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SURVEY OF NUCLEAR REACTOR SYSTEM

PRIMARY CIRCUIT HEAT EXCHANGERS

H. A. Nelms
C. L. Segaser

OAK RIDGE NATIONAL LABORATORY
operated by
UNION CARBIDE CORPORATION
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U.S. ATOMIC ENERGY COMMISSION

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General Engineering Division

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APRIL 1969

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SURVEY OF NUCLEAR REACTOR SYSTEM PRIMARY CIRCUIT HEAT EXCHANGERS

Abstract

Problems resulting from excessive tube vibration in certain primary-circuit heat exchangers in reactor systems for which the USAEC Division of Reactor Development and Technology is technically responsible are evaluated in this report. Of the 19 reactor systems surveyed, problems in primary exchangers resulting from tube vibration hydrodynamically induced by the shell-side fluid flow were experienced in 9 of the 17 plants that have been operated. The results of a detailed vibration analysis that was based on well-known procedures and information available in the literature tend to support the general conclusion that if the designs for these exchangers had been critically examined for tube vibration prior to fabrication, the possibility of failure would have been judged too great to accept and the problems could have been averted by design modifications.

1. INTRODUCTION

Damage of heat exchangers attributed to excessive tube vibration has prompted a survey of some of the primary-circuit heat exchangers in reactor systems for which the USAEC Division of Reactor Development and Technology is technically responsible. This survey is part of a continuing program directed toward upgrading the quality of future reactor plants, and its essential purposes are

1. to determine whether the primary-circuit heat exchangers currently installed in nuclear reactor facilities and those now being designed and constructed are vulnerable to damage from vibration, and
2. to determine what further efforts must be taken to prevent failure from vibration in heat exchangers.

The survey was conducted by reviewing data provided by several AEC contractors related to the design, development, manufacture, and testing of

existing and/or planned primary-circuit heat exchangers in their reactor plants. However, no tests or inspections of heat exchangers other than those normally performed by the AEC contractor were authorized for this survey. The reactor plants included in this survey, the nature of the plants, and whether or not vibration problems have been experienced are given in Table 1.

The flow induced vibration problem is not peculiar to a specific type of heat exchanger, but the investigation reported here has been directed primarily toward units with liquids on both the shell side and the tube side. The particular units that were included in the investigation were selected from a list originally compiled by the USAEC Division of Reactor Development and Technology. Subsequently, some units were selected for detailed study, and detailed calculations and reviews of calculations made by others were made on 16 heat exchangers in 12 systems. The number of units receiving detailed review was limited mainly by the time available, and to a lesser degree, by the time at which data became available, and by the uncertainty associated with some data.

In preparation for reviewing the data collected on the various heat exchangers, a thorough search of the available literature was made and the vibration problem in general was analyzed briefly to determine the significant variables to be investigated. This is discussed in Chapter 3 of this report, and a supplementary bibliography of selected references from the literature reviewed is given in Appendix A. The significant variables are readily identifiable, but useful information on structural and fluid damping, the effects of tube supports (baffles) on the frequency of vibration, prediction of shell-side fluid velocities, and lift and drag flow coefficients in tube banks is essentially nonexistent.

The data reviewed and the procedures used to evaluate the data are described in Chapter 4. The analytical relationships used in the vibration analysis are discussed in detail in Chapter 5. The conclusions and recommendations resulting from this survey are presented in Chapter 6, and a brief summary of this material is given in Chapter 2.

Table 1. The USAEC-DRDT Reactor Plants Included in the Heat Exchanger Survey[a]

Name of Reactor Plant	Designation	Category[b]	Location	Purpose	Type	Power kw(t)	Startup-Shutdown	Vibration Problems Experienced
Advanced Test Reactor	ATR	1	NRTS, Idaho	General irradiation test	Tank	250,000	1967	Yes
Experimental Breeder Reactor No. 2	EBR-2	1	NRTS, Idaho	Experimental elec. power	Sodium cooled fast	62,000	1963	No
Experimental Boiling Water Reactor	EBWR	1	Argonne, Ill.	Experimental elec. power	Boiling water	100,000	1956-1967	No
Elk River Reactor	ERR	1	Elk River, Minn.	Cen. Sta. elec. power	Boiling water	58,200	1962	No
Engineering Test Reactor	ETR	1	NRTS, Idaho	General irradiation test	Tank	175,000	1957	Yes
Fast Reactor Core Test Facility	FRCTF	c	Los Alamos, N. M.					No
Brookhaven High Flux Beam Research Reactor	HFBR	1	Upton, N. Y.	High power research and test	Heavy water	40,000	1965	Yes
High Flux Isotope Reactor	HFIR	1	Oak Ridge, Tenn.	High power research and test	Tank flux-trap	100,000	1965	Yes
Hallam Nuclear Power Facility	HNPF	1	Hallam, Neb.	Cen. sta. elec. power	Sodium-graphite	256,000	1962-1964	Yes
Heavy Water Components Test Reactor	HWCTR	1	Aiken, S. C.	Experimental elec. power	Pressurized heavy water	61,000	1962-1964	No
Loss of Fluid Test	LOFT	2	NRTS, Idaho	Safety research and test	Pressurized water	50,000	1972	
"Sturgis" Floating Nuclear Power Plant	MH-1A	1	Ft. Belvoir, Va.	Defense power, remote	Pressurized water	45,000	1967	No
Molten Salt Reactor Experiment	MSRE	1	Oak Ridge, Tenn.	Experimental elec. power	Single region graphite moderated	7,500	1965	Yes
Materials Testing Reactor	MTR	1	NRTS, Idaho	General irradiation test	Tank	40,000	1952	No
Power Burst Facility	PBF	2	NRTS, Idaho	Safety research and test	Tank	20,000	1971	
Piqua Nuclear Power Facility	PNPF	1	Piqua, Ohio	Cen. sta. elec. power	Organic cooled and moderated	45,500	1963-1967	No
Plutonium Recycle Test Reactor	PRTR	1	Richland, Washington	Experimental elec. power	Pressure tube, heavy water moderated and cooled	85,000	1960	Yes
Stationary Medium Power Plant No. 1A	SM-1A	1	Gatun Lake, Canal Zone	Defense power, remote	Pressurized water	20,200	1962	Yes
Sodium Reactor Experiment	SRE-PEP	1	Santa Susana, California	Experimental elec. power	Sodium-graphite	30,000	1957-1966	Yes

[a]Nuclear reactors built, being built, or planned in the United States as of December 13, 1967 (USAEC Report TID-8200, 17th rev., Reactor Technology).

[b]Plants that have been operated are in Category 1, and those in various stages of design and/or construction are in Category 2.

[c]Does not fall in either category since the plant has not been operated and apparently will not be operated.

2. SUMMARY

The results of this survey and vibration analysis indicate that heat exchangers designed and fabricated in accordance with existing standards may be vulnerable to failure as a result of excessive tube vibrations caused by the shell-side fluid flow. The likelihood of failure in primary-circuit heat exchangers in reactor plants is great when strong emphasis is placed on obtaining low shell-side pressure drops. Generally speaking, some methods, such as decreasing the baffle spacing, used to obtain higher heat transfer ratings for a specific shell size will improve the vibration picture; but other methods, such as decreasing the tube size and pattern, used to improve the heat transfer rating of a specific shell will enhance the possibility of failure caused by tube vibration.

The vortex shedding frequencies and the natural frequencies of tube vibration were compared in a number of exchangers, and this comparison strongly suggests that destructive tube vibrations should be expected when these two frequencies are nearly matched. Although the results of this comparison are not absolutely conclusive, this type of comparison appears to be the best indicator of possible vibration problems that is currently available. By using this indicator, it is likely that a designer can arrive at a satisfactory design for exchangers similar to those currently being used if he is given sufficient latitude in which to operate. That it is necessary to provide the designer latitude of operation must be emphasized. It is obvious that one could set any one of a number of restrictions that would preclude finding a practical solution. It may not be quite so obvious that the establishment of two limits such as tube-side volume and shell-side pressure drop, both of which appear reasonable in normal practice, could pose an impossible task for the designer.

Tube vibration precipitated by shell-side flow conditions is the major problem. Other possible excitation sources, such as the tube-side flow, pump fluctuations, and mechanically transmitted power from nearby equipment, do not appear to constitute a serious problem. Shell-side flow conditions, especially at shell entrances and exits, in regions of high bypass flows, through U-bends, and at baffle edges, can precipitate

several modes of failure. The principal modes of failure are stress cycling in the presence of defects or configuration stress risers, impaction of tubes with one another or the shell, and wear at baffles or other supports. In at least two of the cases reviewed, failure could be partially attributed to the simultaneous interaction of vibration and corrosion.

Except for the extreme cases where exceptionally large or small amplitudes of vibration exist, the life of a heat exchanger tube cannot be predicted accurately because the effects of tube supports (baffles and bundle stiffening members), local shell-side velocities, and system damping are not quantitatively defined. Even if the shell-side velocities are well known, vortex production in tube bundles, the mechanism of excitation, is still little understood and accurate stress calculations are precluded. Even if nominal stresses are determined accurately, the detection and evaluation of defects in the tubes and of low cycle thermal fatigue pose a difficult task.

In some cases where attention has been given to the tube vibration problem, it has been assumed that a "moderate" separation between the calculated value of vortex shedding and the natural frequency of the tube would insure against a resonance condition between these two frequencies, thereby eliminating the possibility of destructive amplitudes of tube vibration. The natural frequencies of vibration of the tubes can be calculated without excessive conservatism, but there is considerable uncertainty associated with vortex shedding frequencies and damping. Vortex shedding frequencies are a function of the fluid velocity, but the velocity varies along a tube span in a heat exchanger and the nature of this variation is not known. The amount of damping in a tube bundle environment is completely unknown. Because of this general lack of information or understanding, it is not possible to say with any authority what constitutes a safe separation between the vortex shedding frequency and the natural frequency of a tube without resorting to factors of separation that constitute significant penalties to the heat transfer rating of the unit. A factor of separation between these two frequencies of from 2 to 3 appears to be necessary to assure against tube vibration failures at

locations in the bundle not strongly influenced by inlet and outlet conditions and where the vortex shedding frequency is based on the average shell-side fluid velocity, V_x, obtained by dividing the volumetric flow rate by the total area available for cross flow. The separation factor that must be used therefore depends strongly on the accuracy with which local shell-side velocities can be determined.

The assumption that there is a unique shedding frequency associated with a particular tube in a heat exchanger tube bundle does not appear to be valid except possibly for the special case of resonance. Apparently there is a spectrum of vortex shedding frequencies associated with each set of geometrical and fluid flow rate conditions. That is, each tube is being excited by many impulses of varying intensity and frequency that cause the tube to vibrate at its natural frequency. There is some scatter in the measured frequency of vibration of a tube because the conditions of support of the tube vary. As the fluid flow rate is increased, the amplitude of vibration will increase. Experiments with single members indicate that the amplitude of vibration will increase as the flow rate is increased until the amplitude of excitation (caused by random frequency excitation) is approximately 0.1 of the diameter of the member. At this point, resonance between the vortex shedding frequency and the natural frequency of the tube will occur. Once resonance has occurred,the amplitude of vibration is limited only by the system damping or by physical limits. Increasing the fluid flow rate beyond the rate necessary to induce resonance will destroy the resonant condition, and the amplitude of vibration will again be a function of the spectral content of the turbulence.

If the tubes in a heat exchanger approach the resonant condition in a manner like that just described for a single tube, it is clear that a good deal more information than is available will be required before a heat exchanger can be designed without using fairly liberal safety factors. For units similar to those reviewed in this survey, the use of liberal safety factors does not appear to be unreasonable. This use coupled with a test at 100 to 125% of shell-side flow in which the amplitude and frequency of vibration of tubes in critical areas are measured can virtually eliminate tube failures caused by shell-side flow conditions. Those

responsible for design should be required to perform an analytical evaluation which is as comprehensive as reliable information will permit and submit the data on this evaluation for approval.

For heat exchangers with unusual arrangements and especially for units which must have a high heat transfer rating coupled with a low shell-side pressure drop, the solution to the vibration problem is not in hand. Experimental work is indicated.

We recommend the implementation of an experimental program to obtain reliable data for the design of reactor-system heat exchangers to minimize the potential for destructive tube vibrations. The experimental program should yield correlations between the typical geometrical characteristics that are controlled by the heat exchanger designer and the dynamic response of the parts of the heat exchanger, particularly the tubes. A second but equally important purpose of the experimental program would be to determine appropriate methods for practical pre-operational evaluation of tube response to shell-side flow.

3. THE VIBRATION PROBLEM

Generally, the three performance goals that must be satisfied in the design of a heat exchanger are (1) the transfer of a specified amount of heat, (2) operation within pressure loss criteria, and (3) an operating life compatible with that of the system of which it is a part. Four major limiting factors influence the achievement of these goals, and they are allowable space, system compatibility, maintenance requirements, and cost. Achievement of the first two goals may be influenced by any one or all of these factors. However, achievement of the third goal is influenced by all four of these factors because it involves design, fabrication, and operating limits.

The general design considerations, basic relations, and approaches applicable to the thermal design of heat exchangers are adequately described by Fraas and Ozisik[1] and Kern.[2] Information from these and other sources may be combined with the safety and service requirements of the Tubular Exchanger Manufacturers Association[3] and with those given in the ASME Boiler and Pressure Vessel Code[4] to produce a satisfactory design for a heat exchanger for nuclear service that will meet specific design criteria related to heat transfer, pressure drop, and safety. However, the recently experienced tube failures from induced vibration in some exchangers make it clear that further information is needed if such failures are to be averted in the future.

A considerable quantity of information pertaining to flow-induced vibration phenomena has been published since Strouhal, in 1876, and Rayleigh, in 1896, first began the systematic study of the oscillations excited in circular rods from lateral flow of fluid over the rods. Most of the published literature is directed toward the flow phenomena associated with single stationary cylinders. A review containing 155 references on vortices and the forces they produce was published by Morkovin,[5] and Marris[6] published a similar review containing 52 references. Only a limited number of these references contain data on oscillating cylinders and tube bundles, and very few of these papers contain data that can be directly applied in an engineering type of vibration analysis study.

From our review of the literature, we therefore concluded that the mechanics of the formation of periodic vortices in the wake of cylindrical bodies in a stream is still only partially understood and the vibration phenomena in heat exchanger tube bundles are even less understood.

The problem of vibration of heat exchanger components has been recognized for some time, but because of its complexity, the solution often has been to design around the problem. This type of approach has frequently resulted in the need for corrective measures after the heat exchanger has been installed. In general, the vibration problem has been attacked but not always solved by accepting such characteristics as short baffle spacings (thereby increasing pressure drops and requiring larger pumps), lower flow velocities, bulkier components, etc. These brute force methods of solving the problem only serve to increase costs and lower unit rating.

The vibration problem in heat exchanger components is compounded by the introduction of a corrosive environment. It is customary in the design of heat exchanger tubing, piping, and other components that are to be operated in a corrosive environment to allow for the effects of corrosion by adding excess material. The amount to be added depends upon the corrosion rate and the desired life of the equipment. Such corrosion allowances would ordinarily be adequate for static operating conditions. However, if the tubing in a heat exchanger is simultaneously subjected to vibratory and corrosive conditions, the effects of fatigue and corrosion may interact to drastically reduce the reliability and operating life of the equipment.

Under the simultaneous action of repeated stress and corrosion, the fatigue strength of most metals is drastically reduced, sometimes to a small fraction of the strength determined by fatigue tests conducted in air. The protective film of oxide breaks down under alternating stress and allows the formation of corrosion pits that progresses more rapidly under stress. The pits cause a high concentration of stress at the bottom, and this cycle of the pit increasing the stress and the stress increasing the depth of the pit continues until the stress exceeds the endurance limit of the material. A fatigue crack is initiated and failure ultimately occurs.

To obtain reliable design data, fatigue tests should be conducted in the same environmental conditions to which the unit will be subjected during actual operation. Some corrosion-fatigue data are available in the literature. The results of experiments made on various types of steel with a number of corrosive agents were reported by McAdam.[7] Liquids and gases not usually considered as being especially corrosive will often have a very deleterious effect on fatigue properties. Some data obtained in a freshwater environment indicate that the corrosion-fatigue limit could be reduced to about one-half of its value in air. The important conclusion to be made from these experiments is that the corrosion-fatigue properties depend primarily on the corrosion resistance of the material under the test conditions. The maximum resistance to corrosion fatigue will be exhibited when the metal alloy is in the best condition to resist corrosion. High-chromium steels have improved corrosion resistance and they also exhibit improved corrosion-fatigue properties.

