EXPERIMENTAL AND NUMERICAL INVESTIGATIONS OF THE PERFORMANCE OF THREE CONCENTRIC PIPES HEAT EXCHANGER

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LIST OF ABBREVIATIONS

Q	Heat transfer
A	Area
$\dot{m}$	Mass flow rate
$\dot{V}$	Volume flow rate
T	Temperature
Cp	Heat capacity
U	Overall heat transfer coefficient
P	Contact parameter
Θ	Dimensionless temperature
L_e	Length of element
h	Hot water
c	Cold water
n	Normal water
NTu	Number of transfer units
$R1$	Heat capacity ratio between hot and cold fluid
$R2$	Heat capacity ratio between hot and normal fluid
H	Heat transfer coefficient ratio
$\Theta_{3,in}$	Dimensionless inlet temperature
W	Weighted residual
N	Shape function
K	Stiffness matrix
Θ'	Element dimensionless temperature
f	Friction factor
D	Diameter
Tb	Bulk temperature
h_h	Heat transfer coefficient of hot water
h_c	Heat transfer coefficient of cold water
h_n	Heat transfer coefficient of normal water
Dh	Hydraulic diameter
Re	Reynolds number
Nu	Nusseltt number
Ra	Rayleigh number
k	Thermal conductivity
ρ	Density
γ	Kinematics viscosity
D_o	Outer diameter
U_o	Overall heat transfer coefficient with ambient
T_O	Temperature of surrounding ambient
h_o	Heat transfer coefficient of air
Pr	Prandtl number
L	length
M	Dimensional matrix parameter

ζ	Dimensionless length
ζ^*	Local cross over point
ε	Effectiveness of the heat exchanger
ΔT	Temperature difference
ϑ	Temperature effectiveness
NTu_∞	Number of transfer unit between the air and outer surface of the heat exchanger
c	Co-current flow
cc	Counter-current flow

LIST OF UNITS

Heat transfer	W
Heat transfer coefficient	W/m^2K
Overall heat transfer coefficient	W/m^2K
Heat capacity	$J/kg.K$
Thermal conductivity	$W/m.K$
Density	Kg/m^3
Kinematics viscosity	m^2/s
Temperature	^oC
Volume flow rate	l/s
Mass flow rate	kg/s
Diameter	m
Area	m^2
length	m

Penyiasatan Eksperimental dan Numerik Prestasi Tiga Penukar Haba Paip konsentrik

ABSTRAK

Tiga konsentris paip penukar panas adalah versi sedikit diubah dari penukar panas balang ganda. Walaupun desain penukar panas telah menunjukkan kemajuan yang luas, mereka umumnya terbatas pada beberapa aliran banyak susunan yang mungkin dan banyak terhad pada dua penukar panas fluida. Sebuah tiga konsentris paip penukar panas yang dibuat di mana tiga cecair, iaitu air panas, air sejuk, dan aliran air biasa dengan suhu yang berbeza dan juga dengan laju massa aliran yang berbeza. Percubaan dilakukan untuk laju aliran massa yang berbeza dari cecair panas, sejuk, dan biasa untuk tatacara arus-arus co dan kontra-saat ini di bawah terpencil dan bukan-keadaan terpencil penukar panas. Dua gabungan aliran untuk cecair diambil, yang pertama apabila arus air dingin melalui anulus luar, dan air biasa yang mengalir melalui paip dalam, dan yang kedua ketika arus air dingin melalui paip dalam dan arus air muzik melalui anulus luar , dengan membiarkan air panas mengalir melalui anulus batin dalam gabungan keduanya. Dijumpai bahawa variasi suhu dalam gabungan pertama adalah lebih baik daripada yang kedua di mana penurunan suhu outlet air panas lebih tinggi. Kaedah unsur hingga digunakan untuk memprediksi variasi suhu dari tiga cecair sepanjang penukar panas dengan mengembangkan program komputer menggunakan perisian MATLB.Hal ini mendapati bahawa ramalan berangka variasi suhu dari tiga cecair dengan menggunakan kaedah unsur hingga mengikuti rapat dengan yang diperolehi dari percubaan baik dalam besarnya dan trend. Ekspresi analitis sedia dalam sastera untuk memprediksi titik crossover dalam hal lokasinya dijumpai memuaskan dalam penyiasatan ini. Akhirnya, daripada analisis parametrik dari pertukaran panas terpencil, dijumpai bahawa dan sangat mempengaruhi terhadap prestasi terma, Begitu juga, dari analisis parametrik dilakukan untuk penukar panas bukan-terpencil, dijumpai bahawa ada pengaruh yang diucapkan dari pada variasi suhu dari tiga cecair terutama pada suhu outlet air sejuk untuk rentang suhu persekitaran sekitar dipertimbangkan dalam analisa ini. Untuk $NTu \geq$ 0.05, peratusan perubahan suhu air sejuk outlet dijumpai menjadi 12,42% apabila suhu sekitar berdimensi bervariasi dari -0,25 menjadi 0,5. Hal ini meningkatkan peratusan perubahan untuk 23,29% saat ini NTu_{∞} meningkat menjadi 0,1 sementara ini, untuk parameter desain lain, perubahan peratusan dalam suhu yang hampir konstan.

