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ANALYTICAL DESIGN OF HELICALLY COILED HEAT EXCHANGERS USED FOR TRANSIENT HEAT TRANSFER APPLICATIONS

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ABSTRACT

In the present study, a special application of a helical heat exchanger, i.e. its usage in the cooling process of a reservoir's hot fluid at a specified finite time was considered. An analytical approach was developed to anticipate the required heat transfer area of the heat exchanger. Also, the effects of the various flow and geometry parameters of the heat exchanger like the mass and initial and target temperatures of the reservoir fluid, mass flow rate and inlet temperature of the tube-side fluid, the diameter, curvature and pitch of the coiled tube, etc, and also the finite time allocated to heat transfer were investigated. Next, the effects of different parameters upon the required length were examined in appropriate curves. It was generally observed that transition from laminar to turbulent regime, enhanced heat transfer coefficient and consequently, reduced the required length of the heat exchanger.

INTRODUCTION

Nowadays, coiled tubes are widely used in heat exchangers because of their compactness and heat transfer enhancement due to the centrifugal force resultant from the curvature of the tubes [1, 2]. Therefore, the coiled tube heat exchangers are now used in a wide variety of applications including chemical industries, industrial and marine boilers, power plants, food industries, refrigeration, and air conditioning systems [3-5].

Rabin and Korin [6] analyzed the performance of a helical exchanger for long-term thermal energy storage in soil experimentally and theoretically. They studied the effect of pitch distance of the helical coil and concluded that it has not a significant effect on the seasonal thermal energy storage.

Laminar natural convection from horizontal helical tubes in air was investigated experimentally by Ali [7]. He correlated the average heat transfer coefficients with Rayleigh number using coiled tube diameter and horizontal coil axis distance as characteristic length. He further studied the natural convection heat transfer from vertical helical coiled tubes to glycerol-water solution experimentally [8]. He obtained average heat transfer coefficient, for laminar and transition to turbulent natural convection of shell side and proposed some correlations to calculate the Nusselt number of natural convection.

Prabhanjan et al. [9] investigated the natural convection heat transfer from helically coiled tubes in water experimentally. They correlated the outside Nusselt number to the Rayleigh number using different characteristic lengths and found out that the best characteristic length was the total height of the coil. Natural convection from uniformly heated helical pipes oriented vertically and horizontally was studied experimentally by Moawed [10]. He investigated the geometrical effects of the coils on the average heat transfer

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coefficient and proposed a new correlation to calculate the outside Nusselt number for each orientation.

Salimpour [11] investigated laminar forced convective heat transfer coefficients of both shell- and tube-side of shell and coil tube heat exchangers experimentally. He also, proposed correlations to evaluate the Nusselt numbers of tube-side and shell-side of the heat exchangers.

Mori and Nakayama [12] studied the coiled tube heat transfer theoretically and experimentally for tubes with constant temperature and proposed a correlation of tube-side Nusselt number in turbulent regime.

As is seen from the works done on the coiled tube heat exchangers, these studies were performed in steady state conditions. But, a special kind of coiled tube heat exchangers is for cooling or heating a reservoir fluid. As an example, a petrochemical viscous fluid that shall be pumped may be considered. It is evident that for a highly viscous fluid, the required pumping power is also high. Therefore, as a practical solution, this fluid is heated to a higher temperature which leads to a lower viscosity of fluid and consequently lower pumping power would be consumed. An important point is the time at which the target temperature shall be arrived, i.e. a time limitation is often determined for heating process.

Another application of such heat exchangers is in cooling of a food product in a prescribed time till the final product meets special conditions. In such cases, the product characteristics are strong function of the cooling time. As seen from these examples, heat transfer in a finite time is of great importance in some industries. Reviewing the literature, no special work devoted to this problem was found. Therefore, in the present study, the transient heat transfer in helically coiled tube heat exchangers was duly dealt and an analytical approach to design such heat exchangers was proposed.