In several of the primary-circuit heat exchangers surveyed for the USAEC, there was evidence that heavy wear of the tube walls had occurred at some of the baffle holes. In some cases, the tube wear at the baffles had reduced the wall thickness of the tube by about 50%. In one of the systems surveyed, failure of the tubing was attributed to tube wall wear that reduced the thickness of the wall to such a point that the tubes ruptured because of the pressure differential between the tube-side and shell-side fluids. However, this wear was apparently caused by a resonant vibration condition that caused adjacent tubes to vibrate with sufficient amplitude to make contact, and the resulting tube failure occurred between baffles and not at the baffle plate penetrations. Such failures could be attributed to fretting fatigue, which is also called rubbing or chafing fatigue.

Most observations on fretting fatigue are in relation to mild steel. However, most other metals and alloys also appear to be susceptible to damage by fretting, particularly if they are of the type subject to galling or seizure under contact pressure. It is well known that the nickel-iron-chromium alloys, ranging from Inconel to the Series 300 stainless steels, gall when two members of the same alloy are subjected

to bearing loads. The laws of fretting fatigue are stated to be generally the same as those for notch fatigue and corrosion fatigue. The literature on fretting fatigue phenomena is extensive, and the reader is referred to the bibliography of 87 references on this general subject published in the ASME Handbook of Metals Engineering Design.[8] A similar bibliography of 133 references on the general subject of the mechanical factors that influence corrosion, including corrosion-fatigue, stress-corrosion cracking, erosion corrosion, and cavitation erosion, is also included in this publication.

An analysis of the problem of vibration in heat exchanger components should therefore involve consideration of all the possible sources of vibration. Vibration in a heat exchanger may be induced

1. hydrodynamically by shell-side flow,
2. hydrodynamically by tube-side flow,
3. by oscillations or pulsations in the flow rate of the shell-side or tube-side liquid,
4. by vibration transmitted through piping and supports, and
5. by external mechanical vibration transmitted to the apparatus.

These sources can induce vibratory modes on the tubes, baffles, and shell. The first source cited is most prevalent, particularly with respect to tube vibration failure, and it accounts for the largest part of heat exchanger failures from vibrations. The second source is usually negligible except at velocities much higher than those existing in heat exchanger tubes. The last three sources listed must be treated from a system design standpoint, and they cannot be disregarded if a satisfactory design is to be assured.

High-velocity fluid flowing through the shell side of heat exchangers is the primary source of energy that can create a potential for destructive vibration in the heat exchanger tubing. Therefore, the crux of the vibration problem is whether engineering practice can be sufficiently conservative to avoid flow induced vibrations and remain practical. From this standpoint, some factors that need to be considered are the

1. maximum fluid velocities that are acceptable;
2. maximum spacing tolerable between tube supports or baffles;

3. minimum clearance for assembly, maintenance, and thermal expansion between tubes and tube support holes;
4. rigidity of the tube as affected by wall thickness, internal and external pressure, bundle arrangement, and other physical parameters;
5. fluid flow distribution within the tube bundle; and the
6. structural effects of corrosion, erosion, fatigue, and creep.

An attempt has been made in this survey to determine whether limits can be placed on some of these variables to assure freedom from damage that might be caused by tube vibration.

4. SURVEY PROCEDURES

This survey was conducted by reviewing data provided by several USAEC contractors on heat exchangers in their plants. Some of these units were selected for detailed study, but the number selected was limited by the time available for performing the study and by the uncertainty associated with some data. Analytical evaluations of vibration in the units selected were made to provide a basis for comparing variables in heat exchangers known to have experienced vibration problems with those in exchangers that have not experienced such problems. Some idea of the precision expected in predicting unsafe natural frequencies and amplitudes of vibration of heat exchanger tubes through engineering design calculations should thereby be gained.

Data Reviewed

The data provided for the heat exchanger survey include two categories of units. The heat exchangers that have been operated are in Category 1, and the heat exchangers in various stages of procurement are in Category 2. The information provided for Category-1 equipment included

1. as-built manufacturer's drawings that depict the tubes, tube bundles, baffle and tube sheet arrangement, a cross section of the exchanger assembly, the fluid flow pattern, and the mechanical construction of the exchanger;
2. the specification sheet customarily furnished by the manufacturer that lists performance characteristics and fluid properties;
3. those portions of the manufacturer's design analysis report that treat vibration considerations; and
4. the results of inspection during fabrication, pre-operational testing, and periodic examinations related to damage from vibration.

The information provided for Category-2 equipment included

1. specific requirements issued during procurement of the primary-circuit heat exchanger that were directed toward prevention of destructive

vibrations by delineating the design procedures, reviews, tests, and inspections conducted prior to acceptance of the exchanger for service in the facility; and

2. a tabulation of the results of the vendor's design analysis and tests related to natural vibration frequencies of the tubes, forcing frequencies induced by the shell-side flow and the Strouhal number associated with these flows and tube arrangements, and the load per unit length for a mid-span tube deflection of one-quarter ligament distance.

The operating experiences of the primary-circuit heat exchangers evaluated analytically in this survey are described in Appendix B. A review of the data pertaining to tube vibration in the 19 AEC reactor plant primary-circuit heat exchangers given in Table 1 reveals that problems resulting from vibration have been experienced in 9 of the 17 plants that have been operated. Two of the reactor plants, LOFT and PBF, are in the Category-2 classification and have not yet been operated. Analysis of the original design (dated November 3, 1967) for the PBF heat exchanger indicated that its design was marginal from the standpoint of possible damage from tube vibration. The remaining eight plants have apparently experienced no difficulties from tube vibration although calculations indicate that the primary heat exchanger in the Piqua Nuclear Power Facility could have sustained damage from this source. Failure of the tubing in this exchanger was attributed to corrosion. With the exception of the HNPF, PNPF, SRE-PEP, EBWR, and the HWCTR, most of the reactor plants listed under Category 1 in Table 1 are apparently still in operation.

The available performance and construction data on the primary heat exchangers reviewed analytically are summarized in Tables 2 and 3. The data given include six of the nine Category-1 plants that definitely experienced problems with tube vibration, one of the Category-2 plants (PBF), the PNPF, and four of the plants that did not experience problems with tube vibrations.

The performance data for the heat exchangers presented in Table 2 indicate the service of the unit, its size and type, the heat transfer fluids involved, the shell-side flow rates, temperatures and pressures.

Table 2. Performance Data on Primary Heat Exchangers Reviewed

Reactor Plant	Service of Heat Exchanger	Exchanger Size and Type (TEMA Designation)	Heat Transfer Fluids		Shell-Side Flow Rate		Shell-Side Temperature		Shell-Side Pressure (psig)	Tube-Side Pressure (psig)
			Shell	Tube	(lb/hr)	(ft^3/sec)	Inlet (^oF)	Outlet (^oF)		
Advanced Test Reactor	Water-to-water heat transfer	68-327 CGU-VERT	H_2O	H_2O	4.31×10^6	19.3	111.65 av		<125	<390
Experimental Breeder Reactor No. 2	Intermediate heat transfer (Na to Na)	50-302-CEN-VERT	Na	Na	3.51×10^6	18.5	900	700	<150	<150
Elk River Reactor	Subcooler	16-158 DFU-HORIZ	H_2O	H_2O	1.125×10^5	0.501	350	464	912	1166
	Storage well cooler	11-72 DEU-HORIZ	H_2O	H_2O	1.25×10^5	0.556	90	98	100	100
	Regenerative cooler	3-38 U-Tube U-Shell	H_2O		6008	0.027	128	438	1250	1250
	Non regen. cooler	8-90 CJU-HORIZ	H_2O		7.26×10^4	0.322	90	110	1250	1250
	Decay heat cooler	16-222 CEU-HORIZ	H_2O		3.004×10^5	1.332	90	110	1250	1250
Engineering Test Reactor	Water-to-water heat transfer	42-240 CEN-HORIZ	H_2O	H_2O	1.58×10^6	7.10	78	110	<150	<250
Fast Reactor Core Test Facility	Intermediate heat transfer (Na to Na)	18-350 U-Tube U-Shell	Na	Na	6.258×10^5	3.31	1004	734	<300	<300
High Flux Beam Reactor Modified	Core heat removal	56-246 HORIZ	H_2O	D_2O	2.003×10^6	8.91	85	119	150	300
					1.002×10^6	4.95				
High Flux Isotope Reactor	Core heat removal	48-337 BGU-VERT	H_2O	H_2O	3.334×10^6	14.9	85	120	<150	600[a] 1000[b]
Hallam Nuclear Power Facility	Intermediate heat transfer (Na to Na)	34-156 CEN-VERT	Na	Na	1.40×10^6	7.28	557	895	80	65
Heavy Water Components Test Reactor	Steam generator	48-122 CEU-VERT	H_2O and steam	D_2O	9.83×10^4 (FW)	0.446	142	470		
					9.50×10^4 (ST)	23.6	470	470		
Molten Salt Reactor Experiment	Core heat removal	16-72 all-welded con. VERT	Fuel salt	Coolant salt	1.483×10^6	2.67	1225	1175	55	90
Power Burst Facility	Water-to-water heat transfer	33-310 CGN-HORIZ	H_2O	H_2O	3.50×10^6	15.6	65.2		<150	<150
Piqua Nuclear Power Facility	Steam superheater	40-156 CJU-VERT	Polyphenyl organic	Steam	5.5×10^6	26.8	575	571	105	440

[a] Actual operating pressure

[b] Design pressure

Table 3. Construction Data on Primary Heat Exchangers Reviewed

Reactor Plant	Tube Material	Number of Tubes	Tube OD and BWG (in.)	Tube Length (in.)	Tube Pitch[a] (in.)	Shell ID (in.)	Type Baffle	Baffle-to-Tube Hole Clearance (in.)	Baffle Spacing Maximum (in.)	Baffle Spacing Minimum (in.)	Baffle Thickness (in.)
Advanced Test Reactor	Inconel B-163-61T	1600 U	3/4 - 18	331	1 1/32	68	Split-cross flow segmental	0.010 to 0.020	30	30	5/8
Experimental Breeder Reactor No. 2		302	5/8 - 18		13/16	50 1/4					
Elk River Reactor											
Subcooler	304 SS	70 U	3/4 - 16	158	15/16	16	Split-flow vert-cut seg.	1/32	12 1/2	12 1/2	1/4 & 1/2
Storage	304 SS	154	1/2 - 20	216	11/16	12	Horiz-cut segmental	1/32	11 3/16	11 1/4	1/8
Regenerative cooler	304 SS	31 U	1/4 - 20	38	7/16	3	Horiz-cut segmental		1 7/8	1 1/4	1/16
Non-regen. cooler	304 SS	51 U	3/8 - 20	90	17/32	7.6	Horiz-cut segmental		6 7/8	3 1/4	1/8
Decay heat cooler	304 SS	218 U	1/2 - 18	222	21/32	16	Horiz-cut segmental		16	15 1/4	1/4
Engineering Test Reactor	A213-304 SS	1700	5/8 - 18	240	25/32	37	Split-cross flow seg.	0.032	19 9/16	19 1/2	3/8
Fast Reactor Core Test Facility	A213-304 SS	134U	1 - 14	350	1 3/8	18	Cross flow segmental	0.032	17	17	1/2
High Flux Beam Reactor	304 SS	1350 U	3/4 - 16		15/16	56	Double-cut segmental	0.0156	29 1/8	29 1/8	5/8
High Flux Isotope Reactor	A249-304 LSS	1190 U	5/8 - 20	337	13/16	48	Split-flow segmental	0.016	25 1/4	25 1/4	5/8
Hallam Nuclear Power Facility	A213-304 SS	1395	5/8 - 22M	156	13/16	34 3/8	Double segmental	0.020	20 5/8	18 5/16	3/8
Heavy Water Components Test Reactor	A210-CS	611 U	3/4 - 12	121 3/4	1 1/8	48	Double segmental	0.016	21 5/8	21 5/8	1/2
Molten Salt Reactor Experiment	Hastelloy-N	159 U[b]	1/2 - 19	72	0.775	16	Horiz-cut segmental	0.031	12	12	1/4
Power Burst Facility	A213-304 SS	468	1 - 16	310	1 3/8	33	Split- cross flow seg.	0.016	28 5/8	8 1/2	3/8
Piqua Nuclear Power Facility	A192-CS	382 U	3/4 - 15	156	1	40	Segmental	0.016	29 3/4	29 3/4	1/2

[a]Tubes have △ orientation in all exchangers except for those in PNPF which have ◇ orientation.

[b]There were 163 tubes in the original design. After flow tests were conducted, 4 tubes were removed.

It may be seen in Table 2 that except for the PNPF and the HWCTR, the heat exchanger designs are generally of the liquid-to-liquid type. The Piqua Nuclear Power Facility heat exchanger is classified as a steam superheater with the steam in the tube side and organic fluid in the shell side, and the Heavy Water Components Test Reactor heat exchanger is a steam generator with feedwater and steam on the shell side. Most of the heat exchangers serve to transfer heat from the reactor-core coolant fluid, which may be sodium, steam, light water, heavy water, or molten salt, that usually flows through the tubing to a secondary coolant on the shell side. The secondary coolant, which varies from case to case, dissipates the heat from the tube side (the reactor-core coolant fluid) to a cooling tower or this heat may be used to generate steam if the reactor is for electric power purposes.

Many different sizes and types of heat exchangers are represented in the survey. In order to describe the sizes and types concisely, a method of designation was adopted from the 1968 edition of the TEMA Standards.[3] For example, the heat exchanger designation for the Engineering Test Reactor is 42-240 CEN (HORIZ), which indicates a fixed tube sheet exchanger with stationary and rear heads integral with the tube sheets, a single-pass 42-in.-ID horizontal shell, and tubes 20 ft long. Since the temperature and pressure in the tubes may affect the natural frequency of vibration, these data are also given in Table 2.

The tube-side and shell-side construction data required to compute natural frequencies of vibration are given in Table 3. The tubes were fabricated from a variety of materials, including carbon steel and Type-304 austentic stainless steel, and they range in outside-diameter size from 1/4 to 1 in. The standards followed in fabricating the heat exchangers generally comply with those of the Tubular Exchangers Manufacturers Association with regard to tube pitch, baffle hole-to-hole clearances, and baffle spacing.

Vibration Analysis

The analytical procedures that are available for predicting the heat transfer capacity and the pressure drops in a heat exchanger are adequately developed. Therefore, the design of a heat exchanger will be initially directed toward determining how these performance criteria are to be met. Once the heat transfer and pressure drop criteria are satisfied, a vibration analysis can be performed. Tubes in any heat exchanger will normally vibrate to some extent, but this usually will not result in early failure unless the amplitudes of vibration become very large. The large amplitudes of vibration resulting from coincidence of the natural frequency and the vortex shedding frequency of a tube may lead to early failure through such mechanisms as tubes impacting on other tubes or on baffle plates. Even if there is no coincidence between natural and vortex shedding frequencies, eventual failure might occur because of cycling stresses at tube vibration amplitudes less than that required to cause the tubes to impact with one another. Impact of a tube against a baffle plate might also lead to failure when a resonant vibration condition does not exist.

The procedure used to evaluate the vibration characteristics of the heat exchangers reviewed was to make the best possible estimate of vortex shedding frequencies and natural frequencies of the tubes by using well-known techniques and information available in the literature. The steps involved in the vibration analysis are outlined in the following paragraphs, and the details of some of the analytical relationships that were employed are discussed in Chapter 5. Of the ten possible steps in a vibration analysis that are outlined here, only Steps 1 through 5 were normally considered in the calculations made for this survey. Because of the uncertainties associated with lift and drag forces (Step 6) and with damping, it is difficult at this time to make meaningful calculations with regard to the parameters listed in Steps 7 through 10.

1. The shell-side cross flow velocities at typical positions in the tube bundle must be estimated. Consideration should be given to shell-side inlet and outlet conditions, sections near baffle edges, and U-bend

regions. Critical regions other than those just listed may arise in any particular case, and a special effort should be made to identify them. Bypass areas that present relatively small resistance to flows should be investigated.

