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Design of Evaporator and Condenser for Solar Adsorption Refrigeration System

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Abstract

Solar adsorption refrigeration systems provide a promising alternative for sustainable cooling in regions with limited grid access and high solar availability because they can convert low-grade thermal energy into refrigeration without mechanical compression. The performance of such systems depends strongly on the thermal behavior of the evaporator and condenser, since these components govern the refrigerant evaporation rate, condensation effectiveness, and overall cycle stability. This paper presents focus on the design of evaporator and condenser units for a solar adsorption refrigeration system, with emphasis on material selection, heat-transfer enhancement, geometric refinement, and compact heat-exchanger configuration. Literature on adsorption cooling shows that finned structures, improved heat-transfer paths, and suitable adsorbent–refrigerant working pairs can raise coefficient of performance (COP), improve specific cooling power (SCP), and reduce cooling time. The analysis indicates that condenser augmentation through finned designs and evaporator optimization through improved thermal conductivity and geometry can significantly enhance cooling capacity and operational effectiveness in solar-driven refrigeration systems.

Keywords: Solar adsorption refrigeration, evaporator, condenser design, finned heat exchanger, heat transfer enhancement

INTRODUCTION

Cooling demand continues to grow in food preservation, medical storage, and thermal comfort applications, while conventional vapor-compression systems remain dependent on electricity and refrigerants associated with environmental concerns [1–2]. Solar adsorption refrigeration has emerged as an attractive alternative because it can operate using low-grade solar heat, has relatively simple mechanical construction, and is suitable for decentralized cooling applications [3–4]. In a typical solar adsorption cycle, the adsorbent bed acts as a thermally driven compressor, while the condenser rejects heat and the evaporator provides the useful cooling effect [5]. The evaporator and condenser are especially important because their thermal resistances directly influence refrigerant phase change and hence the coefficient of performance of the cycle [6–7]. Prior studies report that adsorption-system performance deteriorates when condenser temperature rises, while improved heat transfer surfaces can reduce cycle time and improve refrigeration output [7–8]. Experimental work on solar adsorption systems has also shown that fin-tube evaporator and condenser structures can be effective in compact

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configurations [9]. Accordingly, the present paper focuses on the academic development of evaporator and condenser design principles for solar adsorption refrigeration systems and evaluates how these design strategies can improve overall system performance.

LITERATURE REVIEW

Published research consistently shows that adsorption refrigeration performance depends on the interaction among adsorbent properties, bed heat transfer, evaporator behavior, and condenser effectiveness. Yuan et al.reported an experimental solar adsorption refrigeration system using a concentrated collector in which the evaporator and condenser employed fin-tube compact heat exchangers, and the system achieved a maximum COP of 0.169 using SAPO-34 zeolite with water as the refrigerant [3]. Saravanan and Rathnasamyshowed through thermodynamic modeling that condenser temperature has a negative effect on COP, while evaporator temperature has a smaller but still measurable influence on system performance [10]. The design of the heat exchangers themselves has also been studied in related adsorption systems.

Khanafiah et al. examined evaporator performance in a water-refrigerant adsorption cooling system and emphasized the importance of heat exchanger tube diameter and thermal transport conditions in obtaining high specific cooling power [11]. Other studies summarized by Yuan et al. reported that finned adsorber or heat-exchanger configurations can significantly improve heat transfer and reduce cycle limitations associated with poor thermal conductivity in adsorption devices [3]. Broader design studies of silica gel/water adsorption chillers likewise show that condenser and evaporator area sizing is fundamental for achieving target cooling loads and stable operation.

SYSTEM DESCRIPTION

A solar adsorption refrigeration system as shown in Figure 1 consists of a solar collector, an adsorbent bed, a condenser, an evaporator, and a cooled chamber or chilled-water tank. During the daytime, solar heat raises the temperature and pressure of the adsorption bed, causing refrigerant vapor to desorb and flow to the condenser where it liquefies after rejecting heat to the surroundings. After desorption, the bed is cooled; once its pressure falls below the evaporator saturation pressure, the refrigerant in the evaporator vaporizes and produces a cooling effect while the vapor is re-adsorbed in the bed [2, 6].

