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Effect of non-linear flow behavior on heat transfer in a thermoacoustic engine core

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ABSTRACT

The non-linear behavior of the temperature field in a thermoacoustic engine core is explored using computational fluid dynamics (CFD) simulations; the effect of that behavior on heat transfer is estimated. With respect to heat transfer in a thermoacoustic core (TAC), the unsteady behavior of this temperature field and its influence has not been discussed sufficiently so far. In the present study, to understand this non-linear behavior in oscillatory flows, both CFD simulation and numerical heat transfer analysis, which is combined with standard thermoacoustic linear theory, are performed. The simulated environment is a standing-wave acoustic field in a straight-channel thermoacoustic device. With a comparison of the CFD and heat transfer analyses, differences in the temperature field behavior are discussed. Whereas the acoustic field is sinusoidal in the TAC for both calculations, only the CFD result shows non-linear behavior in the unsteady temperature field. This arises from the interaction between the fluid motion and the fluid temperature, which varies spatially in the streamwise direction. This feature reflects the heat flux on the walls of the heat exchanger. Ultimately, this effect causes around 10% of the difference in estimating the heat transfer in the TAC.

1. Introduction

Since Swift et al. [1] demonstrated its feasibility, the thermoacoustic device has been promoted for its applications in renewable heat energy and in recent years is also attracting attention in regard to CO_2 gas emission reductions. In particular, the rapid advances in thermoacoustic technology have prompted practical applications for both the thermoacoustic engine and refrigerator. In these devices, the thermoacoustic phenomena produce or amplify acoustic power; such sources are essentially induced by temperature gradients in the fluids in engine units. In the thermoacoustic linear theory developed by Rott [2,3] and Tijdeman [4], the acoustic power can be calculated only if the temperature gradient is given for a TAC. From this point, their theory is quite useful for thermoacoustic device design and is applied to actual developments of devices. However, the actual temperature gradient is achieved by thermal interaction between working fluid and heat exchangers. Although an accurate estimate of the heat transfer in heat exchanges is the key to development and design of thermoacoustic device, its mechanism is still not fully understood.

In addition, correctly estimating the amount of heat input/output within the engine is crucial in this kind of research. From this point of view, constructing the heat transfer model plays an important role. Many of the concepts used in the study of heat transfer in oscillatory flows assume steady-flow conditions. Poese and Garrett [5] proposed one such concept, specifically, modifications of standard laminar flow for convective heat transfer using the time-averaged velocity over half a period in the sinusoidal oscillation. Swift [6,7] suggested that a root-mean-square (rms) Reynolds number model is applicable in estimating heat transfer. This is the correlation model obtained by substituting the rms acoustic Reynolds number into the steady-flow correlation. Another concept is based on the time-averaged steady flow equivalent (TASFE) approximation [8]. Zukauskas [9] applied this approximation to a steady cross-flow correlation of a single tube. In the TASFE approximation for oscillatory flow, the steady-flow heat transfer correlation of forced convection is employed through a period-averaged velocity of sinusoidal acoustic oscillation. Furthermore, with respect to the design environment for low-amplitude thermoacoustic engines (DELTAE), which was developed by Swift et al. [10], a boundary layer conduction model is employed. In this model, the coefficient of heat transfer is defined by the thermal conductivity divided by the minimum value of the hydraulic radius of the heat exchanger spacing and the thermal penetration depth.

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Nomenclature

ρ	density [kg/m^3]
u	flow velocity [m/s]
$\langle u \rangle$	velocity amplitude [m/s]
p	absolute pressure [Pa]
$\langle \mathrm{p} \rangle$	pressure amplitude [Pa]
φ	phase of acoustic wave [–]
T	absolute temperature [K]
n_i	normal vector of surface element [–]
t	time [s]
Δt	period of acoustic wave [s]
f	frequency of acoustic wave [Hz]
$\mathscr{V}$	volume element of control volume [–]
S	surface element of control volume [–]
A	area of cross section [m^2]
Br	blockage ratio of engine unit [–]
R	gas constant [J/kg K]
τ	viscous stress tensor [Pa]
q	heat flux vector [W/m^2]
μ	viscosity [Pa s]
ν	kinematic viscosity [m^2/s]
λ	thermal conductivity [W/m K]
α	thermal diffusivity [m^2/s]
γ	specific heat ratio [–]
Pr	Prandtl number [–]
δ_{ij}	Kronecker delta [–]
j	imaginary unit [–]
ω	angular frequency [–]
χ_ν, χ_α	thermoacoustic function for viscous and thermal diffusive term [–]
I	work flow [W]
W_ν, W_p	kinetic and potential energy dissipation [W/m]
W_{prog}	travelling wave components of work source [W/m]
W_{stand}	standing wave components of work source [W/m]
e	internal energy [J]
ϕ	viscous dissipation function [Pa/s]
C_p	specific heat at constant pressure [J/kg K]

