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Thermal and flow characteristics of helical coils with reversed loops

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ABSTRACT

In this experimental study, flow and heat transfer characteristics of water through a newly designed helical coil heat transfer test section were investigated. A structural modification of the traditional helical coil has been implemented to improve its thermal effectiveness. Specifically, a 360° plastic reversed loop was added after each 180° of the main helical coil loop to redistribute the flow's velocity profile, which should have a direct effect on the thermal performance of the helical coil heat transfer test section. Results reveal that reversed loops do enhance heat transfer rates by more than 20%, while pressure drop penalty also increases but not at the same rate. The curvature of the reversed loops with respect to the main coil is a critical factor in terms of thermal performance. Furthermore, as the size of the reversed loop decreases, the heat transfer enhancement and pressure drop penalty increment for the modified helical coil test section increase as well. Accordingly, it is suggested that there may exist an optimum helical coil structural modification for which the best tradeoff between pressure drop penalty and heat transfer enhancement can be found.

1. Introduction

As thermal loads keep increasing in many energy-transport related engineering applications, greater amounts of heat transfer fluids as well as larger heat transfer units are needed to meet the continuously growing cooling demands. One way to fulfill that demand is to utilize advanced heat exchanging devices. Heat exchangers are very effective in terms of thermal energy transport and in particular, coil heat exchangers (CHX) have been considered as the most efficient due to their compactness and good heat transfer performance. It is found that the secondary flows induced by the centrifugal forces due to the curvature of the coil, have a positive effect on thermal performance. Hence, a newly designed helical coil configuration, which includes a 360° reversed plastic coil added to the main helical loop every 180°, may lead to potentially even further heat transfer enhancements.

Many researchers have investigated hydrodynamic and convective heat transfer performance of flows in helical coiled tubes and in CHX, accordingly, as reported selectively and briefly below.

1.1. Flow characteristics in coiled tubes

The flow characteristics in coiled tubes have been studied over the years. Dean [1] first discovered the presence of a secondary flow field in coiled tubes, and formulated a non-dimensional (Dean) number to account for the effects of curvature ratio and Reynolds number on secondary flows. The Dean number is defined as follows:

$$De = Re \cdot \sqrt{\frac{r}{R}} = \frac{\rho \cdot u \cdot d}{\mu} \cdot \sqrt{\frac{r}{R}} \tag{1}$$

where ρ, μ, u, d, r and R are the density, dynamic viscosity of the fluid, fluid mean velocity, inner diameter of the tube, inner radius of the tube, curvature radius of the coil, respectively.

Dravid et al. [2] further emphasized the existence of a secondary flow field in coiled tubes. Numerical studies were conducted under laminar flow regime with Dean Numbers above 100. Their results revealed that cyclic oscillations in coil wall temperature occurred along the tube axis, due to the presence of secondary flows, and those oscillations would die out downstream of the coil.

Pakdaman et al. [3] characterized the performance of nanofluids flowing inside helically coiled tubes. It was found that the heat transfer enhancement is more significant than pressure drop, for

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Nomenclature

A	empirical constant
CHX	coil heat exchanger
CH1	chiller
d	inner tube diameter, m
D	coil tube diameter, m
DAQ	data acquisition
De	Dean number
DPT	differential pressure transducer
f	friction factor
FM	flow meter
h	heat transfer coefficient, kW/m^2-°C, K
HX	heat exchanger
k	thermal conductivity of the fluid, W/m-°C, K
L	tube length, m
Nu	Nusselt number
p	pitch of the coil, m
P	pump
PH	preheater
Pr	Prandtl number
ΔP	pressure difference between inlet and outlet of the tube, kPa
q''	heat flux, kW/m^2
r	tube inner radius, m
R	coil radius, m
Re	Reynolds number
R^2	coefficient of determination
SS1	sample station
T	temperature, °C, K
u	fluid velocity, m/s
x	axial position along the tube, m

Greek symbols

ρ	fluid density, kg/m^3
η	parameter ratio between regular helical coil and modified coil

Subscripts

b	bulk
c	coil
cri	critical condition
r	radius of test section with reversed loops
S	straight pipe
w	tube wall/surface

a helically coiled tube compared to a straight tube. Huttl et al. [4] numerically studied turbulent flows in straight, curved and helically coiled pipes. It was found that a secondary flow would be generated due to the curvature of the pipes, which make the flow structures different from that in straight pipes.

Mishra and Gupta [5] experimentally measured pressure drop of water flowing through helical coils in both laminar and turbulent regimes. For turbulent flows, the following correlation was obtained based on the least-square analysis of the data.

$$f_c = f_s + 0.0075 \cdot \left(\frac{d}{D_c}\right)^{0.5} \tag{2}$$

$$D_c = 2R_c\left[1 + \left(\frac{p}{2\pi R}\right)^2\right] \tag{3}$$

for $4000 < Re < 100{,}000$,

$0.00289 < r/R < 0.155$

$0 < p/D_c < 25.4$

where *d* is the inner diameter of the coil tube, *Dc* is the effective curvature diameter of the coil defined in correlation (3), *p* is the pitch of the helical coil. The Fanning friction factor (f_s) for a straight tube under turbulent flow condition was defined as,

$$f_s = \frac{0.079}{Re^{0.25}} \tag{4}$$

where $4000 < Re < 100{,}000$.

Srinivasan et al. [6] conducted an experimental study comparing twelve coils with curvature ratios ($\frac{d}{D}$) from 0.0097 to 0.135. The experiments were carried out with water and oil under both laminar and turbulent conditions. A correlation for friction factor for laminar flows was defined based on Seban and McLaughlin's approach [7], as follows:

$$f_c = \frac{6.7\sqrt{\frac{d}{D}}}{De^{0.5}} \tag{5}$$

where *D* is the coil diameter, $30 < De < 300$, $0.0097 < d/D < 0.135$. Seban and McLaughlin [7], also provided a correlation for the friction factor in the transitional region, as follows:

$$f_c = \frac{1.8}{\left(Re\sqrt{\frac{D}{d}}\right)^{0.5}} \tag{6}$$

where $300 < De < De_{cri}$, $0.0097 < d/D < 0.135$, and De_{cri} is the critical Dean number. The critical Reynolds number was defined by Ito et al. [13], as follows:

$$Re_{cri} = 2100\left(1 + 12\left(\frac{r}{R}\right)^{0.5}\right) \tag{7}$$

In addition, the friction factor for turbulent flow conditions can be predicted based on Huttl et al. [3], as follows:

$$f_c = \frac{0.084 \cdot \left(\frac{d}{D}\right)^{0.2}}{De^{0.2}} \tag{8}$$

where $De_{cri} < De < 1400$, $0.0097 < d/D < 0.135$.

1.2 Heat transfer characteristics in coiled tubes

Seban et al. [7] carried out heat transfer experiments for laminar flow of oil and turbulent flow of water in coiled tubes with coil-to-tube diameter ratio of 17 and 104 for a Reynolds number range from 12 to 65000. For turbulent cases, the heat transfer coefficients on the outer wall surface were found to be 2 and 4 times greater than those of inner wall for curvature ratios of 104 and 17, respectively. It was found that the peak of the mean velocity profile in the coil shifted to the outer side of the tube, which led to higher velocities near the outer side than near the inner side of the coil. An empirical correlation for the local Nusselt number of laminar flows in coiled tubes was proposed based on their experimental results, as follows:

$$Nu_c = A\left[\left(\frac{f_c}{8}\right) \cdot Re^2 \cdot \left(\frac{d}{x}\right)\right]^{1/3} \cdot Pr^{1/3} \tag{9}$$