TECHNICAL MEMORANDUM

ANL-OTEC-78-3
Distribution Category UC-64

MASTER

FLOW-INDUCED VIBRATION IN SHELL-AND-TUBE HEAT EXCHANGERS
FOR OCEAN THERMAL ENERGY CONVERSION (OTEC)

by

J. J. Lorenz and D. Yung

Components Technology Division

Argonne National Laboratory

USDOE Division of Solar Technology

August 1978

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

The facilities of Argonne National Laboratory are owned by the United States Government. Under the terms of a contract (W-31-109-Eng-38) between the U. S. Department of Energy, Argonne Universities Association and The University of Chicago, the University employs the staff and operates the Laboratory in accordance with policies and programs formulated, approved and reviewed by the Association.

MEMBERS OF ARGONNE UNIVERSITIES ASSOCIATION

The University of Arizona	Kansas State University	The Ohio State University
Carnegie-Mellon University	The University of Kansas	Ohio University
Case Western Reserve University	Loyola University	The Pennsylvania State University
The University of Chicago	Marquette University	Purdue University
University of Cincinnati	Michigan State University	Saint Louis University
Illinois Institute of Technology	The University of Michigan	Southern Illinois University
University of Illinois	University of Minnesota	The University of Texas at Austin
Indiana University	University of Missouri	Washington University
Iowa State University	Northwestern University	Wayne State University
The University of Iowa	University of Notre Dame	The University of Wisconsin

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately-owned rights. Mention of commercial products, their manufacturers, or their suppliers in this publication does not imply or connote approval or disapproval of the product by Argonne National Laboratory or the U. S. Department of Energy.

TECHNICAL MEMORANDUM

ANL-OTEC-78-3

Distribution Category UC-64

MASTER

FLOW-INDUCED VIBRATION IN SHELL-AND-TUBE HEAT EXCHANGERS
FOR OCEAN THERMAL ENERGY CONVERSION (OTEC)

by

J. J. Lorenz and D. Yung

Components Technology Division

Argonne National Laboratory

USDOE Division of Solar Technology

August 1978

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

Table of Contents

	<u>Page</u>
List of Tables and Figures	i
Nomenclature	iii
Abstract	v
1.0 Introduction.	1
2.0 Methods for Predicting Flow-Induced Vibration	2
2.1 Natural Frequency of Tube Vibration.	2
2.2 Vortex Shedding.	3
2.3 Turbulent Buffeting.	6
2.4 Fluidelastic Instability	7
2.5 Acoustic Excitation.	8
2.6 Other Methods.	10
3.0 Results and Discussion.	10
4.0 Summary and Conclusions	15
Acknowledgement	18
References.	19

List of Tables and Figures

<u>Table No.</u>	<u>Title</u>	<u>Page</u>
I	Comparison of Predictions With Experimental Data from HTRI and TEMA Data Banks (from Ref. [14]).	21
II	Wall Thicknesses for Aluminum and Titanium Tubes	22
 <u>Figure No.</u>		
1	Strouhal number as a function of pitch-to-diameter ratio for different tube field layouts (from Ref. [14]).	23
2	Definition of longitudinal and transverse pitch for turbulent buffeting model, Eq. (8)	24
3	Tube span length as a function of crossflow velocity showing region of tube vibration	25
4	Heat exchanger shell diameter as a function of crossflow velocity showing the region of acoustic noise.	26
5	Heat exchanger shell diameter as a function of tube span length showing the region of acoustic resonance.	27
6	Influence of tube diameter on the maximum allowable tube span length.	28
7	Comparison of regions of acoustic noise for different tube diameters.	29
8	Influence of tube diameter on acoustic resonance	30
9	Tube vibration map for $P/D = 1.5$ (compare with Fig. 3).	31
10	Region of acoustic noise for $P/D = 1.5$ (compare with Fig. 4)	32

List of Tables and Figures (contd.)

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
11	Tube vibration map for aluminum (compare with titanium in Fig. 3).	33
12	Region of acoustic resonance for aluminum (compare with titanium in Fig. 5)	34
13	Tube vibration map for 45° configuration.	35
14	Tube vibration map for 90° configuration.	36
15	Tube vibration map for 60° configuration.	37
16	Region of acoustic noise for 45° configuration. . .	38
17	Region of acoustic noise for 90° configuration. . .	39
18	Region of acoustic noise for 60° configuration. . .	40

Nomenclature

c	speed of sound, m/s
D	tube outer diameter, m
D_i	tube inner diameter, m
E	modulus of elasticity, N/m^2
f_a	acoustic frequency, 1/s
f_n	natural frequency (fundamental mode), 1/s
f_{ns}	stressed natural frequency, 1/s
f_t	dominant buffeting frequency, 1/s
f_v	vortex shedding frequency, 1/s
g	gravitational acceleration, m/s^2
g_c	gravitational coefficient, $1 \text{ kg}\cdot\text{m}/\text{N}\cdot\text{s}^2$
I	cross-sectional moment of inertia, m^4
ℓ	longitudinal tube pitch, m
L	tube span length, m
$L_{\max}$	maximum allowable tube span length, m
m	mode number
P_a	axial load, N
P_s	system pressure, N/m^2
P/D	pitch-to-diameter ratio
Re	Reynolds number
S_v	Strouhal number, Eq. (5)
t	transverse tube pitch, m
V	crossflow velocity in transverse gap, m/s
V_c	critical velocity for fluidelastic instability, m/s

Nomenclature (contd.)

