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Analysis of Coiled-Tube Heat Exchangers to Improve Heat Transfer Rate With Spirally Corrugated Wall

Prof. Pravin R. Ingole[1], Prof. Swapnil L. Kolhe[2]

[1](Department Of Mechanical Engg., D.M.I.E.T.R, Wardha, India)
[2](Department Of Mechanical Engg., D.M.I.E.T.R, Wardha, India)

ABSTRACT: *Steady heat transfer enhancement has been studied in helically coiled-tube heat exchangers. The outer side of the wall of the heat exchanger contains a helical corrugation which makes a helical rib on the inner side of the tube wall to induce additional swirling motion of fluid particles. Numerical calculations have been carried out to examine different geometrical parameters and the impact of flow and thermal boundary conditions for the heat transfer rate in laminar and transitional flow regimes. Calculated results have been compared to existing empirical formula and experimental tests to investigate the validity of the numerical results in case of common helical tube heat exchanger and additionally results of the numerical computation of corrugated straight tubes for laminar and transition flow have been validated with experimental tests available in the literature. Comparison of the flow and temperature fields in case of common helical tube and the coil with spirally corrugated wall configuration are discussed. Heat exchanger coils with helically corrugated wall configuration show 80–100% increase for the inner side heat transfer rate due to the additionally developed swirling motion while the relative pressure drop is 10–600% larger compared to the common helically coiled heat exchangers. New empirical Co-relation has been proposed for the fully developed inner side heat transfer prediction in case of helically corrugated wall configuration.*

Keywords: *Heat exchanger, swirling motion, corrugated, regimes, boundary condition*

I. Introduction

Helically coiled-tube heat exchangers are one of the most common equipment found in many industrial applications ranging from solar energy applications, nuclear power production, chemical and food industries, environmental engineering, and many other engineering applications. Heat transfer rate of helically coiled heat exchangers is significantly larger because of the secondary flow pattern in planes normal to the main flow than in straight pipes. Modification of flow is due to the centrifugal forces (Dean roll cells) caused by the curvature of the tube. Several studies have been conducted to analyze the heat transfer rate of coiled heat exchangers in laminar and turbulent flow regimes..Numerical study of laminar flow and forced convective heat transfer in a helical square duct has been carried out by Jonas Bolinder and Sunden .Many authors investigated experimentally the turbulent heat transfer in helical pipes .Further enhancement of heat transfer rate in coiled pipes has great importance in several industrial applications mainly where the flow regime is in the laminar or transitional zone like hot water solar energy applications. There are basically two different concepts to increase the rate of heat transferred, the first one is the active and the other one is the passive method. Many different active techniques exist to increase the heat transfer rate mostly for straight pipes. In case of passive techniques heat transfer enhancement by chaotic mixing in helical pipes has great importance and investigated by Kumar and Nigam and Acharya et al. Helical screw-tape inserts have been investigated in straight pipes experimental by Sivashanmugam and Suresh. There is considerable amount of work reported in the literature on heat transfer augmentation in straight pipes with different corrugation techniques; helical tape Experimental investigation of thermosyphon solar water heater with twisted tape inserts has been carried out by Jaisankaretal. According to the author's knowledge a few examinations are considered in helically coiled tubes with different passive heat transfer augmentation techniques like inside wall corrugation, helical tape inserts and this question is not studied numerically at all in the available literature. Experimental investigations have been conducted in a helical pipe tube possibly increases the heat transfer rate because of the developed swirling motion. Basic aim of this study is to investigate the impact of different geometrical parameters of the corrugation for the inner side heat transfer rate in case of helical tube.

Fig1. Schematic figure of the corrugated coiled-tube heat exchangers with the following geometrical parameters dp = 20 mm, pc = 40 mm, p = 44.5 mm and = 2 mm

II. Mathematical Formulation

This section provides the basic equations that must be solved to describe the velocity field and the temperature distribution inside the heat exchanger coils. It is well known that, the transition from laminar to turbulent flow in curved pipes occur much higher critical Reynolds number(Re) than in straight pipes. The critical Reynolds number for smooth helical pipes can be estimated by the following formula found in. $Re_{crit=}2100(1+ 12\sqrt{\delta})$.

