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LETTERS TO THE EDITOR

ASSESSMENT OF FLUID-INDUCED DAMPING IN REFRIGERATION MACHINERY MANIFOLDS

1. INTRODUCTION

The attenuation of small amplitude plane acoustic waves in a fluid at rest, contained in a tube with rigid walls, due to the viscous and heat conduction losses at the boundary, is well understood and can be predicted theoretically. In the presence of mean fluid flow, however, and with higher acoustic amplitudes, additional losses can occur [1–5]. These may be due to the finite amplitudes and/or acoustic/mean flow interaction.

In high speed refrigeration compressor (positive displacement type) manifolds, the situation is further complicated by the following:

(i) the flow medium is always laden with lubricating oil vapors; it is often not possible to assess oil circulation rates precisely, and thus the prediction of thermodynamic properties is difficult;

(ii) one often encounters irregular orifice-like elements, such as connecting passages, in the manifolds; these may exhibit large dissipation characteristics.

Because of the complexity of the nature of flow-induced damping and unavailability of mathematical models (most of which have been developed with air as the medium [1–5]), experimental means must be adopted to assess the damping values.

2. ASSESSMENT

For a unidirectional plane wave, the pressure $p(x,t)$ is

$$p(x,t) = (P_R\,e^{-\alpha x}\,e^{-j2\pi f x/c} + P_L\,e^{+\alpha x}\,e^{j2\pi f x/c})\,e^{j2\pi f t}, \qquad (1)$$

where P_R and P_L are amplitudes for right-hand and left-hand traveling waves, respectively, x is the longitudinal co-ordinate, t is time, f is the frequency, c is the sonic speed and α is the damping factor which characterizes the wave attenuation per unit path length.

Figure 1 shows schematically the discharge manifold of a reciprocating refrigeration compressor. Some theoretical and experimental pressure energy spectra ($20\log_{10}|p(nf_e)|$,

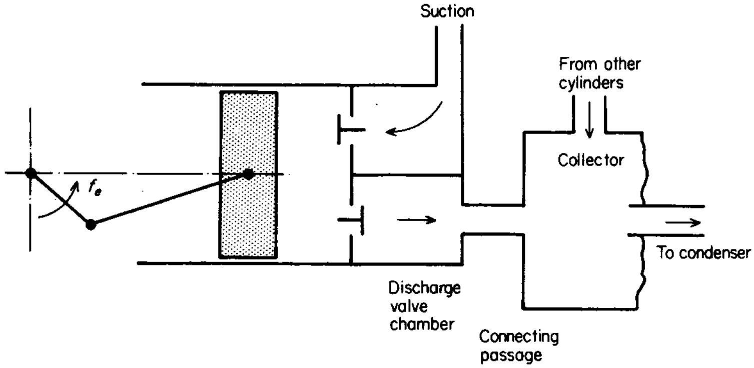

Figure 1. Schematic of a reciprocating compressor discharge manifold.

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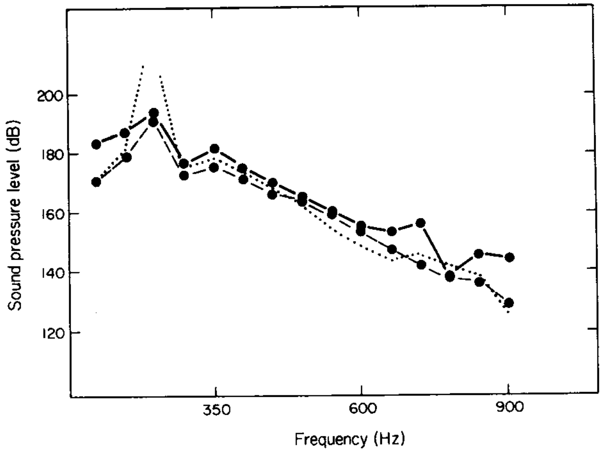

Figure 2. Discharge valve chamber pressure line spectra, given at the harmonics of 60 Hz. The reference pressure is 2×10^{-5} Pa. ——, measured; · · · · · · ·, computed with viscous and thermal wall damping ($\zeta = 1$); – – – –, computed with measured empirical damping coefficient ($\zeta = 150$).

where n is the harmonic index and f_e is the running speed (60 Hz)) in the valve chamber are shown in Figure 2.

