STUDY ON DYNAMIC OF DOUBLE-PIPE HEAT EXCHANGER

A Thesis

Submitted to the College of Engineering of

Nahrain University in Partial Fulfillment

of the Requirements for the Degree of

Master of Science

in

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by

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CERTIFICATION

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Abstract

In this work it has been attempted to study the dynamic behavior of a double pipe heat exchanger both experimentally and theoretically. The double pipe heat exchanger configuration simulated consisted of two concentric pipes, the outer pipe carries the hot fluid and the inner pipe carries cold fluid and the circulation of the two fluids is made by two pumps.

The influence of the following variables studied are the flow rate of hot and cold streams as follows: effect of flow rates of hot stream of 500, 700, 900, and 1100 L / h on the exit hot fluid temperature and also the same range of flow rates for the hot stream is studied as an effect on the exit cold fluid temperature. Firstly the results were obtained from the experimental work for steady state and dynamic results and secondly a mathematical model was constructed to get the transfer function for both the effect of cold fluid flow rate on the exit hot temperature and effect of hot fluid flow rate on the exit cold fluid temperature at which a step in the flow rate is applied to obtain the following expressions for the exit temperatures:

$$\overline{Tc}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right)$$

where $\alpha = BL(Th_i - Th_L)$, $\beta = v_c + v_H + AL + BL$

$$\overline{Th}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right)$$

where $\alpha = AL(Tc_i - Tc_L)$, $\beta = v_c + v_H + AL + BL$

The effect of increasing hot fluid flow rate on exit cold fluid temperature is to increase the temperature as for the effect of increasing cold fluid flow rate on the exit hot temperature is to decrease the temperature.

A comparison between experimental and mathematical model is made and an agreement is obtained if small duration of time is studied because of the approximation made on the model which leads to make the response to maintain constant increasing manner and frequency response plots showed this behavior very clearly.

Variable Notation

Variable	Notation	Unit
A	Constant for unsteady state hot heat balance equation	[s^{-1}]
$A^{'}$	Constant for steady state hot heat balance equation	[m. s^{-2}]
A_i	cross sectional area of the inner pipe	[m^2]
A_a	cross sectional area of the annulus	[m^2]
B	Constant for unsteady state cold heat balance equation	[s^{-1}]
$B^{'}$	Constant for steady state heat cold balance equation	[m. s^{-2}]
Cp	Heat capacity	[J /Kg.^oC]
d_i	diameter of inner pipe	[m]
h_o	heat transfer coefficient of outer fluid	[W/m^2.^oC]
h_i	heat transfer coefficient of inner fluid	[W/m^2.^oC]
k	thermal conductivity	[W/m.^oC]
L	length of heat exchanger	[m]
M	hold-up mass	[kg]
t	time	[s]
Tc	instantaneous cold fluid temperature	[^oC]
Th	instantaneous hot fluid temperature	[^oC]
U	overall heat transfer coefficient	[W/m^2 °C]
Ui	overall heat transfer coefficient for inner fluid	[W/m^2 °C]
Uo	overall heat transfer coefficient for outer fluid	[W/m^2 °C]
x	instantaneous length	[m]

Greek Letters

Variable	Notation	Unit
α	Constant defined by $AL(Tc_i - Tc_L)$	[-]
β	Constant defined by $\nu_c + \nu_H + AL + BL$	[-]
γ	Constant defined by $\nu_C + BL + \nu_H{}^s + AL$	[m/s]
λ	Difference between unsteady and steady velocities	[m/s]
μ	water viscosity	[kg/m.s]
ν	water velocity	[m/s]
ρ	water density	[kg/m^3]
ω	frequency	[rad/sec]

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Chapter One
Introduction

In chemical processes, an obvious way of conserving energy is to recover more of the heat that is currently dissipated to the surrounding. Most of this heat is at a relatively low temperature and therefore can not be usually economically recoverable. However, some of this heat could be recovered with the aid of high efficiency heat exchanger which can operate economically, with a close temperature approach, at a relatively low pumping power. One type of heat exchanger that is particularly suited for this duty is the gasketted double pipe heat exchanger. For many applications, this equipment can transfer heat with almost true counter current flow, which coupled with high heat transfer rates, gives efficient and cheap heat transfer [1].

The heat transfer equipment is defined by the function it fulfils in a process. Exchangers recover heat between two process streams. Steam and cooling water are utilized and are not considered in the same sense as recoverable process streams. Heaters are used primary to heat process fluids, and steam is usually employed for this purpose. Coolers are employed to cool process fluids [2]

The heat exchanger is usually considered to be an accessory piece of equipment in the modern chemical plant [3].Heat exchange is a fundamental and important operation in chemical processing. Control of process conditions is critical to the production of on-specification products, efficiently, economically and safely [4].

1.1 Classifications

There are two general types of heat exchangers, which are: the shell and tube heat exchanger and the special (proprietary) type.

There are further classification according to their mechanical configuration, installed position, and heat transfer process function.

1.1.1 Shell and Tube Exchangers

These exchangers are normally named in accordance with their mechanical configuration [5].

- Fixed tube sheet
- Return bend (U – tube)
- Floating tube sheet
- Bayonet
- Double pipe

1.1.2 Special Types

Special types of heat exchangers may be applied for solving special problems.

This group, is representing less than 15% of all units, containing equipment processing unique features and costing less than conventional exchangers, these types are listed below[6]:

- Reboilers, evaporators, and vaporizers
- Condensers
- Scraped surface
- Spiral types
- Plate types

1.2 Different Types of Heat Exchangers

There are other types of heat exchangers described extensively below in addition to those types presented in chapter one.

1.2.1 Air Cooled heat Exchangers[8]

1.2.2 Froth Contact Heat Exchanger

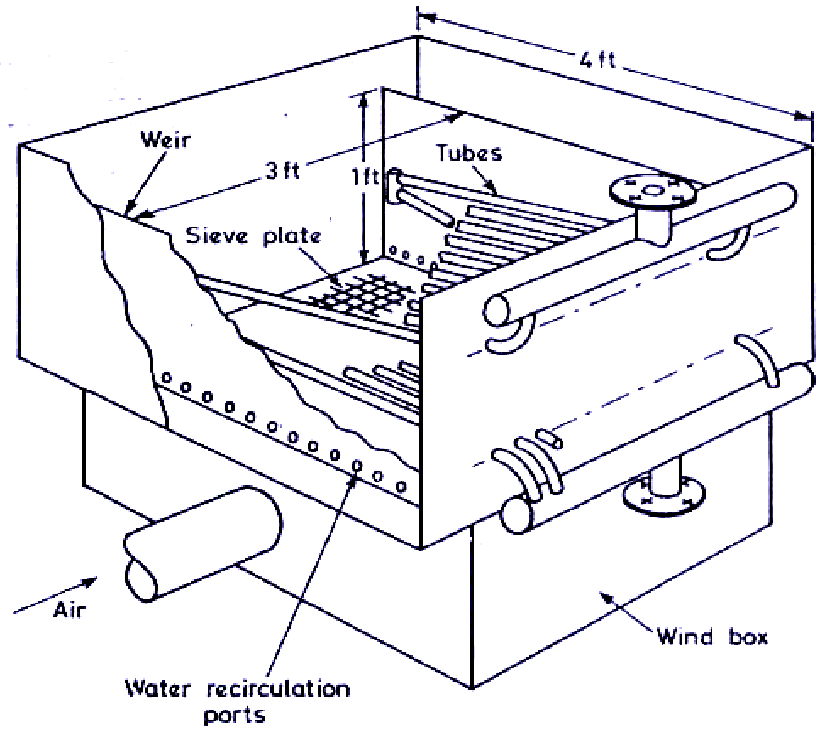


Fig. 1.1 Froth heat exchanger, Ref [9]

1.2.3 Votator Heat Exchangers

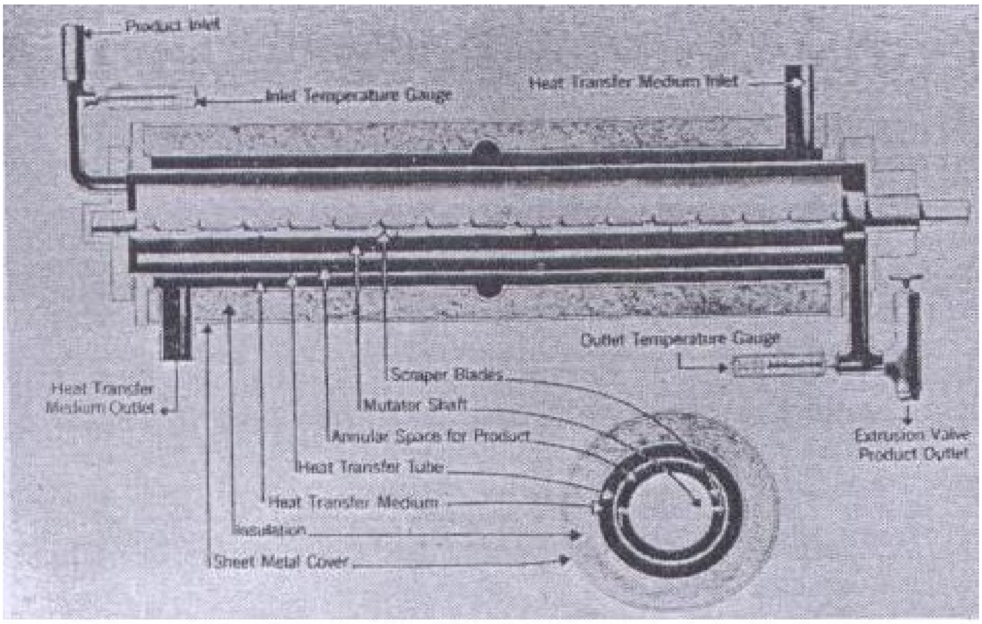


Fig. 1.2 Votator heat exchanger, Ref [10]

1.2.4 Ramen's Heat Exchangers

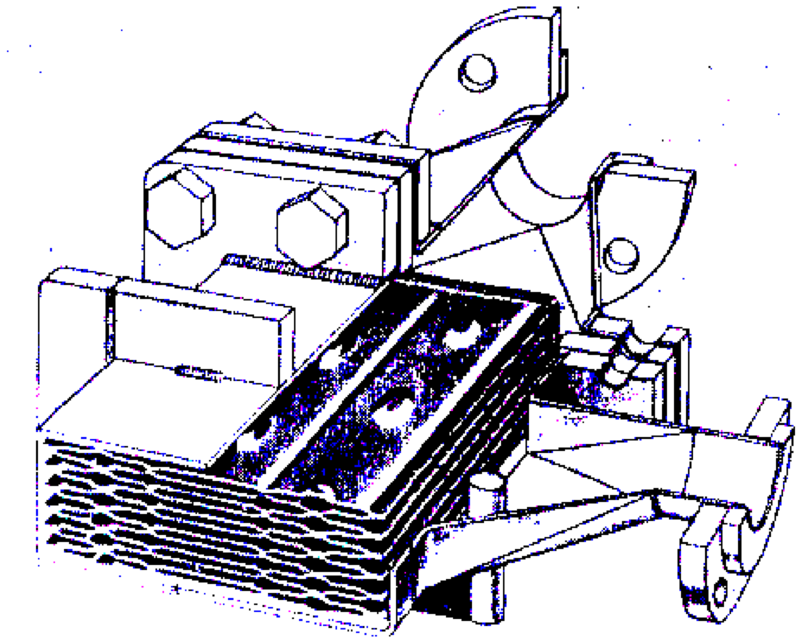


Fig. 1.3 Ramen's heat exchanger, Ref [11]

1.2.5 Rosenbland Spiral Heat Exchangers

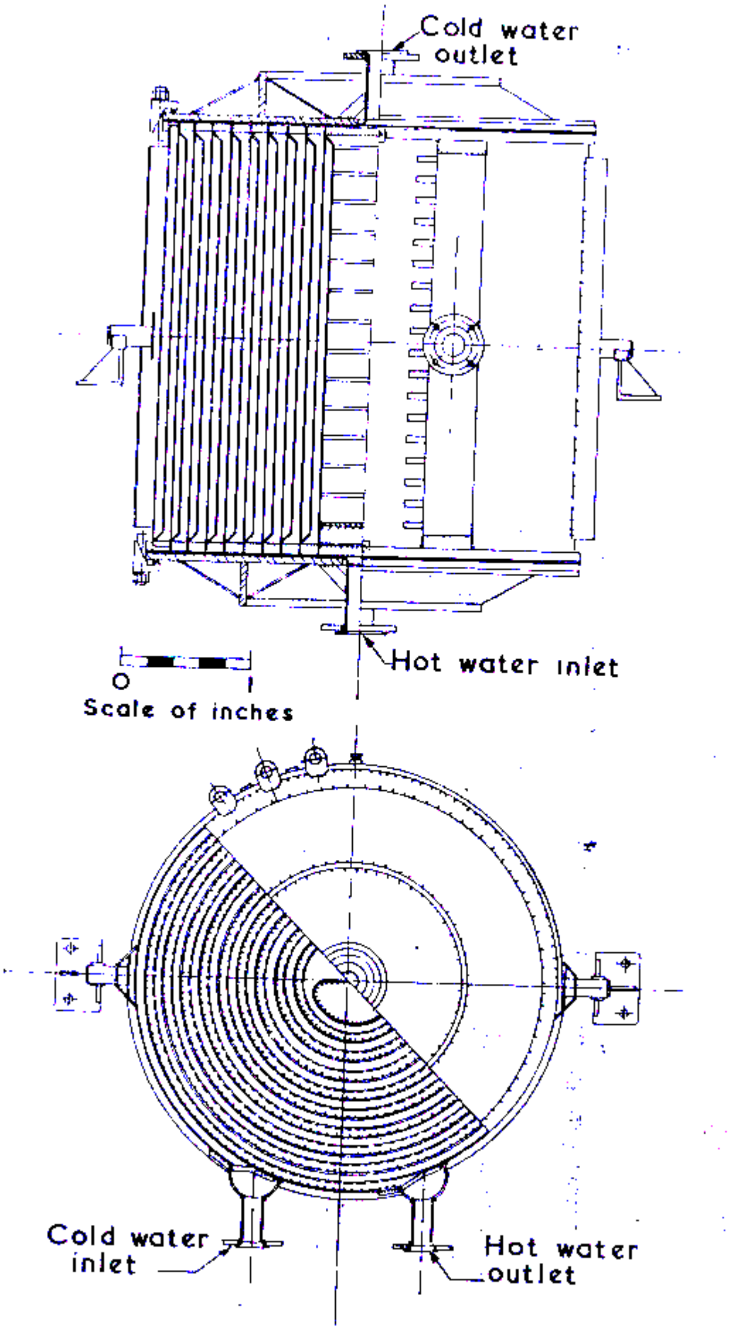


Fig. 1.4 Rosenbland heat exchanger, Ref [12]

1.2.6 Graphite Block Heat Exchanger

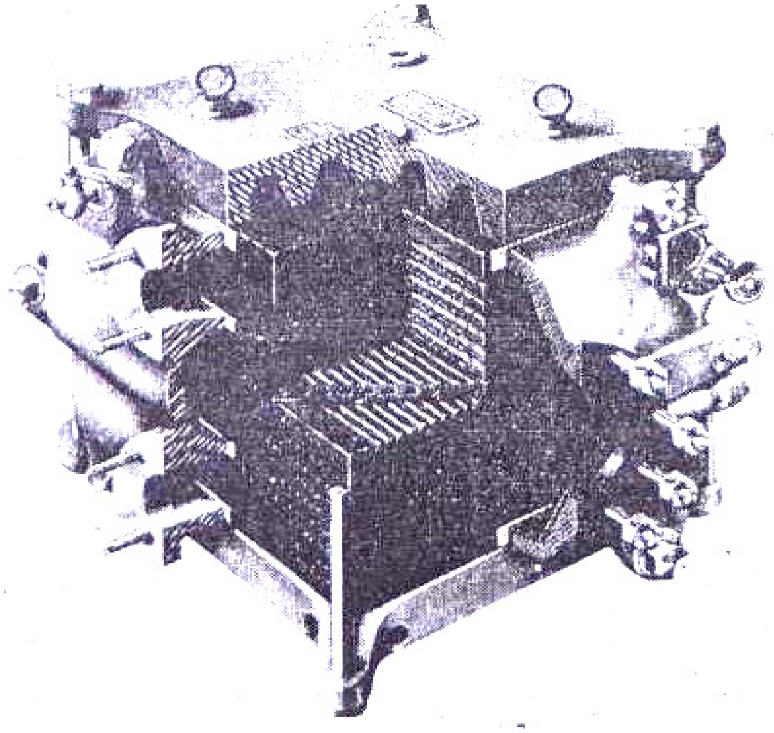


Fig. 1.5 Graphite block heat exchanger, Ref [13]

1.3 Scope of Present Work

The following parameters are to be studied in the present work. These variables are as follows:

1. Studying the dynamic response due to the effect of cold liquid flow rate on the exit hot liquid temperature.
2. Studying the dynamic response due to the effect of hot liquid flow rate on the exit cold liquid temperature.
3. Setting up a mathematical model for the double pipe heat exchanger.
4. Comparing the data obtained from the experimental work and the mathematical model.
5. Studying the frequency response for points 2 and 3, above.

