube Load (Lbs)

N/A - Not available due to inconsistencies in data (See Appendix IV)
or gage malfunction.

NOTE: Negative numbers indicate a compressive load; positive numbers
are tensile loads.

5 WATER CHAMBER STRUCTURAL TEST

5.1 Objective

The purpose of this test was to develop an analytical model based on the finite element technique which will accurately predict condenser water chamber loads, stress levels and displacements.

5.2 Background

Conventional technology for structural analysis of shell and shell-like structures found in shell and tube heat exchanges (of which condenser water box structure is an example) basically rely on the differential equation solution approach pioneered by Timoshenko and developed by countless technical papers investigating particular loading conditions, geometries, temperature ranges, etc. Analysis of an individual structure as an engineering problem involves making certain simplifying assumptions (perhaps erroneously) to force the real hardware to resemble some classical textbook case [7,8]. Often involved is a time consuming literature search through technical papers for a similar solution, or development to some closed form solution which may or may not adequately represent reality. All too often, time constraints for design, analysis and manufacture lead to designs with marginal safety factors or overdesigned, costly hardware.

With the advent of higher pressure requirements for condenser water chambers with complex geometries, alternate means of design analysis may be necessary to produce a product with a highly reliable structural integrity.

As an example of the complex nature of waterbox design, consider the point at which the end closure - shell seam intersects the nozzle attachment weld. This is often called a triple-point and is critical from a manufacturing standpoint because weld problems and hydrotest leakage are most likely to occur there. This is so because of the positioning and "fitting up" problems that are encountered during the welding process. This situation is aggravated by "weld blow" inherent in the arc welding process and the fact that triple-points usually present difficult accessibility for the welder. Furthermore, the triple-point, for three surfaces under internal pressurization is subjected to the combined bending moment of all three surfaces, as well as shear and tension forces arising from local displacements of all three surfaces.

Finite element analysis of complex structures "breaks down" the structure into a series of simple structural shapes (in this case, flat triangular plates) whose structural properties are well defined. Then, by using the computer as a tool to perform vast numbers of simple calculations, the simple structures are analyzed in a set

of simultaneous equations to approximate the full-sized complex structure. By using finite element analysis as a design tool certain "weak" areas of a structure may be pinpointed which may have been otherwise overlooked. Conversely, this method can show by stress level prediction where structures may have been overdesigned in some areas and allow cost effective of design changes to be made. In general, the finite element method of representing structures extends engineering capabilities beyond conventional analysis.

5.3 Test Plan

The test plan was to obtain actual stress data from strain gages on a prototype condenser. The stresses would result from hydraulically pressurizing the circulating water side of the prototype condenser. The results of the test would then be compared with an analytical technique used to predict the stress levels.

5.4 Strain Gage Placement

Strain gages were placed on the water chambers in the positions shown in Figures 10 and 11. These figures show that the condenser water chambers were instrumented with 90 strain gages consisting of biaxial and triaxial types for a total of 244 strain elements. Triaxial strain gages were placed along the weld seams on both inlet and reverse chambers, approximately one inch from the weld. Biaxial strain gages were placed on the chambers away from weld seams, oriented so that the two strain elements on each gage were parallel and perpendicular to the axis of the cylindrical surface.

On the inlet chamber, 18 triaxial gages were placed along the shell-nozzle interface, as shown in Figure 10, and two pairs of gages (115 and 116; 215 and 216) were placed adjacent to the "triple-point". There are also two pairs of gages (117 and 118; 217 and 218) located on the nozzle interface closest to the waterbox flange. All of these areas are difficult to analyze by the more classical techniques.

As shown in Figure 11, the reverse chamber was instrumented with three element gages at 14 locations along the shell-end closure juncture with matching gages on the inside and outside of the shell. An additional 26 biaxial gages were placed on the cylindrical surfaces of the shell lateral surface and end closure shells, sufficiently removed from surface boundaries to be unaffected by local stress concentrations. Again, matching gages were placed on both inside and outside shell surfaces.

5.5 Test Procedure Summary

See Section 3.5 on Page 10.

NOTE:

- (1) 200 SERIES GAGES LOCATED ON INSIDE OF WATERBOX DIRECTLY OPPOSITE CORRESPONDING GAGES SHOWN ABOVE
- (2) WATER CHAMBER IS SYMMETRICAL ABOUT VERTICLE CENTERLINE
- (3) STRAIN GAGE ELEMENT "B", SEE DETAIL IS PERPPENDICULAR TO WELD SEAM AT ALL GAGE LOCATIONS.

FIGURE 10
TRI - AXIAL STRAIN GAGE LOCATIONS INLET CHAMBER

NOTE:

400 SERIES GAGES LOCATED ON INSIDE OF WATERBOX
DIRECTLY OPPOSITE CORRESPONDING GAGES SHOWN ABOVE.

TYP - LOCATION TRI - AXIAL GAGE 301 TO 314

VIEW A - A

TYP - LOCATION BI - AXIAL GAGE 321 TO 327

VIEW B - B

TYP - LOCATION BI - AXIAL GAGE 315 TO 320

VIEW C - C

FIGURE 11
STRAIN GAGE LOCATIONS REVERSE CHAMBER

5.6 Analytical Modeling

The finite element stress analysis using the ANSYS program, as described in Appendix III, was performed for an applied internal pressure of 60 psig with no correction for variation in pressure due to water level.

The inlet chamber consists of the water chamber shell, "orange peel" end closures and twin nozzles for cooling water inlet and discharge. In the computer model, advantage was taken of symmetry about the water chamber shell lateral centerline so that only one-half of the entire structure was modeled.

The inlet chamber model, consisting of water chamber shell, one end closure, and one nozzle connecting the condenser with the inlet piping, is shown in Figure 12. The nozzle section terminates at the nozzle flange. Advantage is taken of the symmetry about an internal baffle plate (on the plane of symmetry) to maintain the number of elements used at a reasonable level. The inlet chamber model consists of 720 flat triangular plate elements. The base of the finite element model is the flange connection to the main condenser shell. Both flanges are relatively large compared to the shell and nozzle thickness of 0.5 inches. Therefore, the nozzle flanges and the large chamber flanges are considered rigid, that is, they resist both translation and rotation in the six degrees of freedom for each node on these boundaries.

Like the inlet water chamber, the reverse chamber is constructed of 352 flat triangular plates, representing one-half of the entire structure. The triangular plate element is a constant strain element with both bending and membrane capabilities. Again, the boundary representing the water chamber flange is considered rigid; however, the boundary where the cylindrical shell is "cut" (by the plane of symmetry, shown in Figure 13) is only partially restricted to represent structural continuity of the remainder of the shell (i.e., the shell is allowed to "grow" under pressure).

For both chambers, the grid sizes are chosen to generally correspond to strain gage spacing of about 8 inches. Departures from this criterion were sometimes required by the rules of good simulation practice for structural modeling by finite elements. For more accurate stress results, the aspect ratio (length to width ratio of the element) was chosen as close to unity as possible.

