FEASIBILITY STUDY OF A PLASTIC HELICAL COIL HEAT EXCHANGER FOR A DOMESTIC WATER STORAGE TANK

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ABSTRACT

The main goal of this study is to investigate whether it is possible to use a polymeric helical coil heat exchanger as an alternative to conventional metallic helical coil. More specifically this work focuses on a helical coil design for a domestic water storage tank application. Corrosion and fouling resistance, scarcity of the materials, low weight and cost are the driving forces to consider designing polymeric heat exchangers rather than metallic heat exchangers. However, simply replacing the metallic material by the polymer material and applying the traditional design methods used for metallic heat exchangers do not lead to an acceptable design. If one wants to design a good polymer heat exchanger, heat transfer and structural problems have to be solved first. In addition, the pressure drop limitation should not be neglected. In order to reach this goal, a model is developed to predict the optimal design of a helical coil heat exchanger immersed in the water storage tank for a certain water mass flow rate and temperature. This design compensates for the low thermal conductivity and strength of the polymer.

INTRODUCTION

Due to high thermal performance, the metallic helical coil heat exchangers are widely used in the water storage tanks to transfer the heat from hot water in the tank to the domestic water in the tube. Extensive numerical and experimental research has been carried out to investigate the heat transfer, pressure drop and thermal performance of metallic helically coiled tube heat exchangers [1].

A schematic of a helical coil is shown in Figure 1. The inner diameter of the tube is d_i . The coil has a mean coil diameter of D_c. The pitch, p is the distance between two adjacent turns.

The heat exchanger market is an enormous market which consumes mostly precious materials such as aluminum. In 2012, the market size was US$42.7 billion and it was predicted to reach US$57.9 billion by 2016 and to increase to US$78.16 billion by 2020. This means a growth rate of approximately 7.8% per year [2].

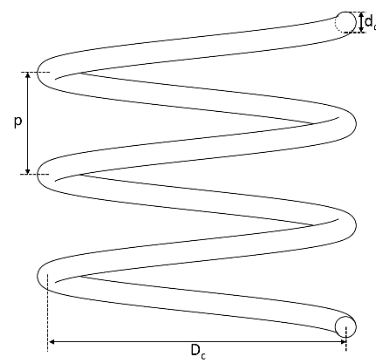

Figure 1. Helical coil geometry

The 2020 targets of EU, scarcity of precious materials high weight, huge capital investment, fouling and corrosion issues related to metallic heat exchangers provide intensives to consider polymer heat exchangers as an alternative for metallic heat exchangers in the water storage tank [3].

In recent years, significant interest has been shown to use of plastic heat exchangers in different applications, HVAC&R [4], heat recovery, sea water and water desalination, material separation and purification [5], electronic cooling [6], automotive [7] and etc.

Most polymer materials exhibit a thermal conductivity of between 0.1 and 0.4 $W.m^{-1}.K^{-1}$ [8] which is an order of magnitude lower than steel (50 $W.m^{-1}.K^{-1}$) or two orders of magnitude lower than copper (400 $W.m^{-1}.K^{-1}$) and aluminum (200 $W.m^{-1}.K^{-1}$) [9]. The low thermal conductivity of a pure polymer is a major impediment for the application of polymers in heat exchangers. The low wall thermal conductivity (K_w) increases the thermal resistance of the wall which is one of the contributor to the total thermal resistance of the heat exchanger. With the fouling resistances neglected, the total thermal resistance (R_{tot}) of the heat exchanger is equal to the sum of three thermal resistances in series: outer tube convection resistance (R_o), wall thermal resistance (R_w) and inner tube convection resistance (R_i) (Eq. 1-2).

$$R_{tot} = R_o + R_w + R_i \tag{1}$$

$$R_w = \frac{\ln \frac{do}{di}}{2\,\pi\,K_w\,L} = \frac{\ln \frac{1}{(1-2/SDR)}}{2\,\pi\,K_w\,L} \quad (2)$$

In order to overcome this impediment, a thin wall should be considered in the design of the heat exchanger. Still, the wall thermal resistance cannot be neglected, in contrary to the metallic heat exchanger design. However there is a limitation on how far the thickness of the wall can be decreased as a very thin wall cannot withstand the high stress. Hence, the low thermal conductivity and strength of polymer materials should be taken into account when designing the heat exchangers.