Chapter Two

Literature Survey

2.1 Introduction

The continuous search for greater economy and efficiency has led to the development of many different types of heat exchanger other than plate exchanger. Double pipe heat exchanger introduced many years ago to meet the hygienic demands of the dairy industries. After that, the use of double pipe heat exchanger increased in other applications of chemical engineering. Dynamic characteristics of heat exchanger can be found in excess in the literature but methods of evaluating the dynamic characteristics depend upon the kind of heat exchanger . For example, finding dynamics of steam-heat exchanger is simpler than liquid-liquid exchanger because in the first one, one side is distributed but in other type both sides are distributed [7]. It is useful to compare the advantages and disadvantages of double pipe heat exchanger and plate exchanger.

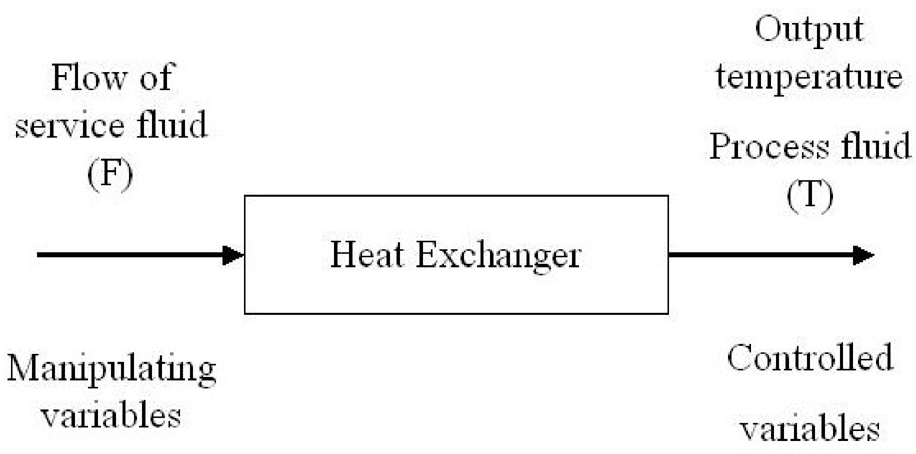
2.2 Comparison of Double Pipe Heat Exchanger and Plate Heat Exchanger

The double pipe heat exchangers are the best design for withstanding high temperature and pressure but cannot render optimum heat transfer surface [14]. The advantages and disadvantages are discussed in details elsewhere and are listed in the following:

1. For liquid / liquid duties the double pipe heat exchanger will give over-all heat transfer coefficient three times less than those of plate heat exchanger for the same pressure drop.[15]

2. The effective mean temperature difference for double pipe heat exchanger is usually less than that of plate heat exchangers.
3. Although the double pipe heat exchanger is the best shape of flow conditions for withstanding pressure, it is entirely the wrong shape for optimum heat transfer performance since it has the smallest surface area per unit cross sectional area of flow.
4. A double pipe heat exchanger usually occupies considerably larger floor space than plate heat exchanger for the same duty.
5. For many materials of construction, sheet metal is cheaper per unit area than tube for the same thickness, which reduces the cost of a plate heat exchanger relative to the double pipe heat exchanger.
6. The plate heat exchanger is limited by the necessity of the gasket being elatsomeric; the limiting temperature and pressure are 260° C, 25 atm, respectively.
7. Table 2.1 gives a good comparison in numerical values of important variables in design for double pipe heat exchangers and plate heat exchangers for the same duty [16].
8. The response of plate heat exchanger is faster than the equivalent duty double pipe heat exchanger because of its small holdup and larger heat transfer coefficient. The following analysis demonstrates this fact [17].

The diagram below shows the input – output relationship to the heat exchanger.



From the above, one may write:

$$\text{Sensitivity of exchanger} = \frac{\text{change in controlled variable}}{\text{change in manipulated variable}} \qquad (2.1)$$

i.e.

$$\text{Gain} = \frac{\Delta T}{\Delta F} \qquad (2.2)$$

Considering data from table (2-1), the holdup of double pipe heat exchanger is ten times more than the plate heat exchanger and the heat transfer coefficient is nearly three times less than that of plate heat exchanger, but the heat transfer area is three times larger than the plate heat exchanger if considering first order lumped system. The ratio of time constant of plate heat exchanger to double pipe heat exchanger is as follows [18]:

$$\frac{T_{plate}}{T_{tubular}} = \frac{0.1V/3hA/3}{V/hA} = 0.1 \quad (2.4)$$

The ratio of time constants is of the order of the ratio of the holdup volumes. The holdup volume affects both gain and lag, which means controllability, i.e. system will be more amenable to automatic control if it had high steady state gain and minimum lag.

Table 2.1 Numerical comparison between PHE and tubular one for the same duty (water / water duty) [16]

	Hot side	Cold side
Flow	50 m^3 / h	50 m^3 / h
Temperature in / out	80 / 40 ^oC	20 / 60 ^oC
The following are the results of the calculations for both types		
Variable's	Plate type	Tubular type
1. Heat transfer area	25 m^2	85 m^2
2. Overall heat transfer coefficient (clean)	6047.6 W / m^2 ^oC	2035.25 W / m^2 ^oC
3. Fouling factor	0.000516 m^2 ^oC / W	0.000086 m^2 ^oC / W
4. Overall heat transfer coefficient (service)	4605.48 W / m^2 ^oC	1744.5 W / m^2 ^oC
5. Pass system (hot / cold)	1 / 1	8 baffled
6. Calculated pressure drop	0.4 atm	0.6 atm
7. Pumping power, HP	1.1	1.65
8. Weight (empty)	615 kg	2400 kg
9. Weight (fully)	720 kg	3100 kg
10. Overall size	1.5 x 0.7 x 1.4 m	7 x 7 x 0.7 m
11. Floor area required	1 m^2	5 m^2

2.3 Studies of Dynamic Characteristics

The dynamic response studies are mainly concerned with outlet temperatures of either cold or hot stream. It is evident that the heat transfer resistance and capacitance of the fluids flowing in exchanger paths are distributed. The concerned variable, of outlet temperature, is function of space and time, results in pertinent differential equations in partial differential form. The solution of such system is quite complex and means lengthly calculations to obtain open loop frequency response [7].

For exchangers with several passes or exchangers where large changes in velocity or physical properties occur, digital computer would have to be used to determine the response.

In general heat exchangers are fairly easy to control except where very close control is needed. Simplified methods of dynamic analysis approximate the actual situation within acceptable limits which are accurate enough for the control engineer.

The analysis expressed elsewhere is to present exact transfer functions for a few simpler heat exchangers in order to show the parameters that determine the lags and to explain the response effects, which are inherited with distributed system [19].

2.3.1 Dynamic Studies of Plate Heat Exchangers

The plate heat exchanger is being increasingly used in the chemical and process industries on account of its flexibility low cost, high overall heat transfer coefficient, high effective mean temperature difference, high surface area per unit volume, less floor space, high degree of turbulence, good velocity profile, smooth heat transfer surface and low fouling resistance[5,20,21].

Maknight and Worely (1953) [22] investigated the dynamic characteristics of plate heat exchangers using frequency response analysis. They reported that the plate heat exchanger could be best approximated under dumped second order system with natural frequency and dumping factor of 10.7 and 0.645 rad / min respectively. Plate exchanger was used as cooler using brine as the coolant with the following flow arrangement, 3- passes, 7 channels/ pass for the water and single pass of 22 channels for the brine solution.

Masubuchi and Ito (1977) [23] analyzed the dynamic characteristics of water plate heat exchanger by actually taking the experimental frequency response. Three flow patterns (series, parallel, and complex flow pattern) were used among four –passage plate exchanger, and each flow pattern combined with various flow types according to the combination and the arrangement of plates with hole or a blank suitably placed at each corner. The flow rate of cold side and hot side streams were $2.7x10^{-5}$ m^3 / s (1.6 L / min) and $4x10^{-5}$ m^3 / s (2.4 L / min) respectively.

Finally the experimental responses were obtained by introducing a sinusoidal change of the inlet temperature in the hot side steam and measuring the outlet temperature in the cold side stream.

The results showed that two plate exchangers with the same characteristics didn’t always have the same dynamics if the flow types of the two were different and that the dynamics was affected more than static by the condition whether the inlet / outlet flow pattern of this heat exchanger was parallel or counter flow. They formulated a mathematical model to express the real situation. The equations were solved to provide theoretical results, which showed a favorable agreement with experimental results.

Masubuchi and Ito (1977) [24] developed the study of dynamic behavior of water – water plate exchanger having a number of heat transfer plates using the same previous pattern and flow rates.

The results were numerically compared under the condition that each flow rate of the hot and cold stream, respectively, remained constant in any flow type. The significant results were noted as:

1. In almost every flow type, the temperature effectiveness increased as the number of passages increased.
2. Dynamic responses (frequency responses) might greatly depend on the relation between the inlet and outlet passages where each fluid flow in a counter – or parallel current.
3. Series flow types where the inlet and outlet passages were adjacent and the flow was cocurrent, had the best statics and dynamics among all types when the number or passages was the same.

Zaleski and Tedszerski (1980) [25] presented a mathematical model simulating transient operation of the plate heat exchanger and a computer program by which they solved the problem for the given inlet and arbitrary structural and process parameters, was suggested.

2.3.2 Dynamic Studies of Double Pipe Heat Exchangers

Dynamic characteristics of popular heat exchanger excess in literature, but methods of evaluating dynamic characteristics is dependent upon the kind of heat exchanger, for example, finding dynamic characteristics of steam –heat exchangers is simpler than liquid-liquid exchanger because in the first one the side is distributed but in other type both sides are distributed [26]

Cohen and Johnson (1956) [27] presented a study of dynamic characteristics of double pipe heat exchanger as well as building an experimental heat exchanger where they used steam as hot fluid condenser in the annulus which heats the cold water flowing inside the inner pipe. They obtained an expression for the outlet cold-water temperature as a function of steam temperature.

$$\overline{\theta_L} = \left[\overline{\theta_o} - \frac{b}{a}\overline{\theta_s}\right] e^{-\frac{La}{v_f}} + \frac{b}{a}\overline{\theta_s} \tag{2.5}$$

where

$$a = s + \frac{1}{T_1} - \frac{T_{22}}{T_1\left(T_{12}T_{22}s + T_{12} + T_{22}\right)} \tag{2.6}$$

and

$$b = \frac{T_{12}}{T_1\left(T_{12}T_{22}s + T_{12} + T_{22}\right)} \tag{2.7}$$

Mozely (1956) [28] developed a simplified mathematical model for the purpose of rapid design calculations. He implemented two methods for approximation to get rid of difficulty of the solution of the heat balance equation.

The first method assumes that the fluid in each side of the exchanger is well mixed, which give, as a result, the following transfer function:

$$\frac{T_2}{T_1}(s) = \frac{w_1 C_1 UA}{[M_1 C_1 s + w_1 C_1 + UA][M_2 C_2 s + w_2 C_2 + UA] - (UA)^2} \tag{2.8}$$

The second method assumes the bulk temperature of the fluids in each side of the heat exchanger is equal to the arithmetic mean of the inlet and outlet fluid temperatures

$$\frac{T_{2o}}{T_{1i}}(s) = \frac{w_1 C_1 UA}{[M_1 C_1 s + w_1 C_1 + UA][M_2 C_2 s + w_2 C_2 + UA] - (UA)^2} \tag{2.9}$$

where the two methods give acceptable agreement with experimental results.

Less and Hougen (1956) [29] worked on steam - water of double pipe heat exchanger by applying sinusoidal variation in pressure signal which was applied to the control valve diaghram, thus causing a similar variation in the valve-stem position and flow of water through the exchanger while the steam pressure and inlet water temperature were maintained constant. The effluent water temperature and valve stem position both varying sinusoidally were recorded as function of time.

Gerard (1974) [30] formulated a model of counter flow double pipe heat exchanger, which has taken into account variation of the heat transfer coefficient with respect to fluid flow rates and temperatures. His approach involved the derivation of transcendental transfer matrix based on Linearization of a set of nonlinear partial differential equations, which describes the heat exchanger processes. He conducted experimental investigation using various test signal such as step function, sine wave, Gaussian noise and pseudorandom.

Burn et al. (1981) [31] used lead / lag approximation in modeling to a double pipe heat exchanger system. Their analysis involved extraction of successive real poles and zeros of the system transfer functions for temperature or flow forcing.

Ghanim (1982) [17] studied the dynamics of the heat exchanger using step change technique applied to cold water flow rate, and other variables were maintained as almost constant. Recorded outlet water temperature was analyzed by process reaction curve, which showed that the system can be represented by first order system with negligible time delay (dead time). Time constant is measured for various flow rates and he concluded that time constant is inversely proportional to the flow rate.

He applied feed back control loop to the system and concluded that steady state offset of the controlled value tend to be smaller as the magnitude of the flow disturbance gets smaller for all settings of proportional action. He applied integral controller in combination with the proportional controller to eliminate the offset and getting stable operation for most of the controllers' settings.

Chapter Three

Theoretical Modeling

3.1 Introduction

This chapter contains three main sections, which deal with the dynamic modeling of the double pipe heat exchanger.

The behavior of the output cold-water temperature as a function of hot water flow rate and that of hot water temperature as a function of cold water flow rate are also studied.