FIGURE 12
COMPUTER MODEL OF INLET CHAMBER & NOZZLE
(HALF MODEL SHOWN)

FIGURE 13
COMPUTER MODEL OF REVERSE CHAMBER WATERBOX
(HALF MODEL SHOWN)

Another factor in grid sizing was to generate elements near weld seams such that the element centroids were close to the "character length" of the surface. Near the boundaries, within the character length λ , where

$$\lambda = 0.78 \sqrt{Rt} \quad (R = \text{mean shell radius, } t = \text{thickness}),$$

there is an influence from local bending moments and shear stresses. Thus, choosing the proper element sizes is critical because stress concentration factors should be included in the analytical predictions.

In general, a smaller grid size yields better resolution of the stress field. Element sizing was chosen as a balance between high resolution and limiting the number of elements in the model to keep the size, complexity and computer running time of the analysis within reasonable limits.

In addition to the usual printout of stresses in the local coordinate system for each element, the model analysis also included node deflections in the global x-y-z coordinates and reaction forces and moments for boundary conditions. These are useful as they predict forces and moments on the flanges; however, because the coordinate system used in the model is for each local element, correlation of stress predictions to a global x-y-z coordinate system requires a series of Mohr's circle rotations.

Because the gage locations did not usually coincide with the element centroids, it was usually necessary to average stresses calculated at the centroids of two or three triangular elements to more accurately approximate stresses close to the strain gage locations. Since stresses are obtained for each element in terms of local element coordinate systems, it was necessary in some cases to perform Mohr's circle rotation of the elements to obtain stresses oriented in the direction of the strain gage elements.

In the case of the triaxial strain gages, all stress results were rotated to obtain equivalent stresses along the axes of the ± 45 degree strain elements. For correlation with strain gage locations which required both averaging and Mohr's circle axes rotations, the Mohr's circle axis rotations were performed first, then the results were averaged in an attempt to minimize errors. In cases where the stresses in one direction were considerably higher than those in the perpendicular direction, or where tension and compression stresses were perpendicular, stress resultants in the direction of strain gage axes are influenced by the rotation angle chosen leading to possible errors in orientation. It is felt, however, that such occurrences are infrequent and deviations from stress patterns would be evident when a "string" of data was calculated.

5.7 Comparison of Analytical and Test Results

A validation of predicted stresses, away from surface boundaries and consequent stress concentrations, is desirable. Such regions of the water box surface, where tangential (hoop) stresses were expected to nearly coincide with the classical textbook [8] PR/t value were instrumented with strain gages. The gages were numbered 322 through 327 on the exterior, and 422 through 427 on the interior surface of the reverse chamber (Figure 11).

If these readings (those for gages 322 - 327 and 422 - 427) yield reasonable agreement, that is $\pm 10\%$ error, with predicted stresses from analysis, then a certain credence may be lent to stress levels predicted at shell - nozzle intersections and in the vicinity of flanges, weld seams, etc., where stress concentration factors are essentially unknown.

The inner and outer stresses may be averaged because they vary linearly from the inner to the outer surface. The average of the inside and outside strain readings should correspond to that associated with pure tension (membrane) stresses. Nominally, these pure tensile stresses calculate, from the PR/t equation, to be 3330 psi for $R = 27.75$ in., $t = 0.5$ in. and $P = 60$ psi.

Since inner and outer shell surface stresses, as well as membrane stresses, are included in the computer printout these surface stresses may be compared directly with strain readings from gages located on these surfaces. The average of the strain readings may be compared directly with predicted membrane stresses.

The strain gages measure strains arising from hoop stresses which are constant throughout the thickness of the shell (pure tensile loading) combined with the strains generated from bending moments as shown in Figure 14.

In Figures 15 and 16 predicted inner, outer and membrane stresses are compared with test data where measured strains were converted to stresses by the relation $\sigma = E\epsilon$, where ϵ is measured strain, E is Young's Modulus (30×10^6 psi for carbon steel) and σ is stress (psi).

In Figure 15, the measured and predicted stresses are compared for both interior and exterior strain gage readings. The difference between data for the two surfaces shows that bending moments are present. In addition, the average of apparent stresses (from strain data) and predicted hoop (membrane stresses) are compared with nominal PR/t stress levels. The analytical values of membrane stresses agree almost identically with nominal stress levels, but average measured data appear to be approximately 20% below predicted

$$\sigma_x = \frac{PR}{t}$$

$$\sigma_x = \frac{MZ}{EI}$$

$$\sigma_{\max.} = \frac{PR}{t} + \frac{Mt}{ZEI}$$

$$\sigma_{\min.} = \frac{PR}{t} - \frac{Mt}{ZEI}$$

(- t/z t/z)

NOTE: E is Young's modulus of elasticity and I is the moment of inertia.

FIGURE 14

ADDITION OF STRESSES FROM PURE TENSILE AND BENDING LOADS

Figure 15
COMPARISON OF MEASURED AND PREDICTED HOOP STRESSES ON REVERSING CHAMBER SHELL

Figure 16
COMPARISON OF MEASURED AND PREDICTED HOOP STRESS ON REVERSING CHAMBER END CLOSURE

values. That is because merely multiplying strain gage readings by Young's Modulus does not account for the influence of biaxial stresses on strain gage readings.

To obtain true values for stress derived from the strain data, corrections must be made by considering the biaxial stress conditions which exist on the shell. The following relationships from reference 9 were used.

$$\sigma_x = G\epsilon_x + L (\epsilon_x + \epsilon_y)$$

$$\sigma_y = G\epsilon_y + L (\epsilon_x + \epsilon_y)$$

where

$$G = E/2 (1 + \nu)$$

$$\nu = \text{Poisson's ratio}$$

$$\epsilon = \text{strain reading in } 10^{-6} \text{ in./in.}$$

$$L = \text{Lame's constant} = E\nu/(1-2\nu)(1+\nu)$$

$$E = \text{Young's modulus}$$

In order to obtain true membrane values of σ_x and σ_y from strains, both inside and outside strains for both perpendicular (x and y) directions are required. For the set of strain gages under analysis, only two locations had been functioning satisfactorily to perform the raw data correction: location 323 (and 423) and 324 (and 424). In Table 4, A represents the element reading x direction and B represents the y direction. The nominal stress value (σ_x) for the region is $PR/t = 3330$ psi.

TABLE 4
COMPARISON OF STRAIN GAGE AND PREDICTED STRESSES

Strain Gage	Reading (ϵ) 10 ⁻⁶ in./in.	Corrected Stress	Predicted Stress From Finite Element
323A	38.5	$\sigma_{yTest} = 2094$ PSI (outside)	(PR/t = 3330 PSI)
323B	49.5		
423A	64.0	$\sigma_{yTest} = 4324$ PSI (inside)	
423B	111.5		
		Avg. $\sigma_{yTest} = 3209$ PSI (96.4% PR/t)	$\sigma_{yMemb.} = 3368$ PSI (101.1% PR/t)
324A	55.0	$\sigma_{yTest} = 2452$ PSI (outside)	
324B	52.0		
424A	46.5	$\sigma_{yTest} = 4367$ PSI (inside)	
424B	123.5		
		Avg. $\sigma_{yTest} = 3410$ PSI (102.4% PR/t)	$\sigma_{yMemb.} = 3357$ PSI (100.8% PR/t)

From this comparison both test and analytic data agree with each other within 5% error. This excellent agreement verifies both strain gage sensitivity and analytic method algorithms. This agreement also indicates that biaxial stress field effects are influencing individual strain element readings.

