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Development of experimental techniques for measurement of heat transfer rates in heat exchangers in oscillatory flows

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ABSTRACT

Heat exchangers are important components of thermoacoustic devices. In oscillatory flow conditions, the flow and temperature fields around the heat exchangers can be quite complex, and may significantly affect heat transfer behaviour. As a result, one cannot directly apply the heat transfer correlations for steady flows to the design of heat exchangers for oscillatory flows. The fundamental knowledge of heat transfer in oscillatory flows, however, is still not well-established. The aim of the current work is to develop experimental apparatus and measurement techniques for the study of heat transfer in oscillatory flows. The heat transferred between two heat exchangers forming a couple was measured over a range of testing conditions. Three couples of finned-tube heat exchangers with different fin spacing were selected for the experiment. The main parameters considered were fin spacing, fin length, thermal penetration depth and gas displacement amplitude. Their effects on the heat exchanger performance were studied. The results were summarised and analysed in terms of heat transfer rate and dimensionless heat transfer coefficient: Colburn-j factor. In order to obtain the gas side heat transfer coefficient in oscillatory flows, the water side heat transfer coefficient is required. Thus, an experimental apparatus for unidirectional steady test was also developed and a calculation method to evaluate the heat transfer coefficient was demonstrated. The uncertainties associated with the measurement of heat transfer rate were also considered.

1. Introduction and literature review

In thermoacoustic heat engines, heat as an input energy is supplied from a high temperature source through a hot heat exchanger and waste heat is rejected to a heat sink with low temperature. The presence of the imposed steep temperature gradient in a solid structure called a stack or a regenerator sandwiched between the two heat exchangers produces acoustic power. In thermoacoustic refrigerators, heat is removed from where desired via a cold heat exchanger, transported via a stack or regenerator by supplied acoustic power, and rejected to the heat sink in an ambient heat exchanger. A simple standing wave thermoacoustic refrigerator and a schematic of thermoacoustic effect are illustrated in Fig. 1. The main components in the device as shown in Fig. 1(a) are an acoustic driver, a stack and two heat exchangers placed in the resonator. The driver sets up a half-wavelength acoustic field. This induces an oscillation of fluid elements in the vicinity of the stack and the heat exchangers.

A thermodynamic process takes place as shown in Fig. 1(b) due to expansion and contraction of fluid elements during their displacement cycle. The gas parcel at its largest volume moves leftward while simultaneously experiencing compression. At the leftmost position, it rejects heat to the stack as its temperature is raised above that of the local surface. This results in a decrease of the gas parcel's temperature which is subjected to the thermal contraction under high pressure. When the parcel moves rightward, it experiences adiabatic expansion enlarging its volume and decreasing its temperature below that of the local surface. At the rightmost position, the irreversible heat transfer takes place from the stack plate to the gas parcel causing the expansion of gas parcel volume and the rise of its temperature to the initial condition. All gas parcels behave as a 'bucket brigade' resulting in absorbing heat $\dot{Q}_c$ at temperature T_c and rejecting heat $\dot{Q}_{amb}$ at temperature T_{amb}. For a standing wave thermoacoustic engine, the process can be described in similar manner as given above for the refrigerator. However, the direction of thermodynamic cycle would be reversed.

As indicated, thermal interaction between working fluid and heat exchangers is crucial for the performance of thermoacoustic

