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Experimental investigation of laminar flow of viscous oil through a circular tube having integral spiral corrugation roughness and fitted with twisted tapes with oblique teeth

Pranab Kumar Pal, Sujoy Kumar Saha *

Mechanical Engineering Department, Bengal Engineering and Science University, Shibpur, Howrah 711 103, India

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ABSTRACT

The experimental friction factor and Nusselt number data for laminar flow of viscous oil through a circular duct having integral spiral corrugation 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 corrugation 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

Integral spirally corrugated channels used in heat exchangers are typically sinusoidal channels. Fig. 1 shows the integral spiral corrugation in a circular duct. The effect of corrugation angle in a corrugated duct was investigated by a number of investigators, [1,2]. The corrugation angle ranged from 0° to 90°. Both friction factor and Nusselt number increase monotonically up to a certain value of the corrugation angle. Focke and Knibbe [3] have shown that at corrugation angle 45°, the fluid flow is predominantly along the furrows. Focke et al. [1] suggested similar flow patterns up to corrugation angle 60°. The reason for increase and decrease of the friction factor and Nusselt number is the positive and negative interaction of criss-crossing fluid streams inducing secondary swirl motion, change of flow pattern and accelerating or decelerating effect on them. Stasiek et al. [2] investigated the effect of corrugation pitch to channel height ratio. Abdel-Kariem and Fletcher [4] developed friction factor and Nusselt number correlations.

For laminar regime,

$$f = 15\ \mathrm{Re}^{-0.3}\left(\frac{\theta}{45}\right)^{2.5} \tag{1}$$

$$\mathrm{Nu} = 0.777\ \mathrm{Re}^{0.444}\ \mathrm{Pr}^{0.4}\left(\frac{\theta}{45}\right)^{0.67} \tag{2}$$

Twisted tapes as shown in Fig. 2 cause the flow to spiral along the tube length. Continuous twisted-tape 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 Refs. [5,6]. Later works have been reported in Refs. [7–27]. Fig. 2(b) and (c) show the layout of a circular duct having full-length twisted-tape with oblique teeth. The details and method of making integral spiral corrugation and twisted-tape with oblique teeth have been described in Section 2.

Saha and Dutta [7] have observed that, for regularly spaced twisted-tape elements, thermohydraulic performance of twisted tapes with multiple twists in the tape module is not much different from that with single twist in the tape module. Twisted tapes with gradually decreasing pitch perform worse than their uniform-pitch counterparts. Patil [8] have worked with varying width twisted-tape inserts for which both friction factor and Nusselt number are lower than those with full-width twisted tapes. Saha et al. [9,10] have introduced regularly spaced twisted-tape elements which are better than full-length twisted tapes under certain circumstances. Li et al. [11] have designed an optimal multi-layer spacer with optimal non-woven nets in the outer layers and twisted tapes in the middle layer. Helical screw-tape inserts [12] behave the same way as the twisted tapes. Twin and triple twisted

* Corresponding author. Tel.: +91 33 26685043; fax: +91 33 26682916.
E-mail addresses: ppbiet@gmail.com (P.K. Pal), sujoy_k_saha@hotmail.com (S.K. Saha).

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Nomenclature

A	heat transfer area (m^2)
Ac	axial flow cross-sectional area $= WD - \delta D$, (m^2)
A_o	plain duct flow cross-sectional area = $W{\cdot}D$ (m^2)
Cp	constant pressure specific heat (J/kg K)
D	internal diameter of the plain duct (m)
f	fully developed Fanning friction factor $= (1/2)\left[\Delta P'/(\rho V_0^2)\right](D/z)$, 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 (m)
Pr	fluid Prandtl number = $\mu C_p/k$, dimensionless
Ra	Rayleigh number = $Gr \cdot$ 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 axial 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	axial length, the distance between the measuring pressure taps (m)

Greek symbols

α	corrugation helix angle (°)
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

tapes [13] are also effective enhancement devices. Dewan et al. [14] have reviewed the studies on twisted tapes. Hong et al. [15] have employed evenly spaced twisted tapes in a convergent-divergent tube. Jagged twisted tapes perform better than the classical twisted tapes [16].

Sarac and Bali [17] have observed better performance with vortex generators having propeller-type geometry. Jaishankar et al. [18] have observed better performance of twisted tapes with spacer at the trailing edge. Chang et al. [19] experienced enhanced heat transfer in case of shaker-bored piston cooling channel with twisted-tape insert. Two co-rotating helical vortices superimposed over the main swirling flow exist in twisted-tape generated swirl flow, Cazan and Aidun [20]. The close proximity of the two co-rotating vortices creates a local reversing flow at the pipe centerline. Helical vortices are generated by vortices originating inside the twisted tape swirler. The main rotational flow accelerates the co-rotating vortices and decelerates the counter-rotating vortices. As a result, the counter-rotating vortices disappear while the co-rotating vortices reach the same maximum tangential velocity as the main flow. Thus the tangential velocity near the wall is approximately doubled by the presence of the secondary vortices.

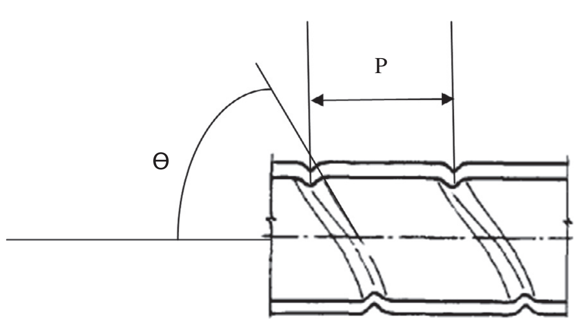

Fig. 1. Integral spiral corrugation in a circular duct.

Ramakrishna et al. [21] have recently worked with twisted-tape having spaces in between. Hans et al. [22] have made a review of various roughness element geometries employed in solar air heaters for performance enhancement.

Saha and co-workers [23–27] have studied experimentally laminar flow through square and rectangular ducts having twisted tapes with oblique teeth, axial corrugations, transverse ribs and wire coil inserts.

It has been observed from the literature review that the combined effect of integral spiral corrugation roughness and twisted-tape with oblique teeth has not been studied in the past. The spiral fluid flow due to integral spiral corrugation 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 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 corrugation 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