Experimental and Numerical Investigations of the Performance of Three Concentric Pipes Heat Exchanger

ABSTRACT

The three concentric pipes heat exchanger is a slightly modified version of double tube heat exchanger. Although the heat exchanger designs have shown extensive progress, they are generally limited to few of many possible flow arrangements and mostly restricted on two fluid heat exchangers. A three concentric pipes heat exchanger is fabricated wherein three fluids, namely hot water, cold water, and normal water flow with different temperatures and also with different mass flow rates. Experiments were conducted for different mass flow rates of the hot, cold, and normal fluids for co-current and counter-current flow arrangements under insulated and non-insulated conditions of the heat exchanger. Two flow combinations for the fluids are taken, first when the cold water flows through the outer annulus, and the normal water flows through the inner pipe, and the second when the cold water flows through the inner pipe and normal water flows through the outer annulus, by allowing the hot water to flow through the inner annulus in both combinations. It is found that the temperature variation in the first combination is better than the second one where the drop in outlet temperature of the hot water is higher. Finite element method is used to predict the temperature variation of the three fluids along the length of heat exchanger by developing a computer program using MATLB software. It is found that the numerical predictions of the temperature variation of the three fluids by using the finite element method follow closely to those obtained from experiments both in magnitude and trend. The analytical expression available in the literature to predict the crossover point in terms of its location is found to be satisfactory in the present investigation. Finally, from the parametric analysis of the insulated heat exchange, it is found that the $R1$ and NTu affect strongly on the thermal performance ,Similarly, from the parametric analysis carried out for the non-insulated heat exchanger, it is found that there is a pronounced effect of NTu_{∞} on the temperature variation of the three fluids especially on the outlet temperature of cold water for the range of the surrounding ambient temperature considered in the present analysis. For $NTu_{\infty} \geq 0.05$, the percentage change in outlet cold water temperature is found to be 12.42% when the dimensionless ambient temperature varied from -0.25 to 0.5. This percentage change increases to 23.29% when NTu_{∞} is further increased to 0.1 while, for other design parameters, the percentage change in temperatures are nearly constant.

CHAPTER 1

INTRODUCTION

1.1 Overview

Heat exchangers have been used in various industries for a wide range of applications (Incropera et al., 1990; Smith, 1997). Some of these applications may be found in space heating, air conditioning, power production, waste heat recovery, and chemical processing. Besides that, heat exchangers are an essential part of the food industry. Pasteurization, sterilization, drying, evaporation, cooling, and freezing are just a few of the purposes that they are being used for (Zuritz, 1990). 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 based on the contact between the fluids (direct or indirect). The type of heat exchanger to be used is determined by the process and the product specifications. Nevertheless, tubular heat exchangers play a major role in accomplishing the heat exchange needs of the food industry. The most common tubular heat exchanger is the double tube heat exchanger (Sunders, 1988). It consists of two concentric tubes of the same length but different diameters. In this configuration, two fluids exchange the heat between them.