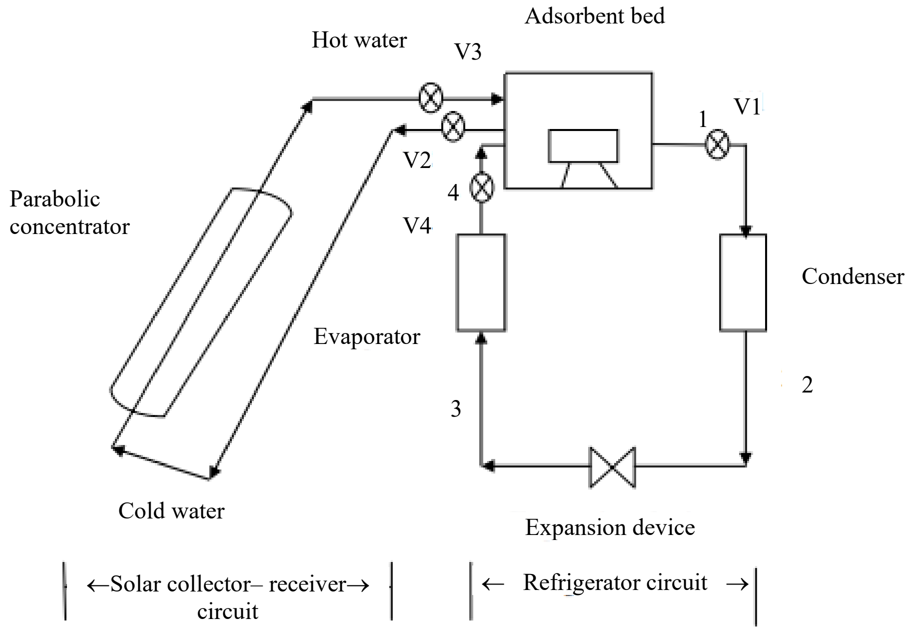

Figure 1. Schematic diagram of solar adsorption refrigeration system [12].

The operation of a single-bed solar adsorption refrigeration system is inherently intermittent due to the cyclic nature of adsorption and desorption processes. Nevertheless, such systems are highly attractive because they utilize environmentally benign working pairs, such as zeolite–water and silica gel–water, and do not require mechanical compression, thereby reducing energy consumption and environmental impact [13–14]. Within this system, the condenser and evaporator serve as critical heat-transfer components rather than auxiliary elements. Their performance directly influences the rate of heat rejection, cooling production, and cycle recovery time. Consequently, the design and optimization of the condenser and evaporator are essential for enhancing the overall efficiency and practicality of solar adsorption refrigeration systems [2, 6].

DESIGN METHODOLOGY

Design Basis

The analysis in this paper is based on a compact solar adsorption refrigeration configuration in which the condenser and evaporator function as phase-change heat exchangers connected to an adsorption bed and solar collector. A water-based refrigerant loop is assumed because water is frequently used with zeolite or silica gel working pairs in solar adsorption cooling research. The design objective is to improve refrigeration performance by reducing heat-transfer resistance at the evaporator and condenser while preserving compactness and operational simplicity [15–16].

Evaporator Design

The design of an evaporator plays a crucial role in determining the overall performance and efficiency of refrigeration systems, particularly in low-flow and laminar flow conditions such as those encountered in solar adsorption refrigeration systems. In the present study, an immersed helical coil heat exchanger is selected for the evaporator design due to its superior thermal-hydraulic characteristics compared to conventional configurations.