2. The Reynolds numbers corresponding to the regions identified in Step 1 should be calculated. Reynolds numbers used in heat transfer and pressure drop calculations are sometimes based on characteristic dimensions or areas other than the tube diameter. Calculating a Reynolds number based on tube diameter will permit direct usage, at least in a qualitative manner, of tabulated and graphical data on Strouhal numbers and lift and drag coefficients.

3. The Von Karman vortex shedding frequencies at the locations being analyzed must be determined. Because of a lack of data, the assumption must be made that a single shedding frequency exists. This is conservative but is not likely to be true unless a resonant condition exists between the vortex and natural frequencies of the tube.

4. Calculate the natural frequency of vibration for tubes at the locations chosen for analysis. The lowest natural frequency can readily be predicted by assuming simply supported end conditions, but this will probably yield results that are too conservative. This problem is treated in detail later.

5. Compare the vortex and natural frequencies of the tubes. It is desirable that these two frequencies be as widely separated as the performance criteria will permit. A separation factor of 3 should insure freedom from destructive vibrations.

6. Calculate the lift and drag forces resulting from shell-side flow. These forces must be known in order to calculate the amplitude of vibration from which stresses can be deduced. There is considerable uncertainty associated with these forces. The remainder of the outline, which covers items that one would like to determine, requires a knowledge of these forces and is therefore inherently limited by the accuracy to which these forces can be determined.

7. Calculate the deflection that will occur as a result of the drag and lift forces, assuming they are static loads. It is assumed that the

drag force is negligible when compared with the lift force so that only the deflection in the direction of the lift force is considered. The accuracy of this assumption is not well established for conditions such as those that exist in tube bundles. Assuming that the forces are uniformly distributed along a span appears reasonable.

8. Estimate the vibration amplitude magnification factor. This requires a knowledge of the forcing frequency, the natural frequency of the component, and the damping in the system. Sometimes a resonant state and viscous damping are assumed so that only a knowledge of the damping is required to find the magnification factor. Assuming that a resonant condition exists is generally a conservative practice, unless of course resonance does exist, but this practice can become excessively conservative because the rate of change of the magnification factor function is high near resonance and the function also peaks near resonance. There are indications that damping in water is negligible. If the shell-side fluid is water, the resonant condition must be avoided.

9. Determine the true amplitude of vibration. Multiplying the amplitude obtained in Step 7 by the magnification factor arrived at in Step 8 gives the amplitude of vibration.

10. Determine the stress induced in the tube as a result of the deflection determined in Step 9. Once the stress caused by vibration is determined and combined with other stresses that may exist, the usual considerations, such as comparing the stress with the endurance limits, can be made.

Some of the values obtained in the analytical evaluations of tube vibration characteristics are given in Table 4, in which the frequencies of vortex shedding are compared with the natural frequencies of the tubes in the heat exchangers analyzed. The vortex shedding frequencies are based on average velocities, and the frequencies of the tubes are based on information reported by Sebald and Nobles[9] and by MacDuff and Felgar.[10] Where the information is available, the mode and location of the failure and the results of independent analyses are given. Experience has indicated that where the vortex shedding frequencies computed using average

velocities correspond closely with the natural frequencies of the tubes, tube vibration problems can be expected. The results of analyses of the several heat exchangers given in Table 4 are in agreement with this observation.

Table 4. Tube Vibration Data for Heat Exchangers Evaluated Analytically

Reactor Plant	V_x [a] (fps)	D_o & BWG (in.)	N_R	N_S	f_s (cps)	L (in.)	f_b [b] "X" (cps)	"Y" (cps)	"Z" (cps)	"M & F" (cps)	Remarks
ATR original	6.05 av	3/4 - 18	1.13×10^5	0.25	24.2	60	41.5	28.6	18.3	19.5	Tube failures attributed to rupture from internal pressure because of thinning of tube walls by abrasion from vibration-caused wear. Typical test frequencies indicated vibrations between 18 and 22 cps. Revised design increased number of cross baffles and stiffened tubes by reducing span from about 60 to 38 in. Tests on revised design indicate f_n at 70 and 80 cps or between calculated frequencies of 71.5 and 103 cps.
ATR revised	3.25 av	3/4 - 18	6.13×10^4	0.19	9.9	38	103	71.5	45.7	45.7	
ERR											
Subcooler	2.2	3/4 - 16	2.5×10^5	0.2	7	25 18 UB				90 300 UB	No vibration problems reported and there appears to be little likelihood that any vibration-related difficulties.
Storage well cooler	0.7 IV 10.9 OV	1/2 - 20	4.3×10^3	0.2	3.3 52.3	22.5 12	148	102		65.2 260	
Regenerative cooler	2.3 IV 4.5 OV	1/4 - 20	$10^4 \rightarrow 10^5$ 2×10^5	0.2	22 43	2.5 1 7/8 23 UB				1847 3280 903 UB	
Non-Regen. cooler	3.1 5.5 OV	3/8 - 20	1.3×10^4 6.9×10^4	0.2	20 35	3 1/4 6 1/2				228 57	
Decay heat cooler	2.7 7.4 OV	1/2 - 18 4.1×10^4	1.3×10^4 4.1×10^4	0.2	12.8 35.5	30.5 16				41.9 153	
ETR	7.4 av 9.7 max	5/8 - 18	8.5×10^4 1.12×10^5	0.17	24.2 31.6	39	71	49	31.3	33	Tube failures attributed to metal fatigue adjacent to tube sheet; also evidence of heavy wear at baffle plate holes. The exciting frequency of 31.6 cps at V_x = 9.7 ft/sec is very near a calculated f_n = 31.3 cps so that resonance could be induced. No measured frequencies were reported.
FRCTF	4.68 av 5.61 max	1 - 14	1.19×10^5 1.43×10^5	0.22	12.4 14.8	34	143	98.5	63.2	74.5	Vibration testing of the exchangers with water instead of sodium indicated that they would be satisfactory. The natural frequency of the tubes is many times that of the exciting frequency.
HFBR original	4.65 4.65 UB[c] 6.8 IV 6.8 OV	3/4 - 16 3/4 - 16	4.1×10^4 4.1×10^4 6×10^4 6×10^4	0.2 0.2	15 15 21.8 21.8	29.125 58.25 116 UB	172 43.6 9.6 OP	119 30		68.3 19.2	Indications of excessive vibration occurred early in the plant operation. Frequencies of vibration in typical straight tube length were about 140, and in the U-bends they were between 30 to 40 and 5 to 15. Use of data published by Sebald and Nobles (Ref. 9) leads to the prediction of frequencies in agreement with the measured values. No tube failures occurred. Significant changes in the flow pattern eliminated the vibration problems. Calculated f_n values listed in this table for HFBR assume no fluid in the tubes.
HFBR revised	2.33 UB[c] 3.4 IV 3.4 OV		2.05×10^4 3×10^4 3×10^4		7.5 10.9 10.9	Calculated natural frequencies unchanged					

Table 4 (continued)

Reactor Plant	V_x[a] (fps)	D_o & BWG (in.)	N_R	N_S	f_s (cps)	L (in.)	f_b[b] "X" (cps)	"Y" (cps)	"Z" (cps)	"M & F" (cps)	Remarks
HFIR	4.12 av 1.5 to 9.97 UB	5/8 - 20	3×10^4 1.1 to 7.2 $\times 10^4$	0.2	15.8	51 3/4 52 UB	39 20.5 OP 96.5 IP	27	17.2	18.5	One tube in each of three separate heat exchangers has failed since startup in 1965. All failures were attributed to vibrational stresses enhanced by defects, and they occurred in the same relative position near the outlet nozzle in each unit. The f_n was measured at 22 cps on Oct. 2, 1968, and the max. amplitude was 0.100 in. peak-to-peak at a flow rate of 4200 to 4300 gpm.
HNPF	6.87 av 10.6 max	5/8 - 22	2.05×10^5 3.11×10^5	0.21	27.7 42.8	36 5/8 38 15/16	69.4 61.2	47.8 42.2	30.6 27	35	Failure attributed to wear at baffle plates. This analysis suggests that a resonance condition can exist. The f_n was measured at 35 to 37 cps at 80 to 90% of full flow.
HWCTR	0.206 FW 10.9 ST	3/4- 12	2.79×10^3 5.5×10^4	0.23	0.76 FW 40 ST	43 1/4	72	49.4	31.7	32.5	No evidence of destructive vibration was detected. This analysis suggests that a possibility, but by no means a certainty, of exciting frequencies at resonance tube frequency assuming a steam flow of 95,000 lb/hr.
MSRE original	5.3 MS 6.1 BE 19.2 IV & OV	1/2 - 19	1.3×10^4 1.5×10^4 4.7×10^4	0.25	30.8 36.6 115	12 24 43 UB	369 92.2 27 IP 23 OP	258 64.3	166 41.4 2.7 IP	214 55	Pre-operational test revealed excessive vibration and pressure drop. The U-tubes vibrated at a frequency between 5 and 10 cps at amplitudes as high as 1/4 in. Vibration of the straight tube segments was less significant but widespread. Extensive lacing of wires through the tube bundle, an enlarged outlet, and an impingement baffle eliminated the problem. The shell-side inlet is in the U-bend region. Frequencies shown assume that tube and shell-side fluids are salts, but tests were made with water. Tabulated frequencies would be about 22% higher if water is the tube and shell-side fluid.
MSRE revised	5.3 MS 6.1 BE 19.2 IV[d] 10.4 OV	1/2 - 19	1.3×10^4 1.5×10^4 2.6×10^4	0.25 0.25	30.8 36.6 0.62	12 24 d	369 92.2 d	258 64.3	166 41.4	214 55	
PBF	4.50 av ~7.0 max	1 - 16	7.43×10^4 1.11×10^5	0.25	13.5 20.2	57.38	50	34.5	22.1	25	A tube f_n of 22 cps is very near a possible f_s of 20.2 cps, and resonance could be induced. The design, based on original submittal dated 11/3/67, is marginal.
PNPF	5.78 av 8.14 max	3/4 - 15	6.70×10^4 9.45×10^4	0.5-0.21	46-19.5 65-27.4	59.5 92 UB	40 16.7	27.5 11.5	17.7 7.4	27	After operation, tubing leaks developed that were attributed to steam-side corrosion. This analysis shows that resonance could exist, but neither of two tests made on superheater indicated any significant resonant frequency problems.

[a]The velocities given were taken at different places in the exchangers, and some of the designations are as follows. BE = baffle edge; FW = feedwater; ST = steam; IV = inlet velocity at nozzle; OV = outlet velocity at nozzle; IP = in-plane frequency; OP = out-of-plane frequency; MS = mid-section; and UB = U-bend.

[b]Methods of calculation are discussed in Chapter 5; "X", "Y", and "Z" represent fixed, fixed/free, and free end conditions, respectively. The "M & F" designation represents values based on data published by MacDuff and Felgar (Ref. 10).

[c]Fluid velocity in U-bend region is probably low, but very low U-bend natural frequencies offset the result of low fluid velocities.

[d]Wire lacing between tubes at each baffle tends to fix ends at straight tube segments. Wire lacing at midpoint of U-bend increases the natural frequencies of the U-tubes significantly. The shell-side outlet is adjacent to the straight tube segments.

5. ANALYTICAL RELATIONSHIPS

Completion of the first five steps of the vibration analysis procedure outlined in the preceding chapter involves the determination of the cross flow velocity of the shell-side fluid, the Reynolds number, the Von Karman vortex shedding frequency, and the natural frequency of tube vibration, and a comparison of the last two values obtained. Other quantities that may have to be considered include the lift and drag forces, vibration amplitude magnification factor, and the stresses in the walls of the tubes. The analytical relationships that were useful in determining these quantities for this vibration analysis and that are of general use in making similar analyses are discussed in this chapter.

Cross Flow Velocity of Shell-Side Fluid

To make a meaningful vibration analysis on a heat exchanger, the shell-side fluid velocity must be known. The interstitial flow velocities on the shell side continuously vary across the tube bundle because the width of the shell and the number of tubes will vary from point to point. Further, the flow distribution along the length of a tube span will undoubtedly be nonuniform. In light of these complexities and others such as those created by leakage flows, it is unlikely that a precise determination of the shell-side flow velocities can be made at this time. However, what appear to be reasonable results can be obtained by using average velocities.

The number of calculations that must be made can be reduced and simplified if use is made of the relationships employed in the heat transfer and pressure drop computations. For instance, Tinker[11] lists relationships for the determination of the average shell-side velocity and recommends their use in rating commercial heat exchangers of the shell and tube variety.

The basic relationship recommended[11] for determining the average shell-side velocity is

$$V_x = \frac{W}{\rho A_x (3.6 \times 10^3)} , \tag{1}$$

where

V_x = average shell-side velocity, ft/sec,

W = mass flow rate through the heat exchanger shell, lb/hr,

ρ = density of shell-side fluid, lb/ft^3,

A_x = average cross flow area within the tube bundle between baffle windows, ft^2.

The relationships recommended by Tinker[11] for determining the average cross flow area at the center line of a circular cylindrical unit are as follows. For tube orientations that are

$$\text{triangular,} \qquad A_x = \frac{1.0C_w}{144}\left(\frac{P - D_o}{P}\right) D_B L ,$$

$$\text{square,} \qquad A_x = \frac{1.0C_w}{144}\left(\frac{P - D_o}{P}\right) D_B L , \tag{2}$$

$$\text{rotated square,} \qquad A_x = \frac{1.414C_w}{144}\left(\frac{P - D_o}{P}\right) D_B L ,$$

where

C_w = that portion of the tube-bundle diameter representing the mean bundle width between baffle windows (Tinker suggests a value of 0.97),

P = tube pitch, in.,

D_o = outside diameter of heat exchanger tubes, in.,

D_B = outer diameter limit of tube bundles, in.,

L = length of tube span between baffle plates, in.

It should be noted that the area between the tube bundle and the shell is neglected in the above equations for A_x and only 97% of the true area between the tubes is used. As previously noted, the equations for A_x as formulated are only good at the midplane (center line of the shell) of the tube bundle. To use these equations at any other plane, the value of D_B should be the chord distance across the bundle at the plane of interest.

Use of the given equations for calculating A_x should yield satisfactory results for large units where the bypass area between the tube bundle and the shell is negligible. For units with small diameters, these equations will tend to be more conservative since an appreciable area available for fluid flow is being neglected. When bypass areas through or around the tube bundle are likely to result in high velocities in the bypass region, the tubes in the vicinity of the bypass region may be of primary interest. There is little available information on methods of predicting bypass flow velocities, but recent work on stream analysis techniques[12] may be helpful.

Reynolds Number

The fluctuating lift and drag force coefficients and the value of the Strouhal number for the flow of fluid transverse to cylindrical tubes are generally given in the literature as a function of the Reynolds number, where the Reynolds number used in the evaluation is as follows.

$$N_R = \frac{(3.6 \times 10^3) D_o V \rho}{\mu} , \qquad (3)$$

where

D_o = outside diameter of heat exchanger tubes, ft,

V = velocity of fluid, ft/sec,

ρ = density of fluid, lb/ft^3,

μ = viscosity of fluid, lb/hr·ft.

Most of the data available on lift and drag force coefficients and Strouhal numbers are for single rigid cylinders where V in Eq. 3 is the mean stream velocity. For the purposes considered here, V should be the V_x calculated in Eq. 1 unless a more accurate value of local velocity is available.

Von Karman Vortex Shedding Frequency

Vortex shedding is a primary cause of tube vibrations perpendicular to the flow velocity. The frequency at which vortices are shed from an un-yawed stationary circular cylinder because of the cross flow of fluid is related in non-dimensional form by the Strouhal number

$$N_S = \frac{f_s D_o}{V} , \tag{4}$$

where

f_s = vortex shedding frequency, cps,

D_o = outside diameter of tube, ft,

V = velocity of transverse fluid flow, ft/sec.

If the fluid approaches a tube at an angle other than perpendicular, the value of V in Eq. 4 can be assumed to be that of the component of velocity perpendicular to the tube.[13,14] However, it is doubtful that there is enough information on flow patterns in heat exchangers to justify estimates of perpendicular and parallel components of flow.