2.1. Conservation equations

The following set of partial differential equations for U1, U2, U3, P and T as functions of x, y, z describes the flow and temperature field inside a helically coiled heat exchanger. The conservation equations are formulated in the Cartesian coordinate system because the applied flow using the Cartesian system to formulate the conservation equations for all quantities. Description of the entire geometry of the studied problem is incorporated into the generated unstructured numerical grid.

2.1.1. Continuity equation

The continuity equation is formulated in the following manner in Cartesian coordinate system$\frac{\partial}{\partial x}$ $(\rho U i)$ = 0.

2.1.2. Momentum equations

The following equation system is the representation of the momentum equations in Cartesian coordinate system where I, j {1, 2, 3},

$$\frac{\partial}{\partial x}\rho UjUi = -\frac{\partial p}{\partial x_i} + \frac{\partial p}{\partial x_i}\left(n\left(\frac{\partial U_i}{\partial X_i}\right)\right)$$

g is the dynamic viscosity and q is the density of the working fluid.

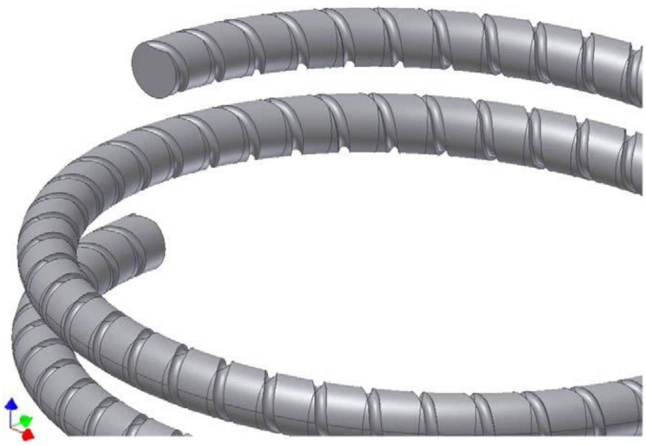

Fig 2. Computational domain with parameters dp = 25 mm, pc = 40 mm, p = 22.25 mm and h = 2.5 mm.

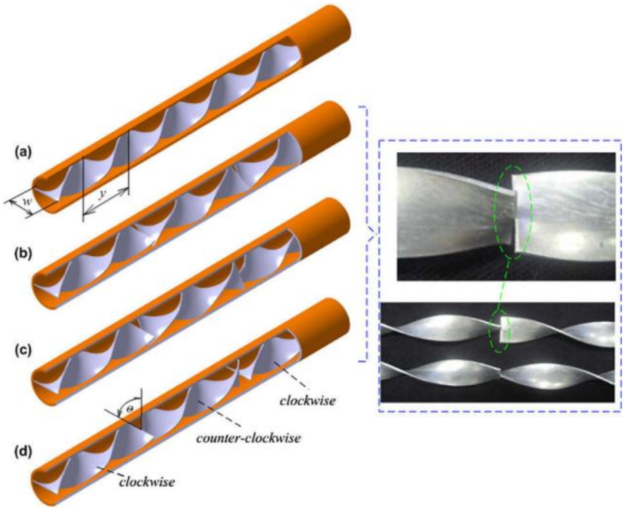

Fig2 (a): horizontal pipes with different tapes

2.1.3. Heat transport equation

The following form of the energy equation is solved to calculate the temperature field

$$\frac{\partial p}{\partial x_i}\left(pc_p U_i T\right) = \frac{\partial}{\partial x_i}\left(\lambda\left(\frac{\partial T}{\partial x_i}\right)\right)$$

Where are the thermal conductivity function and the function of the working fluid at constant?
Fig. 3a shows the generated grid near the outlet region of the corrugated pipe. Fig. 3b presents an enlarged view of the generated grid at the bottom zone of the outlet Diag.

