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Effect of Wall Heat Flux on Flow and Heat Transfer Characteristics in Coiled Corrugated Pipes

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Abstract: The objective of this paper was to study the influence of high-intensity asymmetric heat flux at the wall on pressure drop and heat transfer in the Archimedean spiral coil made of transversely corrugated tube that was exposed to radiant heating and is supposed to represent heat absorber of parabolic dish solar concentrator. The working fluids were water and a mixture of propylene glycol and water (90% and 10% by volume, respectively). An increase in the heat flux density at the pipe wall affects the increase in the pressure drop and peripherally averaged Nu number in the considered geometry in all flow regimes. The effect is most noticeable in the laminar region and monotonically decreases with increasing Reynolds number and curvature ratio. The intensification of heat transfer is less compared to the increase in pressure drop with increasing heat load at the wall. Quantitative comparison of the obtained data with results in the literature is lacking due to significant differences between the considered geometries and experimental conditions.

Keywords: Corrugated spiral coil, Heat flux density, Pressure drop, Heat transfer.

1. Introduction

Spiral pipes or spiral coils were introduced in 19th century and have been widely used in various thermal engineering applications, such as heat exchangers, electronic cooling, chemical reactors, etc. They have better heat transfer performance and compactness in comparison with commonly used straight tube exchangers, which results in occupying less space. The transport phenomena occurring in the spiral tubes are more complicated than those in straight tubes. Secondary flows observed in the flow patterns, induced by centrifugal force, significantly affect the flow field and heat transfer.

Even though the interest in spiral coiled systems is on the rise, there are still very few published articles on spiral coil tubes. They are less popular compared to helical tubes, which have attracted major attention in the study of coiled tubes for heat transfer. Most of the studies of thermal-hydraulic processes in the coils have been dedicated to the helical coils. Much less work has been reported for the hydrodynamics of flow and heat transfer in spiral coils. There is very little information and correlations on the Nusselt number, and in the absence of appropriate correlations, traditional approach is to use correlations developed for circular or helical tubes with an average curvature.

The property of continuously varying curvature along the length makes spiral coil flows never hydrodynamically and thermally fully developed, unlike the helical coil flows. The velocity profile and dimensionless temperature profiles are changing along the flow direction, as well as the friction factor and Nusselt number. The use of the friction factor in correlations is inappropriate, since the magnitude of the pressure drop per unit length is variable along the axial coordinate of the spiral and increases with the curvature ratio.

The presence of the superimposed secondary convection in the curved channel flow suppresses axial propagation of the initial turbulent fluctuations, so that the curvature stabilizes the flow while the transition from the laminar to the turbulent flow is delayed. The spiral coil flow is characterized by two critical Reynolds numbers [1, 2], the first one when the turbulence has set only in the outer turns with lowest curvature ratio, and the second one, when the inertia forces, even in the innermost turns with highest curvature ratio, are sufficient to overcome the damping effect of the secondary flow. In a spiral coil completely filled with laminar flow slow increase of Reynolds number causes the appearance of the turbulent flow zone at the outer end at the approach of the first critical Reynolds number. Further increase of Reynolds number extends turbulent flow zone toward the inner end of the coil until it completely fills the coil with turbulent flow at the second critical Reynolds number. Moreover, there is a possibility of existence of mixed flow inside the spiral coil, where some length of the coil being in the laminar flow and some other length in the turbulent flow. Consequently, the phenomena of the forward and the reverse laminar-turbulent transition can occur in the spiral coil.

On the contrary, the transversal corrugations act as turbulence promoters, since the turbulence level is increased by a separation and reattachment mechanism. The corrugations act as roughness elements and disturb the existing laminar sublayer. There is the similarity of flow phenomena between heated straight pipes and unheated curved pipes, the techniques developed in treating the flow problem in the latter can be applied to study the flow development in the former. Morton [3] found that the flow in a heated straight pipe is similar to that in a curved pipe. For a heated pipe the secondary flow is created by the buoyancy force inside the thermal boundary layer. Heated pipes also develop vortices that result from the combination of the radial-directional and the downward motions of the fluid particles which are induced by the displacement of the boundary layer and develop along the pipe. A favorable pressure gradient is generated on the bottom wall of the pipe, while an unfavorable pressure gradient is induced on the top wall.