1.2 Three Concentric Pipes Heat Exchanger Definition

A three concentric pipes heat exchanger is a slightly modified version of double tube heat exchanger. In this case, there are three concentric pipes and three fluids exchange heat between them, one of the fluids (to be heated or cooled) flows in the inner annulus formed between the inner pipe and outer annulus pipe. Therefore, the three concentric pipes heat exchangers provide better heat transfer efficiencies compared to double concentric pipes heat exchangers.

Because the third pipe improves the heat transfer through an additional flow passage and a larger heat transfer area per unit exchanger length.

1.3 Thermal Design Problem

There are two design problems to be addressed for the heat exchanger under investigation (Sekulic and Shah, 1995). These are known as rating problem and sizing problem. Rating problem discusses the performance of the heat exchanger whereas the sizing problem deals with the design of the heat exchanger. In both problems, heat transfer coefficients are assumed to be known. These two problems are now discussed in the following sub-sections.

1.3.1 Rating Problem

The outlet temperatures for the three fluids flowing through the heat exchanger are obtained depending upon:

1. The type of the heat exchanger (direct or indirect), the configuration of the fluids flowing through it, and the complete dimensions.
2. Number of the communication surfaces whether they are two or three.
3. Mass flow rate of the three fluids, and the flow arrangements.
4. Thermal properties of the fluids.
5. Inlet temperatures of the three fluids flowing through the heat exchanger.

1.3.2 Sizing Problem

To specify the value of heat exchange between the hot fluid and other two fluids in three concentric pipe heat exchanger, there is a need to indicate the heat exchanger type, fluid flow arrangement and the physical size (length, diameters and thickness of the pipes), and calculate the most important parameter *NTu* for the heat exchanger.

1.4 Problem Statement

In a conventional insulated heat exchanger, there is one thermal communication for heat transfer from one fluid to another. When the insulation is removed, there is an additional thermal communication due to the heat loss to the environment (Barron, 1984). The problem gets complicated when the second and /or third thermal communication is introduced by the third fluid stream in the so-called three fluids heat exchanger (Sekulic and Shah, 1995). The practical data of the temperature variation of the three fluids along the length of the heat exchanger in such a heat exchanger is not available in the literature. The thermal performance of three concentric pipes heat exchanger, where one of the fluid (to be heated or cooled) flows in the inner annulus formed between the innermost and intermediate pipe, while the second fluid flows in the annulus between the intermediate pipe and the outer pipe is to be analyzed. The third fluid flows inside the innermost pipe. The direction of flow of these three fluids can be changed manually as desired. The heat exchanger is firstly insulated from the ambient to avoid heat transfer to the surrounding, and then the insulator is removed to indicate the effect of ambient on the performance of heat exchanger. Experiments are conducted for different mass flow rates of the hot, cold, and normal fluids. The thickness of the pipes is small. Therefore, its effect as thermal resistance is neglected in the analysis.

The methodology used to predict the temperatures variation along the length of heat exchanger is by applying the finite element method as numerical solution and then use the analytical solution for the three concentric pipes heat exchanger.

The temperature variations of the three fluids are evaluated using a computer program developed by MATLAB software. The experimental results are compared with those predicted by finite element method for the same operating conditions. The comparison of the three results is given first for the co-current parallel flow conditions followed by the counter-current parallel flow heat exchanger when the heat exchanger is insulated. Then the same is repeated for the case of non-insulated condition to indicate the effect of surrounding ambient on the performance of the three concentric pipes heat exchanger. This procedure is followed once when the cold water flows through the inner pipe and normal water flows in outer annulus pipe, and next when the normal water flows through the inner pipe and cold flows through the outer annulus. It may be pointed out that the hot water flows through the inner annulus in both the above mentioned cases.