W_e	effective mass per unit length, kg/m
W_m	mass of tube per unit length, kg/m
W_s	added mass due to shell-side fluid per unit length, kg/m
W_t	mass of fluid inside tube per unit length, kg/m
Y	characteristic length, m
β	instability threshold constant
δ_o	log-decrement of the tube in still fluid
ζ	constant, Eqs. (6) and (9)
ρ_s	density of shell side fluid, kg/m ³

Abstract

The problem of crossflow-induced vibration in shell-and-tube heat exchangers for OTEC is considered. Models of the dominant excitation mechanisms are discussed: vortex shedding, turbulent buffeting, fluidelastic instability, and acoustic excitation. Criteria are presented for assessing vibration and equations are derived for predicting the maximum allowable span length as a function of crossflow velocity and other system variables. Also regions of acoustic resonance and noise are defined. Among the system variables studied are tube diameter, tube material, tube-field layout, and pitch-to-diameter ratio. All models are presented in a simple form that can readily be applied to the mechanical design of OTEC heat exchangers.

1.0 Introduction

Flow-induced vibration in heat exchangers is a potential cause of acoustic noise and mechanical failure of tubes. Failure due to vibration can result from fatigue, wear, collision of tubes, cutting at baffles, and leaks at tube joints. It has been found that tube vibration can also lead to an increased shell-side pressure drop. Over the years, numerous methods have been developed for predicting flow-induced vibration in shell-and-tube heat exchangers [1-11], and a number of excellent reviews have been published [12-14]. However, at the present time a complete solution to the problem is unavailable.

The present work considers crossflow-induced vibrations in shell-and-tube heat exchangers for Ocean Thermal Energy Conversion (OTEC).^{*} Here the shell-side fluid is a vapor, such as ammonia, and the tube-side fluid is seawater. The purpose of this paper is to assess the problem of flow-induced vibration in light of existing predictive methods. Models of the dominant excitation mechanisms are discussed: vortex shedding, turbulent buffeting, fluidelastic instability, and acoustic excitation. Criteria are presented for assessing vibration and equations are derived for predicting the maximum allowable span length as a function of crossflow velocity and other system variables. Also, regions of acoustic resonance and noise are defined. Among the system variables studied are tube diameter, tube material, tube-field layout, and pitch-to-diameter ratio. All models are presented in a simple form that can readily be applied to the mechanical design of OTEC heat exchangers.

^{*} Flooded bundle evaporators are not treated in this report.

2.0 Methods for Predicting Flow-Induced Vibration

2.1 Natural Frequency of Tube Vibration

Before discussing the various excitation mechanisms, the vibration characteristics of a tube will be considered. For conservatism (and simplicity) in vibration calculations, it is common practice to consider only the fundamental mode of tube vibration. The natural frequency (fundamental mode) of a uniform tube with equal span lengths and simple supports is [15]:

$$f_n = \frac{\pi}{2L^2} \sqrt{\frac{EIg_c}{W_e}} \quad (1)$$

where f_n is the natural frequency; E is the modulus of elasticity; I is the cross-sectional moment of inertia; L is the tube span length; W_e is the effective mass per unit length; and g_c is the gravitational coefficient.

The cross-sectional moment of inertia I is calculated from:

$$I = \pi (D^4 - D_i^4)/64 \quad (2)$$

where D and D_i are the tube outside and inside diameters, respectively. The effective mass per unit length W_e is defined as:

$$W_e = W_m + W_t + W_s \quad (3)$$

where W_m is the mass of tube per unit length; W_t is the mass of fluid inside tube per unit length; and W_s is the added mass per unit length for shell-side fluid. For a single cylinder in an infinite fluid, $W_s = \rho_s \pi D^2/4$, where ρ_s is the density of the shell-side fluid. In a tube bundle, the effect of adjacent tubes tends to increase the added mass. For OTEC heat exchangers, however, the added mass term is insignificant when compared with the other terms because the shell-side fluid is vapor. (The mass of

the thin liquid film and droplets on the shell-side is negligibly small.)

