~~FR-86-03-857~~
FR 8700747

COMMISSARIAT A L'ENERGIE ATOMIQUE

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CEA-CONF -- 8613

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Damping of Multispan Heat Exchanger Tubes

PART I: In Gases

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Communication présentée à : International conference on computers in engineering, pressure vessels and piping
Chicago, IL (USA)
20-24 Jul 1986

ABSTRACT

Flow-induced vibration analyses of heat exchanger tubes require the knowledge of damping. This paper treats the question of damping of multispan heat exchanger tubes in air and gases. The different energy dissipation mechanisms that contribute to tube damping are discussed. The available experimental data are reviewed and analysed. We find that the main damping mechanism in gases is friction between tube and tube-supports. Damping is strongly related to tube-support thickness. Damping values are recommended for design purposes.

NOMENCLATURE

d	=	tube diameter, mm
D_s	=	support diameter, mm
f	=	tube frequency, Hz
ℓ_t	=	tube length
ℓ_m	=	characteristic span length
L	=	support thickness, mm
N	=	number of spans
t	=	tube wall, mm
ζ	=	damping ratio, %
ζ_n	=	normalized damping ratio, %
ζ_{nc}	=	normalized damping ratio corrected, %

INTRODUCTION

Excessive flow-induced vibration may cause heat exchanger tube failures either by fatigue or by fretting-wear. To avoid such problems, a thorough flow-induced vibration analysis should be carried out at the design stage. This involves calculating the tube response to vibration excitation mechanisms, such as periodic wake-shedding and random turbulence, and the prediction of critical flow velocities for fluid-elastic instabilities. These calculations require a knowledge of tube damping.

Part 1 of this paper pertains to damping of multispan heat exchanger tubes with air or gas on the shell-side. Heat exchanger tubes with water or liquid on the shell-side are discussed in Part 2(1). The different energy dissipation mechanisms that contribute to tube damping are discussed. The available experimental data are reviewed and analysed. Damping values are recommended for design purposes.

ENERGY DISSIPATION MECHANISMS

As mentioned in (2), there are several possible energy dissipation mechanisms that contribute to tube damping, namely:

- internal or material damping,
- viscous damping between the tube and fluid,
- squeeze-film damping in the clearance between tubes and tube-supports,
- flow dependent damping due to fluid flow around the tube,
- damping due to fluid flow inside the tube,
- damping due to friction at the tube-to-tubesheet joints,
- damping due to friction between tubes and tube-supports, and
- energy dissipated by tube impact on the support and by the resulting travelling waves.

Material damping is very small in heat exchanger tubing. Haslinger and Thompson(3) measured damping ratios of roughly 0.01% on tubes welded or brazed at both ends with no intermediate support. For such welded tubes, material damping is dominant since the contribution from other energy dissipation mechanisms is minimal. Thus, the contribution of material damping may be neglected.

Since viscosity and density are very small in air and light gases, tube-to-fluid viscous damping and squeeze-film damping are not significant. These damping mechanisms may be appreciable in heavy gases. However, since we have no information at present on viscous damping and squeeze-film damping in heavy gases, it is conservative to ignore them. As discussed in (1), these energy dissipation mechanisms are very important in liquids.

Gas flow through a tube bundle will affect damping. This is difficult to study since it is difficult to separate energy dissipating fluid forces and fluid-dynamic excitation forces. Some researchers talk about negative damping at flow velocities close to critical flow velocities for fluid-elastic instability. In this paper, we assume that flow dependent damping is taken care of in the fluid-elastic instability phenomena since instability is related to the fluid-dynamic forces. However, flow dependent damping could affect the response to periodic wake-shedding or to random turbulence excitation.

Fluid flow inside a tube is unlikely to affect damping unless the flow velocity is very high (i.e., approximately that required for parallel flow fluid-elastic instability). Such internal flow velocities are not normally encountered in heat exchanger tubes. However, two-phase flow or sloshing due to partially filled tubes would contribute to damping. One investigator has studied this phenomenon(4). This source of damping depends very much on flow regime and void fraction. It would be premature to rely on the contribution of this energy dissipation mechanism until it is investigated thoroughly.