NOMENCLATURE

A	surface of coiled tube *(m^2)*
b	coil pitch *(m)*
C_P	specific heat capacity *J/(kg.K)*
d	coiled tube diameter (m)
De	Dean number, $= \mathrm{Re}(d/2R_c)^2$
H	coil height (m)
h	averaged convective heat transfer coefficient, *$W/(m^2.K)$*
k	thermal conductivity, *W/(m.K)*
L	heat exchanger length (m)
$\dot{m}_i$	tube-side fluid flow rate *(kg/s)*
m_o	reservoir fluid mass *(kg)*
Nu	Nusselt number
Pr	Prandtl number, $= \mu C_P/k$
Q	heat transfer *(J)*
R_C	curvature radius *(m)*
Re	Reynolds number
T	temperature *(K)*
$T_{caloric}$	caloric mean temperature *(K)*
$T_{i,i}$	tube-side inlet temperature *(K)*
T_{o1}	reservoir fluid initial temperature *(K)*
T_{ot1}	reservoir fluid target temperature *(K)*
t	time *(s)*
t_1	specified finite time *(s)*
x	spatial coordinate
α	dimensionless parameter, $= d_i h_i / d_o h_o$
β	parameter to simplify Eq. 5, $= \frac{\alpha}{1+\alpha}\pi d_o h_o$
γ	parameter to simplify Eq. 10, $= \beta L/(1+\frac{\beta L}{2\dot{m}_i Cp_i}) d_i h_i / d_o h_o$
γ_p	dimensionless pitch, $= b/(2\pi R_c)$
δ	parameter to simplify Eq. 14, $\gamma / m_o Cp_o$
ave	average value
i	inside of coiled tube
o	outside of coiled tube
w	at tube wall temperature

FORMULATION OF THE PROBLEM

As the analysis of the problem for both heating and cooling is similar, for simplicity only cooling was considered.

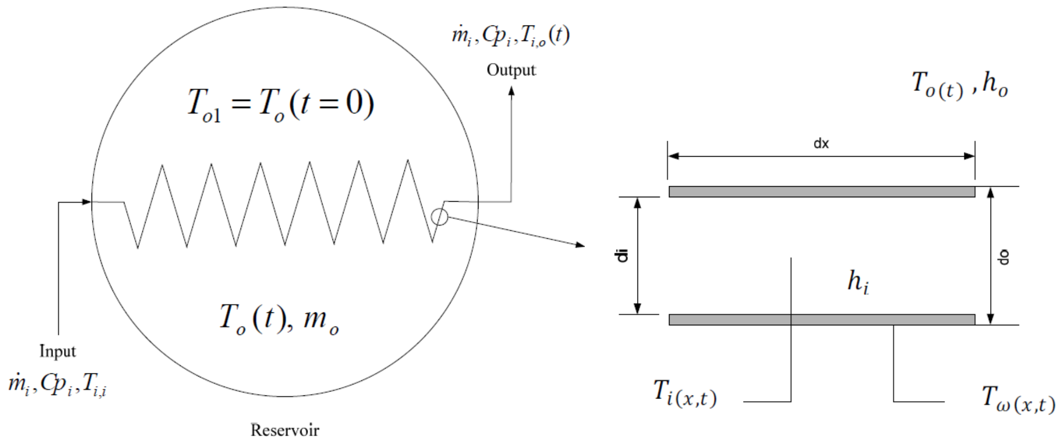

Figure 1. Schematic view of an element of the coiled tube situated in a reservoir.

Figure 1 represents an element of the coiled tube situated in a reservoir. For this element, heat transfer rate can be written as:

$$d\dot{Q}(x,t) = h_o.dA.[T_o(t) - T_w(x,t)] \quad (1)$$

As $dA = \pi d_o dx$, Eq.1 would have the following form

$$d\dot{Q}(x,t) = \pi.d_o.h_o.[T_o(t) - T_w(x,t)]dx \quad (2)$$

Similarly, we can write:

$$d\dot{Q}(x,t) = \pi.d_i.h_i.[T_w(x,t) - T_i(x,t)]dx \quad (3)$$

Where, $T_i(x,t)$ is the bulk temperature of tube-side fluid at distance *x* of the tube inlet and time of *t*.

Combining Eqs. 2 and 3 and defining $\alpha = d_i.h_i/(d_o.h_o)$, we will have:

$$T_w(x,t) = \frac{T_o(t) + \alpha T_i(x,t)}{1+\alpha} \quad (4a)$$

Where, $T_o(t)$ is the average temperature of the reservoir fluid at time of *t*.

The average temperature of wall, $T_{w,ave}$, can be defined as:

$$T_{w,ave} = \frac{T_{o,ave} + \alpha T_{i,ave}}{1+\alpha} \quad (4b)$$

Where, $T_{o,ave} + T_{caloric}$, and $T_{i,ave} = \dfrac{T_{i,i} + T_{i,ot}}{2}$.