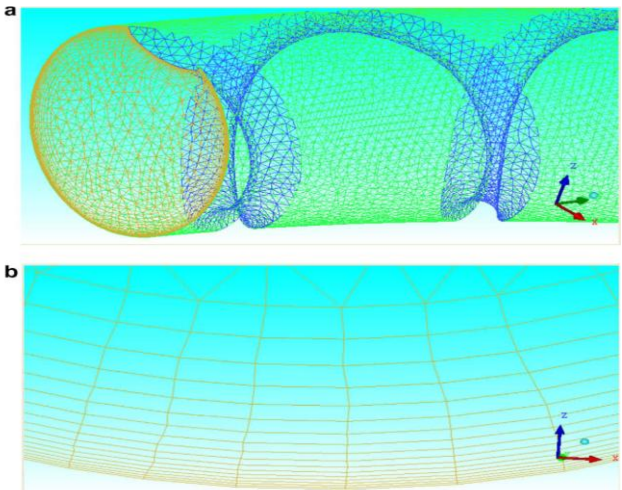

Fig3. One of the applied grids for the calculations and an enlarged view at the bottom of the outlet side of the corrugated coil.

2.5. Calculation of the dimensionless quantities

Representing calculated results the following dimensional andnondimensional quantities have been used. In case of Re number calculations the value of the density and dynamic viscosity of the working fluids have been calculated by averaging for the entire fluid volume. The thermal conductivity of the fluid K (avgas) has been calculated at a specific cross section. It is important to note that the modification of the thermal conductivity between the tube inlet and outlet is not more than 1–2%because of the variation of the fluid temperature.

$$Re = \frac{d_p \rho \bar{v}}{\eta}, \quad Nu = \frac{d_p q_w}{k_{avg}(T_w - T_m)},$$

$$q_w = \frac{1}{A_{wall\,section}} \int \int_A q dA_{wall\,section},$$

$$T_w = \frac{1}{A_{wall\,section}} \int \int T dA_{wall\,section},$$

$$T_m = \frac{1}{\bar{v} A_{wall\,section}} \int \int_A \bar{v} T dA_{cross\,section}.$$

III. Model Validation

Two completely different problems have been tested based on measurements available in the literature. All of the necessary ingredients of the calculations are investigated to extend the validity of the numerical results to the corrugated helical tubes.

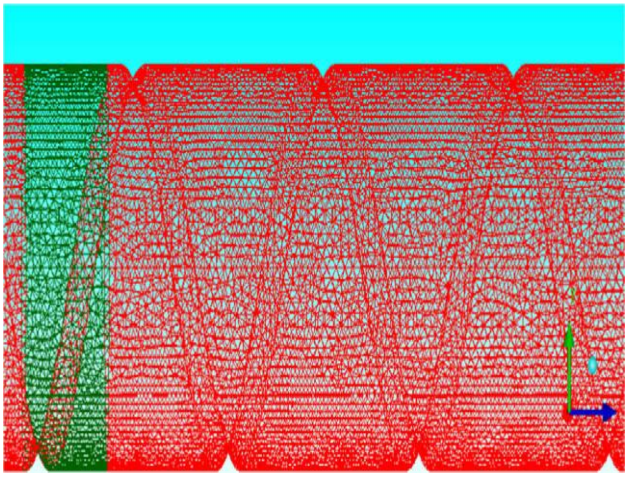

Fig4a. An unstructured grid of a corrugated straight tube with pitch of corrugation p = 15.95 mm and corrugation depth h = 1.03

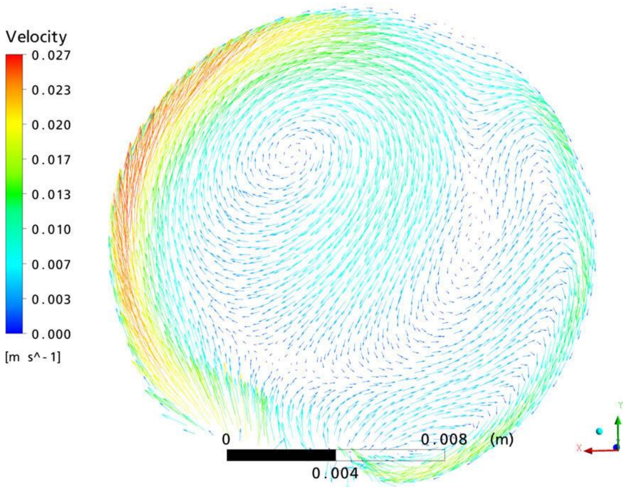

Fig.4 (b): Secondary flow field inside a corrugated straight tube with corrugation parameters (p = 15.95 mm, h = 1.03 mm).