The research group of solar energy at the University of Minnesota, carried out some studies regarding polymer heat exchanger in solar water heating system. They investigated the suitability of different polymer materials for heat exchanger components based on different criteria like water compatibility and tensile strength [10]. Furthermore, thermal and mechanical performance, also cost of solar water heating system for different polymer shell and tube heat exchangers were studied [11][12].

To the authors' knowledge, no application of plastic helical heat exchangers for a water storage tank has yet been the subject of a study. Therefore this study is conducted to develop an integrated design model for this application using thermo-hydraulic and structural considerations.

NOMENCLATURE

A	[m²]	area
d	[m]	tube diameter
D	[m]	diameter
f	[-]	friction factor
F	[-]	service design factor
g	[m²/s]	gravitational acceleration
Gr	[-]	Grashof number
h	[W/m²K]	convection coefficient
H	[m]	height
K	[W/m K]	thermal conductivity
L	[m]	length
ṁ	[kg/s]	mass flow rate
Nu	[-]	Nusselt number
p	[m]	pitch
P	[Pa]	pressure
Pr	[-]	Prandtl number
v	[m/s]	velocity
R	[K/W]	thermal resistance
Ra	[-]	Rayleigh number
Re	[-]	Reynolds number
SDR	[-]	standard dimension ratio
t	[m]	thickness
T	[K]	temperature
TS	[Pa]	tensile strength
U	[W/m²K]	overall heat transfer coefficient
UA	[W/K]	thermal conductance

Greek symbols

β	[1/K]	thermal expansion coefficient
υ	[m²/s]	kinematic viscosity
ρ	[kg/m³]	density
σ	[Pa]	stress

Subscripts

b	balk
c	coil
i	inside
o	outside
tot	total
tw	water in tank
tuw	water in tube
v	von Mises
w	wall
wi	inner wall
wo	outer wall
θθ	hoop
rr	radial

OPTIMIZATION MODEL

A polymer helical coil heat exchanger can be a good replacement for its metallic counterpart in water storage tank when the maximum heat transfer rate is achieved for a given water mass flow rate and inlet temperature. Furthermore, the pressure drop and strength constraints are to be satisfied. Thus an integrated design must be identified with thermal, hydraulic and structural performance.

In order to solve any optimization problem, first it should be translated to a mathematical formulation. The formulation of an optimum design problem involves the definition of objective function, optimization criterion, design variables, constraints and parameters (fixed value variables).

The heat transfer rate is calculated from Eq. (3). To achieve the maximum heat transfer for a given temperature difference the maximum thermal conductance (UA) must be identified or ($-UA$) should be minimized. This is the optimization criterion.

$$\dot{Q} = U.A\,.LMTD \quad (3)$$

As seen in Eq. 4, the goal function (thermal conductance) is a function of helical coil geometry. The design variables are outer tube diameter (d_o), tube thickness (t), mean helical coil diameter (D_c), pitch (p) and helical coil height (H_c) which define the helical coil geometry (Figure 1). Wall thermal conductivity, domestic water bulk temperature, tank water temperature and water mass flow rate are the constant parameters. Pressure drop (dp) and strength (σ_θ) are the constraints.

$$UA = f(d_o, t, D_c, p, H_c) \quad (4)$$

The optimization problem is formulated as a multi variable nonlinear constrained function (Eq. (4)) and is solved using a gradient-based solver. The solver finds a local optimal value of the geometry variables where, for those values, the minimum of the goal function is realized. The local optimal values are found by starting from an initial value and moving towards optimum using information about the gradient.

Three different functions are integrated in the development of the optimization model, thermal conductance (UA), pressure drop (dp) and strength (σ_θ) functions. Each of them are explained further in the following sections.