Helical coil heat exchangers as shown in Figure 2, are widely recognized for their enhanced heat transfer performance, especially under laminar flow regimes. The curvature of the coil induces secondary flow patterns, commonly referred to as Dean vortices, which significantly improve fluid mixing and heat transfer rates even at low Reynolds numbers. This phenomenon leads to higher heat transfer coefficients compared to straight tube or double-pipe heat exchangers operating under similar conditions [17–18].

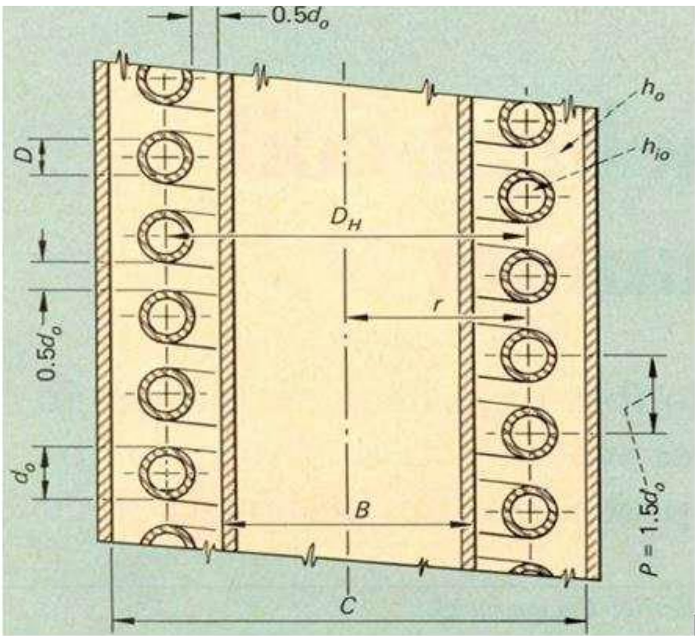

Figure 2. Immersed coil heat exchanger[19].

Figure 3. Developed evaporator inside refrigerated water tank.

Table 1. Physical properties and other data for evaporator and condenser.

Property	Outside fluid (Water)	Inside fluid (Methanol liquid)
Mass flow rate(kg/hr)	170	0.1667
Inlet temperature (°C)	25	14
Outlet temperature (°C)	15	14
Heat capacity - C_p(kcal/kg°C)	1	0.604
Thermal conductivity – k (kcal /m°C)	0.5142	0.1742
Viscosity - μ (kg/mh)	3.204	1.9584
Density – ρ(kg/m^3)	1000	791

In addition to improved effectiveness, the compactness of the helical coil configuration is a key advantage. The coiled geometry allows a larger heat transfer area to be accommodated within a smaller volume, making it highly suitable for systems where space constraints are critical. Compared to double pipe heat exchangers, which require longer lengths to achieve comparable performance, the helical coil design offers a more compact and efficient alternative [20].

Furthermore, the immersed configuration enhances heat transfer by ensuring better contact between the coil surface and the surrounding working fluid. This arrangement minimizes thermal resistance and promotes uniform temperature distribution within the evaporator. Such characteristics are particularly beneficial in adsorption refrigeration systems, where efficient heat exchange is essential for improving system coefficient of performance (COP).Based on these considerations, the immersed helical coil heat exchanger is selected as the evaporator design in this work, owing to its higher effectiveness under laminar flow conditions, reduced size, and improved thermal performance.

Some of the physical properties and other data for evaporator are shown in Table 1. The evaporator is a key component in a refrigeration system, responsible for absorbing heat from the medium to be cooled. In the present work, a helical coil evaporator is designed for cooling water using a shell-and-coil configuration. The design methodology follows standard heat transfer and process design procedures as outlined by Donald Q. Kern [21].

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The evaporator is designed as shown in Figure 3 to provide a cooling capacity for 50 liters of water; however, a higher design capacity of 60 liters (volume) is considered to ensure safe and reliable operation under varying conditions. The analysis assumes steady-state operation throughout the heat transfer process, with fluid properties taken as uniform and constant for simplification. Additionally, fouling effects on both the shell and tube sides are incorporated into the design to account for performance degradation over time. The geometrical parameters and standard dimensions used in the design are selected based on established literature and widely accepted correlations.

