Report 11, 1987

HEAT EXCHANGER SELECTION FOR GEOTHERMAL APPLICATIONS

Edgar C. Gimenez
UNU Geothermal Training Programme
National Energy Authority
Grensásvegur 9
108 Reykjavík
ICELAND

Permanent Address:
Philippine National Oil Company
Energy Development Corporation
Geothermal Division
Merritt Road, Fort Bonifacio
Makati, Metro Manila
PHILIPPINES

ABSTRACT

Heat exchangers are usually the major items of equipment for direct-use projects. All standard types of heat exchangers; shell-and-tube, plate, finned tubes, downhole heat exchangers can be used with geothermal fluids. However, there are several rules which must be considered when designing and selecting equipment for geothermal supplies and the different duties. This project attempts to develop a theory for predicting a typical heat exchanger used frequently in geothermal applications using water-to-water and steam-to-water heat transfer. The theory developed in this report is backed up by experimental measurements taken from the existing two types of heat exchangers operating in Iceland. Furthermore, a computer program is developed to help in minimizing the burden of designing heat exchangers for a given performance using manual calculations.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

1. INTRODUCTION

1.1. Scope of work

This report represents the final outcome of the author's training from April to October 1987 at the UNU Geothermal Training Programme, in Reykjavik, Iceland, under a Fellowship grant from the UNDP.

The training programme started with an extensive five-week introductory lecture course on relevant aspects of geothermal energy. This was followed by a series of specialized seminars and field works for a couple of months, studying the various direct uses of geothermal energy. A field excursion to different geothermal sites in Iceland was carried out for about a week and was followed by another site visits to Nesjavellir and Svartsengi to study and gather pertinent measurements from existing heat exchanger units which are used in the preparation of this report.

The figures and pertinent data used in this report were taken from actual measurements conducted at Nesjavellir, literature surveys and from the references supplied by the National Energy Authority of Iceland.

This report deals with the subject of heat exchangers used in geothermal applications. It briefly surveys the different procedures and criterias used in selecting or designing a suitable unit. Furthermore, the application of computer programs as a method to evaluate the performance and efficiency of the unit has been included.

1.2. Statement of the problem

Predicting heat exchanger sizing and rating performances involves numerous parameters. These parameters foresee considerable efforts in predicting and evaluating the efficiency of the exchanging unit using manual calculations.

The main task of this problem is to develop a theory for typical heat exchangers used frequently in geothermal application and write a computer program to minimize the burden of designing heat exchanger for a given performance.

The theory developed is backed up by experimental measurements taken from the existing two types of heat exchangers operating in Iceland.

1.3. Purpose of the study

The primary purpose of this report is to consider various elements in predicting the actual and theoritical performances of heat exchanger used in geothermal applications. To accomplish this objective, a simple model was used in which actual measurements were taken from an existing pilot plant.

This kind of development uses simple number of thermal units technique in comparing actual and theoritical predictions which could be used as reference in formulating computer programs. The design parameters and the assumptions considered in this study utilizes properties of pure water and steam and the program must be changed if other fluids are used. These are discussed further in the following sections. However, corrosion, scaling and water quality matters are not fully discussed here, neither are the environmental issues.

2. HEAT EXCHANGER FUNDAMENTALS

2.1. Principles of Thermal Design

In general, heat exchangers are designed for many varied applications, and hence may involve different performance criteria. Some of these criteria may be minimum initial cost, minimium initial and operating costs, minimum or maximum heat transfer area, minimum mean temperature difference, and so on.

Heat exchanger thermal design can be classified primarily into two categorical problems, that is, rating and sizing. The determination of heat transfer and pressure drop performance of either an existing exchanger or an already sized exchanger is referred to as a rating problem. Inputs of the rating problem are: the exchanger construction, flow arrangement and overall dimensions, complete details on the material and surface geometries including their heat transfer and pressure drop characteristics, fluid flowrates, inlet temperatures and fouling factors. The fluid outlet temperatures, total heat transfer capability, and pressure drops on each side of the exchanger are then determined in this aspect. This is sometimes called as the performance problem. In contrast, in a sizing problem, the physical size of an exchanger is to be determined to meet the specified heat transfer and pressure drops. Inputs to this are: surface geometries, fluid flowrates, inlet and outlet fluid temperatures, fouling factors, and pressure drops on each side. This is sometimes referred to as the design problem.

One of the simplest ways of solving problems about sizing is to specify tentatively a physical size of the exchanger, analyze it as a rating problem, and compare the computed heat transfer and pressure drops with the specifications. If the computed results do not agree with the specifications, then a new size is assumed and the process is repeated.

2.2. Classification of Heat Exchangers

Heat exchangers are used in the process, power, automotive, air conditioning, heat recovery, alternative fuels, manufacturing industries, as well as key components available in the marketplace. These heat exchangers may be classified according to the transfer process, degree of compactness, construction features, flow arrangements, number of fluids, and fluid phase changes or process function. These classifications are summarized in Figure 1.

