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Numerical investigation of the performance of a triple concentric pipe heat exchanger

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ABSTRACT

The performance of a triple concentric pipe heat exchanger is carried out numerically using finite element method (FEM) under steady state conditions for different flow arrangements and for insulated as well as non-insulated conditions of the heat exchanger. The three fluids being considered are hot water, cold water and the normal tap water. The results are presented in the form of the dimensionless temperature variations of the three fluids along the length of the heat exchanger for their different flow rates. It is found that the numerical predictions of the temperature variations of the three fluids by using FEM follow closely to those obtained from experiments both in magnitude and trend provided correct overall heat transfer coefficients are used. Parametric studies are also carried out to show the effect of the individual design parameter on the performance of the heat exchanger.

1. Introduction

Heat exchangers have been used in various industries for space heating, air conditioning, power production, waste heat recovery, and chemical processing [1,2]. Further, heat exchangers are an essential part of the food industry and are used for pasteurization, sterilization, drying, evaporation, cooling, and freezing purposes [3]. Heat exchangers have been categorized based on flow directions (parallel-flow, counter-flow, and cross-flow), type of construction of the heat exchanger (such as tubular or plate heat exchangers), or on the basis of way of contact between the fluids (direct or indirect). The type of heat exchanger to be used is determined by the process and the product specifications. The performance of the heat exchanger generally depends on the various physical characteristics of the fluid and the material. The relationships between the heat exchanger effectiveness and the thermal resistance (or conductance) are well explained by Guo et al. [4].

Adding an intermediate tube to a double concentric tube heat exchanger converts it to a triple tube one and the latter performs better compared to the prior one [5]. Triple concentric-tube heat exchangers provide better heat transfer efficiencies compared to the double concentric-tube heat exchangers [6]. Principally, the third pipe improves the heat transfer through an additional flow passage and a larger heat transfer area per unit exchanger length. The performance of the non-insulated three fluids heat exchanger will be different from that for the insulated heat exchanger mainly due to the effect of the surrounding ambient present in the former case. It is to be noted that in this case, there are three thermal communication surfaces to be considered for the analysis. Efficiency optimization and energy consumption minimization of a triple concentric-tube heat exchanger was demonstrated by means of a simulated model by Garcia-Valladares [7]. The similar approach was adopted to demonstrate the heat treatment of milk in a helical triple tube heat exchanger by Nema and Datta [8]. Thermal performance for a triple concentric pipe heat exchanger for both cases of co-current and counter-current flow arrangements is evaluated numerically using FEM by Saeid and Seetharamu [9] which is restricted to insulated conditions. Krishna et al. [10] have investigated the effect of heat in leak to the cold fluid in a three-fluid heat exchanger of non-concentric type using both the analytical and finite element methods. They carried out parametric analyses for their seven dimensionless parameters to study their effects on hot fluid behaviour in terms of temperature profile, effectiveness and degradation factor. Quadir et al. [11] have recently reported the performance of a triple concentric pipe heat exchanger investigated experimentally under steady state conditions for different fluid flow arrangements. They expressed their results in terms of temperature distributions of the three fluids along the length of the heat exchanger. The occurrence of cross-over points are also shown and discussed in their results. Krishna et al. [12] have