Some energy may be dissipated due to friction at the tube-to-tubesheet joints. This will obviously depend on the type of tube joint. Friction should be insignificant in welded joints as explained earlier. For a tube carefully clamped at both ends but without intermediate support, we have measured damping ratios of roughly 0.07% over a frequency range of 40 to 150 Hz(2). In a different experiment on a cantilever tube hydraulically expanded in one tubesheet, damping was roughly 0.2%(5). Haslinger and Thompson(3) measured somewhat less damping, 0.2% on large single span tubes with rolled joints at both ends. Measurements on bent tubes (i.e., large bends for thermal expansion) welded at both ends with an intermediate rolled joint, yielded damping values of 0.4 - 0.8%(6). The above damping data is outlined in Table 1. Thus, friction in the tube joints may contribute some damping depending on the type of joints. Since it is relatively small, it is prudent not to consider it at least until it is better understood.

It is now clear that the remaining mechanisms, friction (coulomb) damping together with impacting, are the most important energy dissipation mechanisms in gas heat exchangers. Most of this energy is dissipated between the tubes and the tube-supports. For this paper, we shall not distinguish between friction damping and the energy dissipated by impacts and the resulting travelling waves. The individual investigation of each of these two mechanisms is the subject of further work.

APPROACH

Heat exchanger tube dynamics is inherently a non-linear phenomenon since it depends largely on the dynamic interaction between tubes and tube-supports. This is particularly so for friction type damping which depends entirely on the relative motion between tubes and tube-supports. The analysis of this phenomenon would require a time domain non-linear simulation of the tube dynamics in which the details of sliding friction and impacting between tubes and tube-supports is modelled. Unfortunately, we are not yet at that stage, lacking the detailed information required to model the dynamic interaction between tubes and tube-supports. Furthermore, the required non-linear analysis is difficult and is not yet readily available to the designer. Some progress has been made in this area with the development of codes such as VIBIC(7) and TFDEL-FVICHOC(8) to predict fretting-wear of heat exchanger tubes. These codes could be adapted to study damping of heat exchanger tubes. This could be the subject of interesting future studies.

However, for the time being, we are limited to quasi-linear vibration analyses(9,10) for which we have to obtain an equivalent linear damping value. Values of damping may be obtained experimentally. However, most measurement techniques tend to underestimate non-linear effect since they mostly rely on the vibration being linear. This almost always results in an over-estimate of the damping value if the vibration is non-linear.

The objective of this paper is to recommend appropriate damping values based on available data. There are a few publications devoted to heat exchanger tube damping, i.e., References (11,12,13,14 and 15). In addition, there is some useful damping information buried in a number of other publications related to heat exchanger vibration. We have extracted data from nearly twenty-five references which cover many geometries and include several measurement methods. Most of this information is qualified as being tentative or preliminary. This is not surprising considering the non-linear nature of the problem and the difficulties inherent in measuring damping.

The damping data expressed as the damping ratio, , varied from 0.2 to 8%, which is not very useful information from a design viewpoint. Thus, the data was analysed to find trends and to outline the more relevant parameters. Damping values and important parameters such as tube frequency, f, diameter, d, support thickness, L, span length, diametral clearance between tube and support, D_s-d, wall thickness, t, tube-and-support materials, and number of spans, N, were computerized to facilitate the analysis. Our approach is to establish a conservative but realistic minimum damping criterion based on the analysis of the above data. We propose to take the lower decile of the available data as a minimum damping level. This is reasonable since the smaller damping values measured are usually the more reliable and since vibration problems are usually associated with the lower damping values. On the other hand, taking the lower decile , it is not unduly conservative. The higher damping values reported may have been due to anomalies such as crooked tubes or misaligned supports and are probably not representative of normal heat exchanger tubes.

PARAMETERS INFLUENCING DAMPING

As observed by Geyder(16), there are two principal types of tube motion at the support: rocking motion and lateral motion, as shown in Figure 1. Damping due to rocking motion is likely to be less and thus, may be more relevant in practice. Rocking type motion is predominant in the lower modes. The type of motion at a given support may be a combination of rocking motion and lateral motion.