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Nomenclature

A_{min}	heat exchanger minimum air flow area (m^2)
A_f	fin surface area (m^2)
A_{fc}	fin cross sectional area (m^2)
$A_{t,i}$	water side heat transfer area (m^2)
A_o	gas side heat transfer area (m^2)
$A_{t,o}$	unfinned area on copper tube (m^2)
a	sound speed (ms^{-1})
b	systematic uncertainty
c_p	isobaric specific heat (kJ kg^{-1} K^{-1})
D	fin spacing (m)
D_h	fin spacing hydraulic diameter (m)
D_i	tube inside diameter (m)
D_o	tube outside diameter (m)
F	cross flow correction factor
f	frequency (Hz)
f_s	friction factor
g	half of the gap between hot and cold heat exchanger faces (mm)
HEX	hot heat exchanger
h	heat transfer coefficient (W m^{-2} K^{-1})
j	Colburn-j factor
k	thermal conductivity (W m^{-1} K^{-1})
k_w	wave number (m^{-1})
L	fin length (m)
L_c	corrected fin length (m)
L_T	effective length of copper tube (m)
$LMTD$	log-mean temperature difference (K)
$\dot{m}$	mass flow rate (kg s^{-1})
Nu	Nusselt number
P_a	acoustic pressure amplitude (Pa)
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (W)
R_{tube}	thermal resistance of tube material (m^2 K W^{-1})
Re	Reynolds number
Re_1	acoustic Reynolds number
r_1	inside radius of copper tube (m)
r_2	outside radius of copper tube (m)
s	precision uncertainty
T	temperature (°C)
t	fin thickness (m)
u_1	velocity amplitude (m s^{-1})
V	flow velocity (m s^{-1})
Va	Valensi number
x	distance from pressure anti-node to heat exchanger (m)

Greek symbols

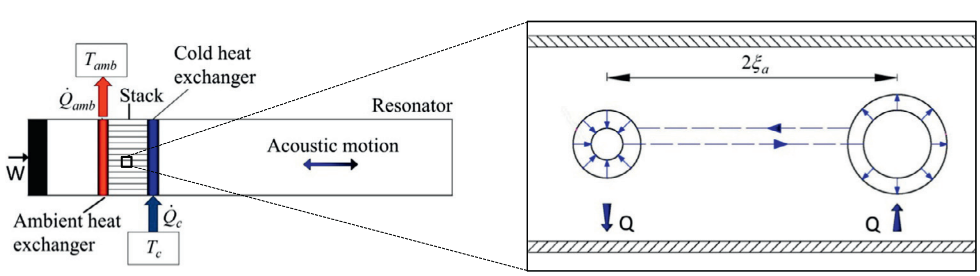
δ_k	thermal penetration depth (m)
ε_s	surface roughness (m)
η_f	single fin efficiency
η_o	overall fin efficiency
λ	wave length (m)
μ	dynamic viscosity (Pa s)
ξ_a	displacement amplitude (m)
Π	fin cross sectional perimeter (m)
ρ_m	mean density (kg m^{-3})
σ	heat exchanger porosity
ω	angular frequency (rad s^{-1})

Subscripts

a	air
amb	ambient
avg	average
c	cold
est	estimation
exp	experiment
i	inlet
max	maximum
OSC	oscillatory flow
o	outlet
STD	steady flow
w	water

devices. Experimental works have been carried out in order to study the heat transfer in oscillatory flow. Temperature profiles at the interface of heated solid surface and oscillating gas were observed by Bouvier et al. [3]. Surface heat flux was determined from the temperature profiles measured on a test section of circular tube. The heat transfer characteristics were analysed as a function of acoustic Reynolds number ($Re_1 = \rho u_1 d/\mu$). Here ρ, μ, d and u_1 are density, dynamic viscosity of working gas, internal diameter of the tube and velocity amplitude, respectively. Mozurkewich [17] carried out tests on simple heat exchanger configurations consisting of parallel tubes located transversely adjacent to the hot end of a thermoacoustic stack. The experimental results were presented in terms of dimensionless heat transfer coefficient ($NuPr^{-0.37}$) as a function of the acoustic Reynolds number (Re_1). It was found that the dimensionless heat transfer coefficient increased with the increase of Re_1. A correlation was also developed to predict the heat transfer coefficient in thermoacoustic heat exchangers.

Fabrication of heat exchangers used in thermoacoustic refrigerators was demonstrated by Garrett et al. [9]. Finned-tube heat exchangers were made and a simple derivation to evaluate their heat transfer performance was reported. Heat transfer coefficient was represented by root mean square value of the ratio of gas



(a) (b)

Fig. 1. Standing wave thermoacoustic refrigerator (a), and a magnified view of thermoacoustic effect taking place in a channel of the stack (b).