IV. Result

Several geometrical parameters of the studied heat exchanger configuration have been investigated numerically. Length of the coil has been specified according to the assumption that the velocity and temperature field is fully developed near the end of the first turn. For this reason 2 turns configuration of the corrugated helical coil has been investigated because it should be enough to test the development of the peripherally averaged Nussle number of d_{ip} = 20 mm the depths are h = {1, 1.5, 2} mm and in case of d_{ip} = 25 mm the corrugation depths are h = {1.25, 1.875, 2.5} mm.

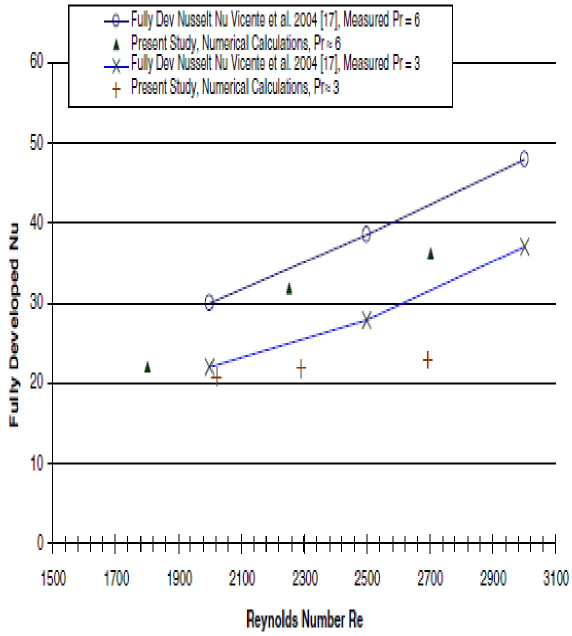

Fig (5): Comparison of the numerical results of a corrugated straight pipe (p = 15.95 mm, h = 1.03 mm) with eased in the numerical calculations

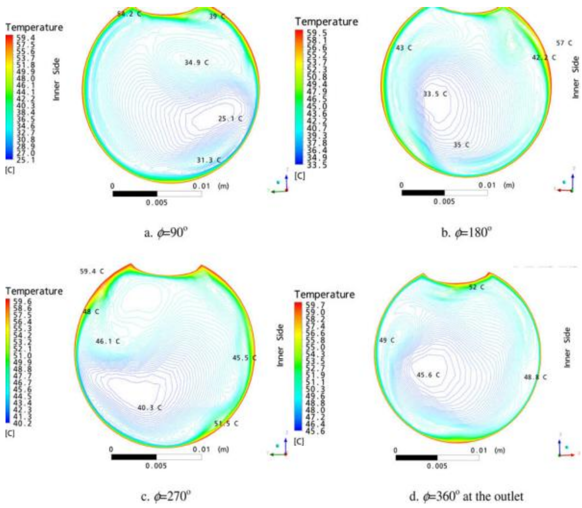

Fig5 (b): Temperature isotherms at different location of a corrugated coiled heat exchanger configuration for De = 1120.

In some cases higher and lower surface temperatures are examined but it was found that the different surface temperatures does not significantly modify the heat transfer rate of the studied heat exchangers. The presented results are valid in laminar and transitional flow regime in the Dean number range 30 < De < 1400 and Prandtl number range 3 < Pr < 30. Fig. presents temperature isotherms at different axial location of a corrugated helically coiled heat exchanger .comparison of the isotherms presented in Fig. With the isotherms of a smooth tube coiled heat exchanger indicates the substantial difference between the two temperature fields. It can be concluded that the temperature field of the corrugated case is far more homogeneous than the smooth tube case helical coil case. This process further increase the cross sectional mixing of the temperature field.

$$Nu = 0.5855 De^{0.6688} Pr^{0.408} \left(\frac{h}{d}\right)^{0.166} \left(\frac{p}{d}\right)^{-0.192}.$$

The formula was obtained via curve-fitting of heat transfer results for the corrugated helical coils. The presented formula is applicable in the following Dean and Prandtl number ranges 30 < De < 1400,