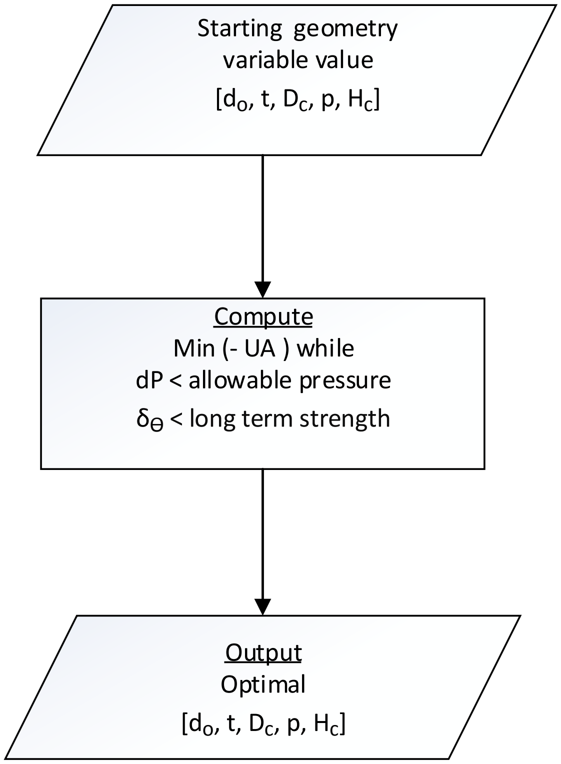

Figure 2 Optimization model

Thermal conductance function

The thermal conductance function is defined based on Eq. 5. It is the reciprocal sum of the inner tube convection resistance R_i, outer tube convection resistance R_o and wall conduction resistance R_W.

$$UA = \frac{1}{R_o + R_w + R_i} = \frac{1}{\frac{1}{h_o . A_o} + \frac{ln\frac{do}{di}}{2\pi\, K_W\, L} + \frac{1}{h_i . A_i}} \tag{5}$$

A_o and A_i are respectively the outer and inner surface area of the helical coil. L is the total length of helical coil.

It is necessary to determine the inner and outer convection heat transfer coefficients (h_i, h_o), to define the thermal conductance. The water properties are obtained from CoolProp Database [13].

The inner heat transfer coefficient is calculated using Eq. (6)-(8). The inner tube diameter (d_i) is chosen as the characteristic length to calculate Reynolds number and inner heat transfer coefficient. Moreover the water properties, Prandtl number (Pr) and thermal conductivity of domestic water in tube (K_{tuw}) are evaluated at the mean bulk temperature. Several correlations are introduced in the literature to calculate the Nusselt number (Nu_i) and friction factor (f). In this case, Gnielinski correlation is used to calculate the Nusselt number for the flow inside tube [14].

$$h_i = \frac{Nu_i\, K_{tuw}}{d_i} \tag{6}$$

$$Nu_i = \frac{(f/2)\, Re\;\, Pr}{1.07 + 12.7 \left(f/2\right)^{\frac{1}{2}} \left(Pr^{\frac{2}{3}} - 1\right)} \tag{7}$$

$$f = (1.58\; lnRe - 3.28)^{-2} \tag{8}$$

Natural convection is the mode of heat transfer, outside of the helically coiled tube. The helical coil height is chosen as the characteristic length to calculate the outer heat transfer coefficient (h_o) (Eq. (9)), also the Grashof number (Eq. (12)). The Nusselt number [15], Rayleigh and Grashof numbers are calculated from Eq. (10)-(11). The thermal conductivity of water in the tank (K_{tw}), Prandtl number (Pr) and coefficient of water expansion (β) are evaluated at film temperature. Since the wall temperature is unknown, the film temperature is unknown as well. Therefore an iterative approach is used to determine the inner and outer wall temperature.

$$h_o = \frac{Nu_o\;\, K_{tw}}{H_c} \tag{9}$$

$$Nu_o = 0.59 Ra^{\frac{1}{4}} \tag{10}$$

$$Ra = Gr . Pr \tag{11}$$

$$Gr = \frac{g\;\, \beta\;\, (T_{tw} - T_{wo})\;\, {H_c}^3}{\upsilon^2} \tag{12}$$

The following steps are used to define the thermal conductance function which computes the thermal conductance for a given geometry d_o, t, D_c, p, H_c (a flow chart is shown in Figure 3).