Another interesting character of the flow in heated pipes, revealed by Mori *et al.* [4], is that the secondary flow generated by heating can suppress the turbulence level when the inlet turbulence level is high and can enhance it when its level is low at the inlet. On the other hand, the curvature of the wall stabilizes the flow by delaying the transition to the turbulent flow.

The substantial features of each of these effects (curvature, corrugated wall and heating) are enhanced heat and mass transfer coefficients due to the cross-sectional mixing of fluid elements, as well as increased frictional losses. Natural convection and variable material properties play important roles in the heat transfer and fluid flow in a heated pipe. Frequently the prediction of the heat transfer by forced convection alone, without considering the secondary flow induced by buoyancy force, could cause significant errors, especially for laminar flow cases.

It is important to distinguish between the vortex flow within the main flow (in the minimum cross section of the corrugated pipe), which is characterized by the existence of Dean vortex cells, and the recirculation flow inside the corrugations, which is a consequence of the sudden expansion of the flow channel. Although both of these secondary flows are recirculating and vortexing, they differ fundamentally in terms of the mechanisms that cause them and their localization. The vorticity vectors of the secondary flow in the basic section of the tube and the vorticity vectors of the secondary flow inside the corrugations are mutually perpendicular.

Several experimental [5-12] and numerical [13-25] studies have been published that examined the flow and heat transfer phenomena in the smooth and hydraulically rough spiral pipes. Most of these papers investigated laminar flow conditions, while those that investigated turbulent flow conditions are less frequent in literature compared to the previous ones.

The literature review reveals that there is a lack of research related to the flow and heat transfer in transversely corrugated straight tubes characterized by a high value of relative roughness and spiral tube coils with transverse corrugations. Existing studies are generally limited to considering the results for relatively narrow ranges of geometric and flow parameters. Moreover, almost all take into account two common thermal boundary conditions - constant wall temperature and constant heat flux, which is significantly different from the high-intensity asymmetric flux considered in this research.

2. Experiment

The objective of this paper was to experimentally study the distribution of the convective heat transfer in a spirally coiled corrugated tube exposed to radiant heating that is characteristic of parabolic dish solar concentrators. Investigation of the influence of hydraulic, physical and thermal conditions, as well as the geometry of the spirally coiled corrugated heat absorber, on the local intensity of heat transfer and pressure drop was conducted using modern experimental methods.

The Archimedean spiral, with a pitch slightly larger than the maximal outside diameter of the corrugated coiling tube, was selected out of different types of spirals in order to achieve the most favorable ratio of active surface area and the total volume of the heat absorber in the parabolic dish receiver. The geometric characteristics and experimental configurations of the transversely corrugated straight pipe and the transversely corrugated Archimedean spiral coil are shown in Figures 1 and 2, respectively, while Table 1 shows geometric parameters of the tested configuration.

Figure 1 Profile of transverse corrugated pipe made of stainless steel AISI 304 [26]

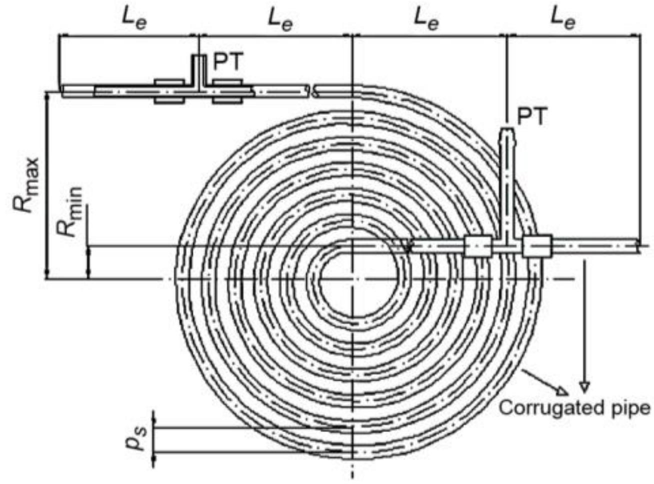

Figure 2 Geometry and configuration of transversely corrugated Archimedean spiral coil

In order to simulate the real working conditions of the heat absorber, an experimental installation was constructed. The installation consists of a spirally coiled corrugated tube of heat absorber with an accompanying flow loop and a radiant heating system. The schematic diagram of the hydraulic system is shown in Figure 2. The entire experimental apparatus enables variations of the following operating parameters: 1) intensity of the incident radiant heat flux; 2) flow rate and flow direction of the working fluid, and 3) angle of inclination of the spiral heat exchanger in relation to the horizontal plane.