It must be emphasized that Eq. 4 applies to stationary cylinders. There are at least four ranges of Reynolds numbers in which different vortex phenomena occur, and it is therefore important to determine the range of the Reynolds number in a specific heat exchanger. For the range $1 < N_R < 90$, vortices are formed in a symmetrical pattern but they do not break away and no oscillating lift or drag forces are produced. In the range $300 < N_R < 2 \times 10^5$, the vortices are formed and released from alternate sides of the cylinder, thus forming a vortex street downstream and producing periodic oscillating forces on the cylinder. For $2 \times 10^5 < N_R$ to over 10^6, the separating boundary layer is turbulent, no regular vortex wake is distinguishable, and the oscillating forces are random with no single dominant frequency. In this range, there is a sudden decrease in the drag coefficient. For Reynolds numbers from approximately 10^6 to 10^7, the drag coefficient increases again and there are indications that the wake is again organized.

For the range of Reynolds numbers from approximately 300 to 2×10^5, which is the range in which most commercial heat exchangers operate, the

Strouhal number with reference to a single non-oscillating cylinder is essentially constant at a value of $N_S \simeq 0.2$. The few studies of vortex shedding frequencies in tube banks that have been made indicate that the surrounding tubes will have considerable influence on the value of the Strouhal number. Correlations have been presented by Chen,[15] who plotted values of the Strouhal number versus the transverse spacing ratio $x_t = T/D_o$ for a range of parametric values of the longitudinal spacing ratio $x_e = L/D_o$ for both in-line tube banks and staggered tube banks by using data from several different investigations. The transverse spacing ratio, x_t, and the longitudinal spacing ratio, x_e, are illustrated in Fig. 1. For some practical combinations of these ratios, the value of the Strouhal number can be of the order of 0.5.

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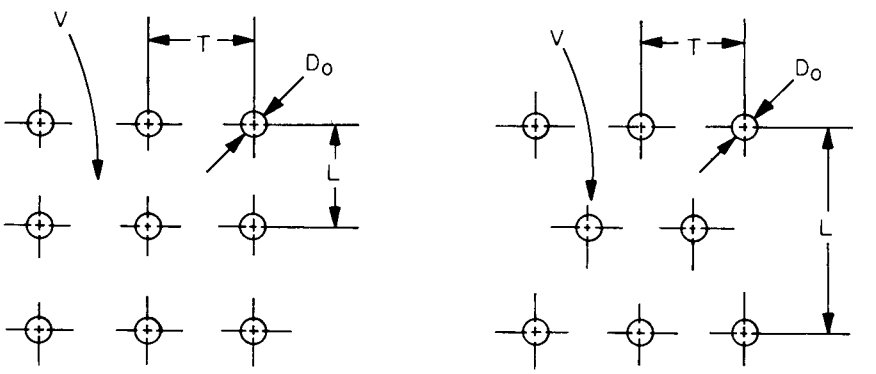

$x_e = L/D_o$

$x_t = T/D_o$

Fig. 1. Tube Pattern Ratios Included in Chen's Correlation.[15]

Although his data are for gas flow and have not been verified for liquids, Chen's correlations were used to evaluate the primary heat exchangers reviewed in this survey. The values used for V in Eq. 4 were the values of V_x obtained from Eq. 1. In all cases except one, the value of N_S obtained by using Chen's correlations was approximately 0.2. The one exception was for the unit in the PNPF, and the value of N_S indicated by Chen's correlations was 0.5.

As stated previously, it does not seem likely that all the vortices being shed from a particular tube will be shed at the same frequency or with the same resulting force on the tube unless a condition of resonance exists between the vortex shedding frequency and the natural frequency of the tube. However, if the best estimate of the average velocity along the tube span is used in Eq. 4, it seems reasonable that this will yield the best estimate of the average vortex shedding frequency and possibly the predominate frequency if one exists.

Natural Frequency of Tube Vibration

A tube in a heat exchanger can be visualized as a continuous beam on multiple supports, the supports being the baffle plates. When calculating the natural frequencies of these tubes by rigorous methods, the section between each pair of supports would be considered as a separate beam with its origin at one of the supports of the section; and the equations of motion from Newton's second law would be applied to each section with appropriate boundary conditions. Procedures are given in the technical literature[16] for solving the resulting set of complex equations, which are very cumbersome unless solved with the aid of a computer.

The application of such rigorous methods to determine the natural frequency of tube vibration in the heat exchangers reviewed was not considered necessary for the purposes of this survey. Considering the unknowns which result from the character of the supports and the changes in tension or compression on the elements that are produced by constructional and operational practices, it is doubtful that one can do better than bracket the potential frequencies by using currently accepted methods. The methods described by Sebald and Nobles[9] and by MacDuff and Felgar[10] were therefore used to evaluate the data reviewed in this survey.

Application of Data Presented by Sebald and Nobles

In their paper on tube vibration,[9] Sebald and Nobles present data on the response to vibration excitation of a single 30-ft length of 3/4-in.-OD tubing divided into 8, 12, or 16 sections by 7/8-in.-thick supports representing baffle plates. The effects of tube hole clearances, mechanically transmitted vibrations, support plate spacing, and of longitudinal stress on the tube vibration characteristics are discussed.

The fundamental (first mode) frequency of bending vibration of a segment of a heat exchanger tube is calculated from the following equation.

$$f_n = C\left[\frac{EIg}{W_E L^4}\right]^{1/2} , \qquad (5)$$

where

f_n = natural frequency of vibration, cps,

E = modulus of elasticity of tube material, psi,

I = moment of inertia, in.^{4},

g = gravitational constant = 386 in./sec^2,

W_E = effective weight of tube per unit length, lb/in.,

L = length of tube span, in.,

C = constant whose values for the three conditions of end support assumed are tabulated below.

Method of Support	Designation	Value of C
Both ends fixed	"X"	3.56
Fixed/simply supported	"Y"	2.45
Both ends simply supported	"Z"	1.57

In Eq. 5, it is shown that the frequency is an inverse function of the square root of the mass. The mass used to compute the natural frequency of vibration of the heat exchanger tube must include the weight of fluid within the tube as well as a portion of the fluid external to the tube that is forced to move as the tube vibrates.

The portion of fluid external to the tube that is accelerated with the tube is defined as added mass and accounts for the apparent increased force to acquire acceleration. In his study of forced vibration, Keane[17]

has shown that added mass is proportional to the volume of fluid displaced by the cylinder and to the mass density of the fluid so that the added mass

$$m_a = k\rho_f v_b \ , \tag{6}$$

where

k = added mass constant,

ρ_f = mass density of fluid, $lb \cdot hr^2/ft$ per ft^3,

v_b = volume of body, ft^3.

The value of k is determined by the geometrical shape of the body. In a study of acceleration of bodies in fluids, Stelson[18] measured the value of k for several bodies and recommends a value of k = 1.00 for cylinders, which value was also confirmed by Keane.[17]

If the length of the tube is much greater than the radius, the value for added mass obtained by using Eq. 6 is valid over most of the tube length and the value of W_E in Eq. 5 becomes

$$W_E = W_t + W_c + W_a \ , \tag{7}$$

where

W_t = weight of tube material per unit length, lb/in.,

W_c = weight of fluid inside tube per unit length, lb/in.,

W_a = weight of added fluid external to tube per unit length, lb/in.

Effect of Tube Hole Clearance. In the test conducted by Sebald and Nobles,[9] the tube-to-baffle diametral clearance ranged from near zero to 0.030 in. At zero clearance where the tube was tightly clamped in the supports, the method of support corresponded to condition "X" with fixed ends. The measured frequency of 44.5 cps at peak amplitude compared very closely with the calculated frequency of 43 cps. The tube hole clearances were then varied systematically from 0 to 0.030 in. and the frequencies were measured. These measured frequencies are compared with the calculated frequencies in Table 5.

The values given in Table 5 are for a tube length of 49 3/8 in. and correspond to a section of tubing adjacent to a tube sheet so that one end is fixed and the other end is supported by a baffle. It should be noted that as the diametral clearance is increased, the measured frequency decreases and corresponds with frequencies between those calculated for

Table 5. Effect of Tube Hole Clearance on Natural Frequency of Tube Vibration[9]

Clearance (in.)	"X" (cps)	"Y" (cps)	"X" (cps)	Measured f_n (cps)	Frequency Range[a] (cps)	Nonresonant Amplitude Range (in.)
0	43	29	19	44.5	2.5	0.01 to 0.02
0.010	43	29	19	38	17.0	0.03 to 0.06
0.020	43	29	19	33.5	18.0	0.06 to 0.08
0.030	43	29	19	28	17.0	0.08 to 0.13

[a]Frequency ranges (about peak) that have significant amplitudes.

conditions "X" and "Y" for diametral clearances from 0 to about 0.020 in. At a diametral clearance of 0.030 in., the measured frequency is between the calculated frequencies at conditions "Y" and "Z". The band of frequencies with significant amplitudes increased markedly at a diametral clearance of 0.010 in. and then it appears to have reached an upper limit.

The simulated baffle-plate thickness of 7/8 in. was held constant during the test to determine the effect of the tube-to-baffle clearance. Since the diameter of the tube, 3/4 in., was also held constant, the question of the extent of the effects of baffle thickness on the natural frequency of the tube arises. When using the data of Sebald and Nobles[9] to estimate the effect of tube-to-baffle clearance on the natural frequency of tubes, the problem should be approached with a degree of caution if the baffle thickness being considered is much less than the diameter of the tube. On the other hand, their test results do suggest that the use of thick baffles might be quite worthwhile. However, the same results might be obtained more economically by spacing two 1/4-in.-thick baffles on 3/4-in. centers to effect the result of a 1-in.-thick baffle.

The major contribution made by the results of these tests on the effects of tube-to-baffle clearance on the natural frequency of tube vibration is to provide information that may be used to reduce the uncertainty associated with calculated values of natural frequencies. Further, the results obtained by Sebald and Nobles suggest that additional tests of a straightforward nature could significantly improve the current capabilities in this area.

Mechanically Transmitted Vibrations. Concern is frequently voiced over the possibility of exciting heat exchanger components by mechanically transmitting vibrations from other equipment. Although this possibility cannot be ruled out in every case, it appears reasonable to assume that a properly functioning piece of equipment will not normally transmit vibrations at energy levels sufficient to cause damaging vibrations. As an example, Sebald and Nobles[9] studied the effect of forces transmitted mechanically through their tube supporting structures. They concluded that mechanically transmitted vibrations at energy levels comparable with those of a steam turbine in dynamic balance do not produce vibration amplitudes of sufficient magnitude to cause damage. As previously stated, no general conclusion can be made with regard to mechanically transmitted vibrations because this source of component excitation will vary from one installation to another and even within a particular installation as the operating cycle varies.

Effect of Support Plate Spacing. Two conditions of symmetrical versus asymmetrical support plate spacing were compared by Sebald and Nobles.[9] In making the comparison, the calculated natural frequencies were compared with measured values for an intermediate span length. The frequencies at peak amplitude fell about midway between the frequencies corresponding to support conditions "Y" and "Z". Their data do not support the general opinion that asymmetrical spacing of supports will produce significant vibration damping.

Effect of Longitudinal Stress. Stresses of 2000 psi in tension and compression were alternately applied to the tube.[9] Measurements of vibration frequencies during application of the stress indicate that the natural frequency of a tube is increased with longitudinal tensile stress and decreased with compressive stress.

In his book on vibrations, Timoshenko[19] states that the effect of an axial force on the lateral vibration of a prismatic bar can be obtained by using the following equation.

$$f_{np} = f_n\left[1 \pm \frac{SL^2}{EI\pi^2}\right]^{1/2} , \qquad (8)$$

where

f_{np} = natural frequency under tension or compression (the value $SL^2/EI\pi^2$ is positive in tension and negative in compression),

f_n = natural frequency of unstressed tube, cps,

S = axial compressive (negative) or tensile (positive) force, lb,

L = tube span, in.,

E = modulus of elasticity, psi,

I = moment of inertia, in.^{4}

Application of Data Presented by MacDuff and Felgar

The paper published by MacDuff and Felgar[10] is a compilation of information formulated in a manner which simplifies the determination of natural frequencies for idealized configurations. The general solutions for finding the natural frequencies of beams on multiple supports with various end conditions are presented in the form of charts, tables, and nomograms. To use these data, one simply selects a frequency constant from the appropriate table. This constant is defined as follows.

$$c = \frac{f_n L^2}{r} , \tag{9}$$

where

c = frequency constant,

f_n = natural frequency, cps,

L = beam or span length, in.,

r = radius of gyration, in.

The tabulated values of c are based on a characteristic density of 490 lb/ft^3 and a Young's modulus of 30×10^6 for steel. When other materials are used, a correction factor is applied to account for the difference in material properties. This correction factor is defined as follows.

$$k_m = \left[\left(\frac{E}{\rho}\right)\frac{\rho_s}{E_s}\right]^{1/2} , \tag{10}$$

where

E = Young's modulus of elasticity for the material, psi,

ρ = density of the material, lb/ft^3,

E_s = Young's modulus for steel = 30×10^6 psi,

ρ_s = density of steel = 490 lb/ft^3.

Using the tabulated data of MacDuff and Felgar, the general equation for the natural frequency is given in Eq. 11

$$f_n = \frac{crk_m}{L^2} . \tag{11}$$

Equation 11 can be put into a form that is more frequently used by combining it with Eq. 10 and substituting the values of E_s and ρ_s into the resulting expression. The equation for the natural frequency then becomes

$$f_n = (4.944 \times 10^{-6})c\left[\frac{EIg}{W_E L^4}\right]^{1/2} . \tag{12}$$

The data tabulated by MacDuff and Felgar[10] are based on the assumption that ideal support conditions exist. The ends of heat exchanger tubes approach the ideal "fixed" condition at a tube sheet, but as shown by Sebald and Nobles,[9] the condition of support at baffles is at least dependent upon the tube-to-baffle clearance. Therefore, if a tube in a heat exchanger is treated as a continuous beam, the logical assumption that a baffle acts like an idealized "hinged" support will normally be made. The subsequent calculation will then yield a value for the natural frequency that will be somewhat less than that predicted when the test results of Sebald and Nobles are used. Use of the data tabulated by MacDuff and Felgar will generally yield conservative results since ideal supports are assumed, and these results can be obtained quickly with simple calculations. The degree of conservatism depends upon the baffle thickness, tube-to-baffle clearance, and the restraints on a tube that can result from imperfect alignment of the holes in successive baffles. For units that must be designed to meet stringent performance criteria, the conservatism resulting from the use of data tabulated by MacDuff and Felgar may be excessive. That is, shell-side fluid velocities less than

those desired may be indicated as a result of a natural frequency of tube vibration that is less than the true frequency of the tube.

U-Bends

So far, nothing has been said about the natural frequency of U-bend segments of heat exchanger tubes or the effects of these segments on the natural frequency of the straight segments. The U-bend region of a tube bundle is highly susceptible to flow induced vibrations, as has been demonstrated by experience. A precise treatment of the U-bend segments of a heat exchanger tube is more difficult than is such treatment of straight segments, but some guidelines can be formulated.

Generally speaking, the U-bend region of a U-tube heat exchanger will consist of U-bends with a large range of bend radii. The bends with the smallest radii will generally be quite stiff as compared with the bends with the largest radii and with the straight segments of the tube bundle. Thus, the small-radii bends generally can be disregarded in themselves, but they do act as short rigid links that couple the two straight segments of the U-tube. On the other hand, the U-tubes with the largest radii can become special problems, and if they are flexible enough, they tend to uncouple the two straight segments of the U-tube.

Determining the natural frequency of the U-bend elements is reasonably tedious and complicated by the usual uncertainties associated with support conditions. To further complicate matters, the actual curved segment may be supported on one or both ends by a straight segment that extends from the last baffle preceding the U-bend region. The presence of such straight segments can make it difficult to determine whether the U-bend will have its first mode of vibration in the plane of the bend or perpendicular to the plane of the bend.

In most situations, it would probably be wise to avoid vibration problems in the U-bend region simply by designing the unit so that there is no appreciable flow in the U-bend region. Since this may not be possible in all cases, some indication of the natural frequencies to be expected is needed.