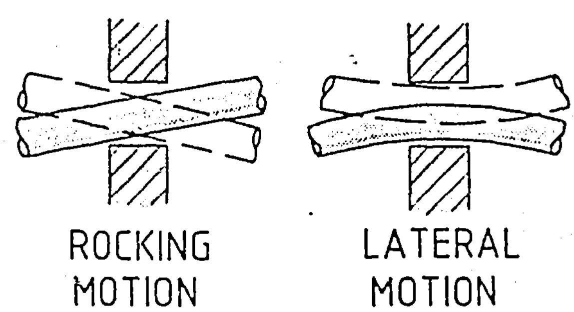

FIGURE 1: Type of Tube Motion at Support Location

The dynamic interaction between tube and tube-supports may be categorized in three main types, namely: sliding, impacting and scuffing which is impacting at an angle followed by sliding (see Figure 2). In practice, the dynamic interaction between tube and tube-support may be a combination of the above. The energy dissipated by sliding is due to friction and is related to the product of contact force and displacement. The energy dissipated by impact is in the form of local deformation of the support followed by stress wave propagation in the support and local deformation of the tube followed by high frequency travelling waves in the tube.

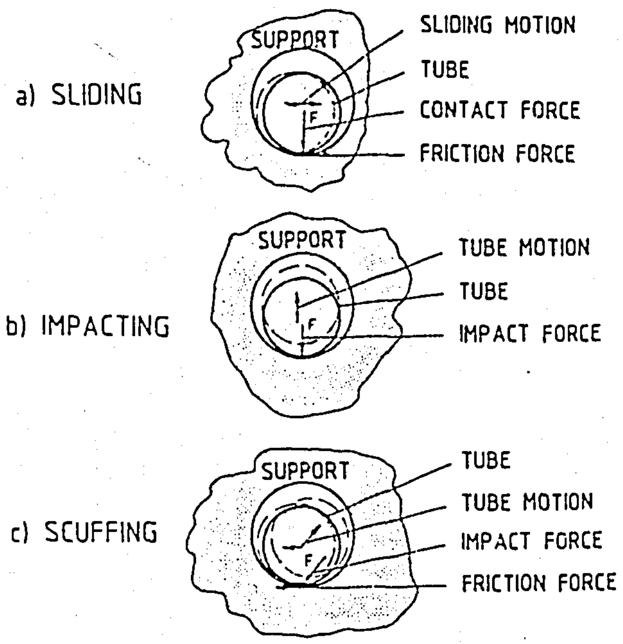

FIGURE 2: Type of Dynamic Interaction between Tube and Tube-Support

The available data was reviewed in an attempt to find trends or significant parameters as follows.

Effect of Number of Supports

Assuming that span length and all other tube parameters are kept constant, the total vibration energy in a tube is proportional to the number of spans. The energy dissipated by friction at the support is obviously related to the number of supports.

In a two-span heat exchanger tube with one support, there is the vibration energy of two spans but only one support to dissipate energy. Thus, damping should be less than for a tube with a large number of spans and a large number of supports. There appeared to be such a trend in the available damping data. Accordingly, the data was normalized such that:

$$\zeta_n = \zeta N/(N-1) \qquad (1)$$

where ζ_n is the normalized damping ratio and N is the number of spans. The normalized damping ratios are shown in Figure 3 against tube frequency.