Table 1. Geometric parameters of the tested configuration

Transversely corrugated straight pipe				Transversely corrugated Archimedean spiral coil			
d	9.3	mm	minimum internal diameter	R_{min}	25	mm	minimum radius of the coil
d_0	11.7	mm	maximum internal diameter	R_{max}	202	mm	maximum radius of the coil
d_{max}	12.2	mm	maximum external diameter	n	13	-	number of coil turns
s	0.25	mm	wall thickness	L	9.324	m	length of the coil
e	1.2	mm	corrugation depth	L_e	0.5	mm	length of entrance section
p_c	4.2	mm	corrugation pitch	p_s	13.6	mm	spiral coil pitch

The incident heat flux upon the tube external surface varies both in the circumferential and axial direction. It was obtained by the radiant heating system, which is specifically designed for the purposes of this experimental research. Detailed analyses of the radiant heat flux produced by the quartz heating system and radiant absorption characteristics of the corrugated curved tubes could be found in our previous papers [27, 28].

3. Results

3.1. Non-isothermal pressure drop

The previous considerations [2] refer to the hydrodynamic characteristics of the flow in the in spirally coiled corrugated pipes under isothermal conditions. There are few studies in the literature that consider the influence of non-isothermal conditions on the pressure drop in curved pipes, as well as in pipes characterized by high values of relative roughness.

The influence of heat flux at the wall on the pressure drop in the spirally coiled corrugated pipe is shown in Figure 3. The dependence of the pressure drop ratio for diabatic and adiabatic conditions on the Re_{ave} number ($Re_{ave}=(Re_{inlet}+Re_{outlet})/2$) and the total absorbed radiation heat flux averaged over the exposed surface of the absorber is shown.

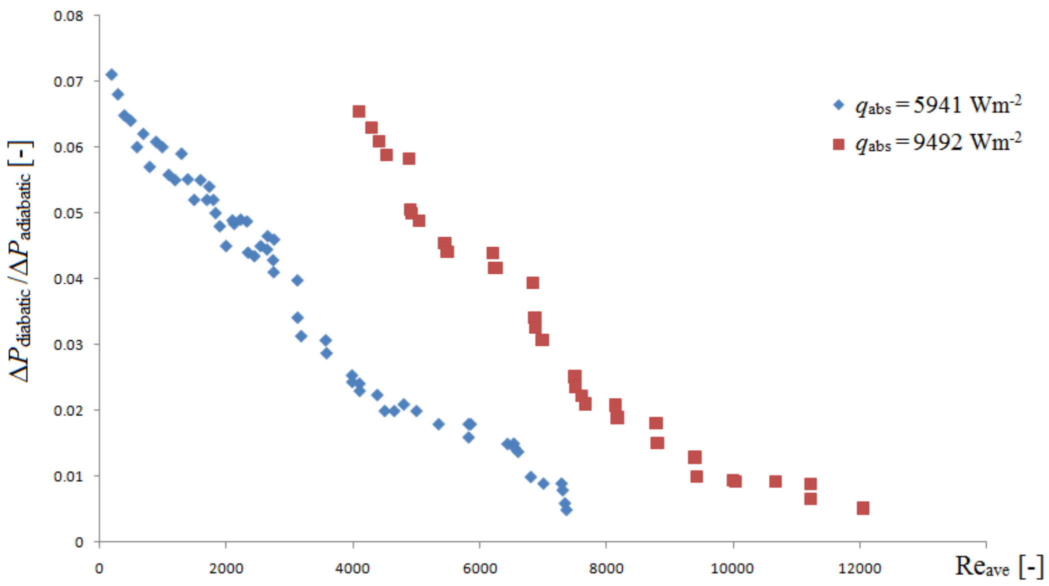

Figure 3. The effect of heat flux at the wall on the pressure drop in the spirally coiled corrugated pipe

Heat transfer causes the increase of the pressure drop in the considered flow geometry in all flow regimes. It could be noted that the largest increase occurs in the laminar region for the smallest values of the Re number. The ratio $\Delta P_{diabatic}/\Delta P_{adiabatic}$ decreases monotonically with increasing Re number for the same values of the heat flux density on the pipe wall. As the heat flux density on the wall increases, the pressure drop also increases.