Geometrical Design of Evaporator

- *Considering, Outer diameter of coil:* $D_o = 0.0127$ m,
- *Inner diameter of coil:* $D_i = 0.0102$ m, Pitch (P) = 1.5 D_o = 0.01905 m, *C:* Inner diameter of shell and H: height of the shell;
- *Volume equation:* $V = (\pi/4) \times C^2 \times H$, where H = 1.5×C gives C = 0.37 m and H = 0.56 m
- *Helix diameter:* $D_h = C - 2 \times D_o = 0.3453$ m
- *Radius:* $r = D_h/2 = 0.1727$ m
- *Inside diameter of helix:* $D_{h1} = D_h - D_o = 0.332$ m
- *Outside diameter of helix:* $D_{h2} = D_h + D_o = 0.358$ m
- *Coil Length:* $L = N\sqrt{((2\pi r)^2 + P^2)} = 1.0853 \times N$ m
- *Heat TransferVolume of annulus:* $V_a = (\pi/4) \times C^2 \times P \times N = 2.056 \times 10^{-3} N$ m^3
- *Volume occupied by the coil:* $V_c = (\pi/4) \times d_o^2 \times L = 1.3748 \times 10^{-4} N$ m^3
- *Volume available for the flow of fluid in the annulus:* $V_f = V_a - V_c = 1.9185 \times 10^{-3} N$ m^3
- *Shell side equivalent diameter:* $D_e = (4 \times V_f)/(\pi \times d_o \times L) = 0.1772$ m
- *Mass velocity of fluid:* $G_s = 4 \times m/[\pi \{C - (D^2_{h2} + D^2_{h1)}\}] = 1805.60$ kg/m^{2}h
- *Reynold's number:* $Re = (D_e \times G_s)/\mu = 99.8382$
- *Prandlt number:* $Pr = (\mu \times C_p)/k = 6.231$
- *Heat transfer coefficient for outer coil:* $h_o = (k/D_e)0.6 \times Re^{0.5} \times Pr^{0.31} = 30.6747$ kcal/m^2°C

For calculating heat transfer coefficient for inner coil (h_i);

- *Fluid velocity:* $u = q/V_f = (m_m/\rho)/\{(\pi/4) \times D_i^2\} = 2.5786$ m/h and $Re = (\rho \times u \times D_i)/\mu = 10.62$

From Donald Q. Kern (1950) for (L/D) = 100 and Re = 10.62; colburn factor for heat transfer; j_H = 1 (dimensional number)

Therefore, $h_i = j_H \times (k/D_i) \times Pr^{0.33} = 32.1716$ kcal/h m^2°C

- *Corrected inner heat transfer coefficient:* $h_{ic} = h_i \times [1 + 3.5 \times (D_i/D_h)] = 35.4978$ kcal/h m^2°C
- *Heat transfer coefficient based on outer diameter of the coil:* $h_{io} = h_{ic} \times D_i/D_o = 28.51$ kcal/h m^2°C
- *Coil wall thickness:* $x = (D_o - D_i)/2$ 1.25×10^{-3} m

If, $R_a$is shell side fouling factor(hm^2°C / kcal), $R_t$is tube side fowling factor (hm^2°C / kcal) with value of 0.0001 hm^2°C / kcal

- *The overall heat transfer coefficient:* $1/U = 1/H_o + 1/H_{io} + x/k_c + R_t + R_a$ where, $k_c$is the thermal conductivity of coil wall with value of 343.94 kcal / h m^2°C gives U= 14.732 kcal / h m^2°C
- *Over all heat transfer:* $Q = U \times A \times \Delta T_m$ gives A = 0.68 m^2 with value of41.79 kcal/hr as Q
- *Therefore, Number of turns:* $N = A/(\pi D_o L) = 15.7 \approx 16$ turns

