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Acoustic pressure losses in woven screen regenerators

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ABSTRACT

In thermoacoustic travelling-wave engines and other Stirling cycle devices, good performance depends on the material of a regenerator being in intimate contact with the gas inside it, so that each particle of gas oscillates in temperature following the adjacent material as it is acoustically displaced. This requires that the passages are small enough for temperature waves to penetrate across the gas path with the frequencies of interest. One type of 'regenerator' that is commonly used for this purpose is composed of multiple layers of woven stainless steel mesh, laid on top of one another in random registration. Associated with the thermal penetration is a viscous loss of pressure and this must be quantified if efficient engines are to be designed.

In the literature, reliance has been placed on the correlation of steady-flow loss data for these meshes, but for the coarser ones operating at frequencies greater than 28 Hz, the assumption of quasi steady-flow is dubious and direct acoustic measurements must be made. This paper reports acoustic pressure loss data for meshes with 34 and 75 wires per inch taken in two configurations of impedance tube, and finds that the dependence on velocity is the same as in steady-flow, but that there is indeed some enhancement of loss for frequencies above 40 Hz. (Separation of the mesh layers is probably responsible for the anomalously low loss coefficients that were recorded in one set of data.) It is shown that the acoustic pressure losses can be correlated in terms that give the acoustic impedance more directly than the friction factor correlations.

1. Introduction

In the performance of thermoacoustic heat engines, the characteristics of the material in the regenerator are of paramount importance [1]. A temperature gradient is imposed on the regenerator by heat exchangers at each end and the function of the material is to ensure that acoustic waves passing through it are in thermal contact to some extent.

In a 'travelling wave' engine, although the wave may be far from a pure travelling one, the contact should be intimate so that all the particles of gas follow the temperature of the adjacent material, and thus undergo a Stirling cycle as the gas is displaced alternately towards the hot end (while compressed) and the cold end (while expanded). This requires that the frequency is low enough for temperature waves to penetrate across the gas path, and hence (even for moderate frequencies) that the gas passages be very small. However, this must be achieved without incurring so much viscous or thermal dissipation that the work input to the acoustic wave is negated. Depending on the acoustic conditions, both of the mechanisms may be important, but in a high impedance regenerator, viscous dissipation tends to dominate. Thus, it is essential to know the acoustic pressure loss characteristics of the material so as to be able to quantify the viscous dissipation.

One type of 'regenerator' that is commonly used in travelling wave devices is composed of multiple layers of woven stainless steel mesh, laid on top of one another in random orientation. This note describes measurements of the acoustic characteristics of a 10 mm thickness of two different mesh layers, and compares the performance with the correlation presented by Swift and Ward [2]. The correlation was derived from the steady-flow data quoted by Kays and London [3] from the experiments of Tong and London [4]. It has proved its relevance to thermoacoustic conditions through incorporation in engine calculations. However, with an acoustic wave there must be differential acceleration of different regions of the gas at the microscopic level within the mesh, so it would not be expected that the steady-flow data can be applied precisely. Direct measurements of the acoustic losses are presented here, thus contributing to the filling of a significant gap in the literature.

2. Pressure loss coefficient measurements

Schematic diagrams of two realizations of the apparatus are shown in Fig. 1. They were both excited by two Pioneer sub-woofer