The natural frequency of clamped ring segments is discussed in a paper by Ojalvo and Newman.[20] The following equation for a 180° arc with fixed ends may be deduced from this paper.

$$f_{ni} = \frac{4.5}{R^2}\left(\frac{1}{2\pi}\right)\left(\frac{EIg}{W_E}\right)^{1/2} , \tag{13}$$

where

f_{ni} = lowest natural frequency in the plane of the ring, cps,

R = mean radius of the ring, in.,

and where the ring material is assumed to have a Poisson Ratio of 0.3. Equation 13 can be written in terms of the arc length as

$$f_{ni} = 2.019\ \frac{22}{2\pi}\left(\frac{EI}{W_E \ell_a^4}\right)^{1/2} \tag{14}$$

$$= 2.019 f_n ,$$

where ℓ_a = arc length = πR in. Thus, a 180°-arc ring with fixed ends will have a natural frequency in the plane of the ring that is approximately twice that of a straight member of equal length.

For the same ring, the relationships of Ojalvo and Newman[20] give the following equation for the lowest natural frequency of the ring perpendicular to the plane of the ring.

$$f_{no} = 0.866 f_n . \tag{15}$$

Combining Eqs. 14 and 15,

$$f_{ni}/f_{no} = 2.33 . \tag{16}$$

From Eq. 16, it may be seen that the U-bend segments will tend to vibrate in an out-of-plane mode. Equation 15 gives some indication of why the large-radii U-bends are vulnerable to flow induced vibrations. It also suggests that an entire U-tube might be adequately analyzed as a continuous beam, treating the U-bend segment as a straight member whose length is approximately $0.93\ell_a$.

When analyzing the U-bend segments of a tube bundle, it is necessary to deal with estimates of fluid velocities in which there are large uncertainties. These uncertainties about fluid flow will normally overshadow the approximate values for natural frequencies of tube vibration obtained

by using the relationships developed above. Unless the main stream of flow is directed away from the U-bend region, a major attempt to stiffen this region will normally be justified. Such stiffening has been done in several units by lacing bars through the U-bend tubing.

Lift and Drag Forces

A dimensional analysis will show that the force exerted by fluid flow transverse to the axis of a stationary cylinder is of the form

$$F = C \frac{\rho A V^2}{2g} . \tag{17}$$

Two components of force are associated with the fluid flow: a lateral or lift force perpendicular to the direction of fluid flow and a drag force parallel to the direction of fluid flow. Using an appropriate value of C, the lift force

$$F_L = C_L A \frac{\rho V^2}{2g} \tag{18}$$

and the drag force

$$F_D = C_D A \frac{\rho V^2}{2g} , \tag{19}$$

where

C_L = coefficient of lateral or lift force,
A = projected area per lineal inch of tube, $ft^2/in.$,
ρ = density of fluid, lb/ft^3,
V = velocity of fluid, ft/sec,
g = gravitational acceleration, ft/sec^2,
C_D = coefficient of drag force.

A suitable value of C_L is not readily determined from the literature. The value used by Ozker and Smith [21] in their work was $C_L = 0.20$; Den Hartog[22] suggests that $C_L = 1.00$; Dockstader, Swiger, and Ireland[23] report experimental and theoretical values of C_L ranging from 0.65 to 1.71; and Steinman suggests $C_L = 1.9C_D$ for low Reynolds numbers. Depending upon the Reynolds number, Chen[15] suggests that the value of C_L for staggered tube banks could be in the order of 0.436. Since this parameter is still

little understood even for single rigid tubes, the heat exchanger designer will be inclined to use a value of 2.00 for C_L. From the preceding discussion, it is apparent that undue conservatism may result from the use of this value of C_L.

Using data obtained from various sources in the literature, Fung[25] plotted the mean drag coefficient and Strouhal number versus Reynolds numbers for a single rigid circular cylinder. A similar plot for the low speed drag of cylinders of various shapes is presented by Delany and Sorensen.[26] These data indicate a constant value of $C_D = 1.00$ for a range of Reynolds numbers from about 300 to approximately 2×10^5.

When the cylinder is oscillating, there is evidence of a different phenomenon of vortex shedding that can apparently result in considerably different coefficients than observed for stationary cylinders. This phenomenon has been studied at some length by Marris,[6] and Den Hartog[22] has also studied the oscillating cylinder. Both seem to concur on the following conclusions.

1. If the Strouhal shedding frequency appropriate to a stationary cylinder is far removed from the natural frequency of the spring-mounted system (vibration heat exchanger tube), very small amplitude vibrations of the forced linear type occur and such vibrations do not appreciably affect the flow regime and the vortex shedding process.[6,22]

2. When linear resonant conditions are approached from either side, the increase in the vibration velocity relative to that of the stream modifies the flow regime to the extent that nonlinear forces are introduced and large-amplitude self-excited vibrations result.[6,22]

3. If the frequency of the cylinder vibration is in the neighborhood of the natural Strouhal frequency, the motion of the cylinder forces the vortex shedding to take place at the vibration frequency of the cylinder and not at the Strouhal frequency.[6,22]

There is some evidence[27] to support the hypothesis that the oscillating motion of the cylinder organizes the vortex shedding along the tube length so that random shedding is locked into synchronism and is no longer of a random nature. It is further assumed that a vortex is shed

at the extremity of the vibration cycle. This implies that the maximum force is exerted at this point and corresponds to a forced nonlinear type of vibration.

The recently published results of a study[28] conducted at the Naval Air Development Center (NADC) in Johnsville, Pa., to investigate the problem of flow induced vibrations in flexible cables towed through water tend to verify the conclusions of Marris[6] and Den Hartog[22] and support the evidence resulting from the study of Macovsky.[27] In the NADC study,[28] it was found that when the amplitude of vibration was less than about 10% of the diameter of the cable, the shedding vortices were not phase correlated along the span of the cable but were shed at random. Under these conditions, large amplitudes do not build up. Further, it was noted that the drag force acting on a vibrating member may be as much as 35% greater than that which acts on a non-vibrating member. When the flow velocity was increased beyond the value where resonance first occurred between vortex shedding and the natural frequency of the cable, the vortex shedding again became random and the resonant condition was eliminated.

The results of the NADC tests[28] appear to shed light on several factors that are pertinent to the heat exchanger vibration problem. With regard to fluid drag forces, the results of the NADC tests suggest that a drag force coefficient of 1.35 rather than 1.00 should be used if one wishes to be sure of a conservative estimate of the effect of drag when a vibrating tube conditions exists. Since drag forces may be as much as 35% greater when the tubes are in a vibrating state, one might assume that the pressure losses associated with tubes might also be increased by that amount. Finally, since vibration amplitudes of as much as 10% of the diameter of a tube may occur before resonance exists between vortex shedding and the natural frequency of the tube, the approach of simply avoiding the resonant condition is not necessarily sufficient to avoid tube failures.

Vibration Amplitude Magnification Factor

It is apparent that the state of vibration in a heat exchanger is not well understood and is a complex result of many factors. Damping is one of these factors that has a strong influence on the amplitude of vibration and a lesser effect on the frequency of vibration. Damping forces in a vibrating tube system are caused by the physical properties of the tube, tube supports, and the fluids involved.

It was previously suggested that to find the true amplitude of vibration, the static deflection that would result from the application of the fluid drag and lift forces should be multiplied by a magnification factor. The nature of this magnification factor can best be studied by examining the equations of motion in an idealized case.

The vibration of a heat exchanger tube is usually categorized as a self-excited oscillation with the energy of excitation being supplied by the fluid flowing past the tube. When a fluid flows by a cylindrical obstacle, it is known that alternating vortices will be formed. Assuming that these vortices will be shed from the cylinder in a regular manner with a frequency related to the Strouhal number, harmonically varying forces on the cylinder in a direction perpendicular to the velocity of the fluid flow will result. Where the drag force is neglected and vortex shedding is assumed to be uniform along the length of a tube span, the intensity of this force

$$F = F_L \sin \omega t \ , \tag{20}$$

where

F_L = lift force on the tube parallel to the direction of flow, lb/in.,

ω = circular frequency of tube vibration, radians/sec,

t = time, sec.

For a first approximation, the vibrating heat exchanger tube can be considered as a single-degree-of-freedom spring-mass system excited by a harmonic force $F_o \sin \omega t$. The differential equation of motion can then be written as

$$m \frac{d^2y}{dt^2} + C \frac{dy}{dt} + Ky = F_o \sin \omega t \ , \tag{21}$$

where

m = mass,

C = damping coefficient,

K = spring constant,

y = amplitude of vibration.

Thomson[29] shows that the steady-state solution of Eq. 21 with viscous damping can be reduced to the form

$$y = \frac{\frac{F_o}{K}}{\left[\left(1 - \frac{m\omega^2}{K}\right)^2 + \left(\frac{C\omega}{K}\right)^2\right]^{1/2}} . \tag{22}$$

Equation 22 can be further reduced to

$$\frac{y}{y_o} = \frac{1}{\left\{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]^2\right\}^{1/2}} , \tag{23}$$

where

y/y_o = the magnification factor

where y = the amplitude of vibration

y_o = zero frequency deflection of the spring-mass system under the action of a steady force F_o,

ω_n = natural frequency of undamped oscillation, radians/sec,

$= (K/m)^{1/2}$,

ζ = damping factor $= C/C_o$

where C_o = critical damping coefficient $= 2m\omega_n$.

The term y/y_o represents the factor by which the zero frequency deflection can be multiplied to determine the amplitude of vibration y. If the magnification factor y/y_o is plotted as a function of the frequency ratio ω/ω_n and the damping factor ζ, a series of response curves will result. Such curves are shown in Figs. 3.1 and 3.2 on page 54 of Ref. 29. When the ratio of the exciting frequency ω to the natural frequency of undamped oscillation ω_n equals unity, the condition of resonance occurs and Eq. 23 reduces to

$$y = \frac{y_o}{2\zeta} = \frac{F_o}{C\omega_n} . \tag{24}$$

Equation 24 can be used to determine the amplitude of vibration of a viscously damped spring-mass system at resonance if the ratio of the viscous damping coefficient to the critical damping coefficient is known.

The derivations of Eqs. 20 through 24 were based on the assumption of a linear forced harmonic vibration of a spring-mass system with viscous damping where the damping force is proportional to the first power of the velocity of motion. Frequently, the damping resistance is more correctly expressed by a term proportional to the square of the velocity,[30] in which case the equation of motion becomes

$$m \frac{d^2y}{dt^2} + a\left|\frac{dy}{dt}\right|\frac{dy}{dt}\right| + Kx = F_o \sin \omega t \ . \tag{25}$$

Exact solutions for forced vibration of nonlinear systems are virtually nonexistent.

However, the resonant amplitude can be estimated with sufficient accuracy by using an equivalent viscous damping factor, C_{eq}, in the previous equations that is evaluated on the basis of harmonic motion and equal energy dissipation. The amplitude of resonance for a damping force that is proportional to the square of the velocity can be determined from the following equation.[29]

$$y = \left(\frac{3\pi F_o}{8a\omega_n^2}\right)^{1/2} , \tag{26}$$

where a is the coefficient of damping in the damping force equation

$$F_D = \pm a\left|\frac{dy}{dt}\right|\frac{dy}{dt} \ . \tag{27}$$

Since $y_o = F_o/K$ and $\omega_n = (K/m)^{1/2}$, the equation for amplitude can also be expressed in terms of initial deflection y_o and effective mass m.

$$y = \left(\frac{3\pi m y_o}{8a}\right)^{1/2} \ . \tag{28}$$

There is very little data on which accurate estimates of magnification factors can be based. Some data indicate that the damping caused by water which surrounds a tube is proportional to the velocity squared, and some data suggest that damping caused by water surrounding a tube is negligible.[31] Further, the amount of damping caused by internal friction in the tube metal and tube supports is not known. Before accurate

estimates of the amplitude of vibrations can be made, a systematic study of the damping that is present will have to be made.

Tube Wall Stresses

There are three factors that significantly affect the degree of accuracy to which tube stress calculations can be made. They are

1. the conditions of support,
2. the nature of the loading resulting from the shell-side flow, and
3. the system damping characteristics.

The effects of internal and external fluid pressure and the temperature differential across the tube wall are reasonably well understood, and the uncertainties associated with the three factors listed above have been discussed.

Each tube span of a heat exchanger tube can be thought of as a beam element that is pinned on both ends, pinned at one end and fixed at the other, or fixed at both ends. The deflection and bending moment equations for a tube span with these support conditions are tabulated below where

y_o = maximum deflection, in.,
F = uniform loading (F_L and/or F_D), lb/in.,
L = unsupported length of tube span (two times baffle spacing), in.,
E = modulus of elasticity, lb/in.^{2},
I = moment of inertia, in.^{4},
M = maximum bending moment, in.-lb, and
k_1, k_2 = factors to account for support conditions.

Pinned-Pinned Support Condition	Pinned-Fixed Support Condition	Fixed-Fixed Support Condition
$y_o = k_1 FL^4/EI$	$y_o = k_1 (FL^4/EI)$	$y_o = k_1 (FL^4/EI)$
$k_1 = 1/384$	$k_1 = 1/185$	$k_1 = 5/384$
$M = k_2 FL^2$	$M = k_2 FL^2$	$M = k_2 FL^2$
$k_2 = 1/12$	$k_2 = 1/8$	$k_2 = 1/8$

The assumption that the loading F is uniform is an expedient made necessary by the lack of detailed information on shell-side flow in a heat exchanger. At this time, the loading F can best be determined by using Eqs. 18 and 19 and by employing the best estimate of the average velocity in the vicinity of the tube being analyzed. Accepting the same assumption for the deflection and bending moment, the bending stress at any point in the tube wall

$$S_b = \frac{Mc_x}{I} = \frac{k_2 FL^2 c_x}{I}$$

$$= \left(\frac{k_2}{k_1}\right)\frac{y_o c_x E}{L^2} \quad , \tag{29}$$

where c_x is the distance in inches from the center line (neutral axis) of the tube to the point in the tube wall being analyzed.

There are three components of stress caused by the external and internal fluid pressures. These are the pressure stress in the circumferential direction, S_{pc}; the pressure stress in the radial direction, S_{pr}; and the pressure stress in the longitudinal direction, $S_{p\ell}$.

$$S_{pc} = \frac{p_1 r_1^2 - p_2 r_2^2 + \left(\frac{r_1 r_2}{r_i}\right)^2 (p_1 - p_2)}{r_2^2 - r_1^2} \quad , \tag{30}$$

$$S_{pr} = \frac{p_1 r_1^2 - p_2 r_2^2 - \left(\frac{r_1 r_2}{r_i}\right)^2 (p_1 - p_2)}{r_2^2 - r_1^2} \quad , \tag{31}$$

$$S_{p\ell} = \frac{p_1 r_1^2 - p_2 r_2^2}{r_2^2 - r_1^2} \quad , \tag{32}$$

where

p_1 = internal pressure, psi,

p_2 = external pressure, psi,

r_1 = inside radius of tube, in.,

r_2 = outside radius of tube, in.,

r_i = radius to the point of investigation, in.

There are two components of thermal stress caused by the temperature differential across the tube wall. They are the thermal stress in the

circumferential direction, S_{tc}, and the thermal stress in the longitudinal direction, $S_{t\ell}$. For thin-wall tubes where the outside temperature is T and the inside temperature is T + ΔT and where both S_{tc} and $S_{t\ell}$ are compressive on the inner surface and tensile on the outer surface of the tube,

$$S_{tc} = S_{t\ell} = \frac{\frac{1}{2}\Delta T\alpha E}{(1 - \upsilon)} , \tag{33}$$

where

ΔT = temperature differential across the tube wall, °F,

α = coefficient of thermal expansion, in./in.·°F,

υ = Poisson's ratio.

If the longitudinal thermal growth of a tube is impeded, the resulting longitudinal stress

$$S_{tF} = (\Delta T\alpha E)_t - (\Delta T\alpha E)_R , \tag{34}$$

where S_{tF} is the uniformly distributed stress caused by relative thermal expansion and where the subscripts t and R refer to tube conditions and support conditions, respectively.

Combining the values given by Eqs. 29 through 34 will give the stresses in the tube wall when the loading F is assumed to be statically applied. Obviously, the deflection and stress that will result from a dynamically applied force F will be somewhat larger than indicated by the above equations, and the extent of this dynamic magnification is a function of the system damping. Since the amount of damping will usually be unknown, at least for the present, and dynamic magnification can be quite large in resonant conditions of vibration, the only path of action that presently can be taken is to assure that the system is well removed from the resonant condition; that is, one must be sure that the amplitudes of vibration are obviously negligible.