1.5 Objectives

The present research has the following objectives:

1. To design and fabricate the three concentric pipes heat exchanger.
2. To investigate the temperature variation along the length of three concentric-pipes heat exchanger for each fluid experimentally.
3. To investigate the effect of ambient (non-insulated condition) on the performance of three concentric pipes heat exchanger experimentally.
4. To investigate the temperature variation along the length of heat exchanger numerically by applying the finite elements method and compare the predicted result with those obtained from the experiments.
5. To study the performance of three concentric pipes heat exchanger by applying the analytical method and compare the results with the experimental performance.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

There are a large number of engineering applications of heat exchanger, such as in heating and cooling the space, power plant, food manufacturing and chemical fabrication. The three concentric pipes heat exchanger offer better heat exchanging this mainly due to the addition of the third pipe into the double pipe heat exchanger. The addition of the third pipe increases the thermal contact between the three fluids. There are many researches explaining the design and performance of two fluids heat exchangers for different flow arrangements. An impressive list of information is available in relevant references during the past 80 years for the design and analysis of double pipe heat exchanger. With addition of third fluid stream increases the complexity of design for the new category of heat exchanger.

2.2 Thermal Analysis of the Heat Exchanger

Morley (1933) analyzed the three concentric pipes heat exchanger in steady state and investigated the temperatures difference through the heat exchanger for all fluids. Based on the theory of energy balance and rate equations, he obtained third differential equation for the temperatures distribution along the length of heat exchanger. In addition he offered the formulas for all three fluids temperatures distributions throughout the heat exchanger. Furthermore, he obtained the solution in terms of unknown coefficients of integration, which he suggested in terms of boundary conditions specified for a given problem.

Hausen (1950) addressed Morley's work and defined an explicit form of temperature difference for three fluids heat exchanger for the case of counter-current flow arrangement. His solution is algebraically well organized, but he did not identify that one can determine in some cases the size of the heat exchanger without a trial and error method. It is important to note that Hausen described how to perform the calculation procedure in the general case of variable heat capacities of the fluids and/or the variable heat transfer coefficients.

Rao (1977) analyzed the co-current and counter-current parallel flow three fluids heat exchanger in the form of temperature variations between the fluids as dependent variables. The heat exchanger had three communications. The general solution for the temperature differences along the length of heat exchanger under steady-state operation and for constant proprieties was presented in dimensional form without defining related design parameters.

Kancir (1980) presented the temperature distributions for the parallel flow heat exchanger in counter-current flow arrangement. He used a matrix algebra process. The form of the solution is like the existing solutions but neither detailed quantitative nor qualitative comparisons were prepared.

Skulic et al. (1995) offered in detail a review on thermal design theory of three fluids heat exchanger, where they have allowed the third fluid temperature to vary according to the main thermal communications while neglecting interaction with the ambient. They classified the heat exchanger in several ways according to the heat transfer process, where the flow arrangement, basic construction, the heat transfer mechanisms, and the number of fluids involving are considered for the heat exchanger.

Unal (1998) conducted a theoretical study for the three concentric pipes heat exchange. His model involved a set of equations derived for adiabatic three concentric pipes heat exchanger using some suitably defined parameters such as heat capacity, number of transfer units NTu, and some other dimensionless parameters. The physical model of the system was modified for counter-current flow arrangement. To reduce the difficulty and make the derivation more easy the following assumptions were considered:

1. All working fluids are incompressible.
2. No phase change occurred through the heat transfer process.
3. Properties of the fluids are under steady-state.
4. There is no heat loss to the surrounding ambient from the heat exchanger. Thus the heat exchanger was fully insulated.

