Experimental Thermal and Fluid Science 60 (2015) 173–181

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Laminar fluid flow and heat transfer through a circular tube having spiral ribs and twisted tapes

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ARTICLE INFO

Article history:
Received 9 April 2014
Received in revised form 9 September 2014
Accepted 11 September 2014
Available online 22 September 2014

Keywords:
Laminar forced convection
Integral spiral rib roughness
Twisted tapes with oblique teeth
Heat transfer enhancement
Swirl flow

ABSTRACT

The experimental friction factor and Nusselt number data for laminar flow of viscous oil through a circular duct having integral spiral rib roughness and fitted with twisted tapes with oblique teeth have been presented. Predictive friction factor and Nusselt number correlations have also been presented. The thermohydraulic performance has been evaluated. The major findings of this experimental investigation are that the twisted tapes with oblique teeth in combination with integral spiral rib roughness perform significantly better than the individual enhancement technique acting alone for laminar flow through a circular duct up to a certain value of fin parameter.

1. Introduction

Laminar flow is encountered in many industrial applications. In case of laminar flow, there is major thermal resistance in the bulk flow in addition to the dominant thermal resistance in the thin boundary layer adjacent to the flow. Spiral ribs are also turbulators. Spiral rib as shown in Fig. 1 is made as integral surface to the duct wall. The dimensionless geometric parameters that influence the heat transfer and friction characteristics are the rib height and rib pitch. Integral rib-roughness has been used for the enhancement of tube-side heat transfer coefficient in low-flow automotive radiators and in this case, the flow Reynolds number is small and may be <2000. Farrel et al. [1] tested one fully-ribbed and two broken ribbed flat radiator tube. They obtained friction factors for $200 < Re < 11{,}000$. However, the heat transfer coefficients were obtained only for turbulent flow with $Re > 2000$. The broken-ribbed tube with the highest e/D yielded the highest heat transfer coefficient as well as the highest friction factor. Olsson and Sunden [2] tested two ribbed radiator tubes with airflow. The air heat transfer data were taken with constant wall temperature and the data provide the spirally averaged heat transfer coefficient over the tube length. The enhanced tubes showed higher friction factors than the smooth tube in both laminar and turbulent regions. However, as the Reynolds number decreased in the strictly laminar region, the friction factors tended to converge and approach the smooth tube value. Also, the laminar-turbulent transition Reynolds number decreased as the friction factor increased. Similar to the friction behaviour, the Colburn *j* factors also tended to converge at low Reynolds numbers, and approached the smooth tube value. However, in contrast to the friction factors, the *j* factors did not show a clear laminar-turbulent transition. Olsson and Sunden [3] investigated the effect of rib configurations for the multiple V-ribbed channel.

Helical screw-tape inserts, Eiamsa-ard and Promvonge [4], cause the flow to spiral along the tube length. Twisted tapes are similar to helical screw-tape inserts. Continuous twisted-tape as shown in Fig. 2(a) has been extensively investigated. Variants of twisted-tape that have been evaluated include short sections of twisted tapes at the tube inlet, or periodically spaced along the tube length. Early works on twisted tapes have been reported in Date [5] and Hong and Bergles [6]. Later works [7–49] have been reported.

Fig. 2(b) and (c) shows the layout of a circular duct having full-length twisted-tape with oblique teeth. The details and method of making integral spiral rib and twisted-tape with oblique teeth have been described in Section 2.

It has been observed from the literature review that the combined effect of integral spiral rib roughness and twisted-tape with oblique teeth has not been studied in the past. The spiral fluid flow due to integral spiral rib roughness coupled with twisted-tape with oblique teeth-generated swirl flow is likely to give larger swirl intensity and vortex in the flow. Also there may be enhanced fluid

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http://dx.doi.org/10.1016/j.expthermflusci.2014.09.005
0894-1777/