There are two factors which affect stress calculations that have not yet been mentioned. They are the tube sheet joints and defects in the tubes caused by manufacturing and fabrication errors. Both of these factors will result in higher stresses than those determined by the methods described in this section, and as has been shown by experience, these

factors coupled with what would otherwise be nominal amplitudes of vibration can lead to tube failures early in the operating life of a heat exchanger.

6. CONCLUSIONS AND RECOMMENDATIONS

A number of designs for heat exchangers have been studied in detail, and the study has provided some insight into the problem of tube vibrations in heat exchangers. Analytical reviews of the units which have not experienced vibration difficulties do not suggest the possibility of problems in the future. The analytical reviews of those units in which problems have been experienced indicate that the difficulties would probably have been suspected had a critical assessment of the designs been made. Some conclusions concerning design procedures and recommendations for an experimental program to obtain more accurate design data are presented in this chapter.

Conclusions

The results of the detailed reviews of the primary-circuit heat exchangers evaluated in this survey show that nine of the Category-1 installations definitely exhibited flow induced vibration problems, one of the Category-2 installations was marginal in this respect, and that two of the heat exchangers designed primarily for superheat and steam generation were damaged by corrosion. In each of the latter two cases, calculations show that tube vibrations could have interacted with corrosion to enhance the mechanism of damage.

In practically all of the cases studied, the designers had apparently relied on adherence to rules in the TEMA Standards[3] to eliminate or minimize problems that might result from vibration. The data reviewed in this study show that such standards alone cannot always be relied on to produce vibration-free designs for heat exchangers. The consequences of a heat exchanger tube failure in a nuclear system clearly indicate that specific attention to the vibration problem is warranted during the design phase and that design procedures more stringent than those normally employed may be necessary.

The results of this evaluation indicate that if the designs for those units with which problems have been experienced had been critically analyzed prior to fabrication, the nearness of the natural frequencies of tube vibration to the calculated Von Karman shedding frequencies would have been justification for design modifications. If the only purpose of the USAEC is to eliminate the types of problems that have been encountered, there would be no need for further experimental refinement of the technology where reasonably conservative engineering techniques are permissible. Most of the previously experienced problems could apparently be avoided by incorporating several requirements into the design criteria. These requirements are as follows.

1. An inlet impingement baffle should be employed.

2. The tube-to-baffle clearances should be as small as possible, considering the possible thermal stress problems and increased assembly difficulties, and in no case should the TEMA Standards[3] be exceeded.

3. Baffle thicknesses approximately equal to the diameter of the tubes should be a goal. This might become impractical for tubes with large diameters, but other methods, such as using two separated plates at each baffle level, of obtaining the same goal can be employed.

4. Where possible, the main stream of the shell-side flow should be bypassed around the U-bend regions. Where this is not possible, the U-bend tubes should be stiffened by tying the outer rows of tubes together.

5. The maximum practical separation factor between vortex shedding frequencies and the natural frequencies of the tubes should be maintained. A separation factor of 3 should account for the uncertainties associated with predicting these frequencies. The factor of separation need not be so great if fluid velocities can be predicted accurately or if comprehensive testing is planned.

It has not been possible in this study to develop any general guidelines with regard to the effects of corrosion and fatigue under vibration conditions, and it is recognized that the requirements listed above do not guarantee absolute freedom from vibration-related failures. Further, it is recognized that some of these requirements, particularly Item 5, can lead to very conservative designs. Some data indicate that the

requirements of Item 5 could be less restrictive, but other data will have to be obtained before a designer can be assured that vibration failures will not occur.

The conditions of the past could be somewhat improved by information gained through an investigation of one of the principal unknowns: the effects of the tube support conditions. Although this could be accomplished easily, the problem in general would not be solved because it is not possible to predict the conditions which lead to resonance between the vortex shedding and the natural frequencies of the tubes. Solution of the latter problem will require a much greater effort which is probably not justified solely on the basis of the past difficulties.

We must therefore conclude from the knowledge we have developed thus far that the interest of the USAEC in destructive vibration in heat exchangers should be directed toward new design applications where conservatism must be held within close boundaries because of special problems in the application. An example of this is the need for high-velocity fluids to assure high heat transfer ratings in heat exchangers where minimum surface is an important consideration in the system application. Close tube spacing might also be important to minimize the inventory of fluids, and this could also lead to high fluid velocities. Another possibility is that thermal expansion and pressure drop or flow distribution considerations might encourage wide spacing between heat exchanger baffles and large clearances between the tubes and their supports. Such considerations might be of significance in either the fast liquid-metal-cooled reactor systems or in the thermal molten salt reactor systems. It is also important to recognize that most of our experience with vibration damage in heat exchangers has been associated with materials where elastic response is the governing structural limit. The heat exchanger of interest in the breeder systems will probably use materials under conditions where creep may be the governing structural factor.

Although present plans call for us to limit our investigation to designs for liquid-liquid heat exchangers, there will ultimately be a need to investigate the problems in two-phase fluid systems of pressurized water reactors and liquid-metal fast breeder reactor steam generators as well as the steam generators in gas-cooled reactor systems. These

systems may introduce so many engineering complications to the analysis that a tremendous design and experimental program will be needed to establish an effective analytical technique for evaluating tube vibration.

Recommendations

There appears to be adequate data available in the literature on the phenomena associated with fluid flow across single stationary cylinders. However, there is only a limited amount of available data on the vibration characteristics and forces developed by flow of a fluid over a bundle of oscillating cylinders, and these data are inconclusive. Our analysis of the heat exchangers reviewed in this survey revealed the need for further research to augment the existing data.

We therefore recommend the initiation of an experimental program directed toward obtaining sufficient data to develop correlations between a few basic design parameters that are controlled by the heat exchanger designer and some of the fundamental mechanics considerations that are suited to analysis within an applied framework. The variables that affect tube vibration and our appraisal of their significance in heat exchanger applications are given in Table 6. From a practical standpoint, we believe that only the problems associated with hydrodynamically induced vibrations deserve attention. We also believe that Reynolds numbers in the range of 300 to 200,000, the so-called critical range, will cover the operating conditions pertinent to practically all nuclear applications.

The configuration of the heat exchanger has a bearing on the vibrational characteristics, and we suggest that the experimental effort be concentrated on cross-flow heat exchangers containing U-tube bundles. These should be investigated for the effects of velocity distribution as a function of tube size and spacing, baffle spacing and window openings, and as a function of the bypass areas. The investigation of conditions at the entrances and exits of heat exchangers also warrants consideration. Both structural and fluid damping parameters deserve attention as do factors, such as baffle thicknesses and tube-to-baffle clearances, that

Table 6. Variables Affecting Tube Vibration in Heat Exchangers

Parameters of Vibration Experiment	Remarks
Generalized Problems	
1. Vibration induced hydrodynamically by shell-side flow	Most prevalent problem, particularly with respect to tube vibration failures, and accounts for largest part of heat exchanger failures from vibration
2. Vibration induced hydrodynamically by tube-side flow	Not expected to be significant except at very high velocities
3. Effect of pulsations in flow rate on shell or tube	Items 3, 4, and 5 are usually treated from the system design standpoint since vibrations arise principally from the pumping equipment
4. Vibration transmitted by piping and supports	
5. Externally induced mechanical vibration transmitted to the equipment	
Hydrodynamically Induced Vibration	
6. Effects of shell-side fluid flow (a) Velocity distribution in cross flow (b) Velocity distribution in parallel flow (c) Velocity distribution in baffle-window areas (d) Effects of bypass leakage (e) Effect of fluid angle-of-approach (yawed cylinders)	The interstitial flow velocities through the shell-side tubing continually vary across the tube bundle since the number of tubes will vary from a minimum at the outer periphery to a maximum at the major shell diameter. The effects of bypass leakage, various degrees of cross and parallel flow, and the effect of angle of approach on the effective flow velocity need to be determined. Item "a" and, to a lesser extent, Item "d" are the principal items of concern.
7. Effect of Reynolds number (a) Regular range (b) Critical range (c) Supercritical range (d) Transcritical range	The four ranges of Reynolds numbers in which different phenomena occur are $1 < N_R < 90$: vortices formed in symmetrical manner and do not break away $300 < N_R < 2 \times 10^5$: vortices formed and released alternately from opposite tube sides, range of interest in exchangers $2 \times 10^5 < N_R$ to over 10^6: no regular vortex pattern and forces are random $N_R > 10^6$ to 10^7: indications that vortex wake is again organized
8. Vortex shedding frequency in tube bank arrays (a) In-line tube banks (b) Staggered tube banks (c) Effect of interaction between tubes	The value of the Strouhal number, N_S, for a single non-vibrating cylinder in cross flow is about 0.2 in the critical Reynolds number range. Surrounding tubes in a heat exchanger bundle can have considerable influence on N_S, which can range up to 0.5 for some configurations. Existing data need verification.
9. Conditions at which vortex shedding changes from "random" to "systematic" in tube banks.	Evidence that shedding changes from "random" to "systematic" or "organized" when amplitude of vibration $\geq$ approximately 10% of cylinder diameter (for single cylinder)
10. Effects of flow at inlet and exit regions	Survey of DRDT heat exchangers indicates damage from tube vibration to be more prevalent in these regions. Methods of designing impingement baffles and diffusers need to be studied.
11. Effects of flow in U-bend regions	Not adequately studied, and is a critical region where failures have occurred.

Natural Frequency of Component Vibration	
12. Effect of baffle plate vibration	Should present no problems since baffles can be designed for adequate stiffness
13. Effect of tube lacing (stiffening)	Several DRDT exchangers have been stiffened by inter-tube lacing. Cannot be quantatively evaluated at this time.
14. Natural frequency of vibration of tubing	
(a) Effect of tube hole clearance	Natural frequency increases with decreasing clearance and becomes better defined
(b) Effect of baffle spacing	Symmetry of baffle spacing has no apparent effect
(c) Effect of baffle thickness	Can affect frequency, effect unknown
(d) Effects of longitudinal stress and temperature	Natural frequency increases in tension, decreases with compression
(e) Effects of corrosion, erosion, fatigue, and creep	Can change tube dimensions and thus alter natural frequency
(f) Effects of added mass (external fluid displacement)	Will decrease natural frequency
(g) Effect of internal fluid (mass and velocity)	Internal velocity has no effect except when very high
(h) Natural frequency of U-bend sections	Methods of calculation presently unsatisfactory
	Items "b", "d", "f", and "g" are adequately understood. Item "e" is dependent on particular circumstances and should be treated as a special problem in each instance
Lift and Drag Forces on Tubing	
15. Coefficient of drag force	Magnitude of these forces will determine the zero frequency (static) bending moment, deflection, and stress in tube wall. Value of $C_D = 1.00$ fairly well established for the critical Reynolds number range ($300 < N_R < 2 \times 10^{15}$) for single rigid tubes. A suitable value of C_L is not readily determined from the literature. Values ranging from 0.2 up to 2.00 are variously quoted. Evidence indicates that vibration amplitudes $\geq$ 10% of tube OD will increase the force coefficients significantly.
16. Coefficient of lift force	
17. Effect of vibration on force coefficients	
Fluid and Structural Damping Effects	
18. Viscous versus non-viscous damping	The importance of damping is mainly in the limiting of the amplitude of vibration response. Evidence exists to indicate that damping is non-viscous rather than viscous in light-water systems. Ratio of the coefficient of damping to the critical damping is the damping ratio used to determine the magnification factor. Values which have been assumed for the damping ratio in water range from 0.02 to 0.10, resulting in magnification factors ranging from 5 to 25. Multiplication of the zero frequency deflection by the magnification factor results in the tube maximum deflection, thereby determining the maximum value of tube stress and whether the tubes will impact during vibration. Experimental data are required, particularly for liquid sodium, molten salt, organic fluids, and heavy water. Indications are that damping in water is not significant but this needs study.
19. Damping ratio	
20. Magnification factor	
21. Tube impaction at resonance	
22. Maximum allowable tube stress versus endurance limit	

affect the vibrational frequency of tubes. The effects of drag and lift forces may warrant separate investigation.

An experimental apparatus that could be used to study the flow phenomena associated with vibrating tube bundles would consist essentially of a closed loop with a test section in which various tube configurations could be mounted to simulate conditions in a heat exchanger tube bundle. Water at room temperature would be used as the flow medium since many of the primary-circuit heat exchangers in reactor systems employ water as the coolant, particularly on the shell side. Water at a temperature of 75 to 80°F also has physical characteristics very similar to those of liquid sodium. Most heat exchangers are operated in a Reynolds number range from about 300 to 2×10^5, and the pumping capacity of the loop would be selected to span a significant portion of this range for at least the initial experiments. The experimental facilities required for the recommended investigative program would therefore consist of a fluid circulation system capable of attaining high fluid velocities over a bank of tubes with a bundle diameter of about 24 in. An assembly equivalent to a heat exchanger about 12 ft long will probably have to be used to provide flexibility in baffle spacing and tube support arrangement with effective space for observation and instrumentation.

The number of experimental tube arrays to be used will depend upon economic considerations and will be materially affected by the number of variables to be studied. The ability to change the tube pattern and spacing as well as the baffle clearance and shape would be most desirable. If a suitable combination of heat exchangers can be found, there might be an advantage in buying a number of standard units. This decision must be left to the discretion of the experimenter.

A separate experimental setup that is simple in form may be necessary to investigate the damping characteristics and natural frequencies of tubes to make certain that the values used in the data correlation have been verified experimentally. The instrumentation required for this experimental work will necessarily include equipment to measure tube vibration frequencies, a means for determining the amplitude of vibration, and strain sensing equipment to measure vibrational strains in the tubing.

The data obtained from an experimental program such as that roughly described here will provide only general guidance to the designers of heat exchangers for preventing tube vibration since each design application will have to be considered separately. We anticipate that the program will provide more definitive limits for the Strouhal criterion, and it will also demonstrate investigative techniques that can be incorporated in heat exchanger engineering practice as part of the technological base. Such techniques could become part of the design specificiations and be included in the USAEC RDT Standards for Heat Exchanger Design.

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APPENDICES

Appendix A

BIBLIOGRAPHY OF SUPPLEMENTARY REFERENCES RELATED TO VIBRATION

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Appendix B

OPERATING EXPERIENCE OF PRIMARY-CIRCUIT HEAT EXCHANGERS REVIEWED IN THIS SURVEY

The operating experiences of the primary-circuit heat exchangers evaluated analytically in this survey are given in this section. It has not been possible in all cases to draw definite conclusions from the information received, and many of the descriptions included here are simply a paraphrased version of the operator's own review. In some instances, the observations made by the operators on the results of tests and the adequacy of certain specifications do not appear to be consistent with the operating experience. Some liberty has been taken with the information in the operators' reports in an attempt to present a logical sequence of events.

Advanced Test Reactor

Demineralized and pressurized primary cooling water is recirculated between the reactor and the tube side of the primary heat exchangers in the Advanced Test Reactor (ATR) system. Heat is removed from the primary water by secondary water recirculated from a cooling tower through the shell side of the exchangers at moderate utility pumping pressures.

The primary heat exchanger system consists of four vertically mounted component exchangers with U-tube bundles and two phases in counterflow on the shell side and tube sides of each exchanger. The installation is such that the U-portions of the tubes are at the top (inverted), and all inlet and outlet water connections are at the bottom. There are 1600 inverted U-tubes, 3/4 in. OD by 18 BWG, with a length of 331 in. on a triangular pitch of 1 1/32 in. The shell has an inside diameter of 68 in. and 12 split-type cross baffles on a 30-in. pitch (original design). Primary coolant water is circulated through the tube side at a rate of 29.97×10^6 lb/hr, an inlet temperature of 163.4°F, an outlet temperature of 129.4°F, and at a design pressure of 390 psig. The cooling-tower

(secondary) water flows through the shell side at a rate of 17.237×10^6 lb/hr, an inlet temperature of 85°F, an outlet temperature of 138.3°F, and a design pressure of 125 psig.*

On January 21, 1965, leaks were detected in two of the four primary-water heat exchangers that had been erected at the site of the AEC's Advanced Test Reactor, National Reactor Testing Station, Idaho. All four units (numbered M-2, M-3, M-4, and M-5) had been operated for about 100 hours at the rated secondary flow when leakage was detected. Subsequent investigation revealed that two tubes, one each in heat exchangers M-2 and M-5, had burst because of internal pressure and were allowing the primary water to flow into the secondary system. The Franklin Institute Research Laboratories (FIRL) was employed to determine the cause of tube failures in the two exchangers and to recommend design modifications to correct the problem. Subsequent tests performed by FIRL indicated that the heat exchanger tubes were vibrating with sufficient amplitude to cause approximately 400 tubes in each heat exchanger shell to wear against each other. The two tubes that failed had enough metal abraded from their surfaces to allow failure because of internal pressure.