1) Input: Geometry variables, d_o, t, D_c, p, H_c
2) Input: domestic water inside tube mass flow rate $(\dot{m})$, water inside tube bulk temperature (T_b), water in the storage tank temperature (T_{tank})
3) Compute dimensionless numbers and heat transfer coefficient for inside tube
    a. Water properties are calculated at bulk temperature using Coolprop database
    b. Calculate Reynolds number (Re_i)
    c. Calculate Nusselt number (Nu_i)
    d. Calculate inner heat transfer coefficient (h_i)
4) Input: initial guess for inner and outer wall temperature (T_{wi}, T_{wo})
5) Compute dimensionless numbers and heat transfer coefficient for outside tube
    a. Calculate film temperature
    b. Water properties are calculated at film temperature using Coolprop database
    c. Calculate Grashof number (Gr)
    d. Calculate Rayleigh number (Ra)
    e. Calculate Nusselt number (Nu_o)
    f. Calculate outer heat transfer coefficient (h_o)
6) Compute inner and outer surface area A_i, A_o

7) Compute inner tube convection resistance R_i, outer tube convection resistance R_o and wall conduction resistance R_{wall}
8) Using the sum of the resistances, compute thermal conductance UA
9) Compute overall heat transfer coefficient (U_o) from UA and A_o
10) Compute the new inner wall temperature: $T_{wi} = T_b + \left(\frac{R_i}{R_{tot}}\right).(T_{tw} - T_b)$
11) Compute the new outer wall temperature: $T_{wo} = T_b + \left(\frac{R_i+R_w}{R_{tot}}\right).(T_{tw} - T_b)$
12) Update the inner and outer wall temperature by the new one calculated in step 10 and 11
13) Return to step 5. Repeat until convergence

Pressure drop function

The determination of the pressure drop of the hot water in the coil is necessary as it influences the power required to pump the water through the heat exchanger.

Pressure drop function is integrated as a constraint in the optimization model. This function calculates the core friction which is the most important contribution (90%) to the overall pressure drop over the heat exchanger core for a given geometry, mass flow rate and water bulk temperature. using Eq. (13). Where f, L, ρ, V, d_i are respectively, friction factor based on Gnielinski correlation (Eq. (8)), length of helical coil, density of water in the tube, mean velocity in heat exchanger core and inner tube diameter which is the characteristic length.

$$\Delta p = \frac{2f\, L\, \rho\, V^2}{d_i} \tag{13}$$

Increasing the surface area to improve the thermal performance by increasing the length, raises the pressure drop through the heat exchanger core for a fixed amount of mass flowrate. The pressure drop should be suitable for pump that is used to pass the fluid over the heat exchanger core. This means the pressure drop should not be lower than an allowable value. The allowable pressure drop depends on the application and energy of the pump In this model, the allowable pressure drop is assumed to be 0.3 bar for the range of mass flowrate between 5-25 l/min.