Invoking Eq.4a, one can write Eq. 2 as:

$$d\dot{Q}(x,t) = \pi.d_o.h_o.[T_o(t) - \frac{T_o(t)}{1+\alpha} - \frac{\alpha T_i(x,t)}{1+\alpha}]dx \quad (5)$$

Defining $\beta = \frac{\alpha}{1+\alpha}\pi d_o h_o$, and simplifying Eq.5, we will have

$$d\dot{Q}(x,t) = \beta[T_o(t) - T_i(x,t)]dx \quad (6)$$

Taking integral from the above equation, the following equation will be arrived at:

$$Q(t) = \beta L \int_0^t T_o(t)dt - \beta \int_0^t \int_0^L T_i(x,t)dxdt \quad (7)$$

Assuming that $T_w(x,t)$ is a linear function of *x*, $T_i(x,t)$ can be written as:

$$T_i(x,t) = \frac{\dot{Q}(t)}{\dot{m}_i.C_{p,i}} \times \frac{x}{L} + T_{i,i} \quad (8)$$

Where, $\dot{m}_i, C_{p,i}$ & $T_{i,i}$, are the mass flow rate, specific heat, and inlet temperature of tube-side fluid, respectively; and L is the coiled tube length.

Taking integral from Eq. 8, we obtain

$$\int_0^t T_i(x,t)dx = \frac{\dot{Q}(t)}{\dot{m}_i.C_{p,i}} \times \frac{L}{2} + T_{i,i}.L \quad (9)$$

Considering Eq. 7 and 9, we get

$$Q(t) = \beta L \int_0^t T_o(t)dt - \beta L \int_0^t (\frac{\dot{Q}(t)}{2.\dot{m}_i.C_{p,i}} + T_{i,i})dt \quad (10)$$

Simplifying Eq. 10 and defining $\gamma = \dfrac{\beta L}{1 + \dfrac{\beta L}{2.\dot{m}_i.C_{p,i}}}$, we can write:

$$Q(t) = \gamma \left[\int_0^t T_o(t)dt - T_{i,i}.t \right] \quad (11)$$

Also, the following equation can be written for the fluid in the reservoir:

$$\frac{dQ(t)}{dt} = -m_o C_{po} \frac{dT_o(t)}{dt} \quad (12)$$

Where, m_o and C_{po} are the mass and specific heat of the reservoir fluid, respectively.

Eq. 12 leads to

$$T_o(t) = \frac{-Q(t)}{m_o C_{po}} + T_{o1} \quad (13)$$

Considering Eqs. 11 and 13 one can write

$$\frac{dQ(t)}{dt} = \gamma \left[\frac{-Q(t)}{m_o C_{po}} + T_{o1} - T_{i,i} \right] \quad (14)$$

Defining $\delta = \gamma / (m_o C_{po})$, and resolving this differential equation, we get the following equation:

$$Q(t) = m_o C_{po} [T_{o1} - T_{i,i}][1 - e^{-\delta t}] \quad (15)$$

Considering the reservoir fluid, the total heat transfer from $t = 0$ till $t = t_1$ can be calculated as

$$Q(t_1) = m_o C_{po} [T_{o1} - T_o(t_1)] \quad (16)$$

Comparing Eqs. 15 and 16 we get

$$(T_{o1} - T_{i,i})(1 - e^{-\delta t_1}) = [T_{o1} - T_o(t_1)] \quad (17)$$

Finally we can write:

$$\delta = \frac{\ln\left(\dfrac{T_{o1} - T_{i,i}}{T_{o,t1} - T_{i,i}}\right)}{t_1} \quad (18)$$

where, $T_{o,t1}$ is the target temperature of the reservoir fluid.

We know that $\delta = \gamma / (m_o C_{po})$ and $\gamma = \dfrac{\beta L}{1 + \dfrac{\beta L}{2.\dot{m}_i.C_{p,i}}}$;

hence, we can write:

$$L = \frac{\gamma}{\beta - \dfrac{\beta.\gamma}{2.\dot{m}_i.C_{p,i}}} \quad (19)$$

Knowing the heat transfer time, t_1; the coolant inlet temperature, $T_{i,i}$; the hot fluid initial temperature, T_{o1}; and its final temperature, $T_{o,t1}$; δ will be calculated and consequently γ can be evaluated. After that, according to the definition of *γ*, *βL* can be calculated. As $\beta = \dfrac{\pi.d_o.h_o.d_i.h_i}{d_o.h_o + d_i.h_i}$, the only unknowns for evaluation of *β* are h_i and h_o.

To evaluate the tube side heat transfer, two flow regimes were considered: Laminar and turbulent.

For laminar flow, the following correlation proposed in [11] was adopted:

$$Nu_d = 0.152 De^{0.431} \Pr^{1.06} \gamma_p^{-0.277} \quad (20)$$

Where, *De, Pr,* and γ_p are Dean number, Prandtl number, and dimensionless coil pitch, respectively.