2.3. Types of Heat Exchangers

The choice of heat exchanger type for a particular duty depends on a large number of factors, not all of which can be discussed in detail. However, we can consider the influence of fluid properties, temperature and pressure conditions, maintenance and extension possibilities, and comparative costs. The statements made here are general in nature, and circumstances in individual cases may radically alter the relative importance of these factors which affect type selection. The following are types of heat exchangers commonly used in geothermal applications:

(1) THE - Tubular Heat Exchanger
Shell and Tube
Fluidized Bed
(2) PHE - Plate Heat Exchanger
(3) DHE - Downhole Heat Exchanger

Conventional shell and tube heat exchangers may be utilized for geothermal applications. This type of exchanger is readily available or may be custom designed to meet the specific need. This exchanger normally consists of a series of tubes, normally carrying geothermal water, sorrounded by an enclosing shell, confining the secondary system water around the tubes (see Figure 2.1). The tubes in this type of exchanger can be U-tube configuration or more likely for geothermal application a straight tube with removable heads at both ends to facilitate cleaning of the tubes. In principal the fluidized bed heat exchanger is of shell and tube type standing vertically, in which small solid particles are kept in a stationary fluidized condition by the liquid passing up the tubes. The solid particles regularly break through the boundary layer in the tubes, so that good heat transfer is achieved in spite of comparatively low liquid velocities in the tubes. Further, the solid particles have a slightly abrasive effects on the wall of heat exchanger tubes, removing any deposit from the tube wall at an early stage. Figure 2.2 shows the principle of fluidized bed heat exchanger. It consists of the tube bundle, on inlet section and the outlet chamber. The inlet section is divided into two chambers by a distribution plate, viz.: actual inlet chamber for liquid, the particle distribution chamber from where the particles are distributed equally over all the tubes.

The plate heat exchanger consists of a series of plates with gaskets held in a frame by clamping rods (see Figure 2.3). The gasket material limits the maximum temperature that can be used. The primary and secondary fluids are usually passed through alternating passages between the plates in single-pass counter flow, although other flow paths can usually be arranged by simple external piping. Stamping of the plates provides a variety of flowpath patterns. The counter current flow and high turbulence achieved in plate heat exchangers, provide for efficient thermal exchange in a small volume. Plate heat exchangers are commonly used in geothermal heating situations worldwide since it is relatively easy to clean. The plates are usually made of stainless steel, although titanium is used when the fluids are especially corrosive.

The downhole heat exchanger eliminates the problem of disposal of geothermal fluid, since only heat is taken from the well. However, their uses is limited to small heating loads such as the heating of individual homes. This exchanger consists of a system of pipes or tubes suspended in the well through which secondary water is pumped or allowed to circulate by natural convection. Several designs have proven succesful, but the most popular is the simple hairpin loop, or multiple loops, of iron pipe extending down to the well bottom (Figure 2.4). In order to obtain maximum output, the well must be designed to have an open annulus between the wellbore and the casing and perforations above and below the heat exchanger surface. Natural convection circulates the water down inside the casing, through the lower perforations, and up in the annulus and back inside the casing through the upper perforations.

2.4. Design Considerations

Design considerations of heat exchangers are usually based on the following conditions:

1.0 Heat transfer requirements
2.0 Physical size
3.0 Pressure drop characteristics
4.0 Cost

The heat transfer requirements must be met in the selection or design of any heat exchanger. The way that the requirements are met depends upon the relative weights placed on the conditions above.

Correspondingly, the overall heat transfer coefficient for a heat exchanger may vary substantially through the exchanger if the fluids are such that their properties are strongly temperature-dependent.

This study emphasize on the first three items and does not consider cost of equipment.

2.4.1. Design stages

The design of any heat exchanger should go through a number of stages (Drew, 1986). The main stages are:

1.0 Equipment flowsheet
2.0 Preliminary design
3.0 Process design description
4.0 Detailed thermal and hydraulic design
5.0 Mechanical design
6.0 Cost evaluation
7.0 Bid evaluation
8.0 Contract for fabrication

2.5. Heat Exchanger Selection

There are three main topics to be considered in selecting heat exchanger types. These are the heat duty, temperatures, and the quality of fluids being utilized. Factors such as materials required to be compatible with the water quality, fouling, capacity of the well, space availability for the equipment, corrosion and so on has also to be considered. However, these secondary factors are not fully discussed here.

2.6. Heat Transfer Equations

2.6.1. General

In order to carry out an analysis and calculation of heat transfer at variable physical properties, one should have an adequately accurate equation for heat transfer at constant properties. But even with this information there is still not enough knowledge of the effect of various geometrical and other parameters on the heat transfer performance. When the geometry of the exchanger becomes sufficiently complicated, it is necessary to develop the experimental correlation for heat transfer performance in order to obtain a reliable design procedure. It is then necessary to introduce correction terms, including the geometrical parameters in dimensionless form effecting the unit's performance.