Since the locations of these particular strain gages (the 300 series on the outside of the shell and 400 series at matching locations inside the shell) are sufficiently removed from any shell junctions, the nominal PR/t value of hoop stress is expected to be in good agreement with stresses both derived from strain data and predicted by finite element analysis. The data shows that all results were within 4% of the PR/t values.

Figure 16 depicts a comparison of hoop stresses on the "orange peel" end closure cylindrical surface for the reverse water chamber. As with strain gage data on the lateral surface, strain data which was not self-consistent reduced the number of data points for averages of inner and outer strains.

On this surface the strain gages are 315 through 320 (and 415 through 420) and are located from the proximity of the flange to the apex of the end closure. It is interesting to note that in this short cylindrical surface the divergence of inside and outside predicted stresses is clearly present near the flange due to the local bending moment and diminishes away from the flange. General agreement between both interior and exterior measured strains and analytical stresses is shown. In addition, the stress pattern for both individual points and averaged data (both actual and analytic) shows reasonable agreement throughout.

Again the averaged strain data, uncorrected for biaxial stress effects, appears slightly lower than predicted stresses but the strain data is too sparse for comparison. Agreement of analytical membrane stresses with the nominal PR/t value is evident with the exception of the area near the apex (especially strain gages 320, 420). For data points 320 and 420, both strain gage readings and predicted stresses show a marked drop due to the influence of near-field boundary effects. This effect of both measured and predicted stresses to drop off in the vicinity of the end closure apex (gages 320 and 420) could only have been predicted by finite element analysis.

Figure 17 compares analytical predicted stresses with test data stresses measured perpendicular to the 45° weld seam for the shell exterior and interior. Due to flange corner bending moments, the outer surface is in compression (300 series) while the inner surface is in tension. In this case, the measured stresses are consistently greater than predicted stresses because the strain gages are closer to the weld seam than the finite element centroids and the stress is rapidly increasing. It is felt that local stress concentrations in the area adjacent to the weld may be caused by local reinforcement of the weld over the shell thickness. The weld reinforcement was not considered in the mathematical model.

Figure 17
COMPARISON OF MEASURED AND PREDICTED STRAINS ALONG END CLOSURE - SHELL SEAL WELD
Stress (perpendicular to weld seam)

● Measured
○ Predicted

Figure 18
COMPARISON OF MEASURED AND PREDICTED STRESSES ALONG END CLOSURE - SHELL SEAL WELD

Of particular interest is the marked rise in test data, and to a lesser extent the rise in predicted stress, which occurs between locations 303/403 and locations 305/405. Reference to the predicted deflections shows this point to be an inflection point for the radial distortion. Thus, at an angle on the shell approximately 25° from the flange, stresses both measured and predicted are rapidly increased. Without this boundary effect one might expect a gradual rise in stress to a point midway between the flange and the end closure apex.

Figure 18 is presented to compare the above strain and predicted values with those measured on the water box shell side of the same weld. Again, the curve assumes the same general shape but without the marked rise in stress at 25° as on the end closure. In both Figures 17 and 18 the effect of the shell corner stiffness nearly matches the rigidity imposed by the condenser flange. The rigidity of the flange reduces local bending moments close to the flange.

On the inlet water chamber, triple point strain gages were inoperative so no comparison could be made with the predicted stress levels of 3000 psi and 3430 psi for gage locations 115 (216) and 116 (216) respectively. Although these stress levels were not significantly high for this particular waterbox geometry, analyses of other production water chambers show stresses in the area of the triple point can be 3 to 4 times the stress levels of the surrounding regions. In short, stress levels in this region are unpredictable and highly sensitive to the geometry of a water chamber so that finite element analysis is necessary for reasonably accurate stress predictions.

Stress levels for all other strain gages locations on the inlet water chamber (100 and 200 series) around the nozzle-shell intersection (Figure 10) were low and ranged from 0 to 1000 psi in compression. These data were average strain readings as well as membrane stresses perpendicular to the weld seam. Because of the low stress levels these data were not plotted.

One interesting set of data were for the gages close to the waterbox flange, locations 117 (217) and 118 (218) in Figure 10. While both average strain reading for 118 & 218 were low (750 psi in compression vs. 350 psi compression predicted), the strain data showed a considerable bending moment present while the analytical model did not.

The analysis was performed for only one internal applied pressure of 60 psi. Strain gage readings were taken every 10 psi as pressure was increased to assure linearity of strain readings. Additional readings were taken at increments of 20 psi as pressure was decreased from 60 psi, for checking repeatability of strain gage readings. Realistically, no strain gage can be expected to produce perfectly linear readings, so a reasonable "error bandwidth" was chosen for screening test data and rejecting erratic strain data. This criterion is presented in detail in Appendix IV.

The analytical predicted stresses for waterbox loading at reduced pressures, however, are linearly proportional to applied pressure loading (i.e., stresses from 30 psi loading = 50% of stresses from 60 psi loading). Presentation of stress analyses for reduced loading would be of little benefit to the reader.

5.8 Conclusions and Recommendations

The finite element analytical technique can predict stresses in complex structures. This is especially significant because it is precisely in those applications that the greatest concern exists for adequate strength.

It is therefore recommended that, since the finite element is now widely available (there is no need for each company to write their own computer program), serious thought be given to incorporating these techniques into the industry's design guidelines manuals. It is thought that this technique will be most useful in the analysis of all new designs.

6 VIBRATION TEST

6.1 Objective

The objectives of this test were to determine the fundamental bending frequencies and the damping of the tube-tube support system of the minimum leakage condenser. The experimentally determined frequency results were to be compared with theoretical frequency results. The experimentally determined damping results for these titanium tubes were to be compared with damping results previously obtained for tube-tube support systems which contained tubes made of a different material.

6.2 Background

One of the most important aspects of condenser design is the prevention of damaging tube vibration. The objective is to prevent destructive tube vibration and the associated tube leaks and, therefore, forced outages or power reductions. Although the majority of outages caused by vibration occurs during the first two years of operation of a new condenser, the outage costs, in terms of power not sold, can be substantial. Therefore, there is a need to verify the design procedure used to determine the tube span for operation free from vibration damage whenever a significant change occurs in the unit configuration. A significant change in condenser configuration occurred with the introduction of titanium tubes, and it has been shown in an Electric Power Research Institute report [1] that vibration is a significant cause of titanium tube failures.

All other factors being equal, tubing materials in general use in condenser applications prior to the appearance of titanium are less prone to vibration damage than are titanium tubes. This is because titanium has less internal damping (vibration damping that is dependent on the material itself).

Until this time, there have been two available tube span design rules [10 and 11]. However, each of these rules was formulated before titanium tubes came into wide use and are questionable from the internal friction standpoint. Therefore, calculations based on these rules could yield results which may be suspect.