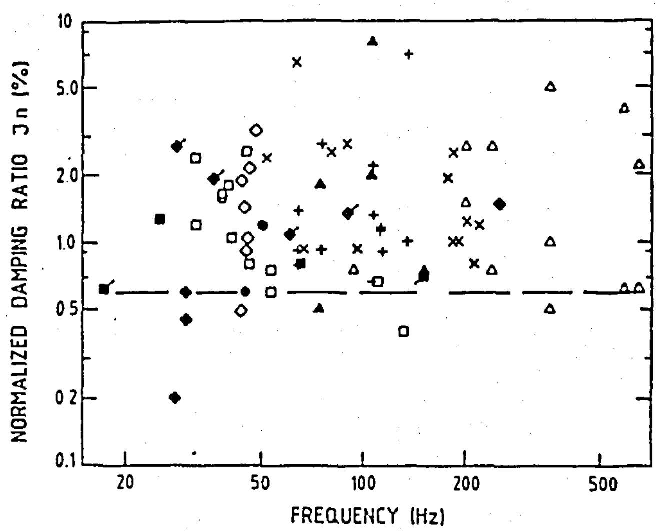

FIGURE 3: Damping of Heat Exchanger Tubes in Air. Symbols: ◘(12), ◆(13), ◇(14), ▲(16), △(17), X(18), +(19), ○(20,21), ◘(22), □(23), ◘(24), ●(25), ◆(26).

Effect of Frequency

From the data of Figure 3, it is not possible to establish a trend with frequency. There is no obvious reason why there should be a trend with frequency. Thus, frequency does not appear to be a significant parameter.

Vibration Amplitude

There is no conclusive trend of damping as a function of amplitude. Very often, the amplitude is not given with the damping measurements. Some researchers say that damping decreases with vibration amplitude. For sliding type damping, this would make sense since friction does not increase with amplitude as would be expected from linear damping. Thus, it would be reasonable to expect damping to decrease with amplitude.

Other researchers have found damping to increase with amplitude. This may be explained as follows. At low amplitude, the tube is in contact with the support and sliding type friction dominates. At higher amplitude, the tube may start rattling within the tube-support. This would cause impact type damping and non-linearities which would tend to increase damping. In practice, both sliding and impacting are possible and a definite trend of damping as a function of amplitude has not been established.

Effect of Diameter or Mass

Large and massive tubes should experience large friction forces and the energy dissipated should be large. However, the potential energy in the tube would also be proportionally greater in the more massive tubes. Thus, the damping ratio which is related to the ratio of energy dissipated per cycle to the potential energy in the tube should be independent of tube size or mass. No meaningful trend over the range of diameters 13 to 25 mm was observed although this parameter was not specifically studied in any of the publications reviewed.

Effect of Side Loads

In real heat exchangers, side loads are possible due to misalignment of the tube-supports or due to fluid drag forces. Side loads may increase or reduce damping. Small side loads may prevent impacting and thus, reduce damping, whereas large side loads may increase damping by increasing friction.

Goyder(17) measured damping ratios as low as 0.2% in an experiment on a carefully aligned tube with a single support. When the support was misaligned the damping increased to 0.8%. In both cases, the tube was impacting against the support in a rocking type of motion.

With lateral type motion, which is uncommon for low frequency modes of vibration, the reversed was observed. Damping increased from 1% for high side load to 8% for zero side load. This is because tube impacting with lateral motion tends to decrease with side load.

In practice and in the majority of the experiment reviewed here, misalignment and side load are not a controlled parameter. This probably explains the large scatter in the damping data. Unfortunately, the designer cannot take advantage of this parameter to increase damping. Therefore, minimum damping values should be used in design.

Effect of Higher Modes

Damping appears to decrease with mode order for mode order higher than the number of spans(13,18). This is not surprising since these higher order modes involve relatively less interaction between tube and tube-support. These higher modes are very rarely the cause of problems in practice.

Effect of Support Thickness

There is a clear indication in the reviewed data that support thickness is a dominant parameter. Two references(13,14) show that damping is roughly proportional to support thickness up to about 15 mm (see Figure 4). Beyond 15 mm, it is not clear that thicker supports increase damping. Blevins(13) found little increase in damping beyond 19 mm thickness.

