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Analysis on the heat transfer characteristics of a micro-channel type porous-sheets Stirling regenerator

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ABSTRACT

To avoid the high flow frictional loss associated with conventional wire mesh Stirling regenerators, a micro-channel type stacked porous-sheets Stirling regenerator is investigated. An analytical solution is derived for the transient heat transfer characteristics of the fully developed reciprocating laminar flow under prescribed wall temperature profiles. The complex Nusselt number (Nu) is expressed as a function of kinetic Reynolds number (Re_ω) and Prandtl number (Pr). At low Re_ω of less than 10, the real part of Nu has an almost constant value of 6.0, approximately equal to the known real-valued Nu for the fully developed unidirectional laminar flow under constant wall heat flux, while the imaginary part is negligible, thus “scaling effect” can be utilized to enhance heat transfer. At higher Re_ω, both the real and imaginary parts of Nu increase with the increase of Re_ω and Pr, and the phase shift between the temperature difference and the heat flux gradually increases and approaches 45°. Approximate analytical solutions are also deduced for the entrance region from the integral boundary layer equations in both cases of “Thermally developing flow” and “Simultaneously developing flow”. The heat transfer is enhanced in the entrance region and the local Nu in the flow direction approaches the corresponding values of fully developed flow. The analytical results are confirmed by dynamic mesh CFD results, and the obtained Nu~ Re_ω data and patterns generally agree with available analytical and experimental data from published literatures. Application of the analytical results to the design and optimization of Stirling regenerator are also shown.

1. Introduction

The application of Stirling engine is promising in the power generation field due to its many advantages including adaptability to versatile heat sources, high thermal efficiency and environmental friendliness [1]. As a central and crucial component of the Stirling engine, the regenerator has received extensive research for improving the engine performance [2]. Up to date the wire mesh type regenerator is most popularly adopted in Stirling engines in virtue of its large heat transfer area, high convective heat transfer coefficient brought by numerous cross flow around cylindrical wires, and low axial thermal conductance. However, the associated high flow friction due to flow separation, wakes, eddies and stagnation zones might impair the power output and engine efficiency [3]. Theoretically a regenerator with heat transfer surfaces parallel to the oscillating flow has better performance [4].

With the emerging micro-fabrication techniques, properly designed regular-shaped micro-channel type regenerator can be fabricated to obtain extremely low flow friction while maintaining high thermal effectiveness. The main features of the regular micro-channel type regenerator include: (1) the heat transfer surface is smooth; (2) the flow acceleration rates are controlled; (3) the flow separation is minimized; (4) the axial thermal conduction is reduced by interrupting the axial continuity of solid structure, for example, using porous sheets with intermediate gaps or clearance. Other advantages include improved structural durability, no gas leakage or short-circuit loss owing to tight tolerance, low cost for mass production [5]. Ibrahim et al. [6] investigated a micro-channel type segmented-involute-foil regenerator under NASA support, in which the oscillating-flow rig test showed the highest figures of merit ever recorded, demonstrating a shift strongly in the direction of the theoretical performance of ideal parallel-plate regenerators.