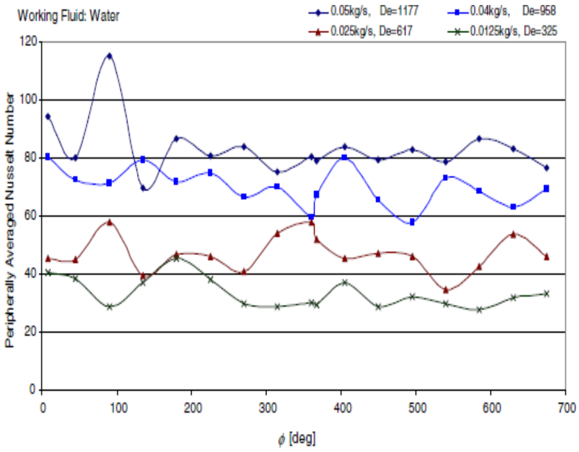

Fig6 (a): Development of the peripherally averaged Nusselt number along the axial direction in case of geometrical parameters dp = 25 mm, p = 22.25 mm, h = 2.5 mm and Pr ≅ 5.

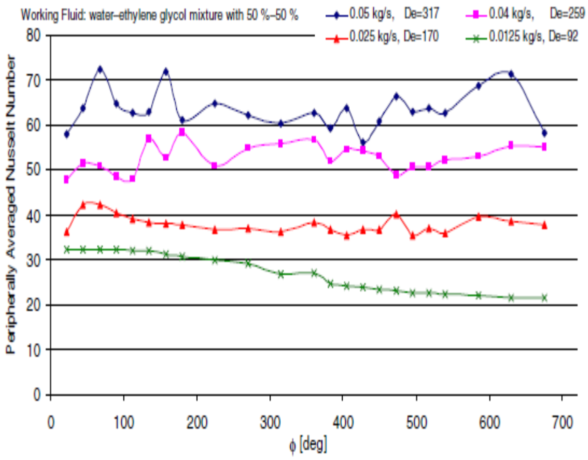

Fig6(b): Development of the peripherally averaged Nusselt number along the axial direction in case of geometrical parameters dp = 20 mm, p = 22.25 mm, h = 2 mm and Pr ≅ 15.

A. Zachár / International Journal of Heat and Mass Transfer xxx (2010) xxx–xxx

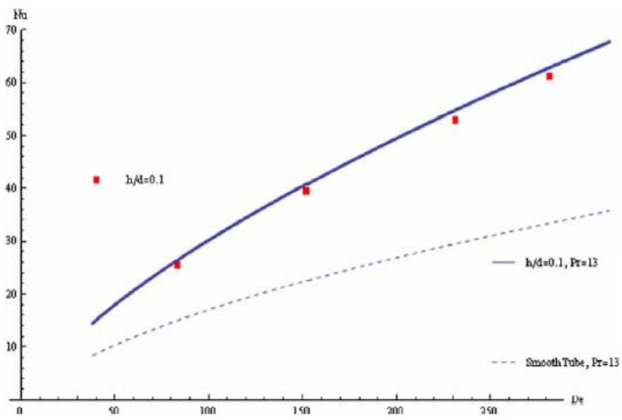

Fig. 14. Nusselt number versus Dean number in case of water–ethylene glycol mixture.

V. Conclusion

Different geometrical parameters of helical corrugation on the outer surface of helically coiled-tube heat exchangers are examined to improve the inside heat transfer rate. Several different in flow rates and temperatures have been studied to test the impact of flow parameters for the efficiency of the heat exchanger. It can be concluded that the heat transfer rate is almost independent from the inlet temperature and the outer surface temperature. As it was expected the volumetric flow rate significantly influences the performance of coiled-tube heat exchangers with and without outer helical corrugation. An empirical formula has been suggested to indicate the dependency of the Nusselt number from the Dean and Prandtl numbers... The presented results show that the ratio of the helical pitch and tube diameter (p/d) and the ratio of the corrugation depth and the tube diameter (h/d) nearly increase or decrease in the same way the heat transfer rate of the studied heat exchangers. The results also show that a spirally corrugated helical tube with corrugation parameters ($p/d = 1$, $h/d = 0.1$) can increase the heat transfer rate nearly 100% larger than a smooth helical pipe in the Dean number range $30 < De < 1400$.

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