For modeling purposes, the measured damping factor α is assumed to be proportional to the theoretically predicted damping factor due to the viscous and thermal dissipation at the tube walls, α_μ [2]:

$$\alpha = \zeta\alpha_\mu. \tag{2}$$

Here ζ is an empirical non-dimensional coefficient. Note that α_μ itself is small, numerically, except for long and narrow tubes, and at high frequencies. Its value is, where μ is the fluid viscosity, ρ is the density and D is the tube diameter,

$$\alpha_\mu = (4\pi\mu f/\rho c^2 D^2)^{1/2}. \tag{3}$$

In the absence of mean fluid flow (e.g., in comparing theory with static bench tests), $\zeta = 1$, i.e., $\alpha = \alpha_\mu$, has been found to be adequate [6]. For the refrigerant fluid R-12 as a medium at $12{\cdot}6 \times 10^5$ Pa pressure and 73°C temperature, α_μ is

$$\alpha_\mu D = 4{\cdot}26 \times 10^{-6}\sqrt{f}, \tag{4a}$$

or the attenuation in decibels is

$$8{\cdot}6\,\alpha_\mu D = 8{\cdot}6 \times 4{\cdot}26 \times 10^{-6}\sqrt{f} = 0{\cdot}00037\sqrt{f}\,\text{dB}. \tag{4b}$$

However, in an operating compressor discharge manifold (Mach number approximately 0·1), $8{\cdot}6\,\alpha D$ has been found to be

$$8{\cdot}6\,\alpha D = 0{\cdot}0055\sqrt{f}\,\text{dB}, \tag{5a}$$

or

$$\zeta = 150. \tag{5b}$$

Since the damping has significance only at a resonance, off-resonance values can be predicted with accuracy without using any empirical damping coefficient, as illustrated in Figure 2. The third harmonic (180 Hz) is in resonance as it is close to a natural frequency which is computed to be 189 Hz. The natural frequency calculation is very sensitive to the dimensions of the connecting passage, which behaves like an orifice. The empirical damping value is applied only to the connecting passage acoustic element; for other elements, linear damping values have been found to be sufficient.

Although only one frequency point is presented here, the authors have used this value for numerous other cases satisfactorily. This is not to say that it will apply to all refrigeration compressors.

3. CONCLUSIONS

The results of the present investigation are compared with those of Ingard and Singhal [4], and Kuhn and Morfey [5] in Table 1. On the basis of these limited investigations, the situation can be summarized as follows:

TABLE 1

Comparison of attenuation coefficients for a circular tube (Mach number ≈ 0·1)

Investigation	Fluid medium	D (mm)	$\alpha_\mu D$ (dB)	αD Attenuation (dB)	αD Frequency	Comment
Ingard and Singhal [4]	air	20	0·00048 $\sqrt{f}$	0·0375 (upstream) 0·0275 (downstream)	independent of frequency	Prediction based on quasi-static perturbation model
Kuhn and Morfey [5]	air	30	0·00048 $\sqrt{f}$	0·0075 0·012 0·012 0·04 0·11	0·5 kHz 1 kHz 2 kHz 4 kHz 8 kHz (octave bands)	Experimental
Present	R-12	11	0·000037 $\sqrt{f}$	0·0055 $\sqrt{f}$	50–1500 Hz	*In-situ* measurement on a refrigeration compressor

(i) damping values in the refrigeration machines are substantially higher than those measured, on a bench test, for similar systems but without mean flow;

(ii) The measured damping in the presence of mean flow is dependent upon frequency; this is in agreement with the results of Ahrens and Ronneberger [3], and of Kuhn and Morfey [5], but is in conflict with those of Ingard and Singhal [4].

The authors believe that a need exists for a more intensive investigation in this area, especially for those fluid media which exhibit properties different from those of air.

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