In general the natural frequency depends on the end conditions, number of spans, and type of intermediate supports. Equation (1) is strictly valid only for simply supported ends, but is a very good approximation even for clamped ends when the number of spans is greater than about 7, see Ref. [14]. The natural frequency is weakly dependent on tube-to-baffle hole clearance [16]. Clearance is important in that it has a strong influence on system damping.

Equation (1) provides the unstressed natural frequency of tube vibration. When a tube is subjected to axial compression, the natural frequency decreases, while under tension the frequency increases. The following equation can be used to compute the change in frequency resulting from axial loading [12].

$$f_{ns} = f_n \sqrt{1 + \frac{P_a L^2}{EI \pi^2}} \quad (4)$$

where f_{ns} is the stressed natural frequency; f_n the unstressed frequency; and P_a the axial load (negative for compression, positive for tension).

2.2 Vortex Shedding

Excitation by vortex shedding has been studied extensively by Chen [1]. The vortex shedding frequency is given by the following expression:

$$f_v = \frac{S_v V}{D} \quad (5)$$

where S_v is the Strouhal number; V is the velocity (free stream for single tube and transverse gap for tube banks); and D is the tube outer diameter.

When the vortex shedding frequency is near the natural frequency of the tube, resonance occurs and large amplitude vibrations are possible. The frequency of vortex shedding from a single cylinder has been investigated by a great number of authors, see review by Chen [1]. The Strouhal number remains nearly constant with a value of 0.20 in the Reynolds number range of 300 to 2×10^5 . Vortex shedding in a heat exchanger tube bank is considerably more complicated than vortex shedding behind isolated cylinders. For a tube bank, the Strouhal number depends on the tube arrangement. Tube fields are generally laid out with equilateral pitches in either square or triangular arrays. The most common arrangements are 30°-triangular, 60°-triangular, 45°-square, and 90°-square. Figure 1 shows a plot of Strouhal number versus pitch-to-diameter ratio constructed by Chenoweth and Kissel [14] from the data of Chen [1]. Unfortunately, Chen obtained little data at the smaller pitch-to-diameter ratios of interest to OTEC. At small P/D values the curves in Fig. 1 were obtained essentially by extrapolation and, therefore, should be regarded as approximate. Recent studies by Chen [2,3] have indicated that for small P/D values the excitation mechanism changes from vortex shedding to other mechanisms such as wake swing, jet swing, and jet instability. Moreover the dynamic forces associated with these latter mechanisms are smaller than those associated with vortex shedding. In the present study where the P/D values are relatively small, we will nevertheless refer to the excitation process as vortex shedding even though we recognize that other mechanisms may come into play.

For a tube bank, the velocity term in Eq. (5) is the transverse gap velocity* and not the diagonal gap velocity. This is in contrast to

*When the vapor crossflow is not truly perpendicular to the transverse gap, which typically occurs in heat exchangers experiencing radial inflow or outflow, there is no universally accepted definition of velocity for flow-induced vibration calculations. For this situation it is suggested that the designer conservatively assume that the entire crossflow is directed perpendicular to the gap.

the practice employed for heat transfer and pressure drop calculations. Although the transverse gap velocity is commonly used for flow-induced vibration calculations investigators have considered other possible definitions [12,14].

Vibration can occur if f_v/f_n is greater than some constant ζ . The condition for the onset of vibration is then:

$$\frac{f_v}{f_n} = \zeta \quad (6)$$

Chenoweth and Kissel [14] compared experimental data with predictions using $\zeta = 0.5$ and $\zeta = 1.0$. For heat exchangers with vapor on the shell-side, their study indicated that $\zeta = 1$ yields the best agreement with experimental data from TEMA and HTRI data banks. Further comments on the selection of ζ will be given later. Upon substituting Eqs. (1) and (5) for f_v and f_n into Eq. (6) and letting $\zeta = 1$, the following expression for the maximum allowable span length, $L_{\max}$, is obtained after some rearrangement:

$$L_{\max} = \left(\frac{\pi}{2}\right)^{\frac{1}{2}} \left(\frac{D}{S_v}\right)^{\frac{1}{2}} \left(\frac{EI g_c}{W_e V^2}\right)^{\frac{1}{4}} \quad (7)$$

where S_v is determined from Fig. 1 for a given tube arrangement and P/D ratio. Reliable heat exchanger design requires that the span length be less than $L_{\max}$ to avoid tube vibration.

Before closing, a few remarks should be made concerning the criterion given by Eq. (6). It was assumed that the vortex shedding and motion of the tubes are uncoupled. Moreover it was implicitly assumed that a tube is forced to vibrate at the vortex shedding frequency. These

assumptions are generally valid as long as the vortex shedding frequency f_v is not greater than the natural frequency f_n , i.e., Eq. (6) is satisfied. In the case when $f_v > f_n$, a tube may selectively respond to some random turbulent excitation component within the flow corresponding to the natural frequency of the tube, despite the fact that organized vortices may be shed at some non-resonant frequency. The tube vibration in turn steers the vortex shedding process, causing it to synchronize with the tube natural frequency.