For turbulent flow, the correlation proposed in [12] was considered:

$$Nu_d = \frac{\Pr^{0.4}}{41.0} \mathrm{Re}^{\frac{5}{6}} (d/R_c)^{\frac{1}{12}} \left\{ 1 + 0.061 [\mathrm{Re}(d/2R_c)^{2.5}]^{\frac{1}{6}} \right\} \quad (21)$$

Where, $h_i = Nu_d . \dfrac{k}{d}$, and R_c is the curvature radius.

Also, the transition from laminar to turbulent flow occurs at the following Reynolds number [13]:

$$\mathrm{Re}_{cr} = 2100[1 + 12(d/2R_c)^{0.5}] \quad (22)$$

To calculate the natural convection heat transfer coefficient, the following correlation proposed in [8] was used:

$$Nu_o = 0.0749 Ra_H^{0.3421} \quad 9 \times 10^9 < Ra_H < 4 \times 10^{11} \quad (23)$$

Where, $h_o = Nu_o . \dfrac{k}{H}$, and H is the coil height which can be calculated as

$$H = \frac{Lb}{2\pi \sqrt{R_c^2 + (\dfrac{b}{2\pi})^2}} \quad (24)$$

Where, L and b are the coil tube length and pitch, respectively.

Having evaluated *βL* and *β*, now the required length of the coiled tube, L, can be calculated.

The flowchart for predicting the required length of the helical heat exchanger is presented in Figure 2.