There are actually three sources of tube damping in shell and tube heat exchangers:

- Tube internal friction
- Tube-to-baffle interaction
- Hydrodynamic inertia

The fluid damping term, hydrodynamic inertia, can be neglected for the condenser application because the damping provided by a low density gas such as steam at approximately one-inch mercury, absolute, which is typical for condensers, is negligible. The remaining two damping components are normally combined. The combined effect is taken as the logarithmic decrement (log dec) for use in the following design rule used by the Westinghouse Steam Turbine Division:

$$L = 21.8 \left(\frac{S_c E I \delta_o}{\rho V^2 D} \right)^{\frac{1}{4}} \quad (1)$$

Where:

L = Tube support spacing for the avoidance of damaging tube vibration, in.

S_c = Severity factor

E = Tube Young's Modulus, lb_f/in^2

I = Tube moment of inertia, in^4

δ_o = Logarithmic decrement (log dec) of vibrating tube, dimensionless

ρ = Steam density in condenser, lb_m/ft^3

V = Continuity velocity of steam at condenser inlet flange, ft/sec

D = Tube diameter, in.

The log dec used in the design relationship is 0.036. Reference 10 describes the source of this value. The number was obtained during shaker tests on 90-10 Cu-Ni tubes in a heat exchanger before the unit was shipped. It was expected that the log dec for a titanium tube condenser would have differed considerably from the log dec for a 90-10 Cu-Ni tube condenser. This prediction was based on the fact that the internal friction of titanium was low relative to the internal friction of 90-10 Cu-Ni [12]. Published results indicated that the internal friction for 90-10 Cu-Ni was an order-of-magnitude greater than the internal friction of titanium for the temperature of this application. It was assumed that the structural (tube-to-baffle) damping of the titanium tube unit would have been the same as that of the Cu-Ni tube unit. Therefore, the combined effect of the internal friction and structural damping should have resulted in a lower log dec for the titanium.

6.3 Test Description

The fundamental bending frequency and damping of a tube which was supported at 33 inch intervals was determined with the frequency response method and with the decay method. The test configuration is shown in Figure 19. The tube was tested both empty and water-filled. In addition, the fundamental bending frequency and damping for a tube supported at 16.5 inch intervals was determined by the frequency response method. This tube was tested both empty and water-filled. See Figure 20 for the test arrangement.

The frequency response method of tube bending frequency determination is based on the assumption that the tube-tube support system is a single degree of freedom, linear system. A shaker is used to effect the forced vibration. At resonance, that is when the shaker (excitation) frequency coincides with a tube-tube support system natural frequency, and with constant peak impressed force, the amplitude reaches a maximum. This effect can be observed from inspection of the solution of the spring-mass-damper system differential equation, found in any standard vibration reference work [13].

The decay method of fundamental bending frequency determination involves excitation of a vibratory system at what appears to be the lowest frequency followed by removal of the excitation. During the resulting free vibration (decay), the system frequency will remain constant if the forced vibration frequency coincides with the lowest bending frequency.

The use of data from the frequency response test for the determination of damping may be found in any standard vibration reference book [14] and involves:

- Determination of the response curve bandwidth at the half-power points.
- Determination of the frequency at which response peaks, at constant peak impressed force.
- Division of the bandwidth by the "peak response" frequency and multiplication of the result by π yields the damping in the form of logarithmic decrement.

The decay method of damping determination involves measurement of the peak amplitudes of several cycles of a vibrating system during free decay. The method may be found in any standard vibration reference book [15]. The particulars of the procedures used for this set of experiments are given in Appendix V.

The relationship between tube-tube support system damping and diametral clearance between tube OD and the tube support holes is currently a topic of intense development effort by heat exchanger specialists; therefore, this clearance was recorded at all of the structural supports for one tube which had 33 inch spans. No attempt was made to make this clearance an independent variable or to note the effect of it on system damping. The clearance was simply recorded. See Table 5 for results.

NOTE

TUBE:

1.00 IN. O.D.
22 BWG (0.028")
ASME SB 338 GR*2

FIGURE 19

TEST CONFIGURATION

NOTE

TUBE:

1.00 IN. O.D.

22 BWG (0.028")

ASME SB 338 GR*2

FIGURE 20

TEST ARRANGEMENT

TABLE 5

DIAMETRAL TUBE HOLE CLEARANCES FOR 33 INCH SPAN TUBES

The diametral tube-in-hole clearance was measured at all eleven supports for the 33 inch span tube. The diametral clearance was calculated as the square root of the sum of the squares of the horizontal and vertical clearances.

Tube Support No. (From Inlet End)	Diametral Clearance Mils
1	13.6
2	12.5
3	13.2
4	14.9
5	11.6
6	11.2
7	11.7
8	14.2
9	14.2
10	14.2
11	12.4

The diametral clearances of the 16.5 inch span tube in the respective supports were too small to measure with the equipment available.

It is unlikely that the two tubes which were tested were under zero axial stress, either compressive or tensile. The effect of axial stress on vibration frequency at significant stress levels is very predictable. However, the effect of such stresses on the frequency and damping in this unit were judged to be negligible.

6.4 Analysis of Test Results

The results of the bending frequency tests are shown in Table 6. The theoretical values are also shown. The theoretical values were determined through the use of the MacDuff and Felgar method [16]:

$$f_{\text{Fundamental}} = (0.04944) (C_n) \left(\frac{EI_g}{M_o L^4} \right)^{\frac{1}{2}}$$

Where:

$f_{\text{Fundamental}}$ = Frequency, Hz

C_n = 32.37 for 12 or more equal spans, dimensionless

L = Unsupported tube span, in.

E = Modulus of elasticity of the tube material, $\text{lb}_f/\text{in.}^2$

I = Tube cross-section moment of inertia, in.^4

g = Acceleration of gravity $386 \frac{\text{lb}_m - \text{in.}}{\text{lb}_f - \text{sec}^2}$

M_o = Tube and tubeside fluid mass per length, $\text{lb}_m/\text{in.}$

The test results exhibited the expected agreement with the theoretical results. The fundamental frequencies determined through testing differed from the theoretical frequencies by a maximum of only several percent.

The damping test data were reduced and analyzed to yield system damping in the form of logarithmic decrements. The resulting log decs are shown in Table 7.

Figure 21 shows a typical response curve obtained during the frequency response method of damping determination. Figure 22 is a typical decay trace from the chart recorder, used in the determination of damping by the decay method.

The average log dec for all of the tests was 0.0842. The lowest log dec, 0.058, differed by almost a factor of two from the highest log dec, 0.1025. The scatter in the data is significant. It is typical, however, of the results reported by investigators for this type of testing. The log decs measured in the test documented in reference 17 ranged from 0.014 to 0.565; in reference 18 the typical log dec observed was 0.04. The reasons why the data typically exhibit so

TABLE 6
TUBE FUNDAMENTAL BENDING FREQUENCIES

Tube Span In.	Tube Condition	Method	Test Fundamental Frequency, Hz	Theoretical Fundamental Frequency, Hz
33	Empty	Freq. Response	95.3	94.9
33	Empty	Decay	93.0	94.9
33	Water-Filled	Freq. Response	58.5	56.6
33	Water-Filled	Decay	58.0	56.6
16.5	Empty	Freq. Response	371.8	379.26
16.5	Water-Filled	Freq. Response	234.0	226.3

TABLE 7
TUBE-TUBE SUPPORT SYSTEM DAMPING

Tube Span, In.	Tube Condition	Method	Logarithmic Decrement (Dimensionless)
33	Empty	Freq. Response	0.1025
33	Water-Filled	Freq. Response	0.0843
16.5	Empty	Freq. Response	0.0769
16.5	Water-Filled	Freq. Response	0.0992
33	Empty	Decay	0.058

AMPLITUDE
(PEAK TO PEAK),
INCHES

FIGURE 21
FREQUENCY RESPONSE AT DRIVING POINT
Y - LAB CONDENSER TUBE
33.00 IN. SPANS - TUBE EMPTY

FIGURE 22
MINIMUM LEAKAGE CONDENSER
FORCED VIBRATION TRACE AND FREE VIBRATION (DECAY) TRACE

much scatter undoubtedly include the intermittency of the linear and nonlinear damping aspects of the interaction of the tube with the tube supports. The tube-tube support system approximates a linear system if the bending deflections are small and if the tube doesn't translate at right angles to the tube longitudinal axis. The only significant nonlinearity is the imperfection of the spring, the tube in this case. Small amplitude bending operation involves only a humming noise. Therefore, it is concluded that only linear and sliding friction effects should be present during small amplitude bending.