The pressure drop increases due to the influence of the secondary flow caused by the buoyancy force, which changes the velocity field and thus affects the shear stress on the pipe wall. The effect of the buoyancy force contributes to the intensification of the secondary flow in relation to isothermal flow conditions in curved pipe, when the secondary flow is only under the influence of the centrifugal force. By intensifying the secondary flow, its kinetic energy also increases, which results in an increase in friction losses.

Heat transfer affects the friction factor, which increases with the amount of added heat, especially in the laminar region. The increase is due to buoyancy induced secondary flow that alters the velocity profile, which influences the shear stress at the pipe wall.

With the increase in the value of the heat flux density at the wall, the naturally convective flow becomes more intense, which causes additional resistance to the axial fluid flow. Consequently, with the increase in the heat flux density, the friction coefficient, i.e. the pressure drop, also increases. By exceeding a certain value of Re number, which is correlated with heat flux density at the wall, the flow becomes completely under the influence of the centrifugal force and it can be considered that there is no difference between the pressure drop for isothermal and non-isothermal conditions.

These findings are qualitatively in agreement with results in the literature [29, 30], but a quantitative comparison is lacking due to significant differences between the flow chanel geometries and considered experimental conditions.

3.2. Effect of wall heat flux on heat transfer

The influence of the intensity of the heat flux density at the outer wall of the tube on the heat transfer, i.e. the value of Nu number, is very difficult to estimate for the given experimental conditions. The reason is very small differences in the experimental values of peripherally averaged Nu numbers, which are within the uncertainty limits of the experimentally determined Nu numbers (up to 3.85% [11]). The peripherally averaged Nu number increases with increasing the heat flux density at the wall, approximately 3% at the beginning of the test range (Re≈200) and approximately 1% at the end of the laminar range (Re≈1,300). Based on the experimental data, it is not possible to determine any influence of the intensity of the heat flux density at the wall on the Nu number in the transitional and turbulent region. The reason for this is not only the obvious reduction of the influence of heat flux density on heat transfer with increasing Re number, but also the inherent oscillation of the experimentally determined Nu numbers with flow intensification. There are few studies in the literature that have considered this aspect, and the results are rather speculative.

The influence of operating conditions outside the experimental range was additionally investigated by numerical simulations [31] in order to assess the influence of the intensity of the radiation heat flux density on the Nu number. The values of Nu number were compared for two different operating conditions, where the distributions of the heat flux density of incident thermal radiation on the outer wall of the absorber are proportional to each other. The first case corresponds to experimental conditions when the average value of absorbed radiation heat flux per exposed absorber surface is 9,492 Wm^{-2}, while the second case corresponds to conditions when the average value of absorbed radiation heat flux per exposed absorber surface is 6.36 times higher, i.e. when the mean value of the heat flux density of incident thermal radiation in the plane of the absorber is approximately 100,000 Wm^{-2}.

In order to evaluate the influence of wall heat flux density, the ratio of the peripherally averaged Nu numbers at higher and lower radiation heat flux density Nu_{II}/Nu_{I} was formed. Its dependance on the Re number and curve ratio δ are shown in Figures 4 and 5. Two curvature ratio values were chosen to make the data comparable: δ=0.033 (5th coil turn) and δ=0.102 (12th coil turn).

The results shown in Figure 4 represent the increase in the peripherally averaged Nu number for a mixture of propylene glycol and water (90% and 10% by volume, respectively) in laminar and transitional flow regimes, while Figure 5 shows the corresponding results for water in the turbulent region.