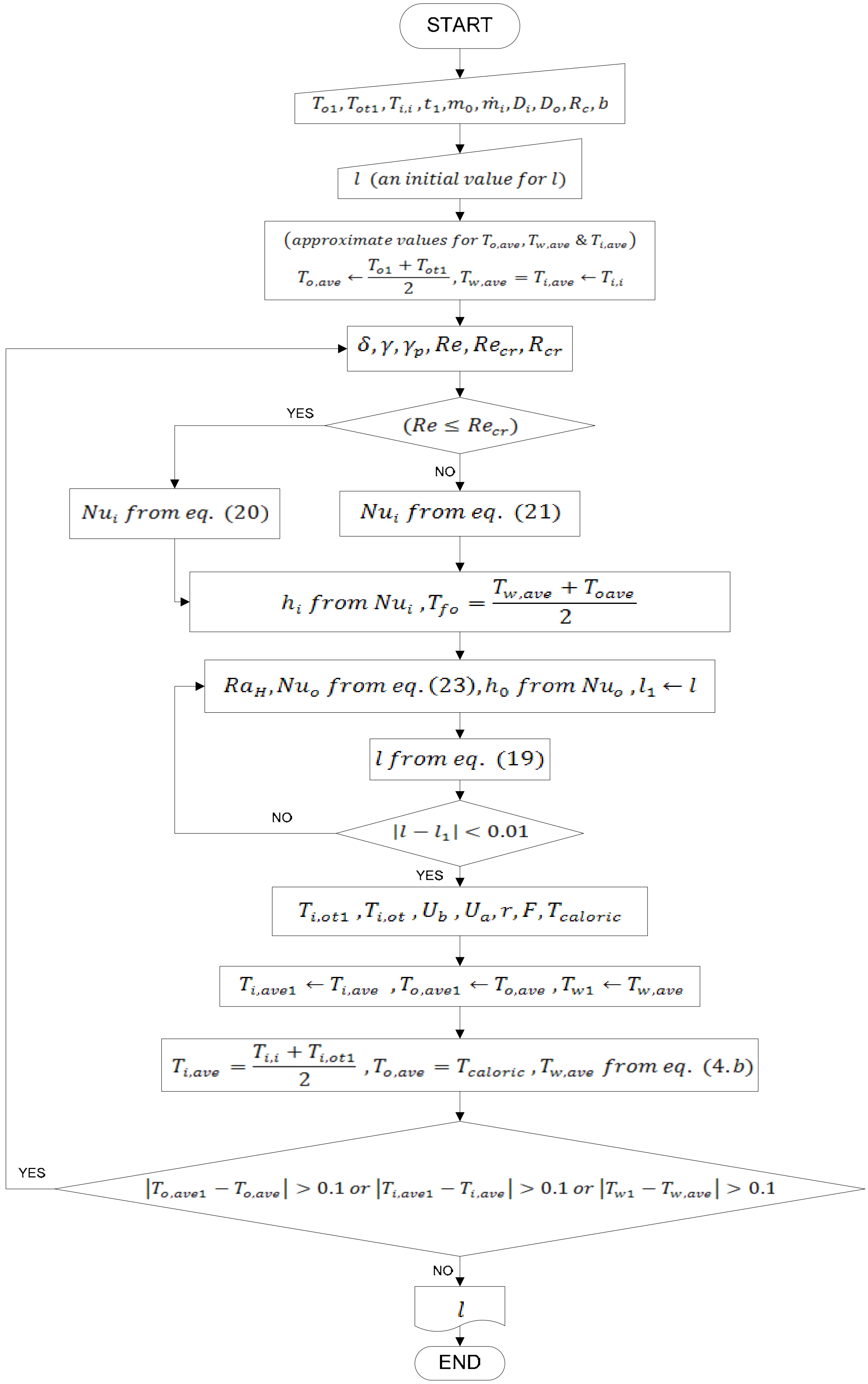

Figure 2. Flowchart to calculate the required length of the helical heat exchanger.

RESULTS AND DISCUSSION

At this section, the effect of various geometrical and flow parameters on the required length of a helical heat exchanger was investigated. As the required length of this kind of heat exchanger and especially for this type of application is a function of various parameters, a base case for different parameters was considered as: $\dot{m}_i = 0.17kg/s$, $m_o = 322kg$, d_i=0.04, t_1=1200 s, R_c=0.35 m, b=0.075 m, $T_{i,i}$=290 K, T_{o1}=365 K, and T_{ot1}=345 K. To generate the diagrams of this section, except the variable parameters of each diagram, the other parameters were considered at base case condition.

Figure 3 represents the variation of the required length of a helical heat exchanger versus the finite specified time of heat transfer.

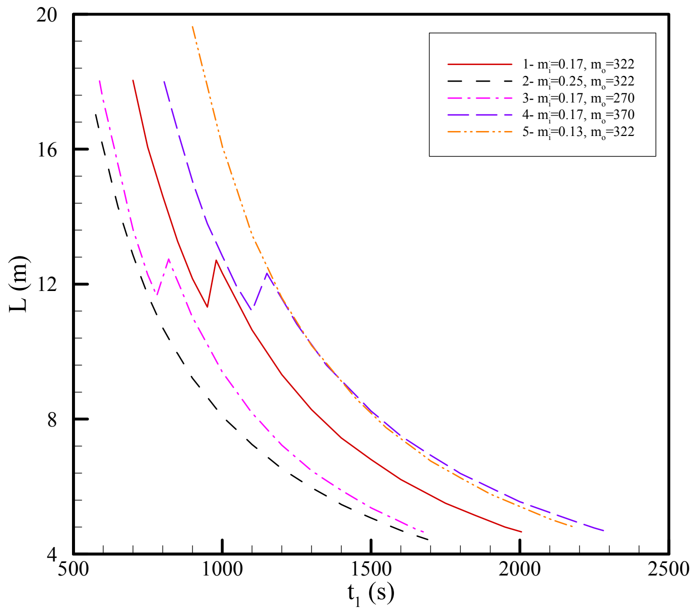

Figure 3. Variation of the required length of a helical heat exchanger with the specified time.

In this figure, curve 1 was considered as a reference and other curves would be compared to that. From this curve, it is observed that increasing the time limit leads to a lower required length of heat exchanger. However, this increase in time results in a leap in the required heat transfer area, because at this point, turbulent to laminar transition occurs, which in turn decreases the tube-side convective heat transfer coefficient. Therefore, the overall heat transfer coefficient falls and more heat transfer area would be required, viz. despite the more allowable time of heat transfer; more heat exchanger length is needed. Thus, in designing such a heat exchanger, special care must be taken to avoid the transition region. At this figure, curves 1, 3, & 4 were plotted for reservoir fluid mass values of 322, 270, & 370, respectively. As is seen from figure 1, these curves have similar trends and only are shifted horizontally which means in the present range, the mass of reservoir fluid does not affect on the tube-side flow regime transition. It is also evident that the more fluid in the reservoir requires more surface area for heat transfer at a given finite time, expectedly.

To study the effect of tube-side mass flow rate, curves 2 & 5 were plotted for $\dot{m}_i = 0.25kg/s$ & $0.13kg/s$, respectively. Comparing curve 2 with curve 1 reveals that increasing the tube-side flow rate causes less required heat transfer area, because the flow regime was fully turbulent (no leap was observed in the required length) which leads to a higher heat transfer coefficient and consequently, less heat exchanger length would be needed. Investigating curve 5 shows a reverse trend for lowering tube-side flow rate, i.e. when tube-side flow rate was reduced, the flow regime became fully laminar which resulted in a lower heat transfer coefficient and more required heat transfer surface area.