If the tube translates transverse to the tube longitudinal axis and relative to the support hole and slides on the tube hole surface, friction damping causes nonlinear response. For this case, if the excitation doubled, the response would not be expected to double. If the tube translates relative to the support hole and does not slide on the hole surface, the tube impacts the hole, exhibiting a shock type of response. Considerable audible noise emanates from the supports at which shocks occur during this type of operation. The most likely combination of effects during large amplitude bending involves all three effects, concurrently and/or intermittently (i.e., linear, sliding friction and shock).

It can be seen that several variables can affect the damping of any given tube-tube support system. These include:

- Tube static deformation, that is, bends which deflect the tube away from the undeformed tube longitudinal axis, acquired during tube manufacture or installation (Figure 23 below).

Figure 23
Tube Static Deformations

- The orientation of the excitation vector relative to the gravity vector. This consideration applies only to horizontal tubes. For example, sliding friction (Coulomb damping) should be reduced or eliminated if the exciting force vector and the gravity vector acting on the tube coincide (Figure 24 below).

Figure 24

Orientation of Excitation Vector to Gravity Vector

- Maximum tube amplitude at the midspan locations. Larger amplitudes reduce the sliding areas during part of the cycle because the contact area approaches zero and defaults to circular line contact (Figure 25). Obviously, wear of the tube or hole would prevent circular line contact and should increase damping during large amplitude bending operation.

6.5 Conclusions and Recommendations

The design method used to determine allowable tube spans for safe operation, from a vibration standpoint (equation 1 page 41), is based on the assumption that the tubes vibrated at the predicted fundamental frequency. The demonstrated agreement between the experimental and theoretical fundamental bending frequencies will ensure that this unit will not exhibit dangerous tube vibration if the design flows and pressures are maintained.

Because of the difficulty of obtaining a highly accurate value for damping of a tube held relatively loosely by numerous supports, conservatism should be used in selecting a design value for damping. A log dec of 0.036, which is a lower value than any determined in this test and which is currently used by Westinghouse for all the tube metals in condensers, is a suitably conservative value for titanium tubes as well.

Case a - Tube flexure sign changes at baffle

Note: No wear on tube or hole

Case b - Tube flexure sign remains constant at baffle

Note: No wear on tube or hole

Figure 25

Maximum Tube Amplitude at Midspan Locations

7 THERMAL TEST OF TUBESHEETS

7.1 Objectives

The objective of this test is to simulate thermal stresses in the integral and double tubesheets that might be encountered in the field from extreme and moderate temperature variations of the cooling water and steam.

7.2 Background

The integrally grooved double tubesheet consists of two tubesheets 1/8 of an inch apart that are attached systematically by ligaments as shown in Figure 4. The tubesheet is also of uniform thickness around the edges of the integral grooves. Part of the thermal test is to verify this design. The major design parameter for the determination of these stresses is the thermal gradient. Once the thermal gradients are known, these thermal stresses can be calculated.

7.3 Test Plan

The test plan is to develop a thermal gradient across the integral and double tubesheet and measure both temperature gradients and tubesheet strains caused by the thermal gradients. To accomplish this, the shell of the condenser was filled with water and heated to a predetermined temperature. Cold water was then circulated through the water chambers and tubes and a temperature gradient was developed in the tubesheet.

7.4 Thermocouple and Strain Gage Locations

Figures 5 and 6 show the locations of the strain gages on both tubesheets. The thermocouple locations are given in Figure 26.

7.5 Test Procedure Summary

The test procedure is summarized here and explained in more detail in Appendix VI:

- Fill condenser shell with ambient water
- Raise water temperature by sparging steam into the water
- Quickly circulate cold water in the water chambers and record thermocouple and strain gage data.

7.6 Analytical Approach to Thermal Stresses in the Tubesheet

A conservative estimate of the thermal stresses in the integral tubesheet near the condenser flange can be made assuming that the tubes cool very quickly when the cool water is circulated during the test. The tubes will decrease in length by $\alpha \times \Delta T$ where α is the coefficient of thermal expansion for the tubes, ℓ is half of the tube length, and ΔT is the temperature difference between the cooling

NOTES:

- 1 - THERMOCOUPLES 1 TO 8 LOCATED ON CIRCULATING WATER SIDE OF DOUBLE TUBESHEET.
- 2 - THERMOCOUPLES 21 TO 28 LOCATED ON STEAM SIDE OF DOUBLE TUBESHEET.
- 3 - THERMOCOUPLES 31 TO 38 LOCATED ON CIRCULATING WATER SIDE OF INTEGRALLY GROOVED TUBESHEET.

FIGURE 28
THERMOCOUPLE LOCATIONS ON TUBESHEETS

fluid temperature and initial temperature. Conservatively assuming that the entire change of the tube length is absorbed by the expansion joint diaphragm and that the bolting flanges do not rotate, the maximum moment that can be transferred into the tubesheet can be determined.

In practice, the flange would rotate slightly and relieves the maximum calculated moment. The maximum moment based on this conservation assumption is:

$$M_{\max} = \frac{6EIy}{2\ell}$$

Where:

E = Modulus of elasticity of diaphragm 30×10^6 psi

I = Moment of inertia 0.0044 in.⁴

ℓ = Diaphragm length 9.5 in.

y = Thermal expansion of tubes, $\alpha \ell_t \Delta T$

α = Coefficient of thermal expansion for tubes 9.5×10^{-6} in./in.^{°F}

ℓ_t = Half tube length 198 in.

ΔT = 160 - 82 = 78^{°F} (Test values)

$$M_{\max} = \frac{(6) (30 \times 10^6) (.0044)}{(9.5) (9.5)} (9.5 \times 10) (198) (78) = 1288$$

The maximum tubesheet stress is:

$$S = \frac{6M_{\max}}{t^2} = \frac{(6) (1288)}{(1.75)^2} = 2523 \text{ psi}$$

Where:

S = tubesheet stress

t = tubesheet thickness 1.75 in.