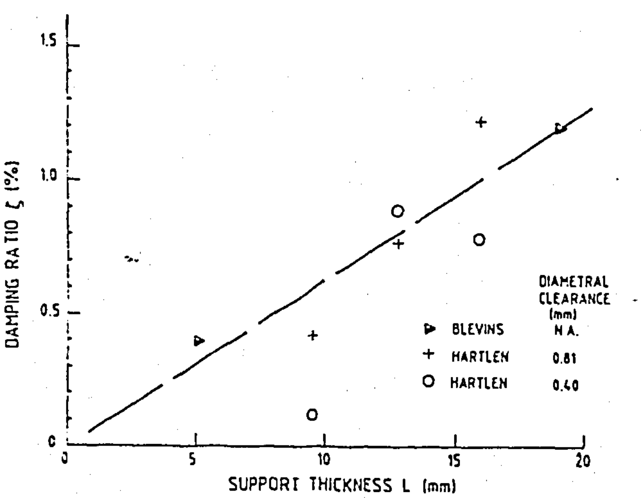

FIGURE 4: Effect of Support Thickness on Heat Exchanger Tube Damping in Air

As a first attempt to take support thickness into consideration, we corrected the damping data linearly for support width less than 12.7 mm such that

$$\zeta_{nc} = \zeta_n (12.7/L) \tag{2}$$

where L is the support thickness in mm and ζ_{nc} is the corrected normalized damping ratio. The value of 12.7 mm (0.5 in.) was chosen for convenience. Many heat exchanger tube-supports are this thickness or wider. The above relationship is illustrated in Figure 5 which shows damping as a function of support thickness. The corrected damping ratio ζ_{nc} is shown versus frequency in Figure 6. Already a reasonable design recommendation emerges from Figure 6. However, it is clear that additional data are required to establish the effect of support thickness with certainty.

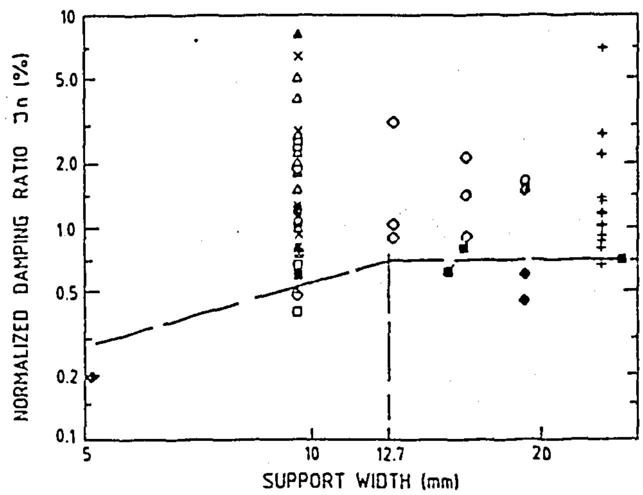

FIGURE 5: Damping Ratio vs. Support Thickness for all Available Data: Damping Correction Curve as a Function of Thickness. Symbols: ◪ (12), ◆(13), ◇(14), ▲(16), △(17), X(18), +(19), ○(20,21), ◼(22), □(23), ◩(24), ●(25), ◈(26).

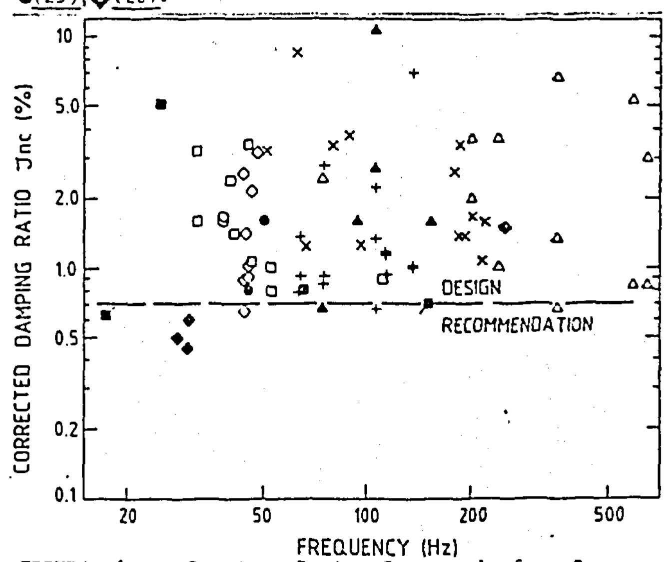

FIGURE 6: Damping Ratio Corrected for Support Thickness. Symbols: ◪(12), ◆(13), ◇(14), ▲(16), △(17), X(18), +(19), ○(20,21), ◼(22), □(23), ◩(24), ●(25), ◈(26).