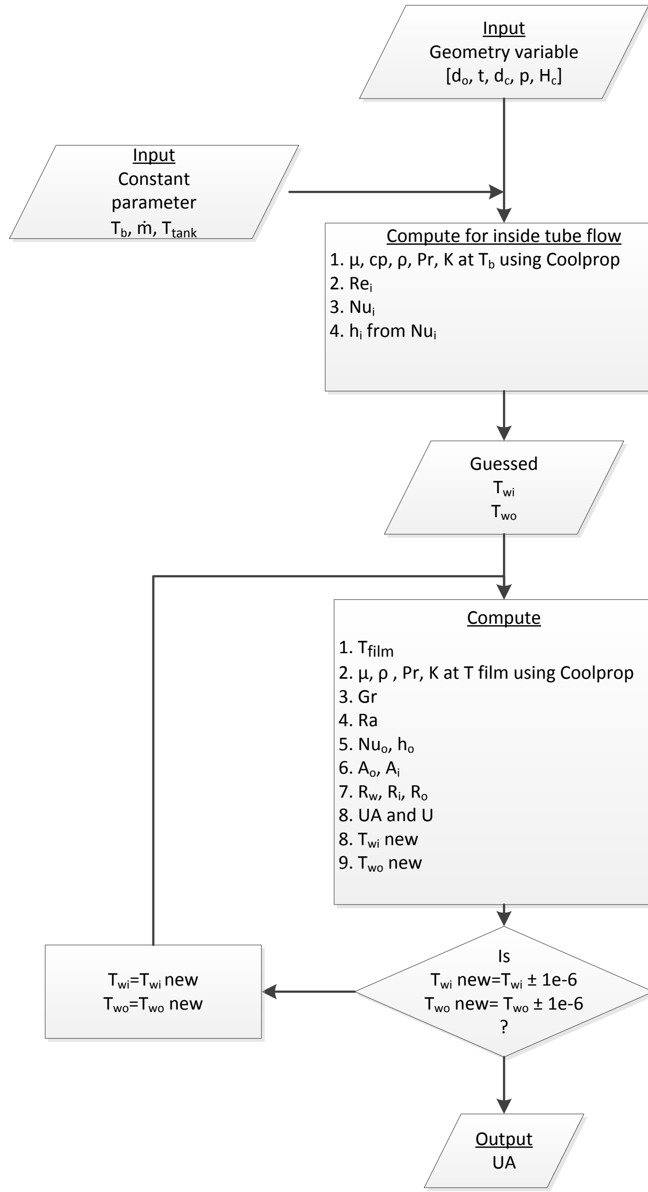

Figure 3 Thermal conductance function

Strength function

The tensile strength and tensile modules of polymeric materials are lower than those of metallic materials. For instance polyethylene has a strength modulus of about 0.1 GPa where for the metal it is in the order of 100 GPa. The tensile strength of polyethylene is 4-38 MPa where this is compared with the values of the order of a few hundred to thousand MPa for the metallic materials [16]. Hence, the structural performance of the polymer should be considered in the helical coil design.

In this study the structural performance of the polymeric helical coil is evaluated based on the von Mises yield criterion (σ_v). The maximum allowable von Mises stress should be lower than the tensile strength including a safety factor. In Eq. 14, the service design factor F is 0.5.

$$\sigma_v < F.TS \tag{14}$$

The von Mises stress is calculated from the following equations. Radial stress (σ_{rr}) and hoop stress ($\sigma_{\theta\theta}$) are calculated at radius r based on Lame equation. Inner and outer pressure loads are shown by P_i and P_o.

$$\sigma_v = \sqrt{\sigma_{rr}{}^2 - \sigma_{rr}\sigma_{\theta\theta} + \sigma_{\theta\theta}{}^2} \tag{15}$$

$$\sigma_{rr} = \frac{P_i r_i^2}{r_i^2 - r_o^2}\left(1 - \frac{r_o^2}{r^2}\right) - \frac{P_o r_o^2}{r_i^2 - r_o^2}\left(1 - \frac{r_i^2}{r^2}\right) \tag{16}$$

$$\sigma_{\theta\theta} = \frac{P_i r_i^2}{r_i^2 - r_o^2}\left(1 + \frac{r_o^2}{r^2}\right) - \frac{P_o r_o^2}{r_i^2 - r_o^2}\left(1 + \frac{r_i^2}{r^2}\right) \tag{17}$$

For this strength function the tensile stress is assumed to be 46 MPa.

RESULTS AND DISCUSSION

The thermal performance of a polymeric heat exchanger with the thermal conductivity of 0.2 $W.m^{-1}.K^{-1}$ is compared to that of the metallic heat exchanger with the thermal conductivity of 200 $W.m^{-1}.K^{-1}$. The polymeric design is identical to the metallic helical coil. Table 1 shows the geometry of both metallic and polymeric heat exchangers. The helically coiled tube has 18 mm outer diameter and 1.2 mm thickness. The coil has the mean diameter of 203 mm and the coil height is 180 mm. The pitch to the outer diameter ratio is 1.

Table 1 Helical coil geometry (All dimensions are in mm.)

d_o	t	d_c	p	H_c
18	1.2	203	18	181

Figure 4 illustrates the ratio of each thermal resistance, inner, outer and wall resistances to the total thermal resistance for the both materials when the water mass flow rate is 15 l/min. For the metallic helical coil, the outer tube side resistance determines the overall heat transfer coefficient as it is about 85% of the total resistance. In addition, the wall thermal resistance is negligible. However in the case of polymeric heat exchanger, the wall thermal resistance dictates the overall heat transfer coefficient.

The change of heat conductance (UA) for different water flow rates is demonstrated in Figure 5 for metallic and for polymeric heat exchanger in Figure 6. These figures show that increasing the water mass flow rate can significantly increase the thermal conductance of the metallic heat exchanger (up to 6 times higher). However in the case of the polymeric heat exchangers, the thermal conductance increases only by factor of 1.5. The thermal conductance of the metallic helical coil (280 $W.K^{-1}$) is 6.3 times higher than the polymeric coil (44 $W.K^{-1}$) for 15 l/min water flowrate.