Evaporator Design Variables

The evaporator is responsible for extracting heat from the cooling chamber or secondary fluid by vaporizing the refrigerant under low-pressure conditions [22–23]. Important evaporator design variables include material thermal conductivity, wall thickness, contact area with the cooled medium, internal flow geometry, and external fin arrangement where applicable [24]. High-conductivity materials such as copper and aluminum are preferred because they reduce conductive resistance between the cooled load and the refrigerant [17]. A practical optimization goal is to increase effective heat-transfer area without introducing excessive material mass that could slow transient response [25].

Condenser Design

The design of a condenser is a critical aspect in determining the overall efficiency and reliability of refrigeration systems, particularly in thermally driven systems such as solar adsorption refrigeration. The condenser is responsible for rejecting heat from the refrigerant vapor to the surrounding environment, thereby facilitating phase change from vapor to liquid. In the present study, an air-cooled finned helical coil heat exchanger is selected for the condenser design due to its enhanced heat transfer characteristics and suitability for compact, low-flow systems.

Helical coil heat exchangers have been extensively studied and are known for their superior heat transfer performance compared to conventional straight tube configurations. Similar to evaporators, the curvature of the coil induces secondary flows, commonly referred to as Dean vortices, which enhance turbulence and mixing even at relatively low Reynolds numbers. This effect significantly increases the convective heat transfer coefficient, making the helical coil configuration highly effective for condensation processes [17–18]. Moreover, during condensation, the continuous removal of the condensate film is improved due to the curvature, which reduces thermal resistance and enhances overall heat transfer efficiency.

In addition to improved thermal performance, the compactness of the helical coil condenser offers a major advantage in system design. The coiled geometry provides a large heat transfer area within a limited space, making it highly suitable for solar adsorption refrigeration systems where space and weight are important constraints. Compared to conventional shell-and-tube or double pipe condensers, the helical coil design requires less material and occupies a smaller footprint while delivering comparable or superior performance [20].

Furthermore, the incorporation of fins on the outer surface of the coil significantly enhances the air-side heat transfer coefficient, which is typically the limiting resistance in air-cooled condensers. The extended surface area provided by fins improves heat dissipation to the ambient air, ensuring efficient condensation even under natural or low forced convection conditions. This feature is particularly beneficial in solar-powered systems where auxiliary power for forced convection may be limited.

Some of the physical properties and other data for condenser are shown in Table 1. The condenser in the present work is designed to reject the heat absorbed in the evaporator along with the heat of compression/adsorption, ensuring complete condensation of the refrigerant (methanol). The design is based on standard heat transfer principles and follows established methodologies as outlined by Donald Q. Kern [21]. The system is designed under steady-state conditions, assuming constant thermophysical properties of the fluids. Fouling factors are incorporated on the refrigerant and air sides to account for long-term performance degradation.

The selection of operating conditions, geometrical parameters, and material properties is based on standard literature correlations and practical design considerations. The condenser is sized to handle the required heat rejection load corresponding to a cooling capacity of 50 liters, with an overdesign margin incorporated to ensure safe and reliable operation under varying environmental conditions.

Based on these considerations, the finned helical coil condenser is selected in this study due to its enhanced heat transfer characteristics, compact design, and suitability for low-flow and thermally driven refrigeration systems. Its ability to efficiently reject heat under varying ambient conditions makes it an ideal choice for solar adsorption refrigeration applications.