Subscripts

m	mean (time-averaged) value
i	vector component
ij	tensor component
pu	difference between pressure and velocity
r	cross section value
Hw, Cw	wall values of hot and cold heat exchangers

To verify the above heat transfer models, many experiments have been performed so far. Mozurkewich [11] measured the heat transfer of wires located at a velocity antinode in an acoustic standing wave. In this experiment, at large acoustic amplitude, the Nusselt number follows the steady-flow and forced-convection correlation using period-averaging of the acoustic oscillation. Furthermore, from heat transfer measurements for a thermoacoustic refrigerator with aluminum heat exchangers, Mozurkewich [12] showed that the TASFE approximation is applicable in estimating heat transfer. As another verification of the TASFE model, Paek et al. [13] performed heat transfer measurements for both a steady flow heat exchanger and a standing wave thermoacoustic cooler investigating specifically the thermal performance of heat exchangers. In their study, they proposed the modified TASFE model by applying the correction factor to the acoustic Reynolds number. In addition to the above experiments, Mao et al. [14] also performed experiments verifying conventional heat transfer models. They measured the heat transfer rate between finned-tube heat exchangers arranged in an oscillatory flow, and compared the conventional heat transfer models, rms Reynolds number model, TASFE approximation model, and a boundary layer conduction model. In their study, a new empirical correlation is proposed as well. They suggested that the correlation determined from the normalized fin spacing and the normalized fin length can be used to evaluate the heat transfer rate.

In addition to the above measurements of heat transfer for oscillatory flows, other experimental approaches also have been performed. Wetzel and Herman [15,16] visualized and quantified the time-dependent temperature field in the neighborhood of the solid plate located in the acoustically driven working fluid. In their experiments, holographic interferometry and high-speed cinematography were employed. From data of heat fluxes close to a heated solid plate, they noted differences between the heat fluxes of steady and oscillatory flows and estimated the local coefficient of heat transfer. In visualizing flow velocity and temperature field, other techniques have been used recently. Using planar laser-induced fluorescence (PLIF) and particle image velocimetry, Jaworski et al. [17,18] measured the time-dependence of the temperature and velocity fields for a oscillatory flow in a parallel-plate channel. With these measurements, detailed behaviors of the oscillatory flow in heat exchangers were shown for both heat transfer and fluid motion. Furthermore, Jaworski et al. [19] measured the temperature field within the heat exchanger channel using PLIF and compared these results with numerical results. The numerical approach was originally developed by Piccolo et al. [20] based on a combination of standard thermoacoustic linear theory and heat transfer analysis.

Distinct from the above experimental approaches, many numerical approaches also have been performed in parallel with advances in computational fluid dynamics (CFD). The results have been compared with experimental data and results from conventional heat transfer models based on the classical steady-flow approach. Cao et al. [21] simulated the thermoacoustic fluid motion within a thermoacoustic couple, a simple channel configuration including one surface of a single plate. Their simulation is based on the numerical solution of the compressible Navier–Stoke equations. In their study, they investigated the time-averaged energy flux density and showed details of the heat sink on the plate. Regarding other investigations using CFD simulations, Worklikar et al. [22] performed a numerical simulation of unsteady two-dimensional flow in a thermoacoustic refrigerator. They solved the thermoacoustic phenomena by combining a low-Mach-number flow equation and energy equation describing the fluid dynamics. In their simulation, the results were compared with theoretical and experimental data. As a simulation based on the full Navier–Stokes equations, Ishikawa et al. [23] as well as Cao et al. [21] investigated the flow and energy fields near a thermoacoustic couple. The results show the effects of heat pumping near the edge of thermoacoustic couple plates. Furthermore, Marx et al. [24] investigated temperature behaviors near the extremities of a thermoacoustic stack plate by solving the two-dimensional compressible Navier–Stokes equations, and by considering temperature non-linearities near the extremities of the plate at higher Mach number.