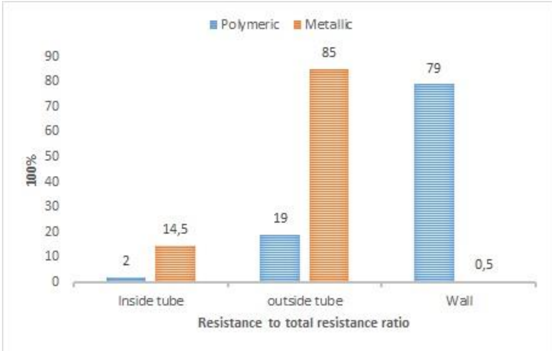

Figure 4 Polymeric & Metallic coil comparison

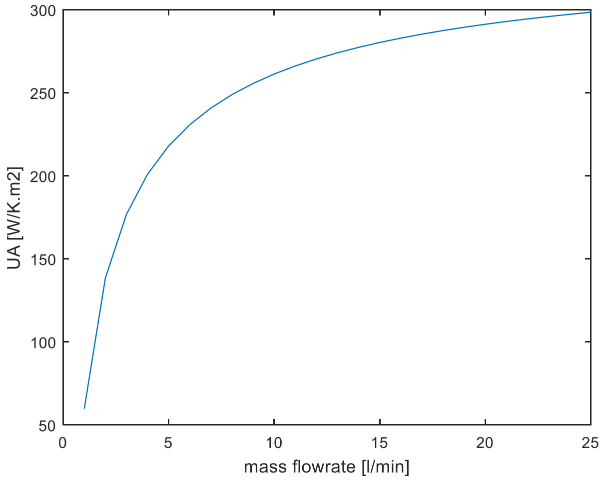

Figure 5 Metallic helical coil thermal conductance to mass flow rate

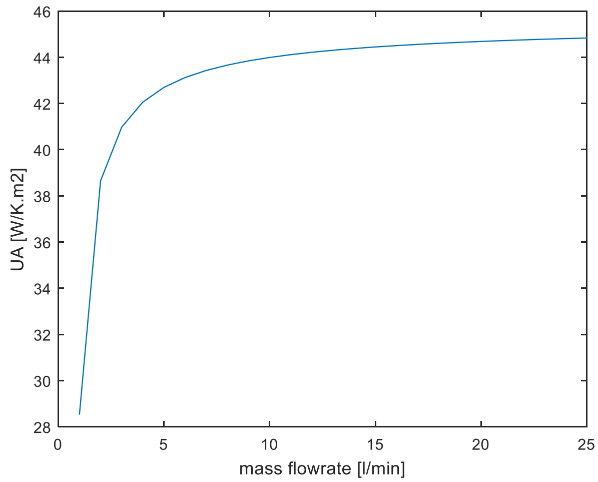

Figure 6 Polymeric helical coil thermal conductance to water mass flow rate

If the same thermal conductance as the metallic heat exchanger (280 $W.K^{-1}$) is desired for this polymeric helical coil, the length must be increased by a factor 7. This means, the polymeric helical coil will be too long to be fitted in a storage tank with the coil height (H_c), 1250 mm and the coil diameter (D_c), 440 mm.

The optimization model previously discussed is used to find an optimal design geometry (d_o, t, D_c, p, H_c). Table 2 shows an optimal geometry for the polymeric helical coil. It is observed that the pitch is at the minimum value which is the outer tube diameter. This increases the surface area and consequently improves the overall heat transfer coefficient. For the same reason, the coil height and coil diameter have the highest possible value. The outer tube diameter is increased and the tube thickness becomes two orders of magnitude smaller. Therefore, SDR is reduced and as it is shown in Eq. 2, the reduction of SDR leads to better heat conductance. At this optimal point, the pressure drop is 0.3 bar and the von Mises stress is 13 MPa. Because of the optimization, the thermal conductance is 1600 $W.K^{-1}$.

Table 2 Optimized polymeric helical coil geometry (All dimensions are in mm.)

d_o	t	d_c	p	H_c
21	0.23	440	21	1250

CONCLUSIONS

This study shows that polymer helical coil can provide thermal output equivalent to conventional metallic helical coil. The design challenge is to overcome the high wall thermal resistance by decreasing the wall thickness and at the same time provide sufficient strength to withstand the pressure. Therefore a design methodology, which considers structural, hydraulic and thermal performance at the same time, is required.

An experimental study for the proposed heat exchanger is needed to better quantify the heat transfer, pressure drop and structural performance.

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