A contract to redesign and modify the exchangers was negotiated, with the major modifications made to the heat exchangers being to increase the number of cross baffles that supported the tubes. The tubes in the original exchangers were supported at 59-in. intervals, and the tubes in the rebuilt exchangers are supported at 38-in. intervals. After the modifications were made to exchangers, M-2, M-3, M-4, and M-5; they were returned to NRTS for reinstallation in the ATR system. A fifth heat exchanger, M-85, was manufactured with the above modifications and added to the system at this time.

The following conclusions were made as a result of the Phase I tests.

1. Tube 60-44 in the M-2 heat exchanger had an irregular axial opening with a maximum width of approximately 3/8 in. and a length of approximately 3 in. The aperture had the form of a rupture caused by excessive internal pressure. The rupture allowed continuous leakage from the primary to the secondary systems.

*From Westinghouse Electric Specification Sheet dated October 1965.

2. The tube rupture occurred because of a loss of metal at the exterior surface of the tube wall that was caused by a rubbing contact with an adjacent tube.

3. At rated secondary flow, tubes both exterior and interior to the bundle vibrated with sufficient amplitude to undergo rubbing contact with one or more adjacent tubes.

4. At rated operating conditions, numerous tubes in all four heat exchangers would progressively fail in a similar manner throughout the design lifetime of the equipment.

5. Failure would occur in tubes at both the exterior of the tube bundle and at a minimum of seven rows of tubes internal to the bundle.

6. Tube vibration sufficient to cause mutual wear of adjacent tubes can occur under the simultaneous operating conditions of

(a) no primary pumps operating,

(b) one secondary pump operating,

(c) rated secondary flow in one heat exchanger, and

(d) primary and secondary flow control gate valves replaced with open spools.

7. Operation of additional pumps and flow in additional heat exchangers caused no increase in maximum amplitude of tube vibration.

8. As the secondary flow was increased, the tube vibration began at approximately 50% of rated flow and increased with flow to a maximum total amplitude of about 3/8 in. Throughout this range of flow, the frequency of vibration was independent of the flow rate, and the typical frequencies were between 18.0 and 22.0 cps.

9. All inlet secondary flow impingement plates were found to be intact. The use of 1/4-in.-thick carbon-steel impingement plate support bars is inconsistent with the specified 1/8 in. corrosion allowance over the life of the unit.

Elk River Reactor

The primary circuit in the Elk River Reactor (ERR) system contains 12 heat exchangers, including two evaporators and two sub-coolers. In

general, the units conform to ASME codes and TEMA specifications. No vibration analyses were requested in the invitations for bid specifications, and it is assumed that none were made. Further, from what can be determined by the operator, no fabrication inspections were conducted, no pre-operational tests were run, and no analyses of the exchangers related to damage from vibration were made. There has been no subsequent testing nor are there plans for tests to determine vibration characteristics or other problems that might arise from possible vibration in the primary-circuit heat exchangers.

There were no new problems uncovered in the survey conducted by the operator, nor is there any information to indicate that vibration is a problem in the ERR system.

Engineering Test Reactor

The primary heat exchangers in the Engineering Test Reactor (ETR) system transfer 175 Mw of reactor thermal energy from the reactor pressurized-water coolant system to the secondary water cooling system. Final heat dissipation from the secondary cooling system is by evaporation to the atmosphere through an induced-draft cooling tower.

The equipment in the ETR primary-to-secondary heat transfer system consists of four parallel banks with three heat exchangers in each bank. The 42-in.-diameter exchangers are conventional shell and tube, single pass, counterflow units containing 1700 Type 304 stainless steel tubes 5/8 in. OD by 18 BWG. The 20-ft-long tubes are rolled into a fixed double tube sheet at each end that is welded to the shell and channel sections. The exchanger shell and shell-side tube sheet (2 1/4 in. thick) are fabricated from firebox quality A-212 carbon steel. The exchanger channels and the channel tube sheet (1/2 in. thick) are fabricated from Type 304 stainless steel. The bolted flat-heat channel covers are A-212 Grade B carbon steel clad with Type 304L stainless steel.

Out of a total of 20,400 tubes installed in the 12 shells, 10 tubes have failed since activation of the system in 1957. Of the tubes that

failed, three were in the middle exchanger in Bank A, five were in the middle exchanger in Bank B, one was in the top exchanger in Bank C, and one was in the top exchanger in Bank A. The first eight tube failures were very similar in nature. Each of these tubes failed next to the north tube sheet, which is near the secondary water inlet, and they had an external scratch at the point of failure. Another similar characteristic of the tubes removed was that heavy wear had taken place at some of the baffles. In some cases, the tube wear at the baffles had reduced the wall thickness of the tube approximately 50%, and this extreme wear only occurred at one baffle on each tube. However, the failures that occurred next to the tube sheets were definitely determined to have been caused by fatigue and not by wear or stress corrosion. The ninth tube did not fail at the tube sheet but at a baffle. This failure was caused by wear between the baffle and the tube, and there was no visible scratch or mark on the tube near the tube sheet.

The leaks resulting from these tube failures were detected by an abnormal reading of the radiation monitors in the secondary water system. If these readings were not too high, reactor operation was continued until normal shutdown for refueling and the leaking tubes were then removed and the tube sheets plugged. However, during cycle 39, four stainless steel tubes installed in one of the 12 primary-circuit heat exchangers failed through what was later determined to be the mechanism of fatigue cracking. In view of the fact that similar failures in heat exchanger tubes caused by chloride stress corrosion cracking had occurred at Savannah River,[1] it was first thought that these failures in the ETR exchangers could have resulted from the same cause. The possibility of chloride stress corrosion prompted complete tube removal to facilitate detailed metallurgical examinations of the austenitic tube material. To have stress corrosion, both stress and corrosive attack are necessary, and no indication of corrosion could be found. However, the following conditions that suggest fatigue failure were found.

[1] R. H. McKane, H. C. Minton, and J. W. Wade, "Stainless Steel Failures in Savannah River Plant Reactor Areas," USAEC Report DP-539, Savannah River Laboratory, November 1960.

1. The tubes failed in the same place, and the point of failure was about 1/2 in. away from the tube sheet. If corrosion had been a factor, the failures would probably have had a random distribution or have been located where crevice corrosion could take place.

2. The location of the tube rupture and the apparent failure through the element of metal fatigue indicates excessive tube vibration. This theory is further substantiated by the evidence of external tube wear (fretting) where the tubes penetrated the heat exchanger baffles. Also, the failures occurred adjacent to the shell-side tube sheet opposite the secondary water inlet nozzle, which is considered the maximum area of water turbulence.

3. Notches that would serve as stress risers were found at the point of failure on the tubes.

The apparent tube vibrations leading to the ultimate failures have caused a reevaluation of the primary and secondary cooling system operation in an effort to reduce the water velocities on the shell side of the heat exchangers. In this case, the physical arrangement of the heat exchangers, with the water approaching the north end of the tubes at right angles, appears to be the cause of tube vibration. This type of failure suggests that water velocities at shell inlet nozzles should be reduced as much as possible and that adequate impingement baffles should be provided.

Fast Reactor Core Test Facility

Testing of the two intermediate heat exchangers in the Fast Reactor Core Test Facility (FRCTF) was completed as part of the close-out effort in the Liquid Plutonium Fuels Program. Tests were performed with both units by using water rather than sodium, for which the equipment was designed, to determine whether or not tube vibration would be a problem under service conditions.

The tube vibration tests were conducted on the 5-Mw and the 15-Mw heat exchangers in a 2500-gpm water loop. The 5-Mw exchanger was subjected to flows of 250, 375, 540, and 650 gpm; and its design sodium flow

rate was 500 gpm. The design sodium flow rate for the 15-Mw exchanger was 1500 gpm, and it was subjected to flows of 1000, 1500, and 2000 gpm in the water loop. Statham strain-gage accelerometers were inserted in the straight runs of selected tubes at various elevations, and the frequency and amplitude of vibration were recorded with an oscillograph. The accelerometer locations were chosen with regard to the flow patterns that could be assumed to exist within the exchangers. The curved portions of the tubes were not tested extensively because flow distribution vanes inserted during fabrication constrained tube motion in this region very effectively. The natural frequencies of the tubes could only be estimated roughly since the support conditions at any particular point in the exchangers are not known with any certainty.

The accelerometer measurements at all flow rates were supplemented by acoustic observations made with both a Mafco Model VT-101 Vibra-Tracer and a stethoscope. These observations were made at the inlet and outlet nozzles, along the straight section of the shell, and at the girth and bend of both heat exchangers. Stethoscopic observations on the 15-Mw exchanger at the 2000-gpm flow rate were more extensive than at other flow rates. The results of these measurements are summarized as follows.

1. There was little if any dependence of vibration frequency or amplitude on flow rate.

2. Many tubes appeared to vibrate at a frequency near their estimated natural frequency. This can be attributed to highly turbulent flow in the staggered tube bank producing random frequency pulsation or to parallel flow effects in certain regions.

3. Amplitudes did not exceed 5 mils peak-to-peak in either heat exchanger, and the larger amplitudes occurred in the vicinity of the shell-side outlet nozzles and the impingement-baffled inlet nozzles.

4. Extensive acoustic observations in the vibration tests and in separate shell-side pressure drop tests detected no sounds that could be attributed to tube vibration.

The general conclusion resulting from these tests was that the two heat exchangers would be satisfactory for their intended use in the FRCTF.

High Flux Beam Reactor

There are two primary coolers in the Brookhaven High Flux Beam Reactor (HFBR) system that transfer the reactor heat from the heavy water to the cooling water. The primary coolers are shell and U-tube units. In the original design, they were counterflow units but were later modified to split counterflow-concurrent flow units. There are 1350 U-tubes, 3/4 in. OD by 16 BWG, in each unit. The average shell-side fluid velocity in the original design was estimated to be 4.65 ft/sec.

During the early stages of reactor operation, rattling noises that originated in the coolers were detected and steps were taken to detect their cause. It was found that these noises could essentially be eliminated by reducing the shell-side flow to one-half of the rated flow. The possibility of tube failure was ruled out because of the absence of Tritium in the secondary water. The design of the primary coolers was modified, and these modifications apparently eliminated the vibration problem without making any significant change in the capacity of the units to exchange heat. The cost of these modifications to the units amounted to approximately 20% of the initial cost of the exchangers.

The alterations to the original design consisted of converting the original outlet port to an inlet port and adding two new ports that are used as outlets. The effect of having an inlet and outlet on each half of the shell pass is to reduce the shell-side velocities to one-half of their original values. The location of the two new outlet ports significantly reduces the flow over the U-bend section of the tube bundle, which is relatively flexible when compared with other parts of the tube bundle. Since vortex shedding frequencies are proportional to the velocity and fluid drag and lift forces are proportional to the velocity squared, a factor-of-2 improvement was made in the mismatching shedding and natural frequencies, and the forces applied are reduced by a factor of 4.

High Flux Isotope Reactor

The cooling requirements for power operation of the High Flux Isotope Reactor (HFIR) are satisfied by two separate cooling loops. The reactor (primary) cooling loop is designed to remove virtually all the energy from the core. In this loop, demineralized water as the primary coolant is pumped through the reactor tank at a design flow rate of 13,800 gpm. It then passes through the tube side of three of the four primary heat exchangers where it gives up its heat to the secondary coolant, which is circulated through the shell side of the exchangers. In the secondary cooling loop, the treated process water (secondary coolant) that circulates through the shell side of the exchangers is then circulated through a conventional induced-draft cooling tower where the heat is dissipated to the atmosphere.

The primary heat exchangers are of the shell and U-tube type, mounted vertically, and designed to permit removal of tube bundles. Each heat exchanger with its associated circulation pump and letdown valve is located in an individually shielded compartment, and each exchanger is designed to transfer approximately 117 million Btu/hr (34.3 Mw) from the primary coolant loop to the secondary coolant loop. Thus, only three heat exchanger and pump combinations are required for full-power operation and the fourth is kept in standby.* The nominal design flow rates for each of the exchangers are 5000 gpm at a temperature of 167°F for the primary coolant and 6667 gpm at 85°F for the secondary coolant.

Each of the heat exchangers is approximately 34 ft long and contains 1190 5/8-in.-OD U-tubes. The tube side of the exchangers is designed to operate at pressures up to 1000 psi at a temperature of 200°F, and the shell side is designed for operation at pressures up to 150 psi at a temperature of 200°F. The tube bundles can be removed with the reactor-bay crane through normally plugged and shielded hatches in the top of each exchanger cell (the reactor bay floor).

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*A new mode of operation instigated in January 1969 requires all four units to be operated at full power.

There have been three tube failures in the primary-circuit heat exchangers of the HFIR. One tube in each of three of the four exchangers has failed, and the circumstances surrounding each of these failures have been quite similar. In each case, the tubes that failed were in the same tube row and the same tube position in that row. This position is just beyond the baffle edge and on the periphery of the tube bundle. This is a position that is likely to have a high bypass flow. All three failures occurred in the month of August during hot, humid weather when it is known that the shell-side flows had been increased to 115-120% of the rated flow on the occasion of at least two of the tube failures. A summary of the operating history of the tubes that have failed in the HFIR heat exchangers is given in Table B.1.[2]

Table B.1. Summary of Data on Tube Failures in HFIR Heat Exchangers

Date of failure	8-15-66	8-6-68	8-13-68
Heat exchanger cell number	111	113	112
Position of failed tube, row	100	100	100
Position of failed tube, side	South, outlet	South, outlet	North, outlet
Axial position of failure	8 in. below baffle No. B-17	1 in. below baffle No. B-15	1 ft below baffle No. B-15
Total Mwd on reactor	5000	51,900	52,600
Total operating time above 6600 gpm,hr	30	220	260
Leakage rate following failure, gpm	Small	100	50
Water flow at time of failure, gpm		7700	6500

Since all of the tube failures occurred in the same general position in the tube bundle, a 10-in.-diameter view port was installed to permit visual observation and access to that area. Direct observations and a

[2] Internal correspondence from A. P. Fraas to J. A. Cox, Oak Ridge National Laboratory.

series of high-speed motion pictures were made at flow rates ranging from 4000 to 7500 gpm. The frequency at which the tube vibration was measured with a Strobatach was 22.5 cps. The flow rate (gpm) threshold for significant vibrations was not the same for all tubes, and this suggests the possibility that flow distribution and/or damping varies with the flow rate. Another possibility which cannot be completely discounted is that for a given flow rate (gpm), the local conditions for a particular tube may be beyond the set of conditions necessary for resonant vibration while the resonant condition does exist for another tube or is being approached as the flow rate is increased. However, in no case was the amplitude of tube vibration large at flow rates less than 6000 gpm. The amplitude of tube vibration generally tended to increase with the flow rate.[2]

After the access port was installed, both in-position and metallurgical investigations were made of a failed tube. Excerpts from the report on the metallurgical investigation[3] follow.