2.3 Turbulent Buffeting

For closely packed tube banks Owen [4] abandoned the idea of a coherent vortex shedding process. Instead, he considers a spectrum of turbulence with a dominant frequency which is proportional to the average flow velocity within the bank. Owen [4] gives the following equation for the dominant turbulent buffeting frequency, f_t :

$$f_t = \frac{VD}{\ell t} \left[3.05 \left(1 + \frac{D}{t} \right)^2 + .28 \right] \quad (8)$$

where V is the transverse gap velocity; ℓ is the longitudinal tube pitch; and t is the transverse pitch, see Fig. 2.

Following the logic of the preceeding section, the criteria for the onset of vibration is given by:

$$\frac{f_t}{f_n} = \zeta \quad (9)$$

Again the constant ζ is set equal to unity. Upon substituting Eqs. (1) and (8) for f_t and f_n into Eq. (9), the following expression for maximum allowable span length $L_{\max}$ is obtained after rearrangement:

$$L_{\max} = \left(\frac{\pi}{2}\right)^{\frac{1}{2}} \left(\frac{\ell t}{D\theta}\right)^{\frac{1}{2}} \left(\frac{EI g_c}{W_e V^2}\right)^{\frac{1}{4}} \quad (10)$$

$$\text{where } \theta = 3.05 \left(1 - \frac{D}{t}\right)^2 + 0.28 \quad (10a)$$

2.4 Fluidelastic Instability

Connors [5] identified a fluidelastic instability which occurs when the work done on the tube system by the fluid exceeds the energy dissipated by system damping. When a critical velocity is exceeded the tubes vibrate in an orbital or whirling manner. Connors [5] gives the following equation for the critical velocity, V_c , beyond which fluidelastic whirling occurs:

$$V_c = \beta f_n D \sqrt{\frac{W_e \delta_o}{\rho_s D^2}} \quad (11)$$

where W_e is the effective mass per unit length [Eq. (3)]; f_n is the tube natural frequency; β is the instability threshold constant; and δ_o the log-decrement of the tube in still fluid. Based on the work of Chenoweth and Kissel [14], the following values were selected for heat exchangers where the shell-side fluid is a gas: $\beta = 9.9$ and $\delta_o = 0.0033$. In general the instability constant β is a function of tube-field layout, and the log decrement δ_o is a function of fluid viscosity, tube material, and bundle geometry. Unfortunately no general methods are yet available for predicting the dependence of β and δ_o on system parameters. In a recent paper, Connors [18] gives a correlation for β as a function of P/D for a 90° tube-field layout. However the correlation is valid only for pitch-to-diameter ratios in the range $1.41 < P/D < 2.12$, and is not applicable to other tube-field layouts. Until more is known about the behavior of β and δ_o , the single values from the work of Chenoweth and Kissel will be employed.

The critical condition for the onset of fluidelastic whirling is when the transverse gap velocity, V , equals the critical velocity for fluidelastic instability, V_c :

$$\frac{V}{V_c} = 1 \quad (12)$$

Upon substituting Eq. (1) for f_n into Eq. (12), the following expression for the maximum allowable span length is obtained.

$$L_{\max} = \left(\frac{\pi}{2}\right)^{\frac{1}{2}} \beta^{\frac{1}{2}} \left(\frac{EI \delta_o g_c}{\rho_s V^2}\right)^{\frac{1}{4}} \quad (13)$$

2.5 Acoustic Excitation

When the shell-side fluid is a vapor, acoustic waves can be excited by either vortex shedding or turbulent buffeting. The acoustic frequency, f_a , of a heat exchanger is given by Chenoweth and Kissel [14] as:

$$f_a = \frac{mc}{2Y} \quad (14)$$

where m is the mode number; Y is a characteristic length; taken as the shell-diameter of the heat exchanger; and c is the speed of sound in the shell-side fluid, which can be approximated by:

$$c = \alpha \sqrt{\frac{P_s g_c}{\rho_s}} \quad (14a)$$

where the constant $\alpha = 1.19$; P_s is the system pressure; and ρ_s is the density of the shell-side gas.