TABLE 8

SUMMARY OF THERMAL TEST RESULTS

Strain Gage Number	Maximum Strain ¹ During Cool Down	Strain Gage Number	Maximum Strain ¹ During Cool Down
IP1	74.5	P19	- 76.5
IP2	82	P20	- 59.5
IP3	- 28.5	P21	-109
IP4	32.5	P22	- 76.5
IP5	89.5	P23	-113.5
IP6	86.5	P24	127
IP7	108	P27	-142.5
IP8	110	P28	-151.5
IP9	29	P29	- 25.5
IP10	24.5	P31	55
IP18A	168	P32	- 28.5
IP18B	189.5	P33	- 71
13	48.5	P34	49.5
19	134	P35	- 14.5
20	124.5	P36	14.5
21	140.5	P37	-146
IP23	119	P38	-133
IP24	121.5	P41	78
IP25	134	P42	89.5
IP26	152.5	P50A	-179.5
22	136.5	P50B	-250
26	147.5	P66A	-106
30	152	P66B	-194.5
P1	80.5	P67A	-104
P2	- 29	P67B	-214.5
P5	67	3	125.5
P7	66	33	99
P8	175.5	34	83.5
P9	55	42	61.5
P10	54.5	45	84
P11	69.5	48	34.5
P12	33	49	36
P13	- 40	P25	95.5
P15	225.5	P26	98.5
P16	86	9	46
		12	51

¹The maximum strain is computed as the largest difference between the gage readings at 160°F and those occurring during cool down.

Negative numbers indicate compression.
Positive numbers indicate tension.

The double tubesheet stresses must be analyzed by a different method because this tubesheet is constructed by pinning together a steel tubesheet and a copper-nickel tubesheet to form a composite material tubesheet. It is convenient to analyze composite materials by the "equivalent width technique" (see reference 7 for a detailed discussion). Using this technique, the maximum stress in the copper-nickel material is 1443 psi and the maximum stress in the steel material is 1843 psi.

7.7 Comparison of Analytical and Test Results

The test results in Table 8 indicate small strains on the average of 100 micro-inches (1800 psi stress) in both the integral and double tubesheet and this is based on the highest test temperature difference of 78°F. In practice, the temperature differences are much smaller. The approximate analysis of section 7.6, which is based on conservative assumptions, indicates stresses of approximately 2500 psi and therefore a more exact analysis does not appear to be justifiable.

The data show that there are peak stresses (see gages P27, P28, P50, P23, P24 in Figure 5) located along the tubesheet centerline due to the waterbox partition. The partition divides the inlet and outlet water flows which are at different temperatures. The maximum stress is located at the center (gages P50 and P51) of the tubesheet and is -4500 psi (-250 micro-inches). The stress at the top of the vertical centerline (gages P27 and P28) is -2727 psi (-151.5 micro-inches) and at the bottom the stress (gages P23 and P24) is 2043 psi (113.5 micro-inches). These stresses are considered low.

7.8 Conclusions and Recommendations

The measured tubesheet thermal stresses are low (1800 psi average stress) and are in general agreement with the approximate calculation method. The thermal stresses are below the fatigue limits for both the steel/copper-nickel and copper nickel tubesheets. If the thermal stresses had occurred in the presence of the highest allowable stresses due to internal pressure as determined by the HEI codes, the low cycle fatigue limits are not exceeded and the design is considered adequate. On this basis, a more exact theoretical analysis is not recommended.

8 LEAKAGE TEST

8.1 Objective

The purpose of this test was to inspect the test condenser for leaks at the tube-to-tubesheet joint before and after it has been subjected to specific mechanical and thermal loads and thereby determine the impact of the loads on the joints.

8.2 Background

Due to recent trends in condenser design (such as increased size, and increased circulating water pressures), the number of failed tube-to-tubesheet joints reported has been rising. For this reason, the prototype condenser was tested for tube-to-tubesheet joint integrity after the testing described earlier in this report was completed. This test could then give an indication of whether or not such loadings were sufficient to cause tubes to break free of the rolled tube-to-tubesheet joint.

8.3 Description of Test

The tube plates were tested for leaks by pressurizing the groove volume of the integrally grooved tube plate and the space between the tubesheets on the double tube plate. The cavity is pressurized with 10 psi of air and then isolated from the pressure source and the rate of decay of the pressure is then measured (the shell and waterbox pressures are at ambient conditions).

8.4 Results

The first results (see Table 9) were obtained before any other testing began, including the shell hydrotest. The second set of results were obtained after all tests (water chamber pressurization, vibration, shell hydrotest, and thermal testing) had been completed. In general, the double tubesheet tube-to-tubesheet integrity did not noticeably change, but there was loss of integrity in the joints in the integrally grooved tubesheet.

8.5 Discussion of Results

Both halves of the integrally grooved tubesheet showed leak rate increases. The outlet half of the integrally grooved tubesheet still had a very slow leak rate. The results for this half of the tubesheet are not too surprising because of the extreme difficulty encountered in rolling the titanium tubes into the 90-10 copper-nickel tubesheet. This difficulty is thought to be caused by a lack of familiarity with the rolling of these two materials. The difficulty was evidenced by joint instability, that is, increasing pressure decay rates with the passage of time, which necessitated several re-rolling operations on some tubes. The results of this test for this half of the integrally grooved tubesheet are therefore inconclusive since it is possible that the observed change could have taken place in the absence of tube loading.

TABLE 9
LEAKAGE TEST RESULTS

Tubesheet	Date	Pressure Loss AT 10 PSI
Double tubesheet	10/76	1 psi/hr.
	7/77	1 psi/3 hr.
Integrally grooved tubesheet, outlet half	10/76	1 psi/20 min.
	7/77	1 psi/55 sec.
Integrally grooved tubesheet, inlet half	10/76	1 psi/15 min.
	7/77	1 psi/sec.

The inlet half of the integrally grooved tubesheet showed a dramatic change in leak integrity. The principal causes of the change were two inserts, called thimbles, that were used to plug holes unintentionally drilled in the tubeplate. One of these plugs was intentionally loosened for a separate test in October 1976 in which a leak detection method using acoustic emissions* was studied. This thimble was never re-rolled securely into place after the testing. In addition, the other thimble leaked during the July 1977 groove pressurization test. It is not known why this thimble became loose. The thimble material is 90-10 copper-nickel.

The rolling of the double tubesheet was much easier and, at the time of installation, showed more stability and greater ability to hold pressure. It is therefore not unexpected that greater stability for this tubesheet is indicated by the data.

8.6 Conclusions and Recommendations

The joint stability indicated by the titanium tube rolled into the double tube plate was better than that for the titanium tube in the integrally grooved tubesheet. This was true at the time of installation and after the stress and vibration testing. However, insufficient information exists to completely explain why these conditions existed.

It is recommended that further experimentation be undertaken to better identify the most reliable materials and procedures for rolled tube-to-tubesheet joints using titanium tubes.

*Note: Acoustic emission tests are not a part of this study.

Figure B
Reverse Water Chamber Dimensions

TUBESHEET LAYOUT

APPENDIX II

TEST PROCEDURE

The following test procedure was used to gather strain gage data for tests described in Sections 3, 4 and 5.