3.2 Theoretical Model Analysis

The mathematical model for the double pipe heat exchanger that is used in the resent work is derived below based on the unsteady state heat balance. The heat transfer coefficient is given by the following equation [32]. (Holman):

$$\mathrm{Nu} = 0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{n} \tag{3.1}$$

where n = 0.4 for heating and 0.3 for dooling, for Reynolds number between 10^4 and 120,000 and Pr from 0.6 to 380 [32].

For the model derivation, the following assumptions are made in the modeling of the double pipe heat exchanger:

1. Heat losses to the surroundings are negligible.

2. The heat transfer within the fluid in the pipes is governed by convection only.
3. The thermal capacity of the pipes wall is negligible.
4. The physical properties of the fluid are constants over the range of temperatures employed. And taken at bulk temperature (average mean of inlet and outlet temperatures).
5. The fluid in each side of the heat exchanger is well mixed, which means that the exit fluid temperature is equal to the bulk fluid temperature.

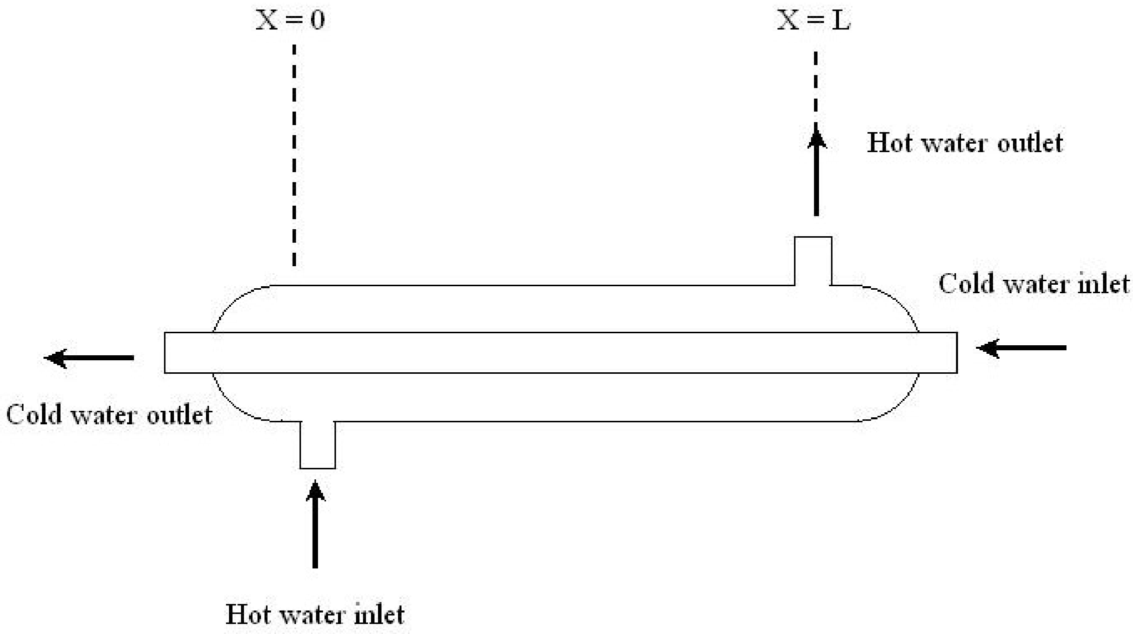


Fig. 3.1 Schematic diagram of the heat exchanger

3.2.1 Heat Balance on Heat Exchanger

The heat balance is made on the hot water and cold water according to the following:

heat in – heat out $\pm$ heat transferred by convection = heat accumulation

Heat balance on the cold water stream.

$$V_c A_i \rho C(T_C - T_r) - V_c A_i \rho C\left(T_c + \frac{\partial T_c}{\partial x}dx - T_r\right) + \pi d_i u_i \Delta x(T_h - T_c) = \frac{\partial}{\partial t}(A_i \rho C \Delta x(T_C - T_r)) \tag{3.2}$$

Where Tr is a reference temperature [33].

Similarly the heat balance on the hot water stream is as follows:

$$v_H A_a \rho C(T_h - T_r) - v_H A_a \rho C\left(T_h + \frac{\partial T_h}{\partial x}dx - T_r\right) - \pi d_0 u_0 \Delta x(T_h - T_c) = \frac{\partial}{\partial t}(A_a \rho C \Delta x(T_h - T_r)) \tag{3. 3}$$

Equations (3.2) and (3.3) are simplified according to the following

$$\frac{\partial T_c}{\partial t} = -v_c \frac{\partial T_c}{\partial x} + \frac{\pi d_i u_i}{A_i \rho C}(T_h - T_c) \tag{3.4}$$

$$\frac{\partial T_h}{\partial t} = -v_H \frac{\partial T_h}{\partial x} - \frac{\pi d_0 u_0}{A_a \rho C}(T_h - T_c) \tag{3.5}$$

where A_a is area of the annulus (area between two pipes) and A_i is the area of the internal pipe.

The steady state equations for both hot and cold streams are as follows:

$$\frac{\partial Tc}{\partial x} = A^{'}(Th - Tc) \tag{3.6}$$

and

$$\frac{\partial Th}{\partial x} = -B^{'}(Th - Tc) \tag{3.7}$$

Where $A^{'}$ & $B^{'}$ are given by the expressions:

$$A^{'} = \frac{\pi d_i u_i}{v_c A_i \rho C} \quad , \qquad B^{'} = \frac{\pi d_0 u_0}{v_H A_a \rho C}$$

The solution of equations (3.6) and (3.7) is obtained by MAPLE 10 by the aid of the following boundary conditions.

$$Th(x=0) = Th_i \quad , \quad Tc(x=L) = Tc_i$$

$$Tc(x) = \frac{A^{'} Th_i - Tc_i B^{'} e^{-\left(A^{'} - B^{'}\right)L} - B^{'}\left(Th_i - Tc_i\right) e^{-\left(A^{'} - B^{'}\right)x}}{A^{'} - B^{'} e^{-\left(A^{'} - B^{'}\right)L}} \tag{3.8}$$

and

$$Th(x)=\frac{A^{'}Th_i-Tc_iB^{'}e^{-(B^{'}-A^{'})L}+A^{'}(Th_i-Tc_i)e^{-(B^{'}-A^{'})x}}{A^{'}-B^{'}e^{-(B^{'}-A^{'})L}} \tag{3.9}$$

After steady state equations are achieved, the solution of the unsteady state equations is needed, since equations (3.4) and (3.5) cannot be solved analytically an approximation (assumption 5) must be made.

3.2.2 Transfer Equation of the Effect of Hot Liquid Velocity on The Cold Liquid Temperature

The heat balance for the hot stream is given by the following equation after making the approximation,

$$L\frac{dTh}{dt}=-\nu_H(Th-Th_i)-AL(Th-Tc) \tag{3.10}$$

Similarly the cold stream heat balance is given by:

$$L\frac{dTc}{dt}=-\nu_C(Tc-Tc_i)+BL(Th-Tc) \tag{3.11}$$

Arrangement of equations (3.10) and (3.11) gives

$$L\frac{dTh}{dt}+\nu_H Th+AL(Th-Tc)=\nu_H Th_i \tag{3.12}$$

and

$$L\frac{dTc}{dt}+\nu_C Tc-BL(Th-Tc)=\nu_C Tc_i \tag{3.13}$$

The steady state form of equations (3.12) and (3.13) are given by the following equations:

$$0+\nu_H{}^s Th^s+AL(Th^s-Tc^s)=\nu_H{}^s Th_i \tag{3.14}$$

$$0+\nu_c{}^s Tc^s-BL(Th^s-Tc^s)=\nu_c{}^s Tc_i \tag{3.15}$$

Linearization of the term (ν_HTh) in equation (3.12) is according to the following formula:

$$z(x,y)=z^s+\left.\frac{dz}{dy}\right|_{x^s,y^s}(y-y^s)+\left.\frac{dz}{dx}\right|_{x^s,y^s}(x-x^s) \tag{3.16}$$

Therefore the linearized term will be as follows:

$$\nu_H Th=\nu_H{}^s Th^s+Th^s(V_H-\nu_H{}^s)+V_H{}^s(Th-Th^s) \tag{3.17}$$

The deviation variables will be defined as $\overline{Th}=Th-Th^s$, and $\overline{\nu_H}=\nu_H-\nu_H{}^s$.

Substitution of equation (3.17) and the deviation variables into equation (3.12) leads to the following:

$$L\frac{dTh}{dt}+\nu_H{}^s Th^s+Th^s\left(\overline{\nu_H}\right)+\nu_H{}^s\left(\overline{Th}\right)+AL\left(Th-Tc\right)=\nu_H Th_i \qquad (3.18)$$

Subtracting the steady state equation (3.14) from equation (3.18) leading to,

$$L\frac{d\overline{Th}}{dt}+Th^s\left(\overline{\nu_H}\right)+V_H{}^s\left(\overline{Th}\right)+AL\left(\overline{Th}-\overline{Tc}\right)=\overline{\nu_H}Th_i \qquad (3.19)$$

Taking the Laplace transform of equation (3.19),

$$sL\overline{Th}(s)+Th^s\left(\overline{\nu_H}(s)\right)+V_H{}^s\left(\overline{Th}(s)\right)+AL\left(\overline{Th}(s)-\overline{Tc}(s)\right)=\overline{\nu_H}(s)Th_i \qquad (3.20)$$

Putting equation (3.5) in deviation variables form,

$$L\frac{d\overline{Tc}}{dt}+\nu_C\overline{Tc}-BL\left(\overline{Th}-\overline{Tc}\right)=0 \qquad (3.21)$$

Taking the Laplace transform of equation (3.21),

$$sL\overline{Tc}(s)+\nu_C\overline{Tc}(s)-BL\left(\overline{Th}(s)-\overline{Tc}(s)\right)=0 \qquad (3.22)$$

Arranging equation (3.22) gives the form,

$$\overline{Tc}(s)[sL+\nu_C+BL]=BL\overline{Th}(s) \tag{3.23}$$

Substitution of equation (3.23) into equation (3.20),

$$\overline{Tc}(s)[sL+\nu_C+BL]=\frac{\overline{\nu_H}(s)B\left[Th_i-Th_{exit}^s\right]+ABL\overline{Tc}(s)}{\left(sL+\nu_H{}^s+AL\right)} \tag{3.24}$$

$$\overline{Tc}(s)=\frac{B\left[Th_i-Th_{exit}^s\right]}{\left(s^2L^2+s\left(\nu_C+BL+\nu_H{}^s+AL\right)\right)}\overline{\nu_H}(s) \tag{3.25}$$

$$G(s)=\frac{\overline{Tc}(s)}{\overline{\nu_H}(s)}=\frac{B\left[Th_i-Th_{exit}^s\right]}{\left(s^2L^2+s\left(\nu_C+BL+\nu_H{}^s+AL\right)\right)} \tag{3.26}$$

$$G(s)=\frac{\overline{Tc}(s)}{\overline{\nu_H}(s)}=\frac{\alpha}{s(s+\gamma)} \tag{3.27}$$

Equation (3.27) represents the transfer function for the effect of hot liquid velocity on the cold liquid temperature.

Assuming that the system is subjected to unit step in cold liquid velocity λ such that

$$\overline{\nu_H}(s)=\frac{\lambda}{s} \tag{3.28}$$

The output cold liquid temperature will be as follows,

$$\overline{Tc}(s) = \frac{\alpha\lambda}{s^2(s+\beta)} \qquad (3.29)$$

Expansion by partial fractions method and inverted the resultant equation becomes

$$\overline{Tc}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right) \qquad (3.30)$$

where $\alpha = BL(Th_i - Th_L)$, $\beta = v_c + v_H + AL + BL$

3.2.3 Transfer Equation of The Effect of Cold Liquid Velocity on the Hot Liquid Temperature

Similar derivation to the above for heat balance,

$$L\frac{dTh}{dt} + \nu_H Th + AL(Th - Tc) = \nu_H Th_i \qquad (3.12)$$

and

$$L\frac{dTc}{dt} + \nu_C Tc - BL(Th - Tc) = \nu_C Tc_i \qquad (3.13)$$

The steady state form of equations (3.12) and (3.13) are given by the following equations:

$$0 + \nu_H{}^s Th^s + AL\left(Th^s - Tc^s\right) = \nu_H{}^s Th_i \qquad (3.14)$$

$$0 + \nu_c{}^s Tc^s - BL\left(Th^s - Tc^s\right) = \nu_c{}^s Tc_i \qquad (3.15)$$

Linearization of the term ($\nu_C Tc$) in equation (3.13) according to the following formula:

$$z(x, y) = z^s + \left.\frac{dz}{dy}\right|_{x^s, y^s}\left(y - y^s\right) + \left.\frac{dz}{dx}\right|_{x^s, y^s}\left(x - x^s\right) \qquad (3.16)$$

Therefore, the linearized term will be as follows:

$$\nu_C Tc = \nu_C{}^s Tc^s + Tc^s\left(\nu_C - \nu_C{}^s\right) + V_C{}^s\left(Tc - Tc^s\right) \qquad (3.31)$$

The deviation variables will be defined as $\overline{Tc}$=Tc-Tc^s, and $\overline{\nu_C} = \nu_C - \nu_C{}^s$.

Substitution of equation (3.31) and the deviation variables into equation (3.13) leads to the following:

$$L\frac{dTc}{dt} + \nu_C{}^s Tc^s + Tc^s\left(\overline{\nu_C}\right) + \nu_C{}^s\left(\overline{Tc}\right) - BL\left(Th - Tc\right) = \nu_C Tc_i \qquad (3.32)$$

Subtracting the steady equation (3.15) from equation (3.32) leading to,

$$L\frac{d\overline{Tc}}{dt} + Tc^s\left(\overline{\nu_C}\right) + V_C{}^s\left(\overline{Tc}\right) - BL\left(\overline{Th} - \overline{Tc}\right) = \overline{\nu_C} Tc_i \qquad (3.33)$$

Taking the Laplace transform of equation (3.12),

$$sL\overline{Tc}(s)+Tc^{s}\left(\overline{\nu_C}(s)\right)+V_C{}^{s}\left(\overline{Tc}(s)\right)-BL\left(\overline{Th}(s)-\overline{Tc}(s)\right)=\overline{\nu_C}(s)Tc_i \tag{3.34}$$

Putting equation (3.4) in deviation variables form,

$$L\frac{d\overline{Th}}{dt}+\nu_H\overline{Th}+AL\left(\overline{Th}-\overline{Tc}\right)=0 \tag{3.35}$$

Taking the Laplace transform of equation (3.35),

$$sL\overline{Th}(s)+\nu_H\overline{Th}(s)+AL\left(\overline{Th}(s)-\overline{Tc}(s)\right)=0 \tag{3.36}$$

Arranging equation (3.36) gives the form,

$$\overline{Th}(s)\left[sL+\nu_H+AL\right]=AL\overline{Tc}(s) \tag{3.37}$$

Substituting equation (3.37) into equation (3.34) and arranging gives,

$$G(s)=\frac{\overline{Th}(s)}{\overline{\nu_C}(s)}=\frac{AL\left[Tc_i-Tc^{s}_{exit}\right]}{\left(s^2L^2+s\left(\nu_C+BL+\nu_H{}^{s}+BL\right)\right)} \tag{3.38}$$

$$G(s)=\frac{\overline{Th}(s)}{\overline{\nu_C}(s)}=\frac{\alpha}{s(s+\gamma)} \tag{3.39}$$

Equation (3.39) represents the transfer function for the effect of cold liquid velocity on the hot liquid temperature.