> "The primary cause of failure of the tube was fatigue. The exact location of failure along the length appears to relate to a local defective area on the inside surface of the tube probably introduced during manufacture as limited visual surface examination of other tubes in this exchanger shows no general deterioration in the form of pits or cracking... . The tube had severed with no evidence of swelling or overall necking down. There was no obvious mechanical damage such as dents or scratches at the abrupt appearing break. Liquid penetrant examination of several inches of the lower half of the broken tube and the adjacent tube revealed no evidence of cracks... .
>
> "The fracture surface revealed evidence of fatigue; distinct beach marks in the metal could not be observed due to the scale, but the scale texture, color, and quantity indicated progressive failure with time. There were chevron marks which assisted in locating the initiation point. Prior to metallographic study, a liquid penetrant examination of the I.D. revealed no general pattern of cracks. A section of the tube was cross-sectioned lengthwise to permit examination of the O.D. and I.D. at and near the fracture. On one side of the tube, the transgranular fracture of grains as occurs by fatigue is evident; 180° away there is local necking down the

[3] Internal correspondence from J. R. McGuffey to R. V. McCord, Oak Ridge National Laboratory.

cross section representing the last region of the tube to fail. A closer look at the structure at and near the break reveals a local deterioration of the tube at the inside diameter ... there is both a transgranular and an intergranular attack. ...at higher magnification a portion of the wall is missing suggesting that a foreign particle was rolled into the metal and then became dislodged. The same plane at the fracture a few thousands of an inch around the perimeter did not show the void nor was deterioration found in any other portion of the tube examined under the microscope. Thus, this appears to be an isolated condition which acted as a focal point for the failure of this tube but may not be prevalent in the remainder of the tubing. The one grain deep intergranular attack is not uncommon in annealed stainless steel and is of no great concern. The transgranular attack starting out of the void is worrysome; this is the mechanism for stress corrosion cracking; however, the fact that visual, boroscope and penetrant examinations show no general pitting or fine cracking in this or related tubes suggests that the one spot in this tube was an abnormality. Should additional failures occur, particularly in regions of the heat exchangers where the vibratory stresses are less severe, a more exhaustive metallurgical study would be warranted. As a precautionary measure, further boroscope as well as probolog inspections and a chemical analysis of the I.D. scale should be performed whenever the manhole of any heat exchanger is opened in the future. [3]

Hallam Nuclear Power Facility

At the Hallam Nuclear Power Facility (HNPF), heat is transferred from the reactor to the sodium-to-water steam generators by means of six heat transfer loops: three primary (radioactive) sodium loops and three secondary (nonradioactive) sodium loops. Heat is transferred from each primary loop to each secondary loop through sodium-to-sodium heat exchangers, which are called intermediate heat exchangers.

The intermediate heat exchangers are of the shell-and-tube counterflow design, with the primary sodium entering at the top of each unit and flowing downward through the tubes. The secondary sodium enters the shell at the bottom and flows upward. Thus, on both the primary and secondary sides, the "cold" (610°F) sodium is at the bottom and the "hot" (945°F) sodium is at the top. The difficulties associated with thermal stratification at low flow rates in a horizontally mounted sodium-to-sodium

heat exchanger previously experienced at the sodium reactor experiment are thereby circumvented.

Each loop at the HNPF incorporates two separate intermediate heat exchanger units piped in parallel and identical in construction. There is therefore a total of six intermediate heat exchangers, and they are designated as 1A, 1B, 2A, 2B, 3A, and 3B. As no valves are incorporated in the piping, each pair of units behaves essentially as one. The units are of all-welded construction fabricated of Type 304 stainless steel. Each unit contains 1395 tubes. A special two-ply bellows is mounted in the shell of each to allow for differential thermal expansion between tubes and shell. The design data for the intermediate heat exchangers are summarized in Table B.2.

Table B.2. Design Parameters of the HNPF Intermediate Heat Exchangers

Pressure, psig	100
Temperature, °F	1000
Tube-side pressure drop, psig	5
Shell-side pressure drop, psig	7
Sodium flow rate	2.82×10^6
Heat transfer coefficient, Btu/hr·ft^2·°F	1050

On November 18, 1962, while the reactor was subcritical, the sodium level in secondary loop 1 decreased. This was investigated and found to be the result of a leak in intermediate heat exchanger 1A between the primary and secondary fluids. The leak was in a tube near the inlet baffle plate. The tube was subsequently removed and examined by personnel at Argonne National Laboratory. The examination indicated that the failure was due to fatigue in the baffle plate area induced by vibration. The International Research and Development Corporation of Worthington, Ohio, was employed to take vibration measurements on the operating intermediate heat exchangers. The information resulting from the test is given in Table B.3.

Table B.3. Results of Vibration Measurements Made on HNPF Intermediate Heat Exchangers

Exchanger Number	Sodium Flow Range (% of full flow)	Tube Frequency (cps)
1B	64 to 74	37
2B	80 to 90	37
2A	80 to 90	35
3B	80 to 90	35

Heavy Water Components Test Reactor

The Heavy Water Components Test Reactor (HWCTR) was first operated at appreciable power in August 1962 and was shut down and put in standby condition in December 1964. The reactor was equipped with two steam generators that transferred heat from the heavy water primary coolant to boiling light water. The primary system was operated at pressures from 1000 to 1200 psig, and the secondary coolant was operated at various pressures up to 600 psig. The steam generators performed satisfactorily except that corrosion of the carbon steel tubes was experienced during a period when inadequate oxygen scavenging existed on the secondary side. No evidence of tube vibration was detected during a number of inspections of the secondary side of the steam generators.

A manufacturer's design analysis report treating vibration considerations was not included in the survey. These heat exchangers were built to the then current du Pont and TEMA Standards in which protection against vibration damage was assumed to be adequate from industry experience. The baffle holes for the tubes were made to give a 1/64-in. clearance for additional protection against vibration problems. The 1/64-in. diametral clearance was the TEMA Standard for tube supports, as compared to a 1/32-in. diametral clearance for baffle holes. A vibration analysis of the tubes in the steam generators was made by the du Pont Engineering Department. Such an analysis must be based on many assumptions. The analysis indicated a possibility, but by no means a certainty, of exciting frequencies at the natural frequency of certain tubes.

Inspection during fabrication revealed no unusual indications. No indications of vibration damage to tubes were seen during pre-operational testing nor after operation. Relatively frequent inspections were made of the secondary side of the steam generators because of the corrosion experienced from the brief period of inadequate oxygen scavenging.

After a period of operation, the feedwater inlet pipes, which entered from both sides of the shell and were cantilevered from the shell nozzles, showed minor wear on the adjacent tubes from vibration of the feedwater pipes. These pipes were removed and a continuous pipe was installed from nozzle to nozzle to enhance the rigidity of the feedwater inlet and to assure clearance from the adjacent tubes. This modification proved adequate.

Molten Salt Reactor Experiment

The primary heat exchanger in the Molten Salt Reactor Experiment (MSRE) is used to transfer heat from the fuel salt to the coolant salt. The unit was designed for low holdup of salts, simplicity of construction, and moderately high performance. The space limitations within the containment and other considerations dictated that the unit be fairly compact.

The heat exchanger was designed in 1961, and its fabrication was completed early in 1963. However, difficulties with excessive vibration in heat exchangers experienced at other nuclear reactor facilities prompted a review of the design for the unit built for the MSRE. This review indicated that vibration could be a problem and that flow tests should be conducted on the exchanger.

Flow tests with water were performed on the exchanger during the winter of 1963-64, and they revealed excessive vibration of the tubes and excessive pressure drop through the shell side of the exchanger. The faults were corrected, and the modified unit was installed in the primary loop of the reactor system in the spring of 1964. From January 1965 through November 1967, the heat exchanger was operated for approximately

14,000 hours with molten salt at temperatures from 1000 to 1225°F without any indication of leakage or change in performance. Operations since November 1967 have apparently been free of tube vibration problems.

Materials Testing Reactor

Data were not submitted on the Materials Testing Reactor (MTR), per instructions from the Idaho Operations Office, because flash evaporators rather than heat exchangers were utilized for the heat removal apparatus. However, as a matter of interest, it is noteworthy that the tube bundles in the MTR flash evaporators have shown no signs of leakage, wear, or deterioration since being placed in operation in March 1952.

Power Burst Facility

The two primary heat exchangers in the Power Burst Facility (PBF) are designed to remove the heat generated in the reactor core during steady-state operation and power bursts. This heat removed from the reactor core is transferred to the secondary coolant and dissipated to the atmosphere when the secondary coolant is circulated through a cooling tower. The flow rate of the secondary coolant through the shell side of the exchangers may be varied from zero to 100% of the design flow rate by control valves that are regulated by the temperature of the primary coolant entering the reactor vessel.

As originally conceived, the two primary heat exchangers are vertical fixed-tube-sheet single-pass counterflow units designed in accordance with TEMA Standards. Each exchanger contains 468 austenitic stainless steel tubes, 1-in.-OD by 16 BWG, on a triangular pitch of 1 3/8 in. Slightly radioactive demineralized water from the reactor core flows through the tubes at a rate of 7500 gpm with an entering temperature of 107.3°F and a design pressure of 150 psig. Treated cooling-tower water flows through the shell side of the exchangers at a rate of 7000 gpm with an entering temperature of 65.2°F and a design pressure of 150 psig.

The PBF is in Category 2 and therefore has no operating experience. The intent is to place the plant in operation in 1971. Vibration analyses of the original heat exchanger designs were made by Struthers Wells Corporation and Franklin Institute Research Laboratories. Our evaluation of the original design concept dated November 3, 1967, indicated that this design appears to be marginal from the standpoint of tube vibration. However, the heat exchangers have been redesigned since we made our evaluation. The new designs were reviewed by Franklin Institute Research Laboratories and Phillips Petroleum Company, and they do not consider the new designs to be marginal.

Piqua Nuclear Power Facility

The Piqua Nuclear Power Facility (PNPF) contains three heat exchangers in contact with the primary fluid. There is a single main heat transfer system containing a boiler and a superheater. The decay heat removal system (always in operation) contains a decay heat exchanger. The boiler and superheater transfer the heat from the organic circulating fluid to the steam system, and the decay heat exchanger transfers heat to a closed-loop water system.

The reactor plant was designed to produce 150,000 lb/hr of steam at a temperature of 550°F and a pressure of 450 psi, and the design power of the reactor is 45 Mw(t). Coolant was introduced into the system in 1962, and pre-operational testing was started. Criticality was achieved on June 10, 1963, and full power was attained in February 1964. During power operation from February 1964 to the last plant shutdown in January 1966, the average power of the reactor was 20 Mw(t). The plant has not been critical since January 1966.

The boiler and decay heat exchanger have no history of vibration analysis or physical measurement of the actual operating conditions pertaining to vibration. No operating problems were encountered with either the boiler or the decay heat exchanger.

Two sets of vibration tests were made on the superheater: one before and one after fuel loading, when the flow through the unit was markedly different. The first was occasioned by an unidentified noise in the vicinity of the superheater during pre-operational testing. The tests were performed in April 1963, and it was concluded from the results of these tests that no resonant frequency problems existed in the superheater. The only definite frequency measured on the unit was 48,000 cpm, which occurred during a coolant flow range of 11,000 to 12,000 gpm. The vibration was associated with a turbulent flow condition that was probably caused by the throttling of a valve. During the test it was noted that as the valve opening increased, the measured amplitude of vibration decreased.

A second set of vibration tests were run on the superheater in September 1963 after the fuel loading. The vibrations observed were of low amplitude and low velocity: less than 2 microinches and up to 0.035 in./sec for vibrations of 60,000 cpm, and up to 80 microinches and 0.04 in./sec for vibrations of 9000 cpm. These figures were interpreted as confirmation of the pre-core loading findings that no significant resonant frequency problems existed in the superheater. Low power operation was then started.

A steam leak developed in the superheater on November 10, 1963. This and subsequent leaks were attributed to steam-side corrosion. Examination of the tubes with a borescope indicated extensive pitting on the inside of the tubes. Corrosion of the superheater tubes is believed to have taken place during the extended periods of time when this unit was performing its preheating function. During these periods of low steam flow, part of the steam is continuously condensing, allowing noncondensables such as oxygen and carbon dioxide to become trapped and build up at the top of the inverted U's. After a while the condensate is in equilibrium with an appreciable partial pressure of oxygen and carbon dioxide, and it becomes acidic and corrosive to carbon steel. Again, it was concluded that vibrations were not involved.

After the failure of the tubes, an analytical vibration study was made of the superheater design. The analysis indicated that the local

inlet region velocity could induce a resonant condition and a vibratory problem could exist within the unit. It was felt that any resonant condition that could occur would be of a low amplitude. Although a resonant vibration problem is termed improbable, the fluid cross-flow velocities within the low entrance region of the unit are higher than those that are normally used in standard heat exchanger design, and it was concluded that a long term "wear" problem may exist.

Bars were installed between the tubes in the superheater to minimize the possibility of tube damage due to lateral vibration of tubes, and another vibration analysis of the superheater was made. This analysis indicated that the lowest possible resonant natural frequency for the modified tube bundle is considerably above the largest potential excitation frequency.

A search of the files indicates that no vibration analysis was done during the design of the heat exchangers. However, the design specifications for the three units state that "TEMA Standards apply". The successful and satisfactory service from units designed to TEMA Standards was precedent for the PNPF designs.

Appendix C

NOMENCLATURE

A = projected area per lineal inch of tube, $ft^2/in.$

A_x = average interstitial cross flow area within the tube bundle between baffle windows, ft^2

C = damping coefficient

C_D = coefficient of drag force

C_L = coefficient of lift force

C_o = critical damping coefficient

C_w = that portion of the tube-bundle diameter representing the mean bundle width between baffle windows, in.

c = freqeuncy constant

c_x = distance from neutral axis to point being analyzed, in.

D_B = outer diameter limit of tube bundle, in.

D_o = outside diameter of heat exchanger tube, in.

E = modulus of elasticity of tube material, psi

F_D = drag force on tube parallel to direction of flow, lb/in.

F_L = lift force on tube perpendicular to direction of flow, lb/in.

f_n = natural frequency of tube vibration, cps

f_s = vortex shedding frequency, cps

g = gravitational constant = 386 in./sec^2 = 32.2 ft/sec^2

I = moment of inertia, in.^{4}

K = spring constant

k = added mass constant

L = length of tube span between baffle plates, in.

M = maximum bending moment on unsupported tube span, in.-lb

m = mass

m_a = added mass

N_R = Reynolds number

N_S = Strouhal number

P = tube pitch, in.

p_1 = internal pressure, psi

p_2 = external pressure, psi

R = mean radius of ring, in.

r = radius of gyration, in.

r_1 = inside radius of tube, in.

r_2 = outside radius of tube, in.

r_i = radius to the point of investigation, in.

S = axial compressive (negative) or tensile (positive) force, lb

S_b = bending stress, psi

S_{pc} = pressure stress in the circumferential direction, psi

$S_{p\ell}$ = pressure stress in the longitudinal direction, psi

S_{pr} = pressure stress in the radial direction, psi

S_{tc} = thermal stress in the circumferential direction, psi

$S_{t\ell}$ = thermal stress in the longitudinal direction, psi

T = temperature, °F

t = time, sec

V = velocity of fluid, ft/sec

V_x = average shell-side fluid interstitial cross-flow velocity obtained by dividing the volumetric flow rate by the total area available for cross flow, ft/sec

v_b = volume of body, ft^3

W = mass flow rate of shell-side fluid, lb/hr

W_a = weight of added mass of fluid external to tube per unit length, lb/in.

W_c = weight of fluid inside tube per unit length, lb/in.

W_E = effective weight of tube per unit length, lb/in.

W_t = weight of tube material per unit length, lb/in.

y = amplitude of vibration

α = coefficient of thermal expansion, in./in.·°F

ζ = damping factor = C/C_o

μ = viscosity of fluid, lb/hr·ft

ρ = density of shell-side fluid, lb/ft^3

ρ_f = mass density of fluid, $lb \cdot hr^2/ft$ per ft^3

υ = Poisson's ratio

ω = circular frequency of tube vibration, radians/sec

ω_n = natural frequency of undamped oscillation, radians/sec

ORNL-4399
UC-38 Engineering and Equipment

Internal Distribution

1. M. Bender
2. C. E. Bettis
3. A. L. Boch
4. H. I. Bowers
5. E. J. Breeding
6. C. D. Cagle
7. Ji Young Chang
8. R. H. Chapman
9. C. J. Claffey
10. W. G. Cobb
11. C. W. Collins
12. J. A. Cox
13. F. L. Culler
14. R. W. Dehoney
15. H. G. Duggan
16. B. C. Duggins
17. W. F. Ferguson
18. F. C. Fitzpatrick
19. A. P. Fraas
20. J. K. Franzreb
21. J. H. Freels
22. C. H. Gabbard
23. W. R. Gall
24. W. R. Gambill
25. D. L. Gray
26. V. O. Haynes
27. R. S. Holcomb
28. A. Houtzeel
29. J. E. Jones, Jr.
30. S. I. Kaplan
31. R. J. Kedl
32. R. E. Lampton
33. M. E. LaVerne
34. D. B. Lloyd
35. M. I. Lundin
36. R. E. MacPherson
37. H. C. McCurdy
38. H. A. McLain
39. J. R. McWherter
40. J. G. Merkle
41. M. L. Myers
42. I. K. Namba
43. F. H. Neill
44-48. H. A. Nelms
49. M. E. Ramsey
50. G. W. Renfro
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