By following these assumption and applying the conservation of energy principle for small control volume with length of (dx) yields the following equation.

$$d\dot{Q}_2 = d\dot{Q}_1 + d\dot{Q}_3 \tag{2.1}$$

In the above equation, the differential values of heat transfer rates $d\dot{Q}_j$, denote heat lost by hot fluid or heat gained by cold fluid flow between locations (x) and (x + dx), and since there is no phase change accrued the differential heat flow rates can be expressed in form of mass flow rates, specific heat and temperature different as shown below.

$$d\dot{Q}_j = \dot{Q}_j(x+dx) - \dot{Q}_j(x) = (\dot{m}C_p)_j \, dT_j \tag{2.2}$$

where j = 1, 2, 3 stand for the fluids flowing through the inner pipe, intermediate and outer pipe, respectively, in the same time, the differential heat gains of cold fluid through the differential control volume can expressed in terms of the difference between bulk temperature of hot and cold fluid streams, $T_2 - T_1$ and $T_2 - T_3$, overall heat transfer coefficients, U_1, U_3, and the corresponding differential heat transfer surface areas,

$$d\dot{Q}_1 = d\dot{Q}_{21} = (T_2 - T_1)U_1 dA_1 \tag{2.3}$$

$$d\dot{Q}_3 = d\dot{Q}_{23} = (T_2 - T_3)U_3 dA_3 \tag{2.4}$$

Again , for the purposed of simplicity, by assuming that the pipe walls are thin, the differential surface areas and overall heat transfer coefficients appearing in the above equations can given as.

$$dA_1 = 2\pi r_1 dx \, , dA_2 = 2\pi r_2 dx \, , U_1 = \left[\frac{1}{h_1} + \frac{1}{h_{2i}}\right]^{-1} \text{ and } U_3 = \left[\frac{1}{h_{2o}} + \frac{1}{h_{3i}}\right]^{-1}$$

where, A = cross section area, r = radii of pipe, h = heat transfer coefficient

For, simplicity using the following definitions:

$$C_1 = (\dot{m}C_p)_2 \, , \; C_2 = (\dot{m}C_p)_2 \, , \; C_3 = (\dot{m}C_P)_3 \, , \;\; A_1 = 2\pi r_1 L \, , \;\; A_3 = 2\pi r_2 L \, ,$$

$$\Delta T = T_{hi} - T_{ci} \, .$$

and, the dimensionless parameters:

$$X = \frac{x}{L} \, , r_1^* = \frac{r_1}{r_2} \, , \; r_2^* = \frac{r_2}{r_3} \, , \; \Theta_1 = \frac{T_2 - T_1}{\Delta T_i} \, , \; \Theta_2 = \frac{T_2 - T_{ci}}{\Delta T_i} \, , \Theta_3 = \frac{T_2 - T_3}{\Delta T_i} \text{ and}$$

$$C_{r1} = \frac{C_1}{C_2} \, , \; C_{r3} = \frac{C_3}{C_2} \, , \; N_1 = \frac{U_3 A_3}{C_1} \, , N_2 = \frac{U_3 A_3}{C_3}$$

The governing energy balance equation can be reduced into:

$$\frac{d\Theta_1}{dX} + N_1(1 - C_{r1})\Theta_1 - N_3 C_{r3}\Theta_3 = 0 \tag{2.5}$$

$$\frac{d\Theta_3}{dX} + N_3(1 - C_{r3})\Theta_3 - N_1 C_{r1}\Theta_1 = 0 \tag{2.6}$$

$$\frac{d\Theta_2}{dX} = (N_1 C_{r1}\Theta_1 + N_3 C_{r3}\Theta_3) \tag{2.7}$$

By replacing Θ_3 and Θ_1, respectively from equations (2.5) and (2.6), the following second order ordinary differential equations were obtained for cold fluid flow:

$$\frac{d^2\Theta_1}{dX^2} + A\frac{d\Theta_1}{dX} + B\Theta_1 = 0 \quad (2.8)$$

$$\frac{d^2\Theta_3}{dX^2} + A\frac{d\Theta_3}{dX} + B\Theta_3 = 0 \quad (2.9)$$

where, the coefficients A and B are defined as:

$A = N_1(1 - C_{r1}) + N_3(1 - C_{r3})$ and $B = N_1 N_3[1 - (C_{r1} + C_{r3})]$.