1. Fill water boxes with water.
2. A sequence of steps is required to be carried out five times with a different set of gages being monitored and recorded each time. The five sets of gages are as follows:
 - a. Integral tube plate, gages #P1-P46, double tube plate gages #IPI-IP8, #321 on outside of reversing chamber, and tube gages 3, 33, 42 and 48 (see Figures 5, 6, 9, 10 and 11 for strain gage locations).
 - b. All tube gages #1-62 and #321 on outside of reversing chamber (2 gage elements)
 - c. Inlet water box gages #101-116 and #201-216 (96 gage elements) plus #321 on outside of reversing chamber (2 gage elements).
 - d. Reversing chamber gages #301-314 and #401-414 (84 gage elements), #321 on outside of reversing chamber (2 gage elements), and inlet water box gages #117, 118, 217 and 218.
 - e. Reversing chamber gages #315 through 327 and #415 through 427 (52 gage elements).
3. The sequence of steps to be followed five times is steps 3-20.
4. Take zero reading on all gages being monitored (set a, b, c, d or e) at zero psig.
5. Increase pressure to 20 psi.
6. Read and record all gages being monitored (set a, b, c, d, or e).
7. Increase pressure from 20 psi to 30 psi.
8. Repeat step 5.
9. Increase pressure from 30 psi to 40 psi.
10. Repeat step 5.
11. Increase pressure from 40 psi to 50 psi.
12. Repeat step 5.

13. Increase pressure from 50 psi to 60 psi.
14. Repeat Step 5.
15. Decrease pressure from 60 psi to 40 psi.
16. Repeat Step 5.
17. Decrease pressure from 40 psi to 20 psi.
18. Repeat Step 5.
19. Decrease pressure from 20 psi to 0 psi.
20. Repeat Step 5.
21. Disconnect one set of gages and connect the next set.

APPENDIX III

DESCRIPTION OF FINITE ELEMENT COMPUTER PROGRAM

A commercially available general purpose finite element computer program, (see reference 19) was used for the analysis described in Sections 3, 4, 5 and 7. This program was selected because it was readily available and because it contains a three-dimensional shell element that was suitable for the analysis of the tubesheet and waterboxes. The ANSYS program has been verified in reference 20. The program contains grid and element preprocessors that simplify program input. There are many other commercially available computer programs that are equally suitable for finite element analysis of this type.

The computer program utilized is a large scale program for the analysis of several classes of engineering problems. Analysis capabilities include static, dynamic, plastic, creep, swelling, large deflections, steady state heat transfer, transient heat transfer and steady state fluid flow.

The ANSYS program is based on the displacement method of finite element analysis. The stiffness matrix for each element is assembled into a system of simultaneous equations for the entire structure. This set of simultaneous equations is solved by a variation of the Gaussian elimination method known as the "wave front technique". The nodal displacements obtained from the solution of a simultaneous equation are then substituted into the stress-displacement matrix relationship for the element and element stresses are determined. A flat triangular shell element and a flat quadrilateral shell element were selected for the analysis. These elements are referred to in the ANSYS Computer Program as elements 13 and 63 respectively and the details of these elements are described below. Both elements have the same properties but triangular elements can model boundaries better and can be used to increase element density for a given node pattern.

The element is defined by three or four nodal points, three or four thicknesses and the material properties. The material properties may be orthotropic but this capability was not utilized. The shear modulus term is optional and if it is not included, it is computed from the other input material properties. The element also has the capability for thermal stress analysis.

The element also has an elastic foundation modulus option that was used in the tubesheet analysis. The elastic foundation of the tubesheet is the tube array and is shown in the sketch on the following page.

K = Spring constant of tubes (lb/in) = AE/ℓ

A = Cross section area in tube (in^2)

ℓ = half tube length (in)

E = modulus of elasticity of tube lb/in²)

The elastic foundation of each element was determined by counting the number of tubes in the element and uniformly distributing the spring constants (load required to elongate a tube by 1 inch) of these tubes over the surface of the element. Also shown in the above sketch are the parameters used to determine the elastic foundation: cross-section area of tube, modulus of elasticity of the tube, half length of tube (also the distance from the tubesheet face to the transverse centerline of the condenser), and surface area of element.

Computer Program Output

The output of the computer program can be summarized as follows:

1. Listing of all input data
2. Node point deflections
3. State of stress at top, middle and bottom surfaces of all elements
4. Principle stresses at top, middle and bottom surfaces of all elements

The output listing is self-explanatory and a sample output appears on the following page.

SAMPLE COMPUTER PROGRAM OUTPUT

FLAT TRIANGULAR SHELL ELEMENT

ELEMENT PRINTOUT EXPLANATIONS

LABEL	NUMBER OF CONSTANTS	FORMAT	EXPLANATION
Line 1			
Shell	1	15	Element number
Nodes	3	315	Nodes I, J, and K
Material	1	13	Material number
Area	1	G13.5	Element Area
Top, TBot	2	2F6.0	Top and bottom surface temperatures
Press	1	F6.0	Normal pressure
Line 2			
NX,NY	2	2F8.0	Shear forces along element sides
MX,MY,MXY	3	3F8.0	Moments in local X and Y directions
YC,YC,Z	3	3G12.4	X,Y,Z global location of centroid of element
Lines 3 - 5			
Label			Top, middle, or bottom
SX,SY,SXY	3	3F8.0	Local coordinate combined membrane and bending stresses
SMAX,SMIN,TAUMAX	3	3F8.0	Principal stresses
A	1	F5.1	Angle of principal stresses relative to local X-y axes
VM	1	F8.0	Von mises equivalent stress

ELASTIC FLAT TRIANGULAR SHELL ELEMENT

The flat triangular shell element has both bending and membrane capabilities. Both in-plane and normal loads are permitted. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes.

Options are available for neglecting the membrane stiffness or the bending stiffness. These options, where applicable, reduce the element formulation time. Another option permits the pressure loading to be concentrated (converted to forces) at the element nodes, giving better stress results for curved shells. The element may also have a variable thickness.

The geometry, nodal point locations, loading and the coordinate system for this element are shown in Figure D. The element is defined by three nodal points, three thicknesses and the material properties. The material x-direction corresponds to the element x-direction. The shear modulus term is optional and if it is not included, it is computed from the other input material properties.

The element may have variable thickness. The thickness is assumed to smoothly vary smoothly over the area of the element, with the thickness input at the three nodal points. If the element has a constant thickness, only one need be defined. If the thickness is not constant, all three thicknesses must be defined.

The element loading can be either surface temperatures or pressure or a combination of both. The positive direction of pressure is along the positive z-axis. The pressure loading may be uniformly distributed over the face of the element, or a curved shell loading consisting of an equivalent element load applied at the nodal points may be used. The latter loading produces more accurate stress results in curved shells because certain fictitious element bending stresses are eliminated. An elastic foundation stiffness may be added to the element.

An assemblage of flat shell elements can produce a good approximation, to a curved shell surface provided that each flat element does not extend over more than a 15° arc. In a rectangular array of nodal points, best results are obtained from an element pattern having alternating diagonal directions. Also, since the element coordinate system is relative to the I-J line, the stress results are most easily interpreted if the I-J lines of the elements are all parallel. Note, if three nodal temperatures are supplied, only the first two are used in computing the average element temperature.

NOTE: x and y are in the plane of the element,
 x is parallel to IJ.