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Nomenclature

A	face area of regenerator
C_p	thermal capacity
G	peak volumetric flow rate at regenerator mid-point
K''	dimensional loss coefficient
L	regenerator depth
L_{eff}	effective depth of regenerator including end-corrections for gas inertia
LC	dimensionless pressure loss coefficient
P	mean pressure
Pr	Prandtl number
Re	Reynolds number based on fixed length
T	mean temperature
T'_a, T_a	acoustic temperature, averaged across the passage
U_{pore}	mean velocity in the pores of the mesh, based on porosity
V_{pk}	peak voltage driving the sub-woofer amplifiers
Z	acoustic impedance (p/u)
c	speed of sound
$c_1(\varphi), c_2(\varphi)$	parameters in the correlation of Swift and Ward [2]
de	equivalent diameter of the gas passage
f	Fanning friction factor, defined by Eq. (1)
f_v	Rott's acoustic parameter obtained by integrating h_v across the passage
f_k	Rott's acoustic parameter obtained by integrating h_k across the passage
$h_v(y, z)$	function describing the variation of the acoustic velocity field
$h_k(y, z)$	function describing the variation of the acoustic temperature field
k	thermal conductivity
p	acoustic pressure, or pitch of woven wire mesh
p_a	acoustic pressure
r_h	hydraulic radius
t	time, or thickness of wire in the mesh
u', u	local acoustic velocity, averaged across the passage
u_{pore}	notional acoustic velocity in the pores of the mesh, based on porosity, at the mid-point of the regenerator
x	axial distance
y_0	channel half-width
y, z	transverse distances
α	thermal diffusivity
γ	ratio of specific heat capacities
δ_v, δ_k	viscous, thermal penetration distances
ζ	specific acoustic impedance $\equiv z/\rho c$
ϑ	parameter in loss coefficient correlation
$\vartheta_{scale}, \vartheta_{vel}$	parameters in proposed variation of ϑ
μ	dynamic viscosity
ν	kinematic viscosity
ρ	mean density of gas
ρ'_a, ρ_a	acoustic fluctuation in the density of the gas, averaged across the passage
τ_{diff}	time constant of diffusion
ϕ	porosity
ω	circular frequency

Subscripts

1–7	position of microphones in Fig. 1

loudspeakers, rated at 1 kW each, mounted in an MDF enclosure so that the majority of the sound emanated from a 250 mm diameter orifice in the side of the enclosure. In the first series of experiments, the sound then propagated through 530 mm of a horizontal 300 mm diameter steel duct, halfway along which was mounted the mesh assembly, before exhausting to the laboratory. The assembly was a 10 mm sandwich of layers of plain weave mesh between two supporting layers of coarse grid. These coarse grids were required to prevent the layers of mesh under test from sagging or springing apart, and they were assumed to make a negligible contribution to the pressure loss. In the second series of experiments, suggested by one of the referees of the first draft of this paper, the duct was extended by 200 mm and the sandwiches of mesh were installed 20 mm from the open end. A second 10 mm sandwich of coarse mesh was placed near the orifice of the enclosure, to enhance the radial uniformity of the acoustic field.

2.1. Technique for the first series

The positions of four 6 mm holes in the 2 mm thick stainless steel pipe are also shown in Fig. 1a. The tube mountings communicating with these holes were designed to support 13 mm B&K microphones. The calibration for these microphones was obtained before each experiment with a 1 kHz source giving 10 Pa, and assumed to apply at the lower frequencies of the experiment, on the basis of the manufacturers' data.

In the first series of experiments, driven by the need for high accuracy in the phases in particular, pressure ratios and phase relationships were recorded using the same microphone. To obtain the phase relationships between positions, the microphone signals were each referenced to the current of the sub-woofer driving signal. Using the ratios of magnitude and differences of phase, the two-microphone method [5] was then applied to deduce the acoustic velocities. Thus, the velocities at stations 1 and 2 for example were given by

$$u_1 = \frac{-p_2 + p_1 \cos\left(\frac{\omega x}{c}\right)}{i\rho c \sin\left(\frac{\omega x}{c}\right)},$$

and

$$u_2 = -i\frac{\sin\left(\frac{\omega x}{c}\right)}{\rho c}p_1 + \cos\left(\frac{\omega x}{c}\right)u_1,$$

where x is the separation distance, ω the circular frequency, c the speed of sound, and ρ the density of air at ambient conditions.

Pressures and velocities at stations 1 and 2, and also 4 and 5, were extrapolated to the centre of the mesh, also using the pipe transmission equations. From these data, the pressure loss Δp and the velocity in the middle of the mesh, before its correction for the porosity to give u_{pore}, were deduced. The velocities deduced by extrapolation from stations 1 and 2 were 5–20% greater than those from 4 and 5. The value from the loudspeaker side was taken to be representative in working out the loss coefficient because the larger phase differences there should yield higher accuracy.

In collecting the data, the voltage generated by the amplifier driving the sub-woofers was fixed. The acoustic wave incident on the regenerator was varied by varying the frequency from 112 Hz down to 28 Hz, the range of interest in the present thermoacoustic studies. A second and higher voltage was then used to provide overlapping and larger values on the velocity scale. Electrical powers up to 500 W were employed, but the maximum transmitted acoustic power in the duct was only about 1 W, because a predominantly standing wave was generated. Typical deduced profiles of acoustic pressure and velocity are shown in Fig. 2.