The relationship with support thickness may be explained. The energy dissipated by friction is proportional to the product of contact force and displacement. The contact force should be independent of support thickness. However, the displacement should be proportional to support thickness for rocking type motion with sliding interaction with the tube-support, as shown in Figure 7a. Sliding interaction between tube and tube-support and rocking motion are expected for lower vibration amplitudes. These are the conditions of interest in practice.

Damping due to rocking type motion with impact interaction should also increase with support thickness. The impact forces are expected to increase with support thickness, as shown in Figure 7b. Damping due to lateral type motion, however, should not be affected by support thickness.

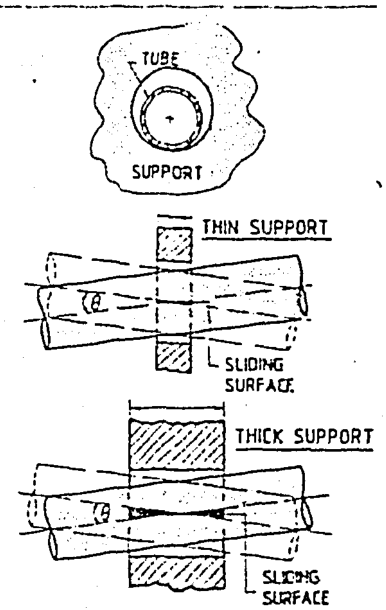

FIGURE 7a: Sliding Interaction: Effect of Support Thickness

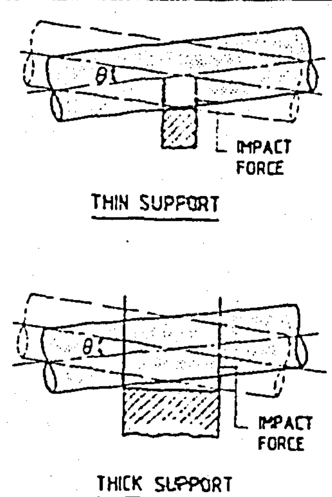

FIGURE 7b: Impacting Interaction: Effect of Support Thickness

The above relationship is simple but not very elegant. We tried more rigorous approaches to correlate damping with support thickness. We know that the total vibrating energy of a tube is related to its length, ℓ_t. On the other hand, the total energy dissipated in the supports is related to their thickness, L. Therefore, damping should be a function of the dimensionless number (L/ℓ_t). For convenience we use a characteristic span length, ℓ_m, to represent tube length. We take ℓ_m as the average of the three longest spans. This is based on the assumption that the lower modes, which involve primarily the longest spans, are the more vulnerable to vibration. Modes involving the shorter spans are less vulnerable since the ratio of energy dissipated at the support over the vibration energy in the tube (i.e., the damping) is larger. This is a crude attempt at weighing the effect of mode shape in the damping analyses.

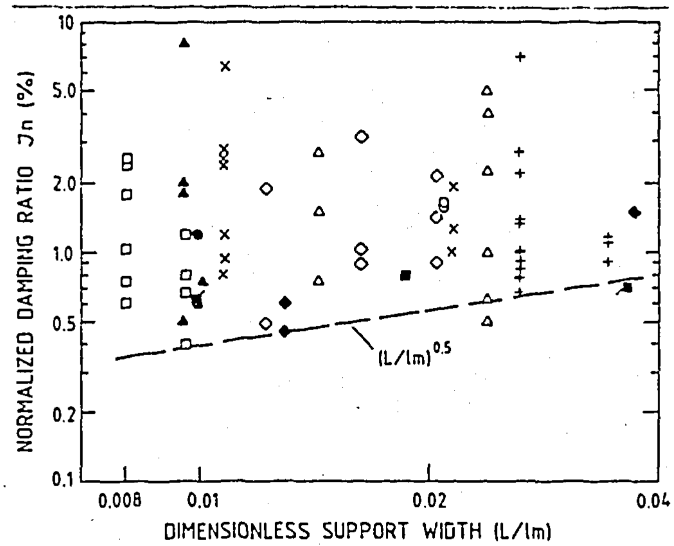

FIGURE 8: Effect of Dimensionless Support Thickness, (L/ℓ_m) on Damping. Symbols: ◘(12), ◆(13), ◇(14), ▲(16), △(17), X(18), +(19), ○(20,21), ◘(22), □(23), ◘(24), ●(25), ◇(26).