Geometrical Design of Condenser

The design methodology adopted for the evaporator can be similarly extended to the condenser, with appropriate modifications based on the operating conditions and heat rejection requirements. In the present study, the condenser is designed for a system capacity corresponding to a refrigerated volume of 120 L, ensuring adequate safety margin and effective heat dissipation. The same systematic approach involving thermal load estimation, selection of geometric parameters, heat transfer analysis, and overall heat transfer coefficient evaluation is followed for the condenser design. However, unlike the evaporator, the condenser operates under heat rejection conditions, where the working fluid undergoes phase change from vapor to liquid, thereby requiring careful consideration of condensation heat transfer coefficients and ambient cooling conditions.

The design begins with the estimation of the total heat rejection load, which includes the evaporator load and the compressor work input. Based on this, suitable condenser dimensions such as tube diameter and coil pitch are selected using standard design correlations and literature guidelines. Number of turns required is 36 which calculated using below equation:

$N = (H-D_o)/P$

The heat transfer analysis is carried out by considering convective heat transfer on the air side and condensation heat transfer on the refrigerant side. Fouling factors and material properties are incorporated to ensure realistic performance prediction. Furthermore, the overall heat transfer coefficient is determined by accounting for thermal resistances on both fluid sides and the tube wall in which ΔT_m is found using following equation:

$$\Delta T_m = \frac{[T_{h1} - T_{c1}] - [T_{h2} - T_{c2}]}{\ln\left(\frac{T_{h1} - T_{c1}}{T_{h2} - T_{c2}}\right)}$$

Condenser Design Variables

The condenser rejects heat from the desorbed refrigerant vapor and must operate effectively at ambient or near-ambient conditions. Its performance depends on external heat-transfer area, fin density, tube arrangement, airflow or cooling-water conditions, and the temperature difference between condensing vapor and the sink medium (Heat Transfer principles; Fundamentals of Heat and Mass Transfer). Finned condenser configurations are especially attractive because they increase area and improve convective heat exchange without a major increase in system complexity [24]. In solar adsorption systems, this is important because higher condenser temperature reduces coefficient of performance (COP) and can diminish overall refrigeration output [6].

RESULTS AND DISCUSSIONS

The geometric parameters of the helical coil heat exchangers designed for both the evaporator and condenser are presented in Table 2. These parameters play a crucial role in determining the heat transfer performance, pressure drop characteristics, and overall system efficiency of the solar adsorption refrigeration system.

From the table, it is observed that the evaporator coil is designed with an outer and inner diameter of 12.7 mm, whereas the condenser coil has comparatively smaller diameters of 10.2 mm. The larger diameter in the evaporator facilitates higher fluid flow capacity and reduces flow resistance, which is essential for efficient heat absorption under low-pressure conditions.

Table 2 Geometric parameters of designed evaporator and condenser.

Sr. No.	Parameter	Evaporator	Condenser
1	Outside diameter of coil - D_o (mm)	12.7	10.2
2	Inside diameter of coil - D_i (mm)	12.7	10.2
3	Pitch - P (mm)	19.05	19.05
4	Mean coil (helix) diameter - D_h (m)	0.3453	0.4416
5	Height of coil - H (m)	0.56	0.7
6	Number of turns - N	16	36

In contrast, the smaller diameter of the condenser coil enhances the heat rejection rate by increasing the surface-area-to-volume ratio, thereby improving condensation performance.

The pitch of the helical coil is maintained constant at 19.05 mm for both components to ensure uniform coil spacing and consistent flow distribution. However, significant differences are observed in the mean coil diameter and coil height. The condenser has a larger mean coil diameter (0.4416 m) and greater height (0.7 m) compared to the evaporator (0.3453 m and 0.56 m, respectively). This indicates that the condenser is designed with a more extended heat transfer surface to effectively dissipate heat to the surroundings.

Furthermore, the number of turns in the condenser (36) is more than twice that of the evaporator (16), which substantially increases the effective heat transfer area. This design choice is justified by the requirement of continuous and efficient heat rejection during the condensation process, which typically demands a larger surface area compared to heat absorption in the evaporator.

