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Heat transfer and thermal performance analysis of a surface with hollow rectangular fins

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Abstract

An experimental study was conducted to investigate the heat transfer and friction loss characteristics in a horizontal rectangular channel having attachments of hollow rectangular profile fins over one of its heated surface. The Reynolds number based on the flow averaged inlet velocity and the hydraulic diameter, ranged from 3000 to 32,000. The hollow rectangular profile fins in 10 cm height and $a \times b = 2$ cm $\times$ 4 cm dimensions with a thickness of 0.2 cm were mounted on a heating surface vertically. Reynolds number, fin arrangement and fin pitch in the flow direction were the experimental parameters. Both in-line and staggered fin arrangements were studied for one-fixed spanwise ($S_x/a = 3$) and four different streamwise ($S_y/b = 1.5$, 1.875, 2.5 and 3.75) distances. Correlation equations for Nu, f and thermal performances were determined for fin configurations and the straight channel case without fins.
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Keywords: Finned surfaces; Heat transfer enhancement; Thermal performance; Hollow rectangular fins; Forced convection heat transfer

1. Introduction

It is well known that a straight fin with a concave parabolic profile provides maximum heat dissipation for a given profile area [1]. Since the concave parabolic shape is difficult and costly to manufacture, the rectangular profile is preferred even though it does not utilise the material most efficiently [2]. For example, in a study performed by Tahat et al. [3], pin fins were employed on the heating surface in a rectangular channel; and Bilen et al. [4] investigated the heat transfer and friction loss characteristics of a surface with cylindrical fins arranged both in-line and staggered in a channel having rectangular cross-section. The maximum amount of the heat transfer occurred at $S_y/D = 2.94$.

A thermal performance analysis is also worthwhile for the evaluation of the net energy gain. One of the ways to evaluate the heat transfer performance is the comparison of the heat transfer coefficients at a constant pumping power [4–6].

Many studies have been done for different types of fin arrays, for example the one reported in [7], but still there is lack of knowledge of the forced convection heat transfer from a surface with vertical hollow rectangular profile fins. The array employed in the present study consists of vertically mounted hollow rectangular profile fins on a surface. The experiments were performed for in-line and staggered fin arrangements. Heat transfer experiments without fins were also conducted, for efficiency comparison. Furthermore, friction loss was determined by measuring pressure drop along the test section. In calculations, two different areas, which were called the total surface area and the projected area, were employed for the average Nusselt number.

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Nomenclature

A	heat transfer area (m^2)
a, b	fin length in spanwise and streamwise direction, respectively
D_h	hydraulic diameter of the channel (m)
f	friction factor
h	mean heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
H	fin and channel height (m)
k	thermal conductivity of air ($W\ m^{-1}\ K^{-1}$)
L	test surface length (m)
N	number of fin
Nu	Nusselt number for finned surface
Nu_s	Nusselt number for smooth channel
$\dot{Q}$	heat transfer rate (W)
Re	Reynolds number
S	distance between the adjacent fins (m)
T	steady state temperature (K)
V	mean inlet velocity ($m\ s^{-1}$)
W	channel or test plate width (m)

Greek symbols

ΔP	static pressure difference ($N\ m^{-2}$)
η	performance efficiency
ν	kinematic viscosity of air ($m^2\ s^{-1}$)
ρ	air density (kg/m^3)

Subscripts

a	finned
axi	axial
bac	back
con	convection
f	film
in	inlet, in-line
loss	losses
net	net
out	out
pro	projected
rad	radiation
s	mean surface, smooth
stag	staggered
tot	total
vol	volt
x, y	spanwise and streamwise directions, respectively

2. Experimental rig

An experimental set up was installed to study the heat transfer performance and friction factor of hollow rectangular profile fins (Fig. 1). Air was the working fluid. The test facility was consisted of a wooden channel set on the suction side of a fan. The cross-section of the channel was rectangular in each section; 10 cm in height and 18 cm in width with a wall thickness of 1.8 cm, total length of the channel was 200 cm. The test section was mounted at the bottom surface. The aluminum hollow rectangular profile fins were placed on an aluminum plate in dimensions of 30 cm length, 18 cm width and 0.2 cm thickness. A plate heater with a 1500 W maximum power, which was approximately the same dimensions as the aluminum plate, heated the lower horizontal wall of the test section to supply a constant heat. The amount of the heat given to the test section was controlled by a variac and a voltage regulator. To reduce the contact resistance to heat flow, a sink compound of high thermal conductivity was applied both between the heater and the test surface and between the test surface and the fins. The backsides of the heater and of the other walls were insulated with glasswool, in order to minimize the heat losses.

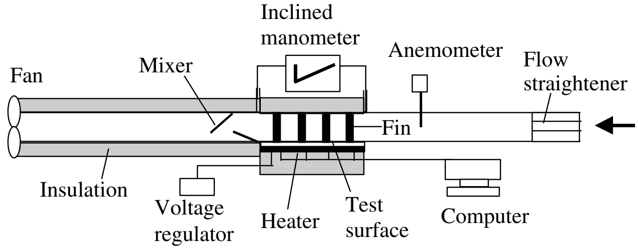

Fig. 1. Schematic diagram of experimental apparatus.

Seven copper–constantan thermocouples were installed along the test section centerline, to measure the steady state temperature of the base surface of the fin array. The average of these readings was taken as average temperature of the test surface at steady state. Analog signals from the experimental system were fed to the data acquisition card (HG 818 advantech), then these signals were amplified and converted to digital signals and both saved and displayed on the computer screen as real temperature values. The temperatures were used the average of ten values collected for a single thermocouple location in two-minute interval at steady state

Table 1
Distance between fins and the number of rectangular profile fins for $a \times b = 2 \times 4\ cm^2$ and $S_x/a = 3$

S_y/b	1.5	1.875	2.5	3.75
N_y	5	4	3	2
N_x	3	3	3	3
N_{tot} (in-line)	15	12	9	6
N_{tot} (staggered)	13	10	8	5

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