The normalized damping ratio is presented as a function of (L/ℓ_m) in Figure 8. It appears that an exponent of 0.5, i.e., $(L/\ell_m)^{0.5}$, best fits the minimum damping values. Here again the trend is not clear. Additional data are required. Figure 9 shows the term $\zeta_n\ (L/\ell_m)^{-0.5}$ against tube frequency. It suggests a minimum reasonable damping level.

Effect of Clearance

For the normal range of tube-to-tube-support diametral clearances (i.e., 0.40 to 0.80 mm), there is no conclusive trend in the reviewed damping data. For very small clearances, in the order of 0.20 mm(14), damping appears to be larger.

Although surprising at first, the relatively weak effect of clearance may be explained. A heat exchanger tube would normally touch most supports on one side or the other. Thus, the dynamic interaction between tube and support is taking place near one side and is not much affected by the proximity of the opposite side which depends on the diametral clearance. This is true in most cases except when the vibration amplitude is very large or the clearance is very small. Therefore,

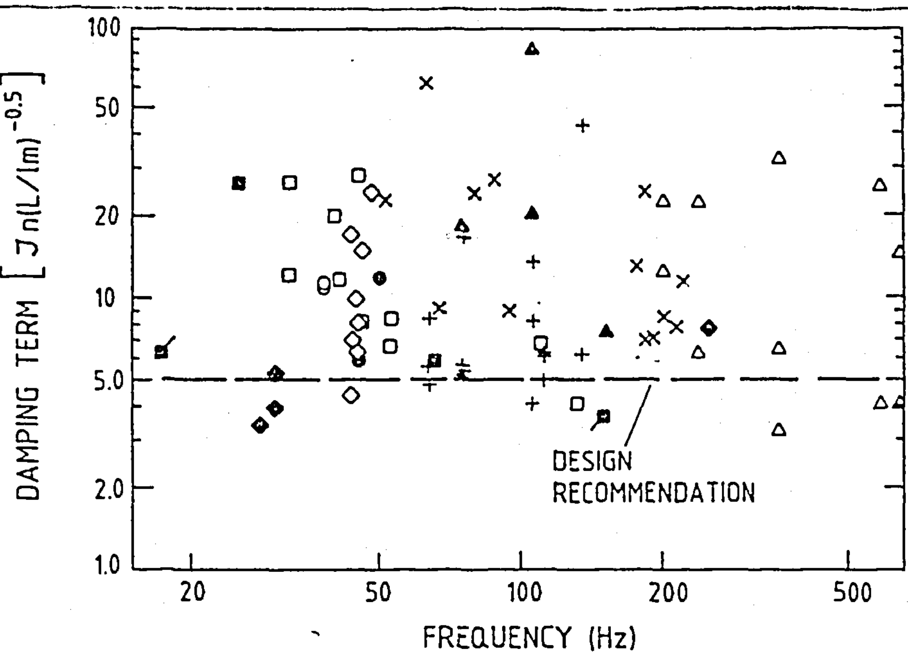

FIGURE 9: Damping Normalized for Support Thickness with Dimensionless Term $\zeta_n(L/\ell_m)^{-0.5}$. Symbols: ■(12), ◆(13), ◇(14), ▲(16), △(17), X(18), +(19), ○(20,21), ■(22), □(23), ■(24), ⊖(25), ◆(28).

damping should not be much affected by clearance. It is possible that side forces in the support be larger for very small clearances thus explaining the somewhat higher damping in the latter case.

DESIGN RECOMMENDATIONS

Following the above discussions, we have derived two expressions to evaluate damping. Our approach is to obtain a reasonable minimum damping value. This is achieved by taking the damping level at roughly the lower decile of the available damping data as already explained.