Assuming that the system is subjected to unit step in cold liquid velocity λ such that,

$$\overline{v_C}(s) = \frac{\lambda}{s} \tag{3.40}$$

The output hot liquid temperature will be as follows,

$$\overline{Th}(s) = \frac{\alpha\lambda}{s^2(s+\beta)} \tag{3.41}$$

Expansion by partial fractions method and inverted, the resultant equation becomes

$$\overline{Th}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right) \tag{3.42}$$

where $\alpha = AL(Tc_i - Tc_L)$, $\beta = v_c + v_H + AL + BL$

3.3 Frequency Response

The frequency response of a system may be found if the transfer function of that system is known [34], see chapter three for derivation of the transfer function

$$G(s) = \frac{\alpha}{s^2 + s\gamma} \tag{3.27}$$

Putting ωj instead of each s where ω is the frequency [33],

$$G(\omega j) = \frac{\alpha}{\omega^2 j^2 + \gamma \omega j} = \frac{\alpha}{-\omega^2 + \omega j \gamma} \tag{3.43}$$

Multiplying by the conjugate to get,

$$G(\omega j) = \frac{-\alpha\left(\omega^2 + \omega j \gamma\right)}{\omega^4 + \gamma^2 \omega^2} \tag{3.44}$$

or

$$G(\omega j) = \frac{-\alpha \omega^2}{\omega^2\left(\omega^2 + \gamma^2\right)} - \frac{\alpha \gamma \omega}{\omega^2\left(\omega^2 + \gamma^2\right)} j \tag{3.45}$$

The amplitude ratio (AR) is defined as follows,

$$\text{AR} = \sqrt{(\text{real part})^2 + (\text{imaginary part})^2} \tag{3.46}$$

Therefore the amplitude ratio is as follows,

$$AR = \frac{\alpha}{\omega} \frac{1}{\sqrt{\omega^2 + \gamma^2}} \tag{3.47}$$

The phase angle (Φ) is given by the following expression,

$$\phi = \tan^{-1} - \frac{\text{real part}}{\text{imaginary part}} \tag{3.48}$$

$$\phi = \tan^{-1}\left(-\frac{\gamma}{\omega}\right) \tag{3.49}$$

3.4 Modeling of Physical Properties

The simulation program was made using Microsoft Excel and in order to make the program flexible, the physical properties (density, viscosity, heat capacity, thermal conductivity, and Prandtl number) are fitted as a function of temperature using the data given by Holman [32], see Table 3.1.

Table 3.1 Properties of water used for curve fitting

Temperature, ^oC	Cp kJ / kg . ^oC	ρ kg / m^3	$\mu \times 10^{-4}$ kg / m . s	k W / m . ^oC	Pr
21.11	4.179	997.4	9.8	0.604	6.78
26.67	4.179	995.8	8.6	0.614	5.85
32.22	4.174	994.9	7.65	0.623	5.12
37.78	4.174	993.0	6.82	0.630	4.53
43.33	4.174	990.6	6.16	0.637	4.04
48.89	4.174	988.8	5.62	0.644	3.64
54.44	4.179	985.7	5.13	0.649	3.30
60	4.179	983.3	4.71	0.654	3.01
65.55	4.183	980.3	4.3	0.659	2.73
71.11	4.186	977.3	4.01	0.665	2.53
76.67	4.191	973.7	3.72	0.668	2.33
82.22	4.195	970.2	3.47	0.673	2.16
87.78	4.199	966.7	3.27	0.675	2.03
93.33	4.204	963.2	3.06	0.678	1.90

The data in Table 3.1 are fitted as follows,

3.4.1 Liquid Density

The best fit for the effect of temperature on the density is given by the following equation,

$$\rho = 990.032 - 0.026\,T - 0.00428\,T^2 - 2.3\times10^{-6}T^3 \qquad (3.50)$$

where ρ is the density in kg / m^3 and correlation coefficient equals to 0.9942.

3.4.2 Liquid Viscosity

The best fit for the effect of temperature on the viscosity is given by the following equation,

$$\mu = 1.67\times10^{-3} - 4.43\times10^{-5}\,T - 6.41\times10^{-7}\,T^2 - 4.8\times10^{-9}T^3 \qquad (3.51)$$

where μ is the viscosity in kg / m . s and correlation coefficient equals to 0.9732.

3.4.3 Liquid Heat Capacity

The best fit for the effect of temperature on the heat capacity is given by the following equation,

$$Cp = 4212.36 - 2.3597\,T + 0.04683\,T^2 - 3.39\times10^{-4}T^3 \qquad (3.52)$$

where Cp is the heat capacity in J / kg . ^{o}C and correlation coefficient equals to 0.9844.

3.4.4 Liquid Thermal Conductivity

The best fit for the effect of temperature on the thermal conductivity is given by the following equation,

$$k = 0.5608 + 0.002471\,T - 0.0000205\,T^2 + 0 \qquad (3.53)$$

where k is the thermal conductivity in W / m . ^oC and correlation coefficient equals to 0.9918.

3.4.5 Prandtl Number

The best fit for the effect of temperature on the Prandtl number is given by the following equation,

$$\mathrm{Pr} = 12.267 - 0.3574\,\mathrm{T} + 0.00544\,\mathrm{T}^2 - 4.1962 \times 10^{-5}\mathrm{T}^3 \qquad (3.54)$$

where Pr is Prandtl number and correlation coefficient equals to 0.9953.

Chapter Four
Experimental Work

4.1 Introduction

This chapter explains the details of the experimental equipment that is used in this study.

4.2 Description of The Experimental Equipment

The general layout of the experimental set-up used in the present work and the block diagram are shown in Fig. 4.1, and its components are listed in Table 4.1. The main items of the figure are discussed in the following section.

4.2.1 The Heat Exchanger

The main part of the experimental rig was an insulated (glass wool) double-pipe heat exchanger containing two concentric pipes assembled in counter –current configuration, single pass. The maximum working temperature is $70^{o}C$.

The specifications of the double-pipe heat exchanger are given in Table (4.2).

Water was employed as circulating fluid due to its availability in addition to its high heat capacity making it the universal cooling medium.

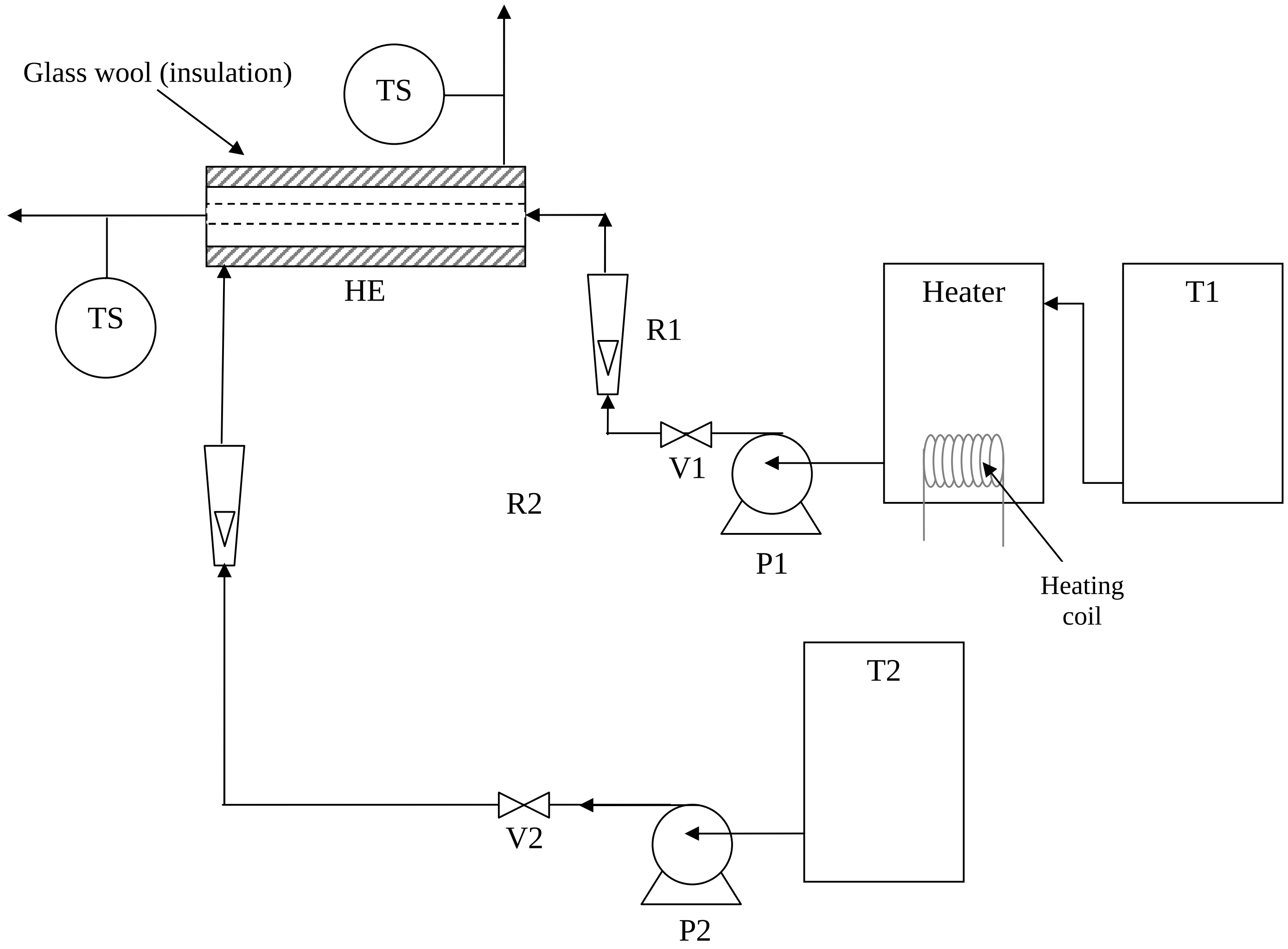


Fig. 4.1 Block diagram of the double pipe heat exchanger

Table 4.1 Description of the experimental rig

Code	Component	Description
HE	Heat exchanger	Double pipe heat exchanger of two concentric pipes
P1	Cold water pump	Centrifugal pump, 3-phase, 210 W, Struat Turner Ltd., 210 W, 200- 2000 L / min
P2	Hot water pump	½ inch, 200 W, 1.6 m^3 / h, Centrifugal pump
R1	Hot water rotameter	“GEC Elliot” rotameter (variable area) type, series 2000, range 300-1100 L / h
R2	Cold water rotameter	“GEC Elliot” rotameter (variable area) type, series 2000, range 300-1100 L / h
T1	Hot water tank	Rectangular tank 60 x 100 x 50 cm
T2	Cold water tank	Galvanized cubic tank, capacity 0.5 m^3
V1	Cold water valve	¾ inch global valve
V2	Hot water valve	¾ inch global valve
TS	Temperature sensor	“Honeywell” resistance bulb temperature transmitter model y - 785309

Table 4.2 Double pipe heat exchanger specifications

Outer diameter	0.0375 m
Inner diameter	0.0127 m
Pipe lengths	1 m

Fig. 4.2 The double- pipe heat exchanger

Fig. 4.3 The experimental rig

4.2.2 Sump Tank

The sump tank supplied by the manufacturer has a capacity of 0.07 m^3. The tank design was altered to increase the capacity to almost double in order to minimize the variation in the inlet temperature of the hot water.

The sump tank outlet is kept as far away as possible from its inlet to avoid short circuits in the flow. The sump tank and the pipes, which carry the hot water, are insulated with glass wool.

4.3 Range of Parameters

The following parameters are studied in the present work:

1. Hot water flow rate range from 300 to 1100 L / h.
2. Cold water flow rate range of 300 to 1100 L / h.
3. Hot water inlet temperature 70 ^oC.
4. Cold water inlet temperature 30 ^oC.

4.4 Experimental Procedure

The following procedure is used in the present work to take the measurements of the results, which is as follows:

1. Switching on the heaters and waiting about 30 min until the desired temperature is reached.
2. Setting the cold-water flow rate at the desired flow rate using cold-water valve (V1).
3. Opening the hot water valve (V2) after setting the desired flow rate.
4. Waiting until steady state is reached by noticing the outlet cold water temperature until a constant value is reached where steady state is reached.

5. After steady state is reached, a step change in hot water flow rate is introduced.
6. Recording the response of outlet cold-water temperature.
7. The effect of the cold water flow rate on the outlet hot water temperature is studied according to the following :
   - Waiting until steady state is reached by noticing the outlet hot water temperature until constant value is reached where steady state is reached.
   - After steady state is reached, a step change in cold-water flow rate is introduced.
   - Recording the response of outlet hot-water temperature.

Chapter Five
Results and Discussion

This chapter deals with the interpretation and discussion of the experimental and the comparison between the theoretical model results and the experimental results.

According to experimental procedure presented in chapter four, the steady state conditions must be evaluated first before studying the system dynamics, then the dynamics are studied by manipulating the flow rates of both hot and cold streams.

5.1 Effect of Hot Water Velocity on Cold Liquid Output Temperature

5.1.1 Steady State Conditions

For the following conditions and applying the procedure presented in chapter four, the steady state is as follows in table 5.1 for the effect of hot water flow rate on the cold liquid output temperature.

Table 5.1 Steady state temperature of hot and cold streams as function of hot and cold flow rates for hot and cold stream temperatures of 70 and 30 ^{o}C

Cold water flow, rate L / h	Hot water flow rate, L / h	Outlet cold temperature ^{o}C
300	300	33.25
300	500	33.36
300	700	35.21
300	900	37.38
300	1100	39.29
500	300	30.32
500	500	32.46
500	700	34.52
500	900	35.72
500	1100	37.31

5.1.2 Dynamic Behavior for Flow Rate for Cold Water 300 l/h and Hot Water 300 L/h

The conditions for cold-water inlet temperature of 30^{o}C and hot water inlet temperature of 70^{o}C steady state flow rates for hot and cold water are 300 L/h for each. A step change rates of 200, 400, 600, and 800 L / h for the flow rate of the hot stream are applied respectively and the response of the outlet temperature of cold water is shown in Table 5.2.

The data presented in Table 5.2 and Fig. 5.1 show that increasing hot stream flow rate leads to increase the temperature of the exit cold stream.

The reason behind this increase is the increase of the heat transfer coefficient of the hot stream caused by the high flow rate of the hot stream resulting in high heat transfer rate from the hot stream to the cold stream, therefore increasing the exit cold stream temperature. The temperature profile obtained seems to be of finite change such that for the flow rate of 500 L /h, the temperature changes from 32.25 to 33.36 ^{o}C and this is due to the relatively short length of the heat exchanger and because of the fact that the type of the heat exchanger is double pipe which results in a small heat transfer area with respect to other types of heat exchangers so that slight change in temperature is expected. And the temperature reaches a value at which it maintains a constant value; this is because of the new steady state condition achieved after changing the flow rate.