If either the vortex shedding frequency or turbulent buffeting frequency is within 20% of the acoustic frequency, a loud noise can be generated within the heat exchanger shell [14]. Although mechanical damage

from acoustic noise is unlikely, the sound may be intolerable and exceed OSHA safety limits. The conditions for acoustic noise are:

Vortex shedding

$$0.8 < \frac{f_v}{f_a} < 1.2 \quad (15)$$

Turbulent buffeting

$$0.8 < \frac{f_t}{f_a} < 1.2 \quad (16)$$

Upon substituting for f_v , f_t , and f_a , the foregoing conditions for acoustic noise can be expressed in an alternate form. The equations defining the upper and lower limits of acoustic noise for each excitation mechanism are:

Vortex shedding

$$Y = K \frac{m}{2} \left(\frac{D}{S_v} \right) \frac{c}{V} \quad (17)$$

Turbulent buffeting

$$Y = K \frac{m}{2} \left(\frac{\ell t}{D\theta} \right) \frac{c}{V} \quad (18)$$

where $K = 1.2$ for the upper limits and $K = .8$ for the lower limits. Acoustic noise is probable when the heat exchanger shell diameter Y lies either within the region bounded by Eq. (17) with $K = 1.2$ and 0.8 or by Eq. (18) with $K = 1.2$ and $K = 0.8$. In acoustic vibration calculations it is usually sufficient to consider only the fundamental mode, i.e., $m = 1$.

A potentially detrimental situation can occur when the acoustic vibration is in resonance with heat exchanger tube vibration. Resonance is probable when the following condition is satisfied:

$$0.8 < \frac{f_n}{f_a} < 1.2 \quad (19)$$

where f_n is the tube natural frequency and f_a is the acoustic frequency. Upon substituting for f_n and f_a , the condition for resonance can be expressed in an alternate form. The equation defining the upper and lower limits for resonance is:

$$Y = \frac{Kmc}{\pi} \left(\frac{W_e}{EIg_c} \right)^{\frac{1}{2}} L^2 \quad (20)$$

where $K = 1.2$ for the upper limit and $K = 0.8$ for the lower limit. The mode number is again taken to be $m = 1$. If the shell diameter lies within the region bounded by Eq. (20) with $K = 1.2$ and 0.8 , it is probable that resonance will occur.

When either acoustic noise or resonance is predicted, appropriate corrective measures should be taken. Generally this problem is remedied by inserting detuning plates parallel to the direction of crossflow. This changes the acoustic frequency by changing the characteristic length, but does not affect the heat transfer performance and pressure drop [14].

2.6 Other Methods for Predicting Flow-Induced Vibration

The foregoing are among the more popular methods for predicting flow-induced vibration, and will be the only ones employed in the present work. Nevertheless some of the other methods which have been developed over the years are noteworthy: Thorngren [6]; Gregorig [7]; Fitz-Hugh [8]; Walker and Reising [9]; and Brothman [10].

3.0 Results and Discussion

Chenoweth and Kissel [14] examined the accuracy of the various predictive methods by comparing predictions with experimental data from TEMA and HTRI data banks. Correct predictions were defined as either when

vibration was predicted where it was reported or when vibration was not predicted where it was not reported. Results are summarized in Table I for heat exchangers with gas as the shell-side fluid. The following values of the constants were employed in the calculations: $\zeta = 1$ (vortex shedding and turbulent buffeting) and $\beta = 9.9$, $\delta_o = 0.0033$ (fluidelastic instability). Other values were tried in Ref. [14] but it was found that the values used herein yield the best agreement with the data. In Table I it can be seen that the models generally predict vibration where it was reported. However, the models tend to be conservative in that they often predict vibration where it did not occur. For example, on the basis of vortex shedding and turbulent buffeting, tube vibration was predicted to occur in nearly all cases although no vibration was reported in 11 out of 20 cases in the two data banks combined. Chenoweth and Kissel [14] point out that possibly a different definition of velocity is needed to improve the predictions. It is hoped that future research in flow-induced vibration will bring forth improvements in all the predictive methods.

In the previous section, equations were derived for predicting the maximum allowable tube span length based on considerations of vortex shedding, turbulent buffeting, and fluidelastic instability. Also, equations were presented for determining regions of acoustic resonance and noise. To illustrate the use of these equations consider a shell-and-tube heat exchanger having titanium or aluminum tubes in a 30°-triangular configuration with $D = 2.54$ cm and $P/D = 1.25$. Tube wall thicknesses for various tube diameters are specified in Table II. The two materials listed in Table II, titanium and aluminum, are leading candidates for OTEC because of

their corrosion resistance and compatability with the working fluid.

For the various excitation mechanisms, Fig. 3 shows the maximum span length as a function of crossflow velocity. Vibration is expected in the region above the fluidelastic curve, while no vibration is expected in the region below the turbulent buffeting curve. The area in between is a region of uncertainty, reflecting the variance in predictions using different models. On the basis of fluidelastic instability, this region appears to be safe; but on the basis of turbulent buffeting, vibration would be expected. For design purposes the span length should be chosen to lie below the region of uncertainty. If the expected crossflow velocity is, say, 1 m/s, the maximum allowable tube span length is 1.8 m. For reference, the TEMA maximum span length is also plotted. As stated in Ref. [14], the 1940 TEMA Standards somewhat arbitrarily specified maximum span lengths for different tube sizes and materials. Common practice is to limit the maximum span length to 80% of the TEMA values. For the particular case considered in Fig. 3, it can be seen that for crossflow velocities less than 1.4 m/s the TEMA maximum span would imply a conservative design. For velocities larger than 1.4 m/s the design would be somewhat uncertain.