The boundary conditions for counter flow arrangement can be specified in dimensionless form as:

$$\Theta_1(0) = \Theta_2(0) = \Theta_3(0) = \frac{T_{ho} - T_{ci}}{T_{hi} - T_{ci}} = \Theta_i \quad (2.10)$$

$$\frac{d\Theta_1}{dX}\Big|_{x=0} = (N_1 C_{r1} + N_3 C_{r3} - N_1)\Theta_i = F_1\Theta_i \quad (2.11)$$

$$\frac{d\Theta_2}{dX}\Big|_{x=0} = (N_1 C_{r1} + N_3 C_{r3})\Theta_i = F_2\Theta_i \quad (2.12)$$

$$\frac{d\Theta_3}{dX}\Big|_{x=0} = (N_1 C_{r1} + N_3 C_{r3} - N_3)\Theta_i = F_i\Theta_i \quad (2.13)$$

The common solution of the above linear uniform second order ordinary differential equation was presented completely with all probable cases.

The physical model for the co-current flow arrangement where both cold and hot fluids enter the heat exchanger at x = 0, was solved with inlet temperatures of $T_{1i} = T_{3i} = T_{ci}$ and $T_{2i} = T_{hi}$, respectively, and the flow in the positive x-direction.

Following the same procedure as mentioned for co-current flow arrangement, the dimensionless governing equations for counter-current flow arrangement, were presented by Unal (1998). The main difference in governing equations comes from the negative temperature gradient in bulk hot temperature with respect to x. The simple energy balance equation for this type of flow arrangement is

$$-d\dot{Q}_2 = d\dot{Q}_1 + d\dot{Q}_3 \tag{2.14}$$

Using the same procedure as discussed for the co-current flow arrangement the differential equations for the three fluids were obtained as:

$$\frac{d^2\Theta_1}{dX^2} + A\frac{d\Theta_1}{dX} + B\,\Theta_1 = 0 \tag{2.15}$$

$$\frac{d\Theta_2}{dX} = -\left(N_1 C_{r1}\Theta_1 + N_3 C_{r3}\Theta_3\right) \tag{2.16}$$

$$\frac{d^2\Theta_3}{dX^2} + A\frac{d\Theta_3}{dX} + B\Theta_3 = 0 \tag{2.17}$$

The boundary conditions are:

$$\Theta_1(0) = \Theta_2(0) = \Theta_3(0) = \frac{T_{hi} - T_{ci}}{T_{hi} - T_{ci}} = 1 \tag{2.18}$$

$$\frac{d\Theta_1}{dX}\Big|_{x=0} = -\left[N_1\left(1 - C_{r1}\right) + N_3 C_{r3}\right] = F_1 \tag{2.19}$$

$$\frac{d\Theta_2}{dX}\Big|_{x=0} = -\left(N_1 C_{r1} + N_3 C_{r3}\right) = F_2 \tag{2.20}$$

$$\frac{d\Theta_1}{dX}\Big|_{x=0} = -\left[N_1 C_{r1} + N_3\left(1 - C_{r3}\right)\right] = F_3 \tag{2.21}$$

It may be noted that the differential equations for cold fluid streams are the same as those of co-current flow arrangement. The solutions for non-dimensional bulk temperature variations are the same except that Θ_i, F_1, and F_3 are different in boundary conditions. This difference occurs in hot fluid temperature due to difference in the sign of hot fluid gradient with respect to x. The equations derived in this study, are useful for both design calculations and performance calculations and also can be used to determine of the bulk temperature variations along the length of heat exchanger but it does not