Table 5.2 Outlet temperature of cold water at different flow rates of hot water for steady state flow rate for cold and hot water of 300 L/h for each obtained from experimental work

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L /h
0	30	30	30	30
2	32.37	32.33	32.41	32.53
4	32.45	32.45	32.65	32.94
6	32.71	32.76	32.93	33.36
8	32.93	32.87	33.33	33.72
10	33.13	33.11	33.61	33.97
12	33.27	33.28	33.92	34.31
14	33.36	33.31	34.32	34.69
16	33.36	33.57	34.57	35.02
18	33.36	33.81	34.81	35.41
20	33.36	34.21	35.17	35.86
22	33.36	34.43	35.43	36.17
24	33.36	34.65	35.52	36.55
26	33.36	34.86	35.74	36.89
28	33.36	35.21	35.97	37.24
30	33.36	35.21	36.29	37.81
32	33.36	35.21	36.61	38.1
34	33.36	35.21	36.93	38.82
36	33.36	35.21	37.38	39.29
38	33.36	35.21	37.38	39.29
40	33.36	35.21	37.38	39.29

Table 5.3 Outlet temperature of cold water at different flow rates of hot water for steady state flow rate for cold and hot water of 300 L/h for each obtained from mathematical model

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L /h
0	30	30	30	30
2	30.074	30.15	30.225	30.299
4	30.198	30.4	30.6	30.8
6	30.328	30.665	30.997	31.328
8	30.46	30.932	31.397	31.861
10	30.592	31.199	31.797	32.395
12	30.724	31.466	32.198	32.928
14	30.856	31.733	32.598	33.462
16	30.988	32	32.999	33.995
18	31.12	32.268	33.399	34.529
20	31.251	32.535	33.8	35.063
2	31.383	32.802	34.2	35.596
24	31.515	33.069	34.601	36.13
26	31.647	33.336	35.001	36.663
28	31.779	33.603	35.402	37.197
30	31.911	33.87	35.802	37.73
32	32.043	34.138	36.203	38.264
34	32.175	34.405	36.603	38.798
36	32.307	34.672	37.004	39.331
38	32.438	34.939	37.404	39.865
40	32.57	35.206	37.805	40.398

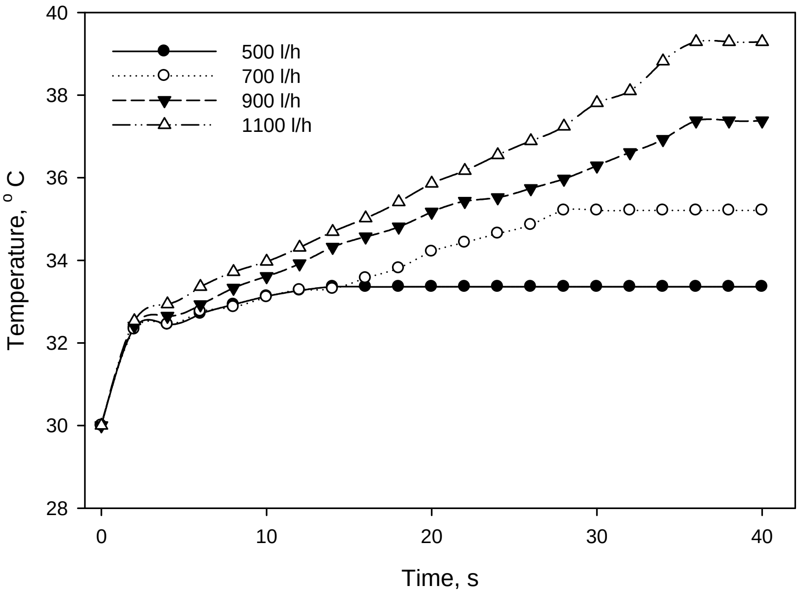


Fig. 5.1 Outlet cold water temperature response for disturbance in hot water flow rate for steady state flow rates of 300 L/h for cold water and 300 L / h for hot water

5.1.3 Comparison Between the results of Experimental Work and Mathematical Model

Comparison between the results obtained from the experimental work and those obtained from the mathematical model are presented in Figs. 5.2 and 5.3. The comparison is made for two flow rate values because of the very close values for the temperature profiles.

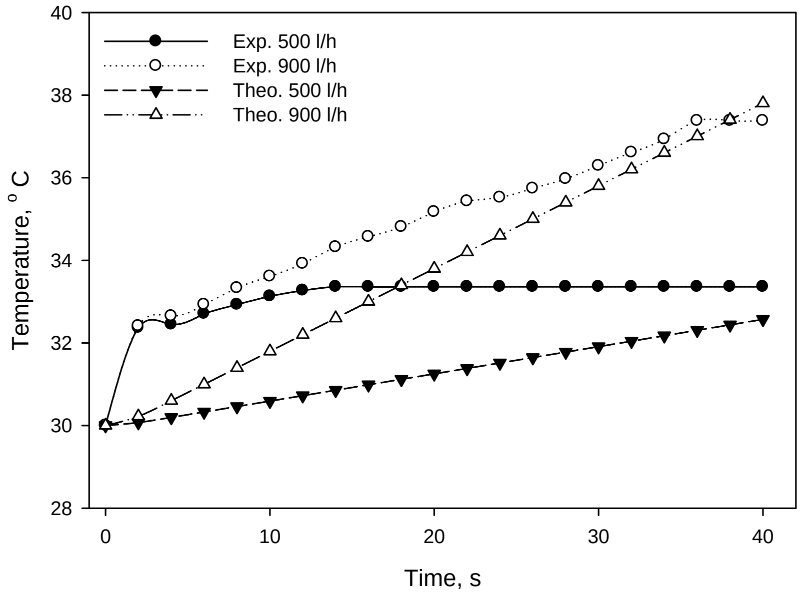


Fig. 5.2 Theoretical and experimental results for 500 and 900 L/h

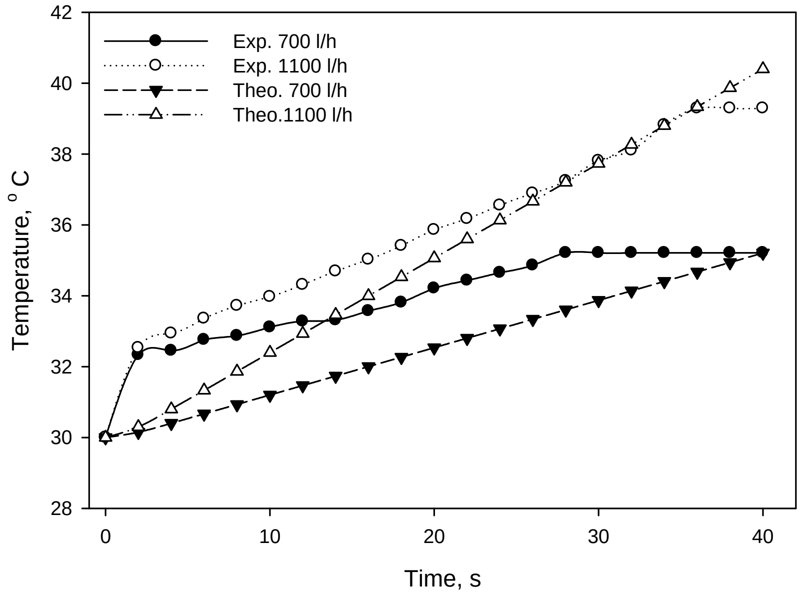


Fig. 5.3 Theoretical and experimental results for 700 and 1100 L / h

Examining Figures 5.2 and 5.3 leads to the fact that the temperatures obtained from the mathematical model for the cold stream is not quite adequate (small error) and the response is exhibiting an increasing manner similar to the response to ramp forcing function and this is due to the approximation made to solve the differential equations of the system. But in general, the results obtained from the mathematical model are satisfactory.

5.1.4 Frequency Response

Table 5.4 Frequency response for the conditions of table 5.3

w	AR			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-0.871	-0.92	-0.967	-1.013
5	-2.65	-2.654	-2.659	-2.664
9	-3.402	-3.404	-3.405	-3.407
13	-3.878	-3.879	-3.88	-3.88
17	-4.226	-4.227	-4.227	-4.228
21	-4.501	-4.501	-4.502	-4.502
25	-4.728	-4.728	-4.728	-4.729
29	-4.921	-4.921	-4.921	-4.922
33	-5.089	-5.089	-5.09	-5.09
37	-5.238	-5.238	-5.239	-5.239
41	-5.372	-5.372	-5.372	-5.372
45	-5.493	-5.493	-5.493	-5.493
49	-5.604	-5.604	-5.604	-5.604
53	-5.706	-5.706	-5.706	-5.706
57	-5.801	-5.801	-5.801	-5.801
61	-5.889	-5.89	-5.89	-5.89
65	-5.972	-5.972	-5.972	-5.972
69	-6.05	-6.05	-6.05	-6.05

Table 5.5 Phase angle for the conditions of table 5.3

w	Φ			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-0.826	-0.875	-0.918	-0.957
5	-0.214	-0.235	-0.256	-0.277
9	-0.12	-0.132	-0.144	-0.156
13	-0.083	-0.092	-0.1	-0.109
17	-0.064	-0.07	-0.077	-0.083
21	-0.052	-0.057	-0.062	-0.068
25	-0.043	-0.048	-0.052	-0.057
29	-0.037	-0.041	-0.045	-0.049
33	-0.033	-0.036	-0.04	-0.043
37	-0.029	-0.032	-0.035	-0.038
41	-0.026	-0.029	-0.032	-0.035
45	-0.024	-0.027	-0.029	-0.032
49	-0.022	-0.024	-0.027	-0.029
53	-0.02	-0.023	-0.025	-0.027
57	-0.019	-0.021	-0.023	-0.025
61	-0.018	-0.02	-0.021	-0.023
65	-0.017	-0.018	-0.02	-0.022
69	-0.016	-0.017	-0.019	-0.021

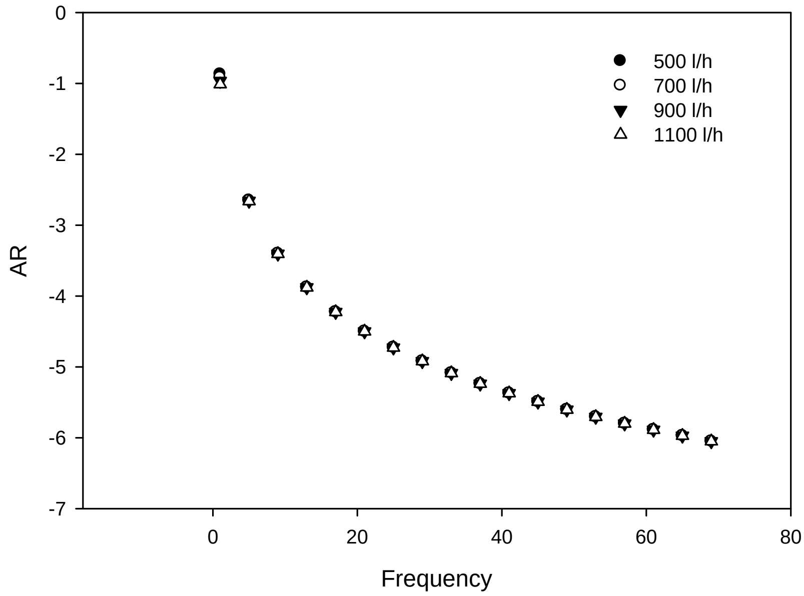


Fig. 5.4 Frequency response for the conditions of table 5.4

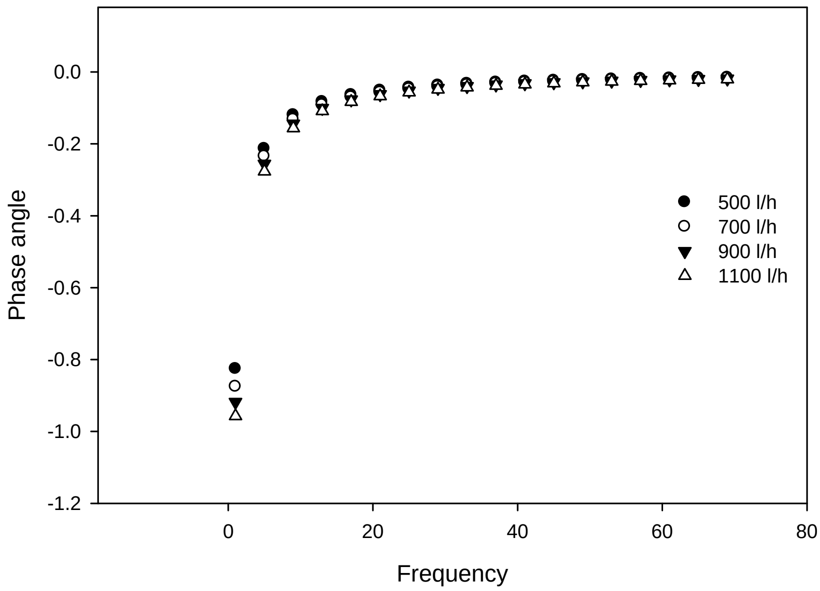


Fig. 5.5 Phase angle for the conditions of table 5.5

5.1.5 Dynamic Behavior for Flow Rate of 500 L/h for Cold Water and of 300 L/h for Hot Water

The conditions for cold-water inlet temperature of 30^{o}C and hot water inlet temperature is of 70^{o}C steady state flow rates for hot water is 300 and for cold water is 500 L/ h. A step change rates of 200, 400, 600, and 800 L / h for the flow rate of the hot stream are applied respectively and the response of the outlet temperature of the cold water is shown in Table (5.6).

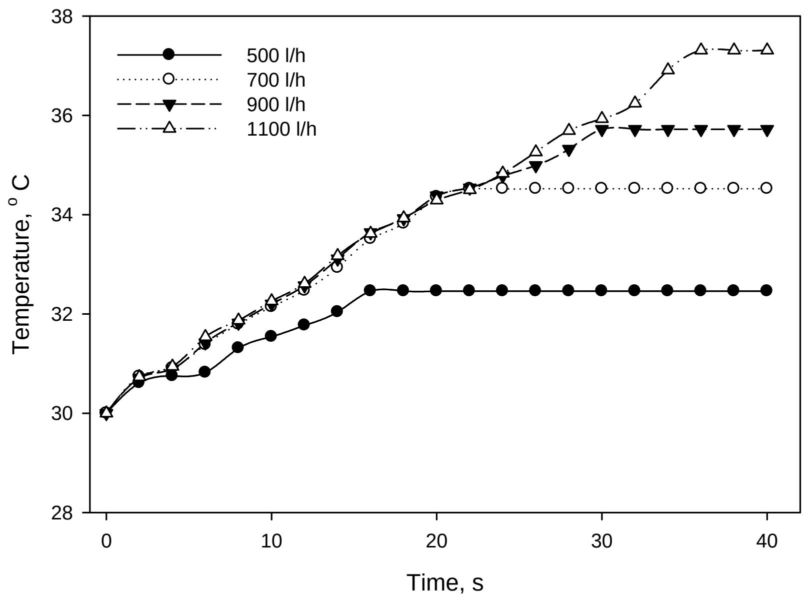


Fig. 5.6 Outlet cold water temperature response for disturbance in hot water flow rate for steady state flow rates of 300 L/h for hot water and 500 L / h for cold water

Table 5.6 Outlet temperature of cold water at different flow rates of hot water for steady state flow rate of 500 L/h for cold water and of 300 L/h for hot water obtained from experimental work

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L /h
0	30	30	30	30
2	30.61	30.74	30.71	30.73
4	30.75	30.91	30.89	30.94
6	30.82	31.38	31.42	31.54
8	31.31	31.79	31.82	31.87
10	31.54	32.14	32.2	32.26
12	31.77	32.47	32.57	32.61
14	32.04	32.93	33.11	33.17
16	32.46	33.51	33.64	33.62
18	32.46	33.82	33.92	33.93
20	32.46	34.36	34.38	34.29
22	32.46	34.52	34.54	34.5
24	32.46	34.52	34.78	34.83
26	32.46	34.52	34.99	35.26
28	32.46	34.52	35.32	35.69
30	32.46	34.52	35.72	35.93
32	32.46	34.52	35.72	36.24
34	32.46	34.52	35.72	36.91
36	32.46	34.52	35.72	37.31
38	32.46	34.52	35.72	37.31
40	32.46	34.52	35.72	37.31

Table 5.7 Outlet temperature of cold water at different flow rates of hot water for steady state flow rate of 500 L/h for cold water and of 300 L/h for hot water obtained from the mathematical model

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L /h
0	30	30	30	30
2	30.061	30.124	30.186	30.247
4	30.151	30.306	30.459	30.612
6	30.243	30.492	30.738	30.984
8	30.335	30.679	31.018	31.356
10	30.427	30.865	31.297	31.728
12	30.519	31.051	31.576	32.1
14	30.611	31.238	31.856	32.472
16	30.703	31.424	32.135	32.844
18	30.795	31.61	32.414	33.216
20	30.887	31.797	32.693	33.589
22	30.979	31.983	32.973	33.961
24	31.071	32.169	33.252	34.333
26	31.163	32.356	33.531	34.705
28	31.255	32.542	33.811	35.077
30	31.347	32.728	34.09	35.449
32	31.439	32.915	34.369	35.821
34	31.531	33.101	34.649	36.194
36	31.623	33.287	34.928	36.566
38	31.715	33.474	35.207	36.938
40	31.807	33.66	35.487	37.31

5.1.6 Comparison Between Results of Experimental Work and Mathematical Model

The data presented in table 5.6 and Fig. 5.6 show that increasing hot stream flow rate leads to increase the temperature of the exit cold stream, but differing from the previous condition for steady state flow rate of cold stream of 300 L /h in the values of the temperature. The reason behind this change is that the lowest residence time for the hot stream and cold stream which cause much lower time of heat transfer between hot and cold stream.