The first expression is derived from Figure 6. This expression takes into account the number of spans and the support thickness. The damping ratio (in percent) of a heat exchanger tube is given by:

$$\zeta = 0.7\left(\frac{N-1}{N}\right)\left(\frac{L}{12.7}\right) \qquad \text{if } L < 12.7 \text{ mm}$$

$$\zeta = 0.7\left(\frac{N-1}{N}\right) \qquad \text{if } L \geq 12.7 \text{ mm} \qquad (3)$$

This is a simple expression based on the results of two researchers(13,14) who observed that damping is proportional to support thickness up to roughly 15 mm as illustrated in Figure 4. Although not very rigorous nor elegant, this expression is practical and satisfies the available data reasonably well. Let us illustrate its application with examples. Suppose the hypothetical tube No. 1 for which N=5, L=15 mm, and ℓ_m=0.6 m. For this tube ζ=0.58%. Suppose now tube No. 2 for which N=10, L=10 mm, and ℓ_m=0.6 m. For this second tube ζ=0.4967.

The second expression is more elegant. It is also based on the dominant effect of support thickness. It is derived from Figures 8 and 9, that is

$$\zeta = 5\left(\frac{N-1}{N}\right)\left(\frac{L}{\ell_m}\right)^{0.5} \qquad (4)$$

This expression also fits the minimum damping data reasonably well. The available data are not sufficient to choose one expression over the other. However, we recommend the second expression for design application since it is more rigorous and less conservative in many cases. It would give for hypothetical tube No. 1, ζ=0.63% and for No. 2, ζ=0.58%.

The above expressions apply to the normal range of heat exchanger tubes, that is: diameters from 12 to 25 mm, support thicknesses from 6 to 25 mm, diametral clearances from 0.4 to 0.8 mm, and frequencies from 20 to 600 Hz. This range of parameters corresponds to the available damping data.

The above expressions are somewhat tentative. Fundamental work is required to establish the trends with more certainty and to confirm the form of the expressions. For example, the effect of clearance, the effect of tube and support materials, the contribution of impact damping and the effect of support thickness, in particular narrow supports, should be studied. Additional measurements on realistic heat exchanger tube configurations are necessary to validate the damping expressions from a statistical point-of-view. However, the above expressions provide a practical and reasonable minimum damping criterion based on the currently available data.

CONCLUSION

Expressions are recommended to evaluate damping of heat exchanger tubes with gases on the shell-side. Support thickness is a dominant parameter whereas diametral clearance between tube and support is much less important. Other parameters such as frequency, mass and diameter do not appear relevant.

ACKNOWLEDGEMENTS

The authors have benefited from discussions with R.T. Hartlen, Ontario Hydro, and R.J. Roger, University of New Brunswick. Their contributions are gratefully acknowledged. This work was done while the first author was a visiting scientist at the Centre d'Etudes Nucléaires de Saclay, France. The Heat Transfer and Fluid Flow Service (HTFS) contributed also to this work. The support of these two organizations is much appreciated.

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TABLE 1: Damping in Tube-to-Tubesheet Joints

f (Hz)	dia. (mm)	Wall (mm)	Span Length Configuration	Mat'l Tube/Tube-sheet	Amp. (μm)	Method	Mode No.	Damping ζ(%)	Reference	Comments
60 - 153	15.9	1.39	0.30-1.33 m	Inconel 600/ Brass	100	Log Dec	1	0.07	(2)	Clamped joints
33	22.0	1.07	←600→	St. St. 304/ Carbon Steel		Log Dec Random	1	0.20	(5)	Hydraulic expansion join
762	16.9	3.45	325	St. St. 304		Log Dec Sweep	1	0.011	(3)	Brazed joint, single tube
220 - 798	15.9	3.45						0.004 - 0.009		Brazed joint tube bundles
123 - 139	44	13	1.5 m	St. St. 304			1	0.026 (0.020 - 0.034)	(3)	Rolled joint
14.5	25			Inconel 800	300	Log Dec Random	1	0.04-0.08	(6)	Welded joints Intermediate rolled joint