Figure 4 shows the region of potential acoustic noise. The upper boundary is set by Eq. (17) with $K = 1.2$ and the lower boundary by Eq. (18) with $K = 0.8$. (It should be pointed out that the regions bounded by Eq. (17) with $K = 1.2$ and $K = 0.8$ and by Eq. (18) with $K = 1.2$ and $K = 0.8$ overlap; consequently it is only necessary to plot the two limiting curves.) For a crossflow velocity of 1 m/s acoustic noise may occur if the heat exchanger shell diameter is between 15 m and 32 m. The more severe problem of acoustic resonance is considered in Fig. 5. Assume that the crossflow velocity is

1 m/s and that a 1.8 m span length is chosen in accordance with Fig. 3 to avoid tube vibration. Then Fig. 5 indicates that resonance can occur if the shell diameter is between 15 m and 23 m. When either acoustic noise or resonance is predicted, appropriate corrective measures should be taken. Generally this problem is remedied by inserting detuning plates parallel to the direction of crossflow. This changes the acoustic frequency by changing the characteristic length, but does not affect the heat transfer performance and pressure drop [14].

The effect of increasing tube diameter from 2.54 cm to 5.08 cm is illustrated in Fig. 6. In these calculations the wall thickness for a given outer diameter is taken from Table II. When the tube diameter is increased from 2.54 cm to 5.08 cm, the predicted maximum allowable span length increases from 1.8 m to 3.5 m, assuming a crossflow velocity of 1 m/s. The increase in allowable span length is essentially due to an increase in tube stiffness. Comparisons of the regions of acoustic noise and resonance are given in Figs. 7 and 8. For a crossflow velocity of 1 m/s, it is found that the heat exchanger will be free of acoustic vibration if the shell diameter is less than about 30 m.

Changing the pitch-to-diameter ratio from $P/D = 1.25$ to 1.5 has little overall effect on the maximum allowable span length as seen by comparing Figs. 3 and 9. The only significant difference is that with $P/D = 1.5$ the vortex shedding curve is shifted downward to a level nearly coincidence with the turbulent buffeting curve. The region of acoustic noise is shown in Fig. 10. By comparing Fig. 4 and 10, it can be seen that the region of acoustic noise is apparently smaller for $P/D = 1.5$ than for 1.25.

This result has no physical significance, however, and is merely a consequence of the similarity of predictions using the vortex shedding and turbulent buffeting models when $P/D = 1.5$. Acoustic resonance is not dependent on P/D so Fig. 5 applies for this case also.

Figure 11 shows the maximum tube span length for aluminum as a function of crossflow velocity. The allowable span length for aluminum is slightly greater than that for titanium, compare Figs. 3 and 11. This is essentially due to the thicker wall of aluminum tubes, see Table II. For a crossflow velocity of 1 m/s the maximum allowable span length for aluminum is 2 m compared to 1.8 m for titanium. In Fig. 11 it is seen that the TEMA maximum span length is conservative over the entire velocity range considered. Acoustic noise does not depend on tube material; therefore, Fig. 4 applies to this case also. The region of acoustic resonance is weakly dependent on tube material, compare Figs. 5 and 12.

Maximum allowable tube span lengths for different tube-field layouts are shown in Figs. 13, 14, and 15. The region of uncertainty is bounded from below by the vortex shedding curve for the 45° and 90° configurations (Figs. 13 and 14) and by the turbulent buffeting curve for the 30° and 60° configurations (Figs. 3 and 15). For all configurations the region of uncertainty is bounded above by the fluidelastic instability curve. Moreover, the fluidelastic curve remains fixed because the dependence of β and δ_0 on geometry is unknown. It is interesting to compare the severity of vibration for the various tube arrangements. Since crossflow velocity depends on tube arrangement, it is first necessary to establish a basis of comparison. An approach similar to that of Ref. [17] will be adopted: Consider four horizontal

tube heat exchanger units, each having a different tube arrangement (i.e., 45°, 90°, 30°, and 60°). Assume that each heat exchanger is designed with the same equilateral pitch-to-diameter ratio, a square bundle cross-sectional envelope, and a fixed overall tube length. Moreover, each unit has an equal number of tubes and hence the same heat duty* and total vapor generation. In each unit the vapor is assumed to flow horizontally outward from the vertical centerline. Under these assumptions, the relative crossflow velocities for the different heat exchangers can be determined. Having determined the crossflow velocities, the vulnerability of each layout to vibration can then be examined. It was found that all of the arrangements were affected about the same. Although differences were not very significant and should not be over-emphasized, it was interesting to note that the 45° arrangement was the best (i.e., least vulnerable to vibration) followed closely by the 30°, 60°, and 90° configurations in that order.