To investigate the effect of reservoir fluid mass on the required length of heat exchanger in more details, figure 4 was illustrated.

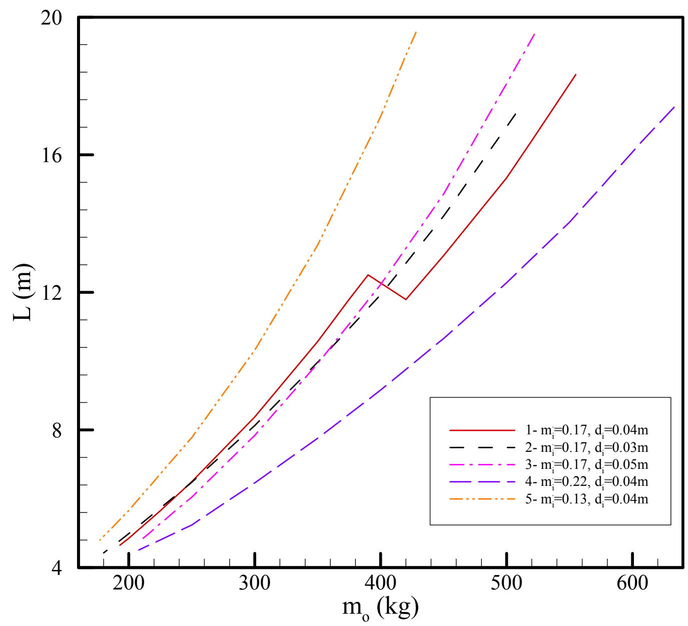

Figure 4. Effect of reservoir fluid mass on the required length of heat exchanger.

From curves of this figure, it is seen that albeit in low reservoir fluid mass, tube diameter and tube-side flow rate have only a slight effect on the required length; however, this effect becomes more discernible in higher fluid mass of reservoir. Also, it is observed that reducing and increasing the tube diameter result in fully turbulent and fully laminar regimes, respectively. The interesting point is that both of these curves, i.e. curves 2 & 3, are very close to each other; and also, at high values of reservoir fluid mass (m>400 kg), tube diameter d_i=0.04 m has the best performance; whilst, this tube at low values of reservoir fluid mass (m<400 kg) needs the most heat transfer surface area. This matter can be explained as, reducing the tube diameter has two opposing effects, viz. reducing the surface area and on the other hand changing the flow regime to fully turbulent; while the former reduces the heat transfer rate, the latter promotes heat transfer. A reverse condition is maintained for increasing tube diameter; therefore, it is reasonable to have optimum values for tube diameter both in laminar and in turbulent flow regimes. However, it can be said that the effect of tube diameter upon the required length of the heat exchanger is not much profound. In curves 4 & 5, the turbulent and laminar regimes were reached by increasing and reducing the tube-side flow rate, respectively; while the tube diameter was maintained constant. It is clear from these curves that the effect of tube-side flow rate changes on the required surface area is very strong. Further, it is seen that at higher values of reservoir fluid mass, the slope of curve 5 increases. This matter can be justified as the reservoir fluid mass is increased, the average temperature of the tube-side fluid is also raised which leads to a lower value of Prandtl number and consequently, less heat transfer coefficient would be resulted. Thus, the required length of the heat exchanger grows with the increase in the reservoir fluid mass.

The effect of tube-side Reynolds number on the required length of heat exchanger was illustrated in Figure 5.

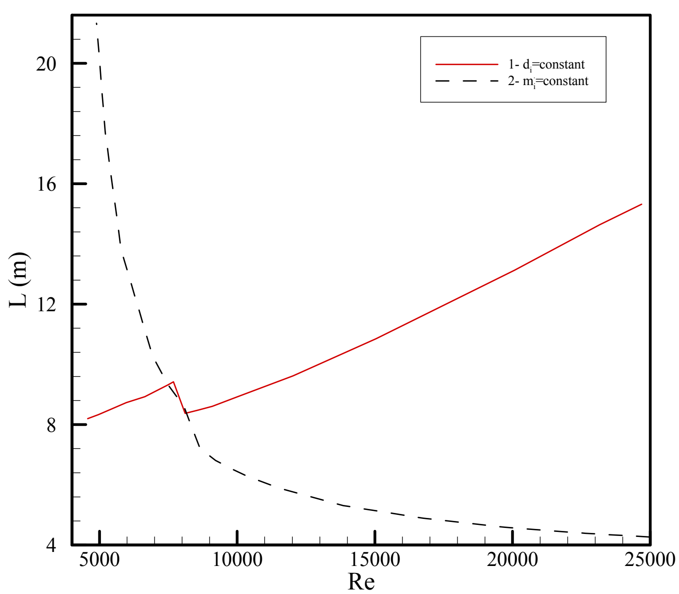

Figure 5. Effect of tube-side Reynolds number on the required length of heat exchanger.