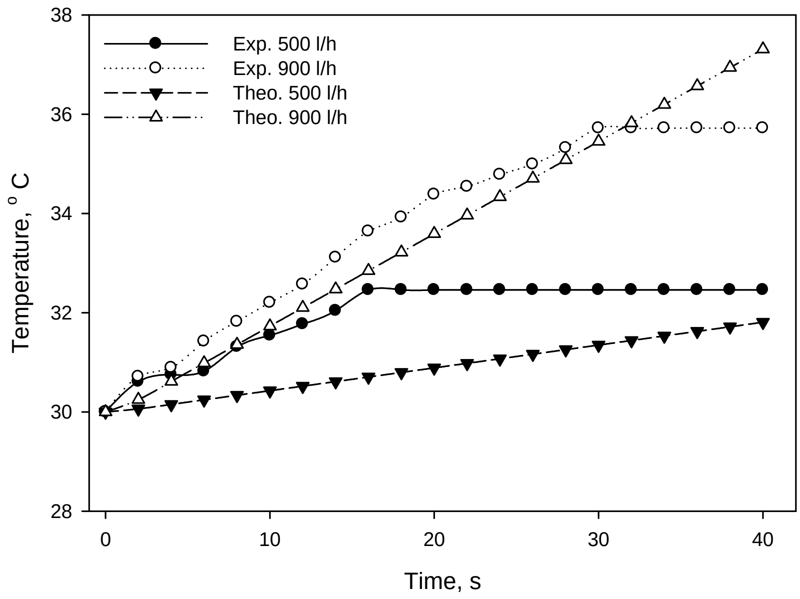


Fig. 5.7 Theoretical and experimental results for 500 and 900 l/h

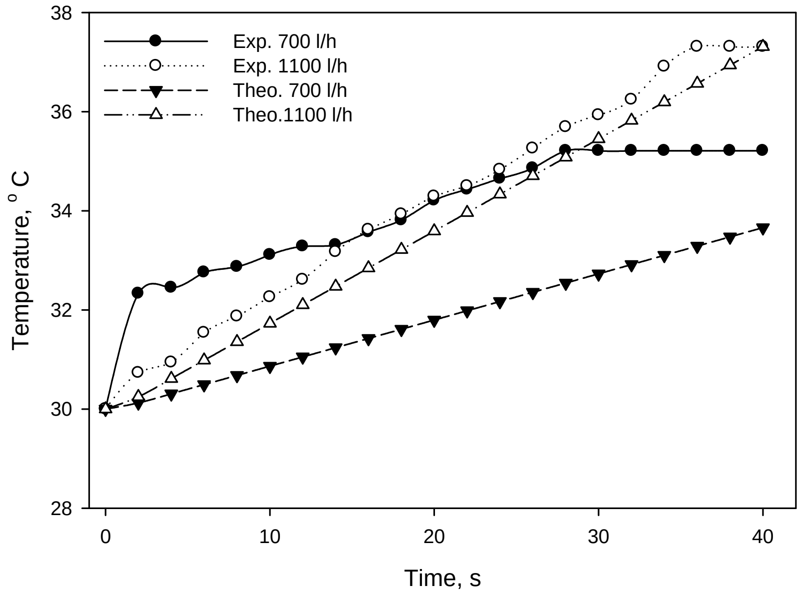


Fig. 5.8 Theoretical and experimental results for 700 and 1100 L /h

Examining Figures 5.7 and 5.8 leads to the same interpretation from the previous conditions with the same reason that the temperatures obtained from the mathematical model for the cold stream are not quite adequate (small error) and the response is exhibiting an increasing manner similar to the response to ramp forcing function, and this is due to the approximation made to solve the differential equations of the system.

5.1.7 Frequency Response

Table 5.8 Frequency response for the conditions of table 5.7

w	AR			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-1.052	-1.097	-1.139	-1.18
5	-2.669	-2.674	-2.68	-2.686
9	-3.408	-3.41	-3.412	-3.414
13	-3.881	-3.882	-3.883	-3.884
17	-4.228	-4.229	-4.229	-4.23
21	-4.502	-4.503	-4.503	-4.503
25	-4.729	-4.729	-4.729	-4.73
29	-4.922	-4.922	-4.922	-4.922
33	-5.09	-5.09	-5.09	-5.09
37	-5.239	-5.239	-5.239	-5.239
41	-5.372	-5.372	-5.372	-5.373
45	-5.493	-5.494	-5.494	-5.494
49	-5.604	-5.604	-5.604	-5.605
53	-5.707	-5.707	-5.707	-5.707
57	-5.801	-5.801	-5.801	-5.801
61	-5.89	-5.89	-5.89	-5.89
65	-5.972	-5.972	-5.972	-5.972
69	-6.05	-6.05	-6.05	-6.05

Table 5.9 Phase angle for the conditions of table 5.7

w	Φ			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-0.988	-1.021	-1.05	-1.076
5	-0.295	-0.315	-0.335	-0.355
9	-0.167	-0.179	-0.191	-0.203
13	-0.116	-0.125	-0.133	-0.142
17	-0.089	-0.096	-0.102	-0.109
21	-0.072	-0.077	-0.083	-0.088
25	-0.061	-0.065	-0.07	-0.074
29	-0.052	-0.056	-0.06	-0.064
33	-0.046	-0.049	-0.053	-0.056
37	-0.041	-0.044	-0.047	-0.05
41	-0.037	-0.04	-0.042	-0.045
45	-0.034	-0.036	-0.039	-0.041
49	-0.031	-0.033	-0.036	-0.038
53	-0.029	-0.031	-0.033	-0.035
57	-0.027	-0.029	-0.031	-0.032
61	-0.025	-0.027	-0.029	-0.03
65	-0.023	-0.025	-0.027	-0.028
69	-0.022	-0.024	-0.025	-0.027

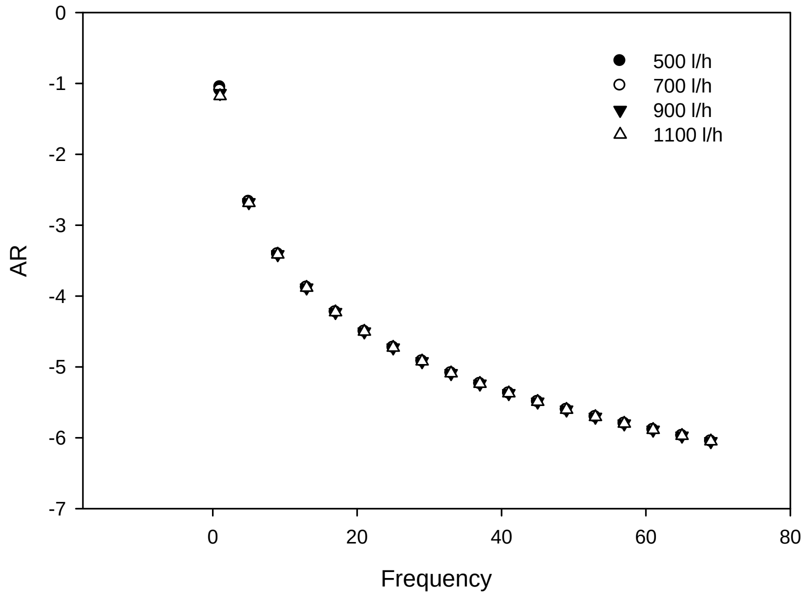


Fig. 5.9 Frequency response for parameters corresponding to table 5.8

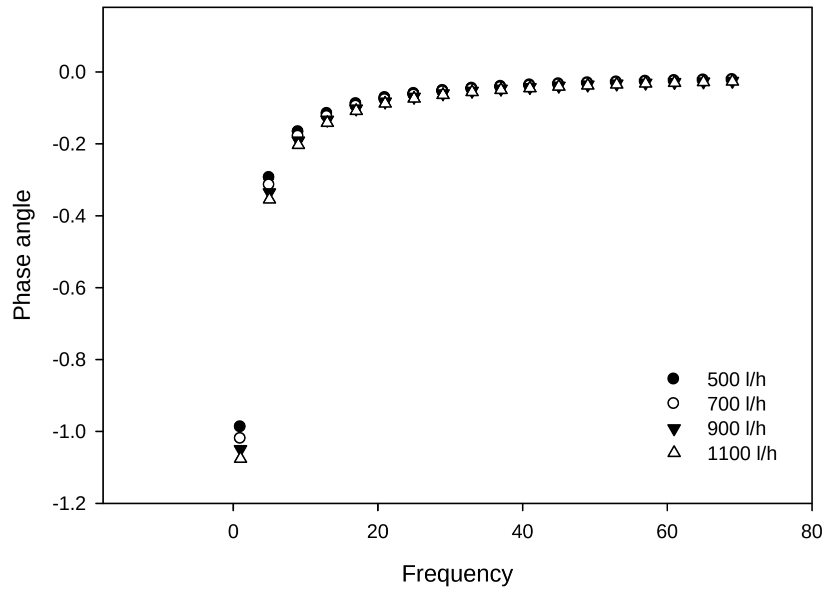


Fig. 5.10 Phase angle for the conditions of table 5.9

5.2 Effect of Cold Water Velocity on Hot Liquid Output Temperature

The conditions for cold-water inlet temperature of 30^oC and hot water inlet temperature of 70^oC steady state flow rates for hot and cold water are 300 L/ h for each. A step change rates of 200, 400, 600, and 800 L / h are applied respectively, and the response of the outlet temperature of hot water is shown in Table (5.10).

5.2.1 Steady State Conditions

Table 5.10 Steady state values of exit hot water temperature

Cold water flow rate L / h	Hot water flow rate L / h	Outlet hot temperature
300	300	69.51
500	300	68.60
700	300	67.89
900	300	66.73
1100	300	64.84
300	500	69.50
500	500	66.71
700	500	65.66
900	500	63.98
1100	500	62.88

5.2.2 Dynamic Behavior for Flow Rate of 300 L/h for Cold Water and 300 L/h for Hot Water

The conditions for cold-water inlet temperature of 30^{o}C and hot water inlet temperature of 70^{o}C steady state flow rates for hot and cold water are 300 L/ h for each. A step flow rates of 500, 700, 900, and 1100 L / h for the flow rate of the cold stream are applied respectively, and the response of the outlet temperature of hot water is shown in Table (5.11).

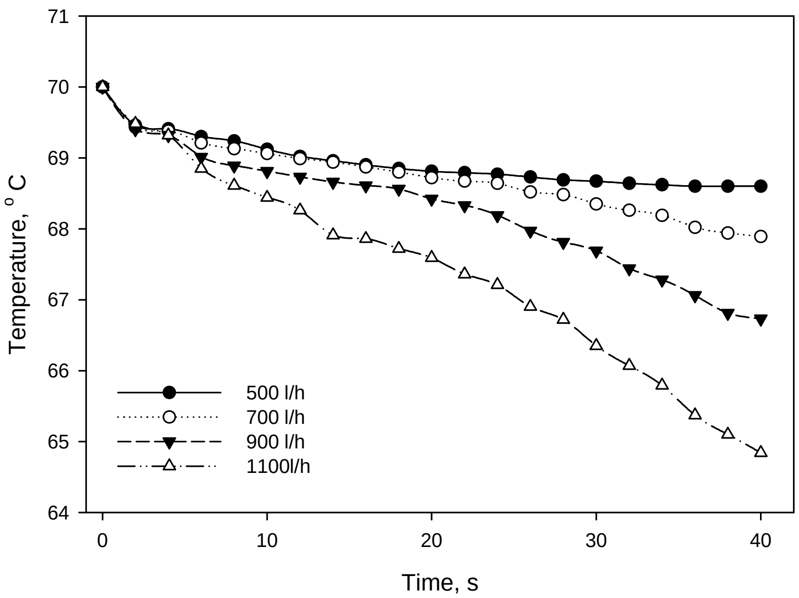


Fig. 5.11 Exit hot water temperature response for disturbance in cold water flow rate for steady state flow rates of 300 L/h for cold water and 300 L / h for hot water

Table 5.11 Outlet temperature of hot water at different flow rates of cold water for steady state flow rate of cold water of 300 L/h and hot water of 300 L/h obtained from experimental work

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L/h
0	70	70	70	70
2	69.46	69.43	69.4	69.48
4	69.41	69.38	69.32	69.32
6	69.3	69.21	69.01	68.85
8	69.24	69.13	68.89	68.61
10	69.12	69.06	68.81	68.44
12	69.02	68.99	68.73	68.26
14	68.96	68.94	68.66	67.91
16	68.9	68.87	68.61	67.86
18	68.85	68.8	68.56	67.72
20	68.81	68.72	68.42	67.59
22	68.79	68.67	68.33	67.36
24	68.77	68.64	68.19	67.21
26	68.73	68.52	67.97	66.9
28	68.69	68.48	67.81	66.72
30	68.67	68.35	67.69	66.35
32	68.64	68.26	67.44	66.07
34	68.62	68.19	67.28	65.79
36	68.6	68.02	67.06	65.37
38	68.6	67.94	66.81	65.1
40	68.6	67.89	66.73	64.84

Table 5.12 Outlet temperature of hot water at different flow rates of cold water for steady state flow rate of cold water of 300 L/h and hot water of 300 L/h obtained from mathematical model

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L/h
0	70	70	70	70
2	69.927	69.851	69.777	69.703
4	69.804	69.603	69.404	69.206
6	69.674	69.34	69.01	68.681
8	69.543	69.075	68.613	68.152
10	69.412	68.81	68.216	67.622
12	69.281	68.544	67.818	67.093
14	69.15	68.279	67.42	66.563
16	69.019	68.014	67.023	66.033
18	68.889	67.749	66.625	65.504
20	68.758	67.484	66.228	64.974
22	68.627	67.218	65.83	64.444
24	68.496	66.953	65.432	63.914
26	68.365	66.688	65.035	63.385
28	68.234	66.423	64.637	62.855
30	68.103	66.157	64.24	62.325
32	67.972	65.892	63.842	61.795
34	67.841	65.627	63.444	61.266
36	67.71	65.362	63.047	60.736
38	67.579	65.097	62.649	60.206
40	67.448	64.831	62.252	59.677

5.2.3 Comparison between Results of Experimental Work and Mathematical Model

Comparison between the results obtained from the experimental work and those obtained from mathematical model are presented in Figs. 5.12 and 5.13. The comparison is made for two flow rate values because of the very close values for the temperature profiles.