The regions of acoustic noise for the various configurations are shown in Figs. 16, 17, and 18. For the 45° and 60° configurations the regions are relatively narrow, reflecting the similarity of predictions based on vortex shedding and turbulent buffeting. Acoustic resonance is not dependent on configuration and, therefore, Fig. 5 applies for these cases also.

4.0 Summary and Conclusions

A study of crossflow-induced vibration in shell-and-tube heat exchangers for OTEC was conducted. Models of the dominant excitation mechanisms were discussed: vortex shedding, turbulent buffeting, fluidelastic instability, and acoustic excitation. Criteria were presented for assessing vibration and

*It was assumed that the thin film heat transfer coefficient is not affected by changing vapor velocity; this assumption is believed to be a realistic one.

equations were derived for predicting the maximum allowable span length as a function of crossflow velocity and other system variables. Also regions of acoustic resonance and noise were defined. Among the system variables studied were tube diameter, tube material, tube-field layout, and pitch-to-diameter ratio. All models were presented in a simple form that can readily be applied to the mechanical design of OTEC heat exchangers.

For 2.54 cm diameter titanium tubes in a 30°-triangular layout with $P/D = 1.25$, the maximum allowable span length for a vapor crossflow velocity of, say, 1 m/s is 1.8 m. For reliable heat exchanger design the span length should be less than 1.8 m to avoid tube vibration. With a crossflow velocity of 1 m/s, it was also predicted that acoustic resonance and noise will occur if the heat exchanger shell diameter is between 15 m and 32 m. For tubes having the wall thicknesses specified in Table II, the predicted maximum allowable span length was found to increase with tube diameter. This behavior is essentially due to an increase in tube stiffness. When the tube diameter was increased from 2.54 cm to 5.08 cm, the predicted maximum allowable span length increased from 1.8 m to 3.5 m, assuming a crossflow velocity of 1 m/s. Changing the tube pitch-to-diameter ratio (from $P/D = 1.25$ to 1.5) and the tube material (from titanium to aluminum) had little overall affect on the results. A crude comparison was made of the severity of vibration for different tube-field layouts (30°-triangular, 60°-triangular, 45°-square, and 90°-square). It was found that all of the arrangements were affected about the same. Although differences were not very significant and should not be over emphasized, it was interesting to note that the 45° arrangement was the best (i.e., least

vulnerable to vibration) followed closely by the 30°, 60°, and 90° configurations in that order.

Acknowledgement

The authors are indebted to Drs. M. W. Wambsganss and T. R. Bump for reviewing this report and providing valuable comments; and to Celeste B. Kinsella for final typing.

References

1. Chen, Y. N., "Flow-Induced Vibration and Noise in Tube-Bank Heat Exchangers Due to Von Karman Streets," Trans. ASME, Series B, J. of Eng. for Industry, Vol. 90, Feb. 1968.
2. Chen, Y. N., "Excitation Sources of The Flow-Induced Vibrations and Noise in Tube-Bank Heat Exchangers," ASME Winter Annual Meeting, Atlanta, GA, Nov. 1977.
3. Chen, Y. N., "The Sensitive Tube Spacing Region of Tube Bank Heat Exchangers for Fluid-Elastic Coupling in Cross Flow," Published in Fluid Structure Interaction Phenomena in Pressure Vessel and Piping Systems, PVP-PB-026, ASME Winter Annual Meeting, Atlanta, GA, Nov. 1977.
4. Owen, P. R., "Buffeting Excitation of Boiler Tube Vibration," J. Mech. Eng. Sci., Vol. 7, No. 4, 1965.
5. Connors, H. J. Jr., "Fluidelastic Vibration of Tube Arrays Excited by Cross Flow," Symp. on Flow-Induced Vibration in Heat Exchangers, ASME Winter Annual Meeting, New York, Dec. 1970.
6. Thorngren, J. T., "Predict Exchanger Tube Damage," Hydrocarbon Processing, Vol. X, April 1970.
7. Gregorig, R., et al., "Vibration Criteria of a Tube in Crossflow, Part 3: Vibration of Tube Bundles," Chemie-Ing.-Techn., Vol. 40, 1968.
8. Fitz-Hugh, J. S., "Flow-Induced Vibration in Heat Exchangers," Paper No. 427, Proc. Int. Symp. on Vibration Problems in Industry, Keswick, V. K., April 1973.
9. Walker, W. M., and Reising, G. F. S., "Flow-Induced Vibrations in Cross-

References (Contd.)

