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Numerical and experimental investigation of heat transfer augmentation potential of wire-loop structures

Subhashis Ray [a,*], Robert Eder [a,*], Thomas M. Wittenschlaeger [a], Ingolf Jaeger [a], Volker Uhlig [a], Dimosthenis Trimis [a,b]

[a] *Chair of Gas and Heat Technology, Institute of Thermal Engineering, Technische Universität Bergakademie Freiberg, Gustav-Zeuner-Straße 7, 09596 Freiberg, Germany*
[b] *Engler-Bunte-Institute, Karlsruhe Institute of Technology, Engler-Bunte-Ring 1, D-76131 Karlsruhe, Germany*

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ABSTRACT

The present paper explores the possibility of using wire-loop structures on the active plate of a parallel plate channel for efficient heat transfer augmentation. For this purpose, numerical simulations were carried out for periodically fully-developed turbulent flow in typical repeating modules using the realizable $k-\varepsilon$ model. Experiments were also conducted in order to validate some of the results obtained from numerical simulations. For both studies, the Reynolds number was varied from 2000 to 20,000. The effect of three different loop-densities on fluid flow and heat transfer characteristics were investigated when wire-loops were placed perpendicular to the main flow direction, whereas the effect of loop-orientations on these parameters were studied for a fixed loop density of 2270 loops/m^2. While for all investigated cases, substantial heat transfer augmentation was observed with wire-loop structures as compared to the empty parallel plate channel under the condition of identical pressure gradient, the thermal-hydraulic performance improved significantly with the increase in loop-density. The maximum attainable loop-density, however, was found to strongly depend on the loop-orientation owing mainly to the geometric as well as manufacturing constraints. It was also observed that loops, oriented diagonally to the main flow direction offer the best performance, although it is less sensitive to loop-orientation as compared to the loop-density.

1. Introduction

High temperature processes are encountered in many industries, such as steel making and ceramics production industries, where they are significant consumers of energy. Data for the year 2011 shows that out of the 2,624 PJ used in the German industrial sector (form 8,744 PJ total energy consumption), 22.6% of the total energy consumption or 1,979 PJ was used for heat processing. This is slightly more than 75% of the total German industrial usage [1]. One of the consequences of such a skewed energy distribution is that even a marginal increase in the efficiency of high temperature processes on the system level would translate into a dramatic reduction of energy consumption on the industrial level. It is definitely not unreasonable to assume that this conclusion would be valid for most other industrialized countries as well.

In such processes, heat loss represents the major loss of energy from the system and is, in fact, the principal measure of the efficiency of a system. In order to substantially improve the efficiency, it is necessary to integrate a heat exchanger into the system for recovering a portion of heat, which would otherwise remain unaccounted for and hence would be lost. For example, in applications involving burners, a recuperator should be placed in the exhaust stream of the burner where the high temperature exhaust gas can be used in order to preheat the incoming oxidizer stream. It is estimated that a 5% reduction in the burner fuel consumption can be achieved for every 100 °C reduction in the exhaust gas temperature in the recuperator [2].

Heat exchangers are themselves bound by the efficiency limitations and improving the efficiency of these devices has been the subject of extensive research over the past several decades [3–18]. In general, the flow through a single heat exchanger passage can be regarded as the flow through a duct of a given cross-section that

* Corresponding authors.
E-mail addresses: Subhashis.Ray@iwtt.tu-freiberg.de, juhp_sray@yahoo.co.in (S. Ray), Robert.Eder@iwtt.tu-freiberg.de (R. Eder).