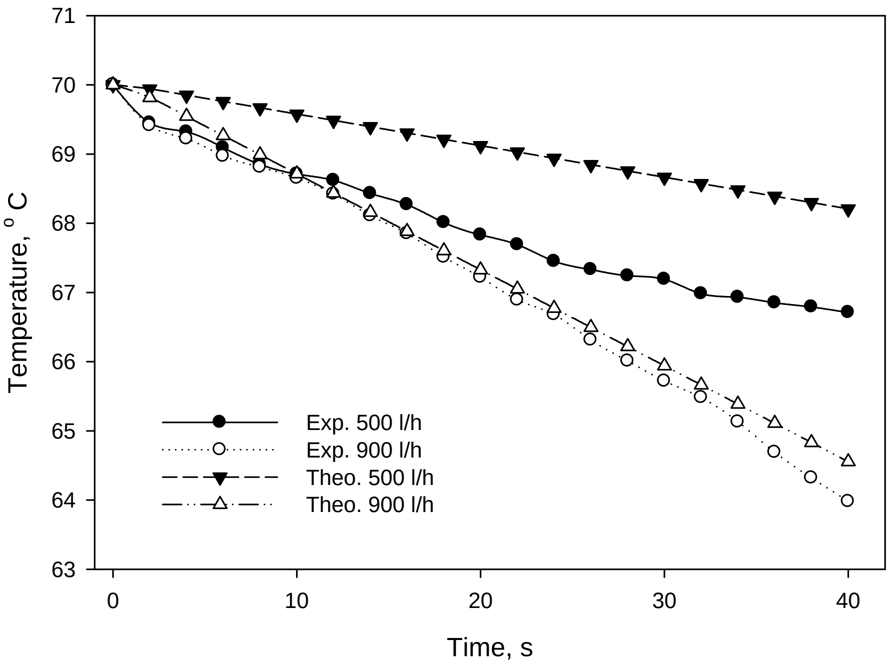


Fig. 5.12 Theoretical and experimental results for 500, 900 L/h

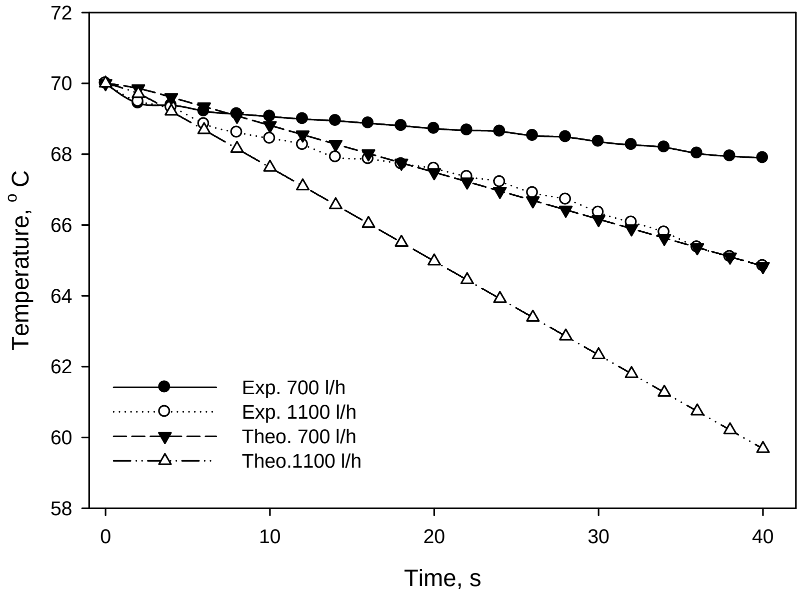


Fig. 5.13 Theoretical and experimental results for 700, 1100 L / h

The data of the exit hot water temperature as a function of cold-water flow rate presented in tables 5.11 and 5.12 (comparison between the experimental and theoretical) show a decrease in the exit hot water temperature as the cold-water flow rate is increased.

The reason behind the decrease in the exit temperature of the hot stream is that the heat transfer rate from the hot stream to the cold stream . The mathematical model results are similar to those presented in section 5.1 in a linear manner, which is because of the approximation made on the differential equations of the system.

5.2.4 Frequency Response

Table 5.13 Frequency response for the conditions of table 5.12

w	AR			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-1.124875	-1.766	-1.914	-2.043
5	-3.221785	-3.277	-3.303	-3.333
9	-3.987602	-4.005	-4.014	-4.025
13	-4.466705	-4.475	-4.48	-4.485
17	-4.816222	-4.821	-4.824	-4.827
21	-5.091533	-5.095	-5.097	-5.099
25	-5.318695	-5.321	-5.322	-5.324
29	-5.512069	-5.514	-5.515	-5.516
33	-5.680417	-5.682	-5.682	-5.683
37	-5.82948	-5.831	-5.831	-5.832
41	-5.963226	-5.964	-5.965	-5.965
45	-6.084512	-6.085	-6.086	-6.086
49	-6.195463	-6.196	-6.196	-6.197
53	-6.297702	-6.298	-6.298	-6.299
57	-6.392499	-6.393	-6.393	-6.393
61	-6.480864	-6.481	-6.481	-6.482
65	-6.563615	-6.564	-6.564	-6.564
69	-6.641422	-6.642	-6.642	-6.642

Table 5.14 Phase angle for the conditions of table 5.12

w	Φ			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-0.953	-1.073	-1.156	-1.216
5	-0.274	-0.352	-0.426	-0.495
9	-0.155	-0.202	-0.247	-0.291
13	-0.108	-0.14	-0.173	-0.205
17	-0.083	-0.108	-0.133	-0.157
21	-0.067	-0.087	-0.108	-0.128
25	-0.056	-0.073	-0.091	-0.108
29	-0.048	-0.063	-0.078	-0.093
33	-0.043	-0.056	-0.069	-0.082
37	-0.038	-0.05	-0.061	-0.073
41	-0.034	-0.045	-0.055	-0.066
45	-0.031	-0.041	-0.05	-0.06
49	-0.029	-0.038	-0.046	-0.055
53	-0.027	-0.035	-0.043	-0.051
57	-0.025	-0.032	-0.04	-0.047
61	-0.023	-0.03	-0.037	-0.044
65	-0.022	-0.028	-0.035	-0.042
69	-0.02	-0.027	-0.033	-0.039

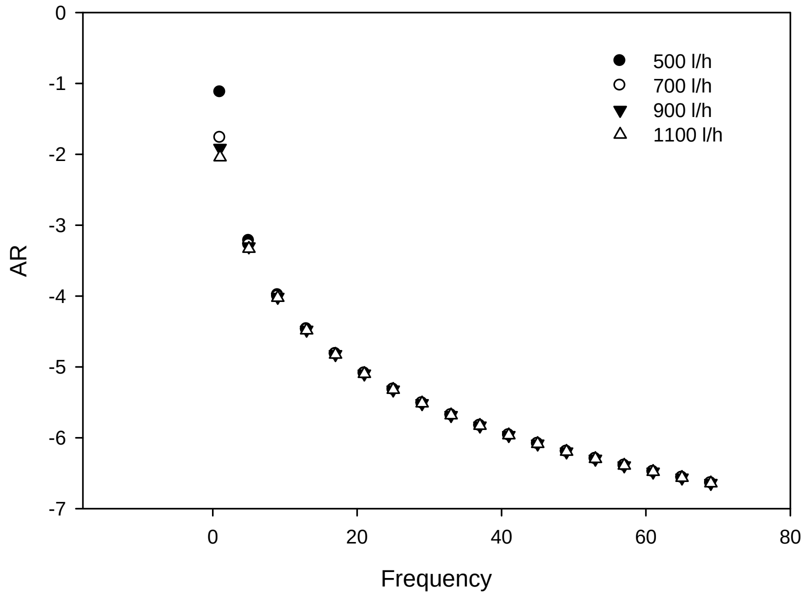


Fig. 5.14 Frequency response for parameters corresponding to table 5.13

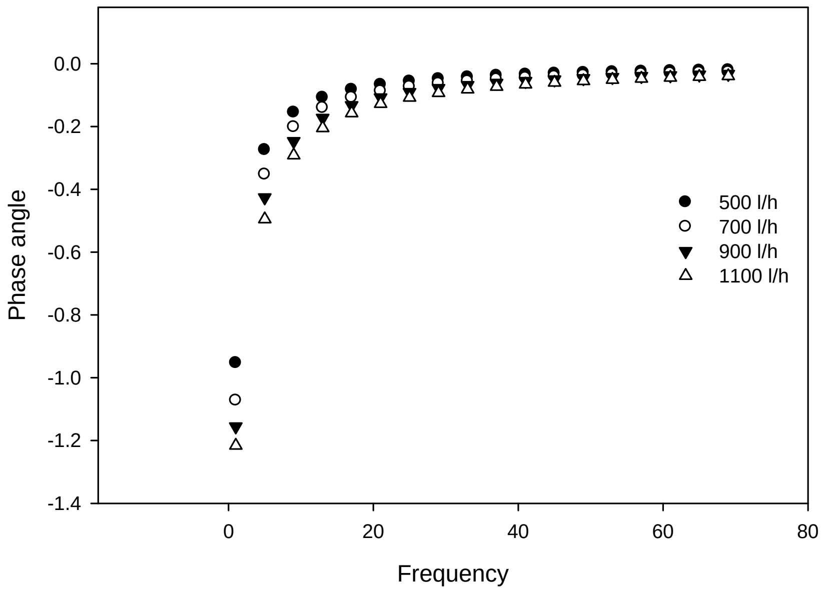


Fig. 5.15 Phase angle for parameters corresponding to table 5.14

5.2.5 Dynamic Behavior for Flow Rate of 300 L/h for Cold Water and of 500 L/h for Hot Water

The conditions for cold-water inlet temperature of 30^{o}C; and hot water inlet temperature of 70^{o}C steady state flow rates for hot water of 500 L /h and cold water is 300 L/ h. A step change rates of 200, 400, 600, and 800 L / h are applied respectively and the response of the outlet temperature of hot water is shown in Table 5.15

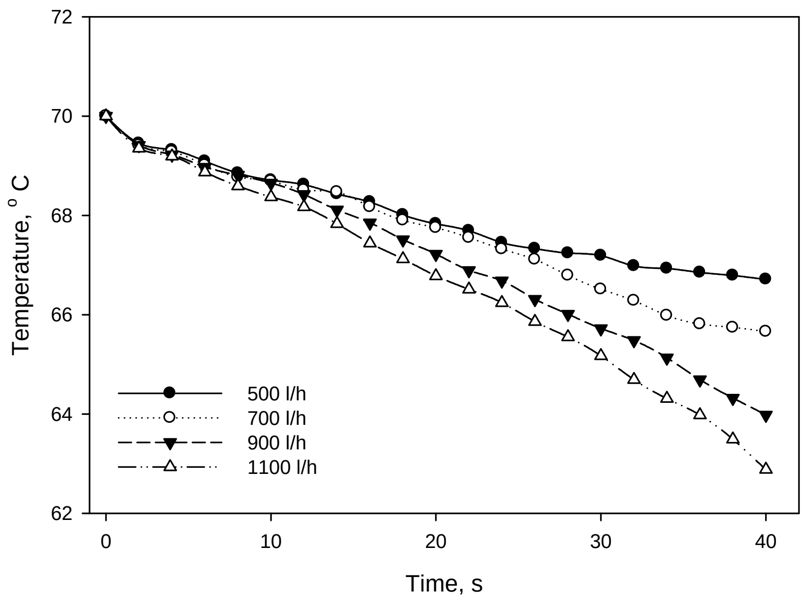


Fig. 5.16 Exit hot water temperature response for disturbance in cold water flow rate for steady state flow rate of 500 L/h for cold water and 300 L/h for hot water

Table 5.15 Outlet temperature of hot water at different flow rates of cold water for steady state flow rates for hot water of 500 L /h and cold water of 300 L/h obtained from experimental work

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L/h
0	70	70	70	70
2	69.45	69.41	69.41	69.35
4	69.32	69.28	69.22	69.19
6	69.09	69.01	68.97	68.87
8	68.85	68.77	68.81	68.59
10	68.71	68.69	68.65	68.37
12	68.62	68.51	68.42	68.17
14	68.43	68.47	68.11	67.83
16	68.27	68.17	67.85	67.44
18	68.01	67.9	67.51	67.12
20	67.83	67.75	67.22	66.78
22	67.69	67.55	66.89	66.51
24	67.45	67.32	66.68	66.24
26	67.33	67.11	66.31	65.86
28	67.24	66.79	66.01	65.55
30	67.19	66.51	65.72	65.17
32	66.98	66.28	65.48	64.69
34	66.93	65.98	65.13	64.31
36	66.85	65.81	64.69	63.98
38	66.79	65.74	64.32	63.49
40	66.71	65.66	63.98	62.88

Table 5.16 Outlet temperature of hot water at different flow rates of cold water for steady state flow rates for hot water of 500 L /h and cold water of 300 L/h obtained from mathematical model

Time (s)	Outlet temperature (^{o} C) at flow rate			
	500 L /h	700 L /h	900 L /h	1100 L/h
0	70	70	70	70
2	69.939	69.877	69.816	69.754
4	69.85	69.696	69.544	69.392
6	69.759	69.511	69.267	69.023
8	69.667	69.326	68.99	68.654
10	69.576	69.141	68.712	68.285
12	69.485	68.956	68.435	67.915
14	69.393	68.771	68.158	67.546
16	69.302	68.586	67.881	67.176
18	69.211	68.401	67.603	66.807
20	69.119	68.216	67.326	66.437
22	69.028	68.031	67.049	66.068
24	68.937	67.846	66.771	65.698
26	68.845	67.661	66.494	65.329
28	68.754	67.476	66.217	64.959
30	68.663	67.291	65.939	64.59
32	68.571	67.106	65.662	64.22
34	68.48	66.921	65.385	63.851
36	68.389	66.736	65.107	63.482
38	68.297	66.551	64.83	63.112
40	68.206	66.366	64.553	62.743

5.2.6 Comparison between Results of Experimental Work and Mathematical Model

Comparison between the results obtained from the experimental work and those obtained from mathematical the model are presented in Figs. 5.17 and 5.18. The comparison was made for two flow rate values because of the very close values of the temperature profiles.