Nomenclature

A	heat transfer area, m^2
A_c	axial flow cross-sectional area, $=WD - \delta D$, m^2
A_o	plain duct flow cross-sectional area, =WD, m^2
C_p	constant pressure specific heat, J/kg K
D	internal diameter of the plain duct, m
e	rib height, m
f	fully developed Fanning friction factor = $(1/2)[\Delta P'/(\rho V_0^2)](D/Z)$, dimensionless
g	gravitational acceleration, m/s^2
Gr	Grashof number = $g\beta\rho^2 D^3 \Delta T_w/\mu^2$, dimensionless
Gz	Graetz number = $\dot{m}C_p/kL$, dimensionless
H	pitch for 180° rotation of twisted-tape, m
h_z	axially local heat transfer coefficient, W/(m^2 K)
k	fluid thermal conductivity, W/(m K)
L	axial length, length of the duct, m
$\dot{m}$	mass flow rate, kg/min
Nu_m	axially averaged Nusselt number = $\frac{1}{L}\int_0^L \frac{h_z D dz}{k}$, dimensionless
ΔP_z	pressure drop, mm
$\Delta P'$	pressure drop, N/m^2
P	wetted perimeter in the particular cross-section of the duct, rib pitch, m
Pr	fluid Prandtl number = $\mu C_p/k$, dimensionless
Ra	Rayleigh number = $Gr \cdot \mathrm{Pr}$
Re	Reynolds number based on plain duct diameter = $(\rho V_0 D_h)/\mu$, dimensionless
T	temperature, K
ΔT_w	wall to fluid bulk temperature difference, K
V_a	mean spiral velocity = $\dot{m}/\rho A_c$, m/s
V_o	mean velocity based on plain duct diameter = $\dot{m}/\rho A_0$, m/s
X	Pr^n, the value of n depends on the exponent of Pr in the correlation
Y	$\left(\frac{\mu_b}{\mu_w}\right)^{-0.14} \times \frac{1}{5.172}$
y	twist ratio = H/D, dimensionless
z	spiral length, the distance between the measuring pressure taps, m

Greek Symbols

β	coefficient of isobaric thermal expansion, K^{-1}
δ	tape thickness, m
μ	fluid dynamic viscosity, kg/ms
ρ	density of the fluid, kg/m^3
θ	twisted-tape tooth angle angle, °

Subscripts

b	at bulk fluid temperature
hl	twisted-tape tooth horizontal length
m	axially averaged
w	at duct wall temperature, with
z	local value

mixing with increased heat and momentum diffusion. This may increase heat transfer even if it may also give increased pressure drop. In this paper, therefore, the laminar flow experimental heat transfer and pressure drop results of combined effect of integral spiral rib roughness and full-length twisted-tape with oblique teeth in circular ducts are presented. Friction factor and Nusselt number correlations are presented. Also the performance of this combined geometry is evaluated.

2. Experimental set-up, operating procedure and data reduction

The heat transfer and pressure drop measurements were taken in a 19 mm ID, 20 mm OD and 2 m long circular stainless steel duct. Fig. 3 shows the self-explanatory experimental rig. The test section was electrically heated by nichrome heater wire giving uniform wall heat flux boundary condition. Nichrome heater wire was having porcelain bead insulation on it. There was no direct contact of the Nichrome heater wire with the duct wall. First, there was fibre glass tape insulation (electrical but not thermal) on the duct wall. Then the porcelain-bead covered Nichrome heater wire was wrapped on the duct wall. Two consecutive turns of the heater wire seated side by side touching each other. The thermal conductivity of the duct wall material was high enough and the duct wall thickness was sufficient to ensure uniform wall heat flux. Asbestos rope and glass wool insulated the heat transfer test section after the heater wire. Finally the test section was covered with jute bag for further thermal insulation.

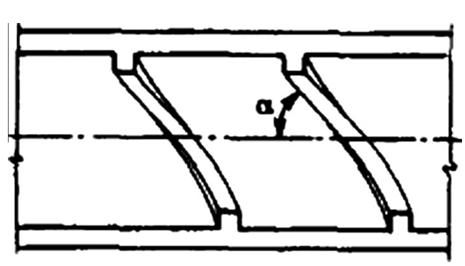

Fig. 1. Circular duct with helical ribs.

Spiral corrugation roughness for heat transfer tests was made of brass. Servotherm medium oil of Indian Oil Corporation was used as the working fluid. Wide Prandtl number range (152–549) was achieved by using this oil. Twisted-tape inserts were placed at the centre of the duct cross-section by SS lugs.

Oil mass flow rate was measured by rotameters. Pressure drops were measured by vertical mercury manometer. The local enhancement due to ribs quickly dissipates due to boundary layer mixing dissipation downstream of the ribs. Hence, rational selection of the rib heights and rib pitch requires knowledge of the local heat transfer coefficient. Therefore, heat transfer test section outer wall temperatures were measured at seven axial locations (each axial station had four thermocouples ninety degrees apart along the duct periphery) by copper-constantan thermocouples and digital multimeter. Typically, there was only 2.3–3.7% of maximum wall temperature variation in peripheral outside wall temperature measured at four locations in an axial station. Similar results have been observed for all cases irrespective of Reynolds number and Prandtl number. The peripheral wall temperature variation is due to buoyancy, effects of rib heights and rib pitch, and tape-induced swirl. However, the effects are not very strong.

Thermocouples were installed on the duct outside wall by brazing. Axial locations of thermocouples were 5 cm, 50 cm, 1.00 m, 1.25 m, 1.5 m, 1.75 m and 1.95 m along the downstream direction from the onset of heating at the upstream end of the duct. Duct inside wall temperatures were evaluated by calculating duct-wall temperature drop from the one-dimensional radial heat conduction equation.