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Influence of entrance geometry on flow field and heat transfer performance in stationary two-pass smooth parallelogram channels

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ABSTRACT

PIV and Infrared Thermometry measurements are respectively conducted to study the effects of entrance geometry and Reynolds number (*Re*) on the detailed flow fields and local temperature distributions in a stationary two-pass smooth parallelogram channel with 180-deg sharp turn. Two entrance geometries, including a fully developed inlet condition (FDI) as well as an asymmetrically and suddenly contracted inlet condition (ASI), are investigated. The smooth parallelogram channel has equal adjacent sides of 45.5 mm in length and two pairs of adjacent angles are 45-deg and 135-deg. Local (Nu_0) and regionally averaged ($\overline{Nu}_0$) Nusselt numbers over entire top and bottom walls along the first and second passages and through the bend region with the associated pressure drop are examined under *Re* ranging from 5000 to 20,000. Moreover, cross-sectional secondary-flow patterns as well as the near-wall streamwise mean velocity components and turbulent kinetic energy are analyzed to correlate the relationship between flow characteristics and heat transfer distributions at $Re = 10{,}000$. The most distinct finding of the present study is that the asymmetric thermal and fluid flow features on the top and bottom wall side, in contrast to symmetric ones in the corresponding square and rectangular channels. The affected top and bottom wall Nu_0 distributions of ASI respectively extends downstream to the mid-turn and the middle of second pass. Compared with FDI, ASI elevates the $\overline{Nu}_0$ about 65.3–70.1%, 14.2–13.7%, and 23.9–14.0% in the first passage, turn region, and second passage, respectively, under constant flow rate condition. Thermal performance factors of the ASI are about 43.5% and 53.7% higher than the FDI at $Re = 5000$ and 20,000, respectively. Moreover, the correlations of Nu_0 and fanning friction factor (f_0) with *Re* are obtained and further compared with those of the corresponding square channels available from the literature.

1. Introduction

Turbomachinery plays an important role in aerospace industry and power generation industry. For high power output applications, such as turbojet engines, the turbine blades of turbo machines are usually working at extremely high temperature and pressure conditions. Furthermore, the higher engine entry temperature (TFT) provides higher thermodynamic efficiency and power output. However, high working temperature also increases thermal loads on the turbine blades. These thermal loads including thermal stress and thermal fatigue will sharply shorten the working life of turbine blades. Although material science may increase the upper thermal limit of turbine blades by developing novel materials that can endure higher temperature, additional turbine blade cooling methods are more available and less expensive way for commercial products. These cooling technologies, such as external film cooling, impingement cooling, and internal cooling channels, are not only used in turbine blade cooling but also in many daily heat exchange applications. In the present study, the fluid flow feature and heat transfer performance of internal cooling channels are the main subject, and affected by the parameters such as the inlet condition, channel cross-section, and turn geometry.

The cross-sectional shape is one of the most important design points of the internal cooling channel. It affects the flow structure, pressure drop and thermal performance of the whole channel. Khalifa and Trupp [1] conducted an experimental investigation of the stationary flow structures of fully developed air flow through a single-pass 60-deg symmetric smooth trapezoidal channel. Pitot tube and hotwire anemometer were applied to the measurements

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Nomenclature

English symbols

A	parallelogram cross-section area (m^2)
AR	aspect ratio $= W_1/(B/\sin 45°) = W_2/(B/\sin 45°)$
B	channel height (m)
C_p	specific heat of fluid ($J\ kg^{-1}\ K^{-1}$)
D_H	hydraulic diameter $= 4A/P$ (m)
f_0	fanning friction factor for each channel wall$= [\Delta P/(0.5\rho U_b^2)]/(4L/D_H)$
f_∞	Blasius equation for turbulent flow $= 0.079Re^{-0.25}$
K	turbulent kinetic energy $= [(u')^2 + (v')^2 + (w')^2]/2$
k_f	thermal conductivity of fluid ($W\ m^{-1}\ K^{-1}$)
L	channel length (m)
Δl	laser light sheet thickness (m)
$\overline{Nu}$	averaged Nusselt number
Nu_0	local Nusselt number of stationary channel
Nu_∞	Nusselt number value evaluated from Dittus-Boelter correlation $= 0.023Re^{0.8}Pr^{0.4}$
P	perimeter of parallelogram (m)
P_T	total input heat flux provided by power supply ($W\ m^{-2}$)
ΔP	pressure drop across scanned section ($N\ m^{-2}$)
Pr	Prandtl number $= \mu C_p/k_f$
q_f	convective heat flux ($W\ m^{-2}$)
q_{loss}	heat loss to the environment ($W\ m^{-2}$)
Re	Reynolds number $= \rho U_b D_H/\mu$
Ro	rotation number $= \Omega D_H/U_b$
S	traveling distance along each section (m)
T_b	fluid bulk temperature (K)
T_w	wall temperature (K)
T_∞	ambient temperature (K)
TPF	channel thermal performance factor$= (\overline{Nu}_0/Nu_\infty)/(\bar{f}_0/f_\infty)^{1/3}$
Δt	shutter of CCD camera (s)
U_b	bulk flow velocity ($m\ s^{-1}$)
U	streamwise mean velocity ($m\ s^{-1}$)
u'	streamwise turbulence intensity ($m\ s^{-1}$)
V	transverse mean velocity ($m\ s^{-1}$)
v'	transverse turbulence intensity ($m\ s^{-1}$)
$\lvert V\rvert$	velocity magnitude traveling through the laser light sheet ($m\ s^{-1}$)
W	spanwise mean velocity ($m\ s^{-1}$)
W_1	width of first-pass duct (m)
W_2	width of second-pass duct (m)
W_d	divider thickness (m)
w'	spanwise turbulence intensity ($m\ s^{-1}$)
X	streamwise coordinate (m)
X^*	normalized streamwise coordinate $= X/D_H$
Y	transverse coordinate (m)
Y^*	normalized transverse coordinate $= Y/B$
Z	spanwise coordinate (m)
Z^*, Z^{**}	normalized spanwise coordinate for all Y^*, (i) $X < 0$, $Z^{**} = Z/[(W_1 + W_2 + W_d)/2]$ (in the turn) (ii) $X \geqslant 0$, $Z < 0$, $Z^* = [Z + (W_d/2)]/W_1$ (in the 1st pass) (iii) $X \geqslant 0$, $Z > 0$, $Z^* = [Z - (W_d/2)]/W_2$ (in the 2nd pass)