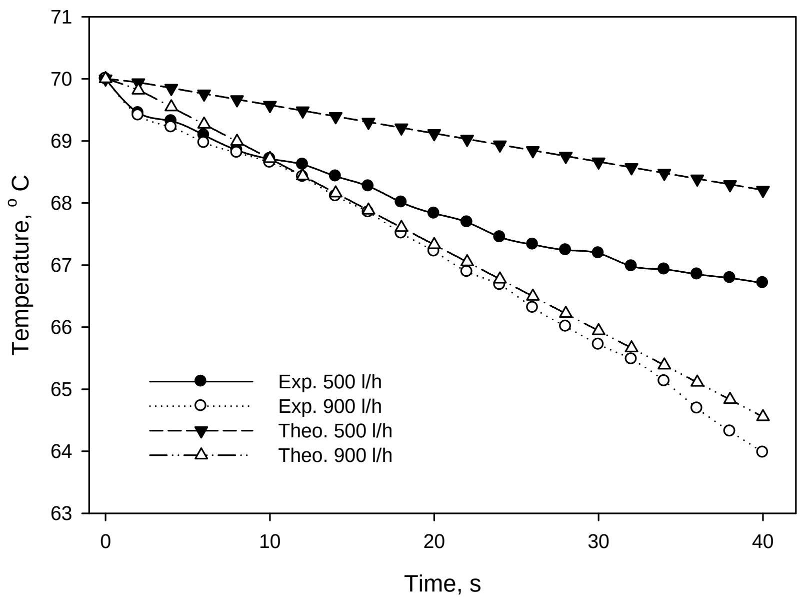


Fig 5.17 Theoretical and experimental results for 500 and 900 L / h

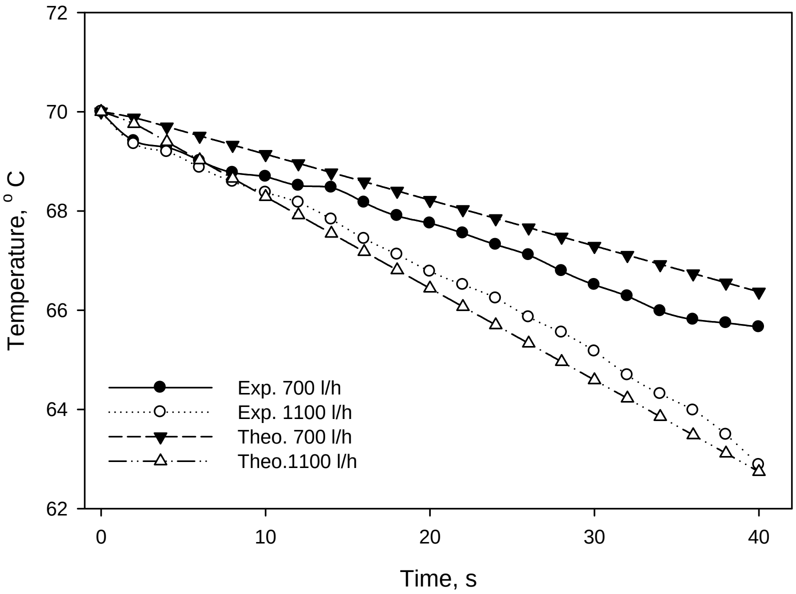


Fig. 5.18 Theoretical and experimental results for 700 and 1100 L/h

The data of the exit hot water temperature as a function of cold-water flow rate presented in tables 5.15 and 5.16 show also a decrease in the exit hot water temperature as the cold-water flow rate is increased.

The reason behind the decrease in the exit temperature of the hot stream is due to the same reason explained in the previous conditions for steady state flow rate of 300 L /h for the hot stream but with the following difference in the values of the temperature where for steady state flow rate of 300 L /h, the exit temperature is higher than that for 500 L / h and this is because of the more decrease in the residence time which causes the heat transfer to increase.

5.2.7 Frequency Response

Table 5.17 Frequency response for the conditions of table 5.16

w	AR			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-1.644	-1.806	-1.948	-2.074
5	-3.26	-3.283	-3.31	-3.341
9	-4	-4.007	-4.017	-4.028
13	-4.473	-4.476	-4.481	-4.487
17	-4.82	-4.822	-4.825	-4.828
21	-5.094	-5.095	-5.097	-5.099
25	-5.32	-5.321	-5.323	-5.324
29	-5.513	-5.514	-5.515	-5.516
33	-5.681	-5.682	-5.683	-5.684
37	-5.83	-5.831	-5.831	-5.832
41	-5.964	-5.964	-5.965	-5.965
45	-6.085	-6.085	-6.086	-6.086
49	-6.196	-6.196	-6.196	-6.197
53	-6.298	-6.298	-6.299	-6.299
57	-6.393	-6.393	-6.393	-6.394
61	-6.481	-6.481	-6.482	-6.482
65	-6.564	-6.564	-6.564	-6.564
69	-6.642	-6.642	-6.642	-6.642

Table 5.18 Phase angle for the conditions of table 5.16

w	Φ			
	500 L / h	700 L / h	900 L / h	1100 L / h
1	-0.988	-1.097	-1.173	-1.229
5	-0.295	-0.372	-0.444	-0.512
9	-0.167	-0.213	-0.258	-0.303
13	-0.116	-0.149	-0.181	-0.213
17	-0.089	-0.114	-0.139	-0.164
21	-0.072	-0.093	-0.113	-0.133
25	-0.061	-0.078	-0.095	-0.112
29	-0.052	-0.067	-0.082	-0.097
33	-0.046	-0.059	-0.072	-0.085
37	-0.041	-0.053	-0.064	-0.076
41	-0.037	-0.047	-0.058	-0.068
45	-0.034	-0.043	-0.053	-0.062
49	-0.031	-0.04	-0.049	-0.057
53	-0.029	-0.037	-0.045	-0.053
57	-0.027	-0.034	-0.042	-0.049
61	-0.025	-0.032	-0.039	-0.046
65	-0.023	-0.03	-0.037	-0.043
69	-0.022	-0.028	-0.034	-0.041

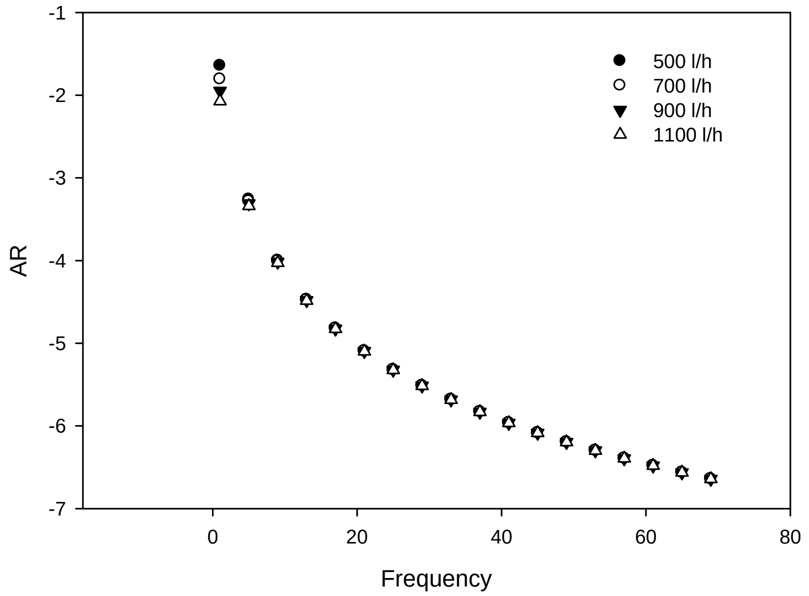


Fig. 5.19 Frequency response for parameters corresponding to table 5.17

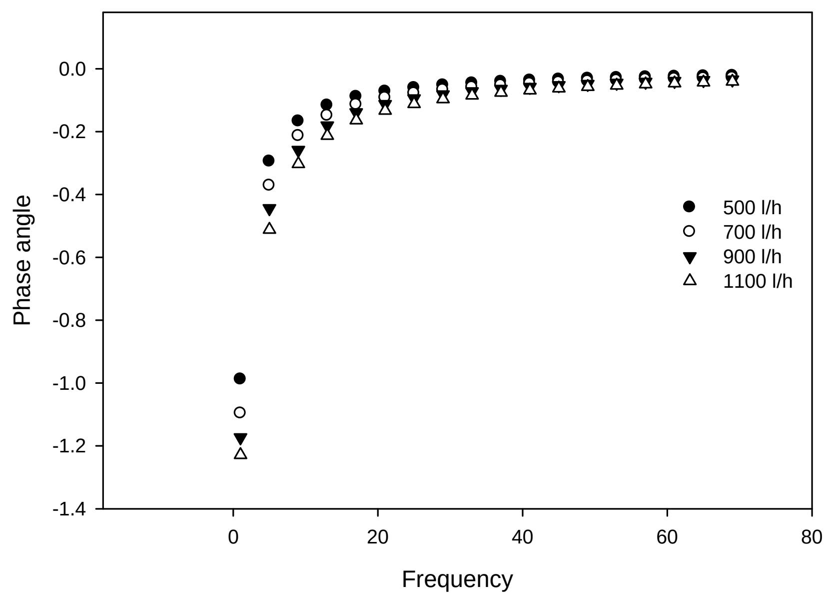


Fig. 5.20 Phase angle for parameters corresponding to table 5.18

Chapter Six
Conclusions and Recommendations for Future Work

6.1 Conclusions

The following conclusions can be written:

1. The dynamic of the double pipe heat exchanger can be described by first order for step change.
2. The real process is non-linear due to the shape of the double pipe, turbulent flow and change in physical properties.
3. Frequency response data from the transfer function, which led to the construction bode diagram would not recommended method for process identification

6.2 Recommendations for Future Work

The following points may considered for future work

1. A control system may be added to the present experimental apparatus
2. An experimental study of another type of heat exchanger can be made as well as the corresponding mathematical model may studied.
3. The mathematical model may be solved rigorously according to any appropriate numerical method.
4. Installment of filter is highly desirable to eliminate noise as well as obtaining constant pumping rates.
5. Different flow pattern must be studies to develop the dynamic characteristics of double pipe heat exchanger.

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Appendix A
Root Square Sum Error (RSSE)

RSSE for experimental data and mathematical results

Flow conditions	RSSE for 500 l/h	RSSE for 700 l/h	RSSE for 900 l/h	RSSE for 1100 l/h
Steady state flow rate for cold and hot water of 300 l/h	0.4178	0.3306	0.286	0.2504
Steady state flow rate of 500 l/h for cold and of 300 l/h for hot water	0.2484	0.3588	0.2524	0.1203
Steady state flow rate for cold and hot water of 300 l/h	0.1146	0.3637	0.5626	0.6614
Steady state flow rate of 500 l/h for cold and of 300 l/h for hot water	0.2643	0.131	0.0571	0.082

Appendix B
Sample of Calculations

L= 1 m

dc= 0.012825 m

dh= 0.02527 m

$$\mathrm{Re} = \frac{\rho U d}{\mu}$$

Rec= 10304.24

Reh= 10186.97

$$\frac{hd}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n}$$

n= 0.3 for cooling

n= 0.4 for heating

hc= 3010.12 W/m^2 K

hh= 1411.05 W/m^2 K

$$Uc = \frac{1}{\frac{1}{hc} + \left[\frac{Ac}{Ah} \times \frac{1}{hh}\right]}$$

$$Uh = \frac{1}{\frac{1}{hh} + \left[\frac{Ah}{Ac} \times \frac{1}{hc}\right]}$$

Uc= 1941.75 W/m^2 K

Uh=500.98 W/m^2 K

الخلاصـــة

لقد تم في هذا البحث دراسة التصرف الديناميكي لمبادل حراري ثنائي الأنبوب دراسة عملية و نظرية. و إن المبادل الحراري الذي جرت دراسته يتألف من أنبوبين متداخلين بحيث أن الأنبوب الخارجي يحمل الماء الحار و الأنبوب الداخلي يحمل الماء البارد و تدوير الماء جرى بأستخدام مضختين.

لقداجريت دراسة تأثير كل من سر عة جريان الماء الحار (٥٠٠، ٧٠٠، ٩٠٠، ١١٠٠ لتر بالساعة) على درجة حرارة الماء البارد الخارجة و كذلك اجريت دراسة تأثير سرعة جريان الماء البارد (٥٠٠، ٧٠٠، ٩٠٠، ١١٠٠ لتر بالساعة) على درجة حرارة الماء الحار الخارجة بإيجاد النتائج العملية في حالة الأنتظام مع الزمن و كذلك التصرف الديناميكي ثم إشتقاق نموذج رياضي لأستحصال صيغة لكل من تأثير سر عة جريان الماء الحار على درجة حرارة الماء البارد الخارجة و كذلك جرى دراسة تأثير سرعة جريان الماء البارد على درجة حرارة الماء الحار الخارجة كلآتي:

$$\overline{Tc}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right)$$

$$\alpha = BL(Th_i - Th_L)\ ,\ \beta = v_c + v_H + AL + BL$$

$$\overline{Th}(t) = \frac{\alpha\lambda}{\beta^2}\left(\beta t - \left[1 - e^{\beta t}\right]\right)$$

$$\alpha = AL(Tc_i - Tc_L)\ ,\ \beta = v_c + v_H + AL + BL$$

إن تأثير زيادة سرعة الماء الحار على درجة حرارة الماء البارد سبب زيادة درجة الحرارة إن تأثير زيادة سرعة الماء البارد على درجة حرارة الماء البارد سبب تقليل درجة الحرارة.

عند إجراء مقارنة بين النتائج المستحصلة من التجارب العملية و الموديل الرياضي حصل توافق بين النتائج خاصة في حالة فترات زمنية قصيرة و ذلك بسبب أن التبسيط الذي أجري على الموديل أدى لجعل النتائج أن تكون خطية والذي يؤدي أن تكون الزيادة أو النقصان مستمرة كما دلت على ذلك مخططات التردد.

شكر وتقدير

اولا وقبل كل شئ الحمد والشكر لله على تمام الصحة وقوة الايمان التي ساعدتني على تخطي جميع الصعاب التي واجهتها طيلة فترة البحث.

اود ان اعبر عن خالص شكري وتقديري وعرفاني بالجميل للمشرف **ا.د. قاسم جبار سليمان** و **م.د خالد مخلف موسى** لما قدماه لي من اهتمام كبيروجهد بالغ ولما ابدياه من توجيهات قيمة ساعدت على انجاز هذا العمل.

اود ان اشكر جميع منتسبي قسم الهندسة الكيمياوية لابدائهم المساعدة اللازمة اثناء هذا العمل.

ولا انسى ان اتقدم بجزيل الشكر والتقدير الى من ساندني وساعدني على تخطي الصعوبات خلال فترة البحث الى الذين لا مثيل لهم في الدنيا الى أبي وأمي الأعزاء.

م. منى منصور حسين

دراسة في ديناميكية المبادل الحراري مزدوج الانابيب

رسالة

مقدمة الى كلية الهندسة في جامعة النهرين

و هي جزء من متطلبات نيل درجة ماجستيرعلوم

في الهندسة الكيمياوية

من قبل

منى منصور حسين

بكالوريوس علوم في الهندسة الكيمياوية ٢٠٠٤

رمضان ١٤٢٨

تشرين الاول ٢٠٠٧