In this figure, Reynolds number enhancement was achieved by two methods; i.e. decreasing the tube diameter while flow rate was fixed (curve 1), and increasing the flow rate while tube diameter was maintained constant (curve 2). As is observed from this figure, these curves have opposing trends; because, increment of flow rate promotes the turbulence of the flow which in turn augments the heat transfer coefficient; but, reduction of tube diameter, as discussed in previous paragraph, has two different effects, viz. decrease in heat transfer surface area, and enhanced heat transfer coefficient, which the former was stronger and resulted in more required tube length. From curve 2, it is concluded that at laminar region, the required length is a strong function of mass flow rate; because tube-side heat transfer coefficient is contingent upon the Reynolds number, and also in this region the tube-side heat transfer coefficient is the dominant term of overall heat transfer coefficient, while passing the transition region and reaching to the turbulent region, the reservoir-side heat transfer coefficient becomes dominant; hence, the variation of required length at this area is slighter.

Figure 6 represents the variation of cooling process specified time versus tube-side flow rate for different values of reservoir fluid mass for a coiled tube of 9.5 m length.

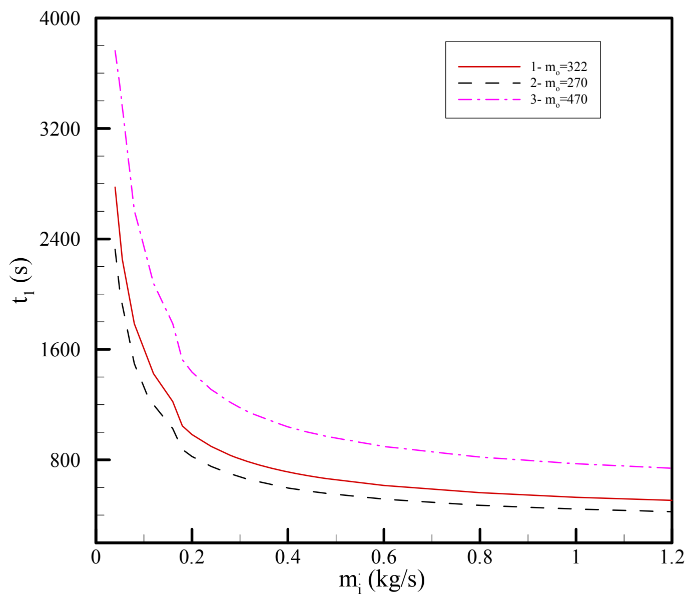

Figure 6. Variation of cooling process specified time versus tube-side flow rate.

As expected, more hot fluid requires more coolant flow rate at a specified time. However, the trends of the variations of different reservoir fluid mass are the same.

The variation of required length with coil curvature radius was illustrated in figure 7.

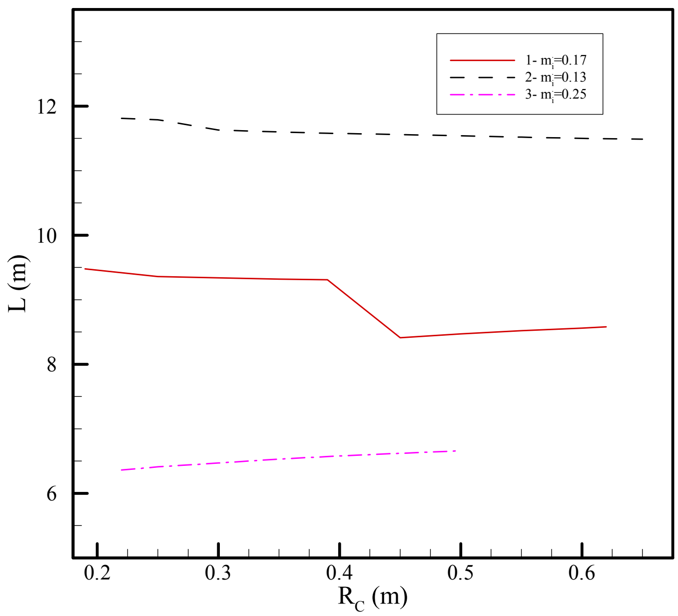

Figure 7. Variation of required length with coil curvature radius.