- Flow Heat Exchangers," Chem. and Process Eng., Vol. 49, No. 11, 1968.
10. Brothman, A., et al., "A Tube Vibration Analysis Method," AICHE Symposium Series 138, Vol. 70, 1974.
 11. Chen, S. S., "Crossflow-Induced Vibration of Heat Exchanger Tube Banks," Nuclear Eng. and Design, Vol. 47, 1978.
 12. Shin, Y. S. and Wambsganss, M. W., "Flow-Induced Vibration in LMFBR Steam Generators: A State-of-the-Art Review," Nucl. Eng. and Design, Vol. 40, No. 2, 1977.
 13. Moretti, P. M., "A Critical Review of the Literature and Research on Flow-Induced Vibrations in Heat Exchangers," AICHE Symposium Series 138, Vol. 70, 1974.
 14. Chenoweth, J. M. and Kissel, J. H., "Flow-Induced Vibration in Shell-and-Tube Heat Exchangers," SAN/1273-1, Feb. 1977.
 15. Timoshenko, S. P., Vibration Problems in Engineering, Van Nostrand, New York, 1955.
 16. Shin, Y. S., et al., "Effect of Tube/Support Interaction on the Vibration of a Tube on Multiple Supports," Argonne National Laboratory Report No. ANL-CT-77-5, Jan. 1977.
 17. Yung, D., Lorenz, J. J., and Ganic, E. N., "Vapor/Liquid Interaction and Entrainment in Shell-and-Tube Evaporators," ASME Winter Annual Meeting, San Francisco, CA, Dec. 1978.
 18. Connors, H. J., "Fluidelastic Vibration of Heat Exchanger Tube Arrays," ASME Paper No. 77-DET-90, 1977.

Table I. Comparison of Predictions With Experimental Data from HTRI and TEMA Data Banks (from Ref. [14])

Summary of Results for Cases with Gas as the Shell-Side Fluid.*

	Fluidelastic Instability, $\frac{V}{V_c} > 1$	Vortex Shedding, $\frac{f_v}{f_n} > 1$	Turbulent Buffeting, $\frac{f_t}{f_n} > 1$	Acoustic Excitation	
				Vortex Shedding $0.8 < \frac{f_v}{f_a} < 1.2$	Turbulent Buffeting $0.8 < \frac{f_t}{f_a} < 1.2$
<u>HTRI</u>					
Vibration	6/14	14/14	14/14	5/14	5/14
No Vibration	5/5	0/5	0/5	5/5	5/5
<u>TEMA</u>					
Vibration	4/4	4/4	4/4	-	-
No Vibration	2/6	2/6	0/6	-	-

*Results are expressed as:

number of correct predictions/number of cases considered

Table II. Wall Thicknesses for Aluminum and Titanium Tubes.

	Outer diameter cm (in.)	Wall thickness, cm (in.)
<u>Aluminum</u>	2.54 (1.00)	0.165 (0.065)
	3.81 (1.5)	0.218 (0.086)
	5.08 (2.0)	0.262 (0.103)
<u>Titanium</u>	2.54 (1.00)	0.071 (0.028)
	3.81 (1.5)	0.091 (0.036)
	5.08 (2.0)	0.122 (0.048)

Fig. 1 Strouhal number as a function of pitch-to-diameter ratio for different tube field layouts (from Ref. [14]).

Fig. 2 Definition of longitudinal and transverse pitch for turbulent buffeting model, Eq. (8).

Fig. 3 Tube span length as a function of crossflow velocity showing regions of tube vibration.

Fig. 4 Heat exchanger shell diameter as a function of crossflow velocity showing the region of acoustic noise.

Fig. 5 Heat exchanger shell diameter as a function of tube span length showing the region of acoustic resonance.

Fig. 6 Influence of tube diameter on the maximum allowable tube span length.

Fig. 7 Comparison of regions of acoustic noise for different tube diameters.

Fig. 8 Influence of tube diameter on acoustic resonance.

Fig. 9 Tube vibration map for $P/D = 1.5$ (compare with Fig. 3).

Fig. 10 Region of acoustic noise for $P/D = 1.5$ (compare with Fig. 4).

Fig. 11 Tube vibration map for aluminum (compare with titanium in Fig. 3).

Fig. 12 Region of acoustic resonance for aluminum (compare with titanium in Fig. 5).

Fig. 13 Tube vibration map for 45° configuration.

Fig. 14 Tube vibration map for 90° configuration.

Fig. 15 Tube vibration map for 60° configuration.

Fig. 16 Region of acoustic noise for 45° configuration.

Fig. 17 Region of acoustic noise for 90° configuration.

Fig. 18 Region of acoustic noise for 60° configuration.