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Nomenclature

a	thermal diffusivity, m^2/s
$A(r^*)$, $B(r^*)$	functions defined in Eqs. (5a) and (5b)
A_o	dimensionless fluid oscillating amplitude, $A_o = x_{max}/d_0$
A_R	relative fluid displacement ratio, $A_R = x_{max}/L$
b	side length of a hexagonal cross section
Bi	Biot number, defined in Eq. (56)
c	specific thermal capacity, J/(kg K)
c_p	specific thermal capacity at constant pressure, J/(kg K)
C_1, C_2	constants defined in Eq. (6c)
C_f	Fanning flow friction coefficient, $C_f = \tau_w/(\frac{1}{2}\rho u_{max}^2)$
Cr	heat capacity ratio of solid matrix to fluid flow, defined in Eq. (64)
d_0, r_0	channel inner diameter and inner radius, m
D_1, D_2	constants defined in Eq. (31b)
d_h	hydraulic diameter, m
E_1, E_2	constants defined in Eqs. (21a) and (21b)
$f(r^*)$, $g(r^*)$	functions defined in Eqs. (22a) and (22b)
Fo	Fourier number, defined in Eq. (60)
h_x	heat transfer coefficient, $W/(m^2\ K)$
Imag[ ]	imaginary component of a complex number
K	variation amplitude of pressure gradient multiplied by $-1/\rho$, m/s^2
k	thermal conductivity, W/(m K)
L	regenerator length, m
L_f	hydrodynamic entrance length, m
L_t	thermal entrance length, m
$\dot{m}$	mass flow rate, kg/s
n	rotational velocity, rpm
N_s	number of porous sheets
N_{TU}	number of heat transfer units, defined in Eq. (62)
Nu	Nusselt number, $Nu = hd_h/k_f$
p	pressure, Pa
Pr	Prandtl number, $Pr = \nu/a$
q_w	wall heat flux, W/m^2
r^*	dimensionless radial coordinate, $r^* = 2r/d_0$, m
Real[ ]	imaginary component of a complex number
Re_ω	kinetic Reynolds number, $Re_\omega = \omega d_0^2/\nu$
S	total heat transfer area between matrix and fluid, m^2
$\dot{S}_g$	total entropy generation rate, W/K
$\dot{S}_{g,f}$	entropy generation rate due to flow friction loss, W/K
$\dot{S}_{g,h}$	entropy generation rate due to irreversible heat transfer, W/K
t	time, s
T	temperature, K
T_b	bulk (mixed mean) temperature, $T_b = \int_0^{r_0} uTrdr / \int_0^{r_0} urdr$, K
t_{blow}	blow time for half cycle, s
T_m	cross-sectional average temperature, defined in Eq. (24)
u	axial velocity, m/s
u^*	dimensionless axial velocity, $u^* = u/u_{max}$
$X(r^*)$	function defined in Eq. (18a)
$Y(r^*)$	function defined in Eq. (18b)
x^*	dimensionless axial position, $x^* = 2x/d_0$
x_{max}	amplitude of fluid oscillation

Greek symbols

α	specific surface area, defined in Eq. (57), 1/m
β	constants defined in Eq. (6c)
γ	modified Womersley number, $\gamma = \frac{1}{2}\sqrt{Re_\omega Pr}$
Δ	$\Delta = \delta/\delta_t = \sigma/\xi$
δ	thickness of hydrodynamic boundary layer, m
δ_s	width of solid rib between adjacent pores, m
δ_t	thickness of thermal boundary layer, m
ε	regenerator effectiveness
ζ	dimensionless y coordinate in flow boundary layer, $\zeta = y/\delta$
η	dimensionless y coordinate in thermal boundary layer, $\eta = y/\delta_t$
Θ	dimensionless fluid temperature, $\Theta = (T - T_c)/(T_h - T_c)$
θ	excess fluid temperature, $\theta = T - T_w$
λ	Womersley number, $\lambda = \frac{1}{2}\sqrt{Re_\omega}$
ν	kinetic viscosity, m^2/s
ξ	dimensionless thickness of thermal boundary layer, $\xi = \delta_t/r_0$
ρ	density, kg/m^3
σ	dimensionless thickness of flow boundary layer, $\sigma = \delta/r_0$
τ	shear stress, Pa
φ	crank angle, $\varphi = \omega t$
Φ	matrix outer diameter, m
ϕ	regenerator porosity
ψ	dimensionless temperature in solid rib, $\psi = (T - T_0)/(T_m - T_0)$
ω	rotational velocity, rad/s
μ	dynamic viscosity, Pa s
Λ_1	phase shift between the mean velocity and pressure gradient
Λ_2	phase shift between heat flux and temperature difference

Superscript and subscript

*	dimensionless parameter
–	cycle averaged parameter
→	complex parameter
Ac	cross section
c	cold end of regenerator
e	external core flow
f	fluid
h	hot end of regenerator
m	mean
max	maximum
s	solid
w	wall
wi	channel inner wall
x	axially local parameters
Γ	the boundary curve of channel cross section

Takizawa et al. [7] developed a 3-kW Stirling engine installed with a porous-sheets regenerator with electrically etched holes, and the engine performance was improved by about 5–10% compared to that with wire mesh regenerator. A series of engine tests were done by Matsuguchi et al. [8] to optimize the geometrical parameters of the porous-sheets regenerator. The recent work of the present authors, via dynamic mesh Computational Fluid Dynamics (CFD) method and experimental validation, indicates that the regular-shaped micro-channel type porous-sheets regenerator has extremely low flow friction while maintaining high thermal effectiveness, thus achieving significantly lower total entropy generation rate and leading to higher comprehensive performance [9].