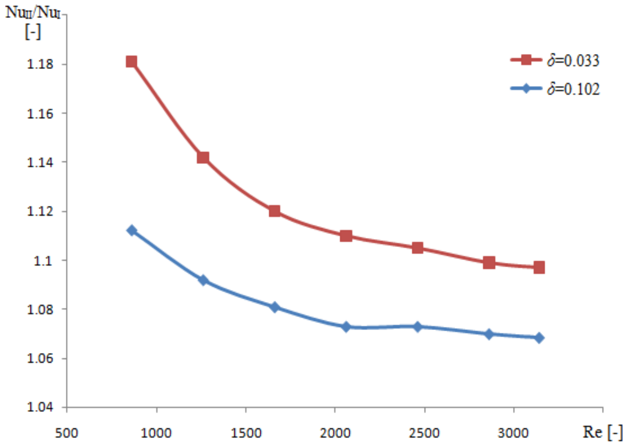

Figure 4. The influence of wall heat flux density on the peripherally averaged Nu number - laminar and transitional flow regimes

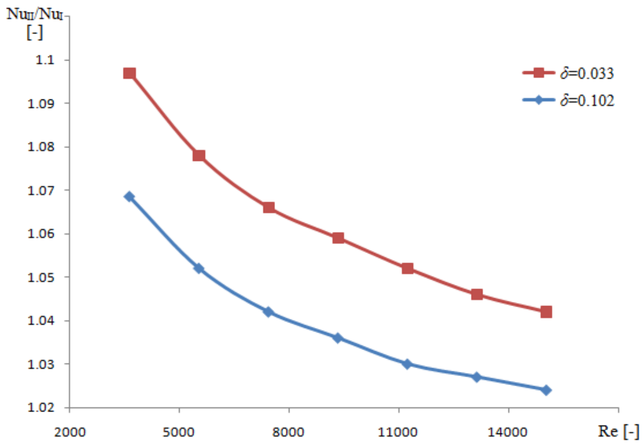

Figure 5. The influence of wall heat flux density on the peripherally averaged Nu number - turbulent flow regime

An increase in the intensity of the heat flux density at the wall affects the increase of the peripherally averaged Nu number in the considered geometry in all flow regimes. It was observed that the influence is greatest in the laminar region for the smallest values of Re number and curvature ratio δ. The Nu_{II}/Nu_{I} ratio monotonically decreases with increasing Re number.

The increase in the peripherally averaged Nu number is a consequence of the influence of the secondary flow caused by the effect of the buoyancy force, i.e. the intensification of the natural convective circulation and the change in the viscosity of the fluid within the boundary layer with the increase in the temperature gradient near the wall. The large heat flux at the wall increases temperature of fluid in the near wall region and significantly decreases the viscosity of fluid along the axial direction. The change in viscosity causes redevelopment of the hydrodynamic and thermal boundary layer, resulting in the formation of the region where the local heat transfer coefficient increases.

The effect of the secondary flow caused by the buoyancy force generally weakens with increasing Re number and curvature ratio δ. Increasing the curvature ratio δ affects the complex superposition of the secondary flows because the centrifugal force dampens the influence of the buoyancy force. On the other hand, by increasing the Re number and the curvature ratio δ, that is, the De number, the vortex flow inside the corrugations intensifies, which additionally suppresses the natural convective circulation.

4. Conclusion

Heat transfer affects the pressure drop increase in the considered geometry in all flow regimes. The influence is the most pronouced in the laminar region and ratio $\Delta P_{diabatic}$ /$\Delta P_{adiabatic}$ decreases monotonically with increasing Re number for the same heat flux density at the pipe wall.

As the heat flux density at the wall increases, so does the pressure drop. For a certain value of the heat flux density at the wall, the value of the Re number can be determined, beyond which the secondry flow is completely under the influence of the centrifugal force and when there is almost no difference between pressure drop for isothermal and non-isothermal conditions.

An increase in the heat flux density at pipe wall affects the increase in the peripherally averaged Nu number in the considered geometry in all flow regimes. The influence is the most noticeable in the laminar region for the smallest values of Re number and curvature ratio δ and monotonically decreases with increasing Re number. The effect of the buoyancy force becomes negligible compared to the effect of the centrifugal force in the considered flow geometry even in the laminar flow regime for higher values of the Re number.

These observations are qualitatively in agreement with results in the literature, but a quantitative comparison is lacking due to significant differences between the considered geometries and experimental conditions.

It can be concluded that with the increase in the heat load at the wall of the considered flow channel geometry, both the pressure drop and the heat transfer increase, but the intensification of the heat transfer is less compared to the increase in the pressure drop.

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