The comparative analysis of the geometrical parameters reveals that the evaporator is optimized for efficient heat absorption with lower flow resistance, whereas the condenser is designed to maximize heat rejection through increased surface area and a compact coil configuration. The evaporator employs a larger tube diameter and fewer coil turns (16 turns) to enhance heat transfer under low-pressure conditions, while the condenser utilizes a smaller tube diameter, larger coil dimensions, and a higher number of turns (36 turns), resulting in improved heat dissipation. These design variations are essential for achieving balanced thermal performance and enhancing the coefficient of performance (COP) of the solar adsorption refrigeration system. The use of compact fin-tube structures further improves area density, contributing to both compactness and enhanced thermal efficiency.

Baiju & Muraleedharan's [12] experimental studies reveal that the performance of solar adsorption refrigeration systems relies heavily on the efficiency of the evaporator and condenser. Their results confirm that using better designed heat exchangers results in increased cooling capacity, improved stability of the system and higher overall efficiency. The results of this study, such as the increased surface area for the condenser and the optimized evaporator configuration, match the experimental data upon which they are based. Additionally the similarities between the analytical results obtained in this paper and those derived from the experimental results confirms the analytical method used and support the viability of the proposed designs.

CONCLUSION

Solar adsorption refrigeration is a promising low-energy cooling technology whose performance is strongly influenced by the design of the evaporator and condenser. This study presents an analytical design approach for both components, highlighting that the evaporator should minimize thermal resistance and enhance heat absorption, while the condenser should maximize heat rejection through increased surface area and compact configurations. The use of helical coil and finned heat exchanger designs improves heat transfer performance and contributes to better coefficient of performance (COP).

The experimental findings reached in the past, for example, studies provided by Baiju and Muraleedharan confirm how well the heat exchanger's effectiveness contributes to system efficiency and stability. The similarity of these two outcomes lends further credibility to the analysis methods used in this investigation.

To improve accuracy of heat exchanger performance estimates, research needs to validate performance by prototype testing and using Computation Fluid Dynamics (CFD) for simulation. Optimizing the materials used and enhancing the design of fins for use within solar adsorption cooling systems is necessary for future testing.

Nomenclature

Symbol	Description
Cp	Fluid heat capacity, kcal/kg.°C
D_i	Inner diameter of coil, m
D_o	Outer diameter of coil, m
P	Pitch, m
C	Inner diameter of shell, m
H	Height of shell, m
D_h	Diameter of helix, m
D_{h1}	Inside diameter of helix, m
D_{h2}	Outside diameter of helix, m
r	Average radius of helical coil, m
L	Length of coil needed to form N turns, m
V_a	Volume of annulus, m³
V_c	Volume occupied by N turns of coil, m³
V_f	Volume available for fluid flow in annulus, m³
D_e	Shell-side equivalent diameter of coil, m
G_s	Mass velocity of fluid, kg/m²h
Pr	Prandtl number, dimensionless
Re	Reynolds number, dimensionless
k	Thermal conductivity of fluid, kcal/(h·m·°C)
kc	Thermal conductivity of coil wall, kcal/h·m·°C
μ	Fluid viscosity at mean bulk temperature, kg/m.h
h_o	Heat-transfer coefficient outside coil, kcal/(h·m²·°C)
u	Fluid velocity, m/h
h_i	Heat-transfer coefficient inside straight tube, kcal/(h·m²·°C)
h_{ic}	Heat-transfer coefficient inside coiled tube, kcal/(h·m²·°C)
h_{io}	Heat-transfer coefficient inside coil (based on outside dia), kcal/(h·m²·°C)
jH	Colburn factor for heat transfer, dimensionless
N	Number of turns of helical coil
Q	Heat load, kcal/h
R_a	Shell-side fouling factor
R_t	Tube-side fouling factor
ΔT_m	Log-mean temperature difference, °C
U	Overall heat-transfer coefficient, kcal/h·m²·°C
x	Thickness of coil wall, m
ρ	Fluid density, kg/m³

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