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Nomenclature

BC	boundary conditions
c_p	specific heat, J/kg K
D_h	hydraulic diameter, m
H	overall heat transfer coefficient ratio
f	load vector in Eq. (15); friction coefficient in Eq. (21)
h_1	heat transfer coefficient of the normal water, W/m^2 K
h_2	heat transfer coefficient of the hot water, W/m^2 K
h_3	heat transfer coefficient of the cold water, W/m^2 K
h_o	heat transfer coefficient of air, W/m^2 K
g	acceleration due to gravity, m/s^2
$[K]$	stiffness matrix
k	thermal conductivity, W/m K
L_e	length of an element, m
$\dot{m}$	mass flow rate, kg/s
N_1	shape function = $1 - X$
N_2	shape function= X
Nu	Nusselt number
NTu_e	number of heat transfer units between the hot and cold water
$NTu_{e\infty}$	number of heat transfer units between the surrounding ambient and cold water
P_1	contact perimeter between the normal and hot water, m
P_2	contact perimeter between cold and hot water, m
P_o	contact perimeter between the surrounding ambient and cold water, m
Pr	Prandtl number
R_1	heat capacity ratio between the hot and cold water
R_2	heat capacity ratio between the hot and normal water
r_{1i}	inner radius of the innermost pipe, m
r_{1O}	outer radius of the innermost pipe, m
r_{2i}	inner radius of the inner pipe, m
r_{2O}	outer radius of the inner pipe, m
r_{3i}	inner radius of the outer pipe, m
r_{3O}	outer radius of the outer pipe, m
Ra	Raleigh number
Re	Reynolds number
T_f	mean film temperature, °C
T	temperature of the fluid stream, °C
U_1	overall heat transfer coefficient between the normal and hot water, W/m^2 K
U_2	overall heat transfer coefficient between the cold and hot water, W/m^2 K
U_o	overall heat transfer coefficient between the surrounding ambient and the cold water, W/m^2 K
$\dot{V}$	volume flow rate, l/m
X	dimensionless length of the heat exchanger
Θ	dimensionless temperature of the fluid stream
β	coefficient of compressibility = $1/T_f$
δ	characteristic length, m
ν	kinematic viscosity, m^2/s

Subscripts

c	cold water
e	element
h	hot water
n	normal water
in	inlet
out	outlet
∞	ambient

investigated the effect of longitudinal heat conduction in the separating walls on the performance of a three-fluid cryogenic heat exchanger involving thermal interaction between all three fluids using the analytical method and FEM.

The temperature distributions of the three fluids along the length of the heat exchanger are not reported experimentally by many researchers in the literature. Further, it is evident from the literature review that the majority of researchers considered the triple concentric pipe heat exchanger being insulated from the surrounding ambient. The current work presents the numerical analysis of three fluids flowing in a triple concentric pipe heat exchanger and exchanging heat transfer between them for insulated and non-insulated conditions of the heat exchanger.

2. Mathematical model

For the non-insulated triple concentric pipe heat exchanger, the heat exchange takes place among four fluids with three thermal communication surfaces as shown in Fig. 1. The fourth fluid is the surrounding ambient.

In order to analyze the thermal performance of the non-insulated three concentric pipe heat exchanger, following assumptions are made:

(1) The heat exchanger operates at a steady state.
(2) Axial conduction in the pipes or in the fluid streams is neglected.
(3) There is no heat source or heat sink present in the heat exchanger or in any of the fluid streams.
(4) The mass flow rates of the different fluids remain constant.
(5) All properties, variables, and parameters are considered to be constant.

Consider the steady-state flow of hot fluid in the inner annulus exchanging heat between steady state flows of two fluids, called as cold water and normal water. For co-current parallel flow arrangement, all the three fluids flow in the same direction. If the direction of flow of hot water is reversed keeping the flow directions of cold and normal water same as before, then the arrangement refers to the counter-current parallel flow as shown in Fig. 1. Another possibility for counter-current parallel flow arrangement is the change in the direction of flow of cold and normal water keeping the hot water flow direction fixed. The following mathematical formulation is carried out for the flow arrangement as shown in Fig. 1. By applying the energy balance between different fluids, the following governing equations for each of the stream are written:

$$(\dot{m}c_p)_h \frac{dT_h}{dx} = \mp\ U_2P_2(T_h - T_c) \mp U_1P_1(T_h - T_n) \tag{1}$$

$$(\dot{m}c_p)_n \frac{dT_n}{dx} = +\ U_1P_1(T_h - T_n) \tag{2}$$

$$(\dot{m}c_p)_n \frac{dT_c}{dx} = +\ U_2P_2(T_h - T_n) + U_0P_0(T_\infty - T_c) \tag{3}$$

where, the negative sign in Eq. (1) is for co-current parallel flow and the positive sign is for the counter-current parallel flow. Using the following dimensionless variables:

$$\Theta = \frac{T - T_{c,in}}{T_{h,in} - T_{c,in}}, \quad \text{and} \quad X = \frac{x}{L_e}, \tag{4}$$