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Nomenclature

A_0	constant in $C_\mu = 4.04$
A_c	cross-sectional area [m^2]
A_s	heat transfer surface area [m^2]
A_S	function of ϕ, appearing in C_μ
C_2	constant in dissipation equation = 1.9
C_p	specific heat at constant pressure [J/kg K]
C_μ	proportionality constant, appearing in Reynolds stresses
d_h	hydraulic diameter of the empty channel = $2l_g$ [m]
d_w	diameter of the loop wire [m]
$(dp/dx)^*$	dimensionless pressure gradient = fRe^2
E	evaluated variable, function of M_j
f	friction factor
G_k	production of turbulent kinetic energy [J/m^2 s]
h	periodic module-averaged heat transfer coefficient [W/m^2 K]
h_{eff}	effective heat transfer coefficient $h(A_{as}/A_{ts})$ [W/m^2 K]
h_x	local heat transfer coefficient [W/m^2 K]
k	turbulent kinetic energy [J/kg]
l	turbulent length scale used for initialization = 0.07 d_h
l_c	distance between loop and upper inactive plate [m]
l_g	gap between two parallel plates [m]
l_{cy}	width of the computational domain [m]
l_{pd}	length over which pressure drop was measured [m]
l_{px}	distance between loops in the direction of flow [m]
l_{py}	distance between loops in the transverse direction to the flow [m]
l_w	half length of wire-loops = $\pi r_l/2$ [m]
m	characteristic of fin (wire-loop) = $(4h/\lambda_s d_w)^{1/2}$ [1/m]
$\dot{m}$	mass flow rate [kg/s]
M_j	jth measured variable
n	total number of measured variables required for E
Nu	Nusselt number = hd_h/λ_f
p	effective pressure including hydrostatic pressure variation [Pa]
P_w	perimeter of the wire = πd_w [m]
$\dot{Q}_c$	corrected rate of heat transfer [W]
$\dot{Q}_s$	simulated rate of heat transfer [W]
r_l	radius of the loop [m]
Re	Reynolds number = $\rho u_{av} d_h/\mu$
S	modulus of mean rate-of-strain tensor [s^{-1}]
S_{ij}	mean rate-of-strain tensor [s^{-1}]
t	time [s]
T	temperature [K]
T_u	turbulence intensity used for initialization = $0.16\,Re^{-1/8}$
u_j	Cartesian velocity components [m/s]
U^*	combined mean-rate-of-strain and rotation [s^{-1}]
$\dot{V}$	volume flow rate [m^3/s]
x_j	Cartesian coordinates [m]

Greek symbols

α_ϕ	relaxation parameter for conservation equation of variable ϕ
Δh	estimated reduction in predicted heat transfer coefficient [W/m^2 K]
$\Delta p/L_{px}$	average axial pressure gradient [Pa/m]
Δt	time step [s]
ΔT	estimated temperature drop along wire-loops $T_w - T_{tip}$, also uncertainty in measured temperature [K]
$\Delta\phi$	uncertainty in ϕ
ε	dissipation of turbulent kinetic energy [m^2/s^3]
η_{fw}	fin efficiency of wire-loops
λ_f	thermal conductivity of fluid [W/m K]
λ_w	thermal conductivity of wire-loop material (solid) [W/m K]
μ	dynamic viscosity [Pa s]
μ_t	turbulent (Eddy) viscosity [Pa s]
ν	kinematic viscosity = μ/ρ [m^2/s]
θ	orientation of wire-loops on active plate with respect to main flow direction
ρ	density [kg/m^3]
σ_k	Prandtl number for turbulent kinetic energy = 1.0
σ_T	turbulent Prandtl number for temperature = 0.85
σ_ε	Prandtl number for dissipation rate of turbulent kinetic energy = 1.2
ψ_A	ratio of active surface area to the total heated surface area = A_{as}/A_{ts}
Ω_{ij}	mean rate-of-rotation tensor [s^{-1}]

Subscripts

a	active plate
av	average
b	bulk
e	at experimental condition
i,o	inlet, outlet
n	at normal condition
t	total
tip	tip of wire-loop
w	wire-loop, also wall

is specific to the considered heat exchanger and the heat transfer enhancement can be realized by either limiting the size or the development of thermal boundary layer. Varieties of schemes and arrangements have been proposed for this purpose. One option is to introduce elbows and kinks into the geometry of heat exchanger. This keeps the flow in the developing region where the thermal boundary layer is expected to be thin. Another option is to disrupt directly the development of flow through the duct in some way.

Cylinders inserted in the cross-flow promote vortex shedding, where the shed vortices disturb the development of thermal boundary layer and promote mixing of the fluid and thereby, ultimately, the heat transfer. The typical horse shoe vortex of the wall mounted cylinder have been studied frequently, for example by Baker [3–5], Eckert and Soehngen [6] and Zdravkovich [7]. In contrast, experimental results of the local heat transfer from the free cylinder with high aspect ratio in cross flow were presented by Sanitjai and Goldstein [8]. For the same configuration, Sabanca and Durst [9] performed a numerical investigation. Similarly, wavy, ribbed, or dimpled channels have been shown to improve heat transfer [10,11].

Jacobi and Shah [12] presented a detailed review of heat transfer enhancement through longitudinal vortices caused by delta wings, rectangular wings, delta winglets, and rectangular winglets. Beside cylinders, other bluff bodies (e.g., cubes, hemispheres, cones, etc.) as turbulators were also presented. In addition, Fiebig [13] provided a survey on wings and winglets for heat transfer enhancement.