Figure D

Flat Triangular Shell Element

ELASTIC FLAT QUADRILATERAL SHELL ELEMENT

The quadrilateral shell element has both bending and membrane capabilities. Both in-plane and normal loads are permitted. The element has six degrees of freedom at each node: translations in the nodal x , y , and z directions and rotations about the nodal x , y , and z axes.

The geometry, nodal point locations, loading and the coordinate system for this element are shown in Figure E. The element input data is initially the same as for the triangular shell element (STIF13) described above. This element, however, is defined by four nodal points instead of three and is restricted to a constant thickness.

The applied thermal gradient is assumed to be linear through the thickness and uniform over the element surface.

The four nodal points defining the element should lie in an exact flat plane; however, a small out-of-plane tolerance is permitted so that the element may have a slightly warped shape. A slightly warped element will produce a warning message in the printout. If the warpage is too severe, a fatal message results and triangular elements should be used.

An assemblage of flat shell elements can produce a good approximation to a curved shell surface provided that each flat element does not extend over more than a 15° arc.

Figure E
Flat Quadrilateral Shell Element

APPENDIX IV

RULES FOR USING STRAIN GAUGE DATA

VI 1 For 20 psi and 40 psi test results:

(average strain at pressure) - (average "zero" reading)

Where:

Average Strain at Pressure =

$$\frac{\text{Initial Strain Reading} + \text{Final Strain Reading}}{2}$$

Average "Zero" Reading =

$$\frac{\text{Initial "Zero" Reading} + \text{Final Zero Reading}}{2}$$

When:

$$\left(\frac{\text{Final Strain Reading} - \text{Initial Strain Reading}}{\text{Initial Strain Reading}} \right) > .25,$$

then INC, meaning inconsistent, is entered in the data sheet, unless

(Final Strain Reading - Initial Strain Reading) < 10 μ in./in.

The above "INC" rule was applied to zero readings.

VI 2 For 60 psi:

(Strain at Pressure) - (Average "Zero" Reading)
unless INC is entered at 40 psi, then INC is automatically entered
at 60 psi.

VI 3 "No" means that both strain readings at the indicated pressure are
missing.

APPENDIX V

Procedures for the Determination of Fundamental Bending Frequency and the Damping of the Tube-Tube Support System of the Test Condenser

Frequency Response Method of Fundamental Bending Frequency

1. Attach a 6 pound shaker to the selected tube at midspan between supports No. 7 and 8 for the 33 in. span tube or between support No. 7 and 7p for the 16.50 in. span tube. The full size supports are numbered from the inlet end of the condenser. The partial supports are numbered from the inlet end of the condenser. Use epoxy to attach the shaker rod to the tube such that the rod longitudinal centerline is coincident with a radius of the tube cross section.
2. Select and apply excitation (shaker) force. Select sinusoidal excitation.
3. Determine approximate fundamental bending resonant frequency by a large variation of the excitation frequency. Observe the frequency at which the response peaks, with excitation held approximately constant.
4. Filter the excitation and response signals as required. Note filler phase shifts.
5. Vary shaker frequency in small steps through the resonant region. Hold excitation (peak-to-peak force) constant. Record response (peak-to-peak amplitude of the tube deflection) and peak-to-peak force.

Decay Method of Fundamental Bending Frequency

1. Excite selected 33 in. span tube at midspan, between supports No. 7 and 8 with 6 pound shaker. Use rubber tip on end of shaker armature.
2. Determine approximate fundamental resonant frequency by a scan of the frequency range by variation of the excitation frequency. Observe the frequency at which the response peaks, with excitation peak force held approximately constant.
3. Turn on chart recorder on which tube response is recorded.
4. Retract shaker from tube. Record decay trace on chart recorder until tube amplitude approaches zero; then turn off recorder.
5. Examine the decay trace. Ascertain that the tube frequency did not change during decay. Determine that this mode is the lowest mode for which the frequency-during-decay remained constant.

Frequency Response Method of Damping Determination

1. Attach a 6 pound shaker to the selected tube at midspan between supports No. 7 and 8 for the 33 in. span tube and between support No. 7 and 7p for the 16.50 in. span tube. The full size supports are numbered from the inlet end of the condenser. The partial supports are also numbered from the inlet end of the condenser. Use epoxy to attach the shaker rod to the tube such that the rod longitudinal centerline is coincident with a radius of the tube cross section.
2. Select and apply excitation (shaker) force. Select sinusoidal excitation.
3. Determine approximate fundamental bending resonant frequency by a scan of the frequency spectrum. Observe the frequency at which the response peaks, with excitation held approximately constant.
4. Filter the excitation and response (accelerometer) signals as required. Note filter phase shifts.
5. Vary shaker frequency in small steps through the resonant region. Hold excitation (peak-to-peak force) constant. Record response (peak-to-peak amplitude of the tube deflection) and force.

Decay Method

1. Excite selected 33 in. span tube at midspan, between supports No. 7 and 8 with 6 pound shaker. Use rubber tip on end of shaker armature.
2. Determine approximate fundamental resonant frequency by a scan of the frequency range by variation of the excitation frequency. Observe the frequency at which the response peaks, with excitation held approximately constant.
3. Turn on chart recorder on which tube response is recorded.
4. Retract shaker from tube. Record decay trace on chart recorder until tube amplitude approaches zero; then turn off recorder chart drive.
5. Examine the decay trace. Ascertain that the tube frequency did not change during decay. Determine that this mode is the lowest mode for which the frequency-during-decay remained constant.
6. Measure peak amplitudes of successive cycles for log dec calculation.

APPENDIX VI

Thermal Test Procedure

After studying alternative methods of simulation, the following method was chosen. The condenser shell is filled with tap water. The condenser then is allowed to reach a uniform temperature distribution at ambient temperature. Steam is then sparged into the shell side to raise the temperature to a predetermined point. After a period of equilibration (during which time the steam is turned off), Delaware River water is introduced to the empty tube side at a rate of 20,000 gpm. Thermocouple and strain gage data is taken while the once-through river water cools the shellside water.

Detailed Test Procedure

1. Set up test by installing strain gages and thermocouples as shown in Figures 5, 6, and 26, and a manometer on cooling water inlet and outlet flow as well as thermocouples in the inlet and outlet water flows.
2. Fill shellside to a point 6 inches over the top of the tubesheet with tap water. Tube side should be drained and let empty until Step 4 below.
3. Inject steam into this shellside water volume until the shellside reaches a temperature T_s . Steam should be injected in such a way as to encourage a uniform temperature distribution throughout. (See below for values of T_s .)
4. Turn off the heating steam to the shellside; allow thermocouple to reach equilibrium condition.
5. As rapidly as possible, bring the tube side water velocity up to V_t with cooling water at temperature T_a . Here the attempt is being made to simulate a step change in condition.
6. Monitor and record all thermocouple and strain gage information during the duration of this transient test. The duration of the test will be established at the time of the test and will be based on the time dependency of the thermocouple and strain gage results.
7. This test should be run for the following values of T_s and V_t . T_a will be whatever is available from the cooling water supply:
$$T_s = 120^\circ \text{ and } 160^\circ\text{F}$$
$$V_t = 1.6 \text{ and } 6.5 \text{ ft./sec.}$$
8. The program above calls for four separate tests. They should be run in duplicate in order to have a check on the repeatability of the data. This will make a total of eight runs.

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