Greek symbols

ε	coefficient of heat flux non-uniformity
ξ	ratio of the out-of-plane displacement to the laser sheet thickness $= \lvert V\rvert\Delta t/\Delta l$
ρ	density of fluid ($kg\ m^{-3}$)
μ	fluid dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
Ω	rotational speed of test duct ($rad\ s^{-1}$)

Subscripts

1	first passage
2	second passage
t	turning region
0	refers to non-rotating situation

over a Re range of 37,000 to 116,000. They pointed out the existence of four unequal sizes counter-rotating pairs of secondary flow that differs from the well-known four perfectly symmetric pairs of vortices in the square channel [2]. Dutta et al. [3] presented the heat transfer results with thermocouple and the numerical predictions with k-ε model in a rotating two-pass smooth triangular channel for Re from 2500 to 20,000. Their experimental results showed that triangular channel provided smaller Nusselt number (Nu) difference between the trailing and leading wall in the first passage but greater in the second passage than those in the corresponding square channel. Their numerical results indicated a prevailing dominant vortex and a smaller vortex at the acute corner in the cross-sectional plane of triangular channel. In contrast, the vortices in the cross-sectional plane are symmetric in the corresponding square channel. The geometry of the turn region also influences the performance of the coolant channel. Wang and Chyu [4] provided the numerical results (k-ε model) of the flow pattern and heat transfer distribution of the turn region in the stationary two-pass smooth square channel at a fixed Re = 75,000. Three different 180-deg turn configurations were chosen: rectangular turn, rounded-corner turn, and circular turn. In the turn region, rectangular corner has the 4.5% and 6.2% higher heat transfer enhancements than rounded-corner turn and circular turn, respectively.

With Particle Image Velocimetry (PIV), Schabacker et al. [5] investigated the 3D mean velocity fields and turbulence characteristics in a stationary two-pass smooth square channel at Re = 50,000. The measurements showed a strong secondary flow consisting of two Dean-type vortices occurred in the bend and impinged on the channel tip, influencing the flow field in the second passage. Liou and Chan [6] provided thermocouples data of a rotating two-pass smooth rectangular channel at Re ranging from 5000 to 50,000. The channel had a 180-deg rectangular turn and a cross-sectional aspect ratio of 1.25. Additional flow visualization was added to get insight into the heat transfer mechanism. The results showed that the secondary flow and turbulent motion induced by sharp turning bend were the main reasons for the heat transfer enhancement in the near turn region of the second passage. Liou et al. [7] presented the measurements of heat transfer and fluid flow in a two-pass smooth square channel at Re of 10,000 with transient liquid crystal thermometry (TLCT) and Laser Doppler Velocimetry (LDV). The detailed local Nu distributions of the wall surfaces at high rotation number (Ro) were shown. With rotating, the symmetric Dean vortices in the turn region of the stationary channel changed into single vortex which impinged on the channel tip and, in turn, leading wall causing local high heat transfer. Elfert et al. [8] used PIV to investigate a stationary two-pass smooth wedge channel at Re = 40,000 (based on the hydraulic diameter of the second passage). The study reported that the complex flow fields in the turn region were affected by the cross-sectional geometry. For example, the separation bubbles near the upper and lower side of the second passage were not symmetric at the dividing walls. Liou et al. [9] researched into the fluid