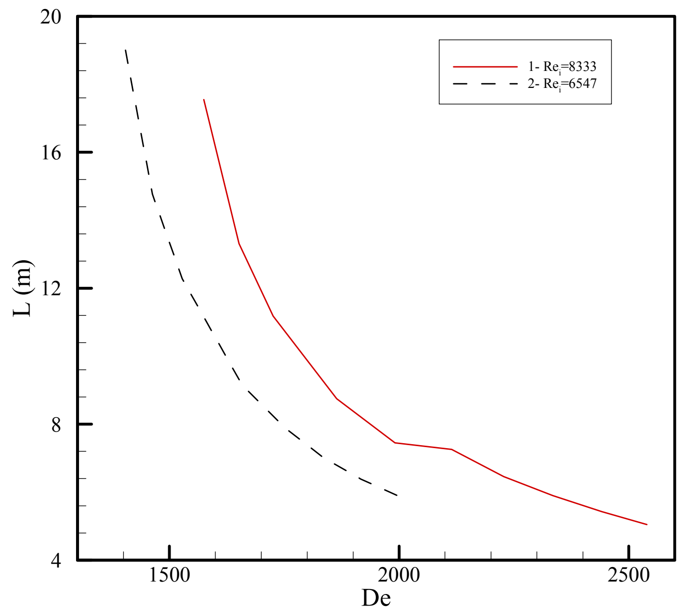

Figure 8. Effect of coiled tube Dean number on the required length of heat exchanger.

In this figure, three curves were plotted for different flow regimes, i.e. curves 1, 2 & 3 were drawn for laminar-turbulent, fully laminar, and fully turbulent regimes, respectively. In the first curve, the sudden reduction in required length is due to the transition from laminar to turbulent regime; because increment of curvature radius reduces the critical Reynolds number (according to Eq. 22), while the Reynolds number is constant. In curve 2, dimensionless coil pitch was reduced with the increase of curvature radius; hence, according to Eq.20, Nusselt number is increased; therefore, less heat exchanger length would be needed. But, in the fully turbulent region, curve 3, different conditions govern; viz. curvature radius in this region weaken the secondary flows in the pipe which leads to lower values of heat transfer coefficient (in the limit, this heat transfer coefficient tends to that of a straight tube). Further, the outside heat transfer coefficient of the tube would be slightly reduced with increase of tube curvature radius; because, $Nu_o \sim Ra^{0.3421}$, and $Ra \sim H^{1.0263}$; thus, $Nu_o \sim H^{1.0263}$; and since $h_o \sim Nu_o.H^{-1}$, one can say $h_o \sim H^{0.0263}$. Therefore, with the increase in curvature radius, both tube-side and reservoir-side heat transfer coefficients reduce, i.e. more heat exchanger length would be required. The interesting point is in curve 1 where both opposing effects are observed, viz. the increment of curvature radius has a favorable slight effect in the laminar region; while, it has an unfavorable slight effect in turbulent region.

Figure 8 illustrates the effect of coiled tube Dean number on the required length of heat exchanger.

As is seen, at this figure, two curves are plotted; curve 1 for Re=8333, and curve 2 for Re=6547. At the first case, as the critical Reynolds number reduces with the increase of Dean number, the transition between laminar and turbulent flow regimes is experienced; but, at the second case, this reduction is not enough to reach the Reynolds number of the flow. Comparing curves 1 & 2 for a given Dean number, as curvature radius is fixed, the greater Reynolds number corresponds to a smaller tube diameter; therefore, in the second case, which has a less Reynolds number, the tube diameter and consequently, the peripheral surface area is greater which leads to a less tube length.

CONCLUSIONS

At the present study, a helical tube was used as a heat exchanger to cool a hot fluid in a reservoir at a specified finite time. For this purpose, an analytical approach was developed to anticipate the required length of such a heat exchanger in accordance to the various parameters of flow and geometry of the coiled tube like the mass and initial and target temperatures of the reservoir fluid, mass flow rate and inlet temperature of the tube-side fluid, the diameter, curvature and pitch of the coiled tube, etc, and also the finite time allocated to heat transfer. Next, the effects of different parameters upon the required length were examined in appropriate curves. It was generally observed that transition from laminar to turbulent regime, enhanced heat transfer coefficient and consequently, reduced the required length of the heat exchanger. Investigating the results revealed that the variation trends in different regimes were not often the same, and in some cases, optimum values existed for different parameters of the coiled tube.

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