The subject of this section is to define the functional heat transfer equations used in designing or rating a shell and tube and plate heat exchangers which are used as models in evaluating its rating and performance. The equations and correlations used in calculating the performance of the exchanging unit are combinations of the relatively basic ones and from the equations used by established manufacturer following the TEMA standards.

These equations are basically the same equations being used in developing computer programs which the author has been formulating as part of this project.

2.6.2. LMTD Method

The Log Mean Temperature Difference (LMTD) as shown in Figure 3 is defined as:

For counterflow,

$$LMTD=(Th2-Tc2)-(Th1-Tc1)/(\ln[(Th2-Tc2)/(Th1-Tc1)]) \quad (1a)$$

For parallel flow,

$$LMTD=(Th1-Tc1)-(Th2-Tc2)/(\ln[(Th1-Tc1)/(Th2-Tc2)]) \quad (1b)$$

The heat transfer between the two fluids can then be calculated from:

$$q = U\ A\ LMTD \quad (2a)$$

where U = overall heat transfer coefficient
A = surface area for heat transfer consistent with definition of U

For heat exchanger of other types than parallel or counter-flow the heat transfer equation takes the form:

$$q = U\ A\ F\ LMTD \quad (2b)$$

where F is the correction factor less than one and LMTD is calculated as for counter flow arrangement.

2.6.3. Effectiveness-NTU method

The effectiveness method offers many advantages for analysis of problems in which a comparison between various types of heat exchangers must be made for purposes of selecting the type best suited to accomplish a particular heat transfer objective. Heat exchanger effectiveness is defined as:

$$\text{Effectiveness} = \epsilon = \text{AHT} / \text{MPHT}$$
(3)

The actual heat transfer (AHT) may be computed by calculating either the energy lost by the hot fluid or the energy gained by the cold fluid.

For counterflow exchanger;

$$q = m_h c_h (Th_1 - Th_2) = m_c c_c (Tc_1 - Tc_2) \quad (4a)$$

For parallel-flow exchanger;

$$q = m_h c_h (Th_1 - Th_2) = m_c c_c (Tc_2 - Tc_1) \quad (4b)$$

To determine the maximum possible heat transfer (MPHT) for the exchanger:

$$q_{max} = (mc)_{min} (Th_{inlet} - Tc_{inlet}) \quad (5)$$

The minimum fluid may be either the hot or cold fluid, depending on the massive flow rates and specific heats. For counterflow exchanger:

$$\epsilon_h = (Th_1 - Th_2)/(Th_1 - Tc_2) \text{ if } mh*ch=(mc)min \quad (6a)$$
$$\epsilon_c = (Tc_1 - Tc_2)/(Th_1 - Tc_2) \text{ if } mc*cc=(mc)min \quad (6b)$$

For parallel-flow exchanger:

$$\epsilon_h = (Th_1 - Th_2)/(Th_1 - Tc_1) \text{ if } mh*ch=(mc)min \tag{6c}$$
$$\epsilon_c = (Tc_2 - Tc_1)/(Th_1 - Tc_1) \text{ if } mc*cc=(mc)min \tag{6d}$$

It may be shown that effectiveness for parallel flow is usually written as:

$$\epsilon = 1 - \exp[(-UA/C_{min})(1+C_{min}/C_{max})]/(1+C_{min}/C_{max}) \tag{7a}$$

where C = mc is defined as the capacity rate.

A similar analysis may be applied to the counterflow case, and the following relation for effectiveness results:

$$\epsilon = 1 - \exp[(-UA/C_{min})(1-C_{min}/C_{max})]/(1-(C_{min}/C_{max}) \exp[(UA/C_{min})(1-C_{min}/C_{max})]) \tag{7b}$$

The effectiveness for other flow arrangement are shown in Table 1.

2.6.4. Shell and Tube Heat Exchanger

The heat transfer equation yields the following relationship:

$$Q = U\ A\ F\ LMTD \tag{8}$$

where U = overall heat transfer coefficient
A = surface area for heat transfer consistent with U
F = correction factor
LMTD = log mean temperature difference

The overall heat transfer coefficient is defined as:

$$U_i = 1/(1/h_i + A_i \ln(r_o/r_i)/2\pi kL + A_i/A_o h_i + R_f) \tag{9a}$$

$$U_o = 1/(1/h_o + A_o \ln(r_o/r_i)/2\pi kL + A_o/A_i h_i + R_f) \tag{9b}$$

where the subscripts i and o pertain to the inside and outside of the smaller inner tube. Fouling factors (R_f) must be obtained experimentally by determining the values of U for both clean and dirty conditions in the heat exchanger. The fouling factor is thus defined as:

$$R_f = 1/U_{dirty} - 1/U_{clean} \tag{10}$$

The pressure drop associated with an ideal crossflow section between two baffles is:

$$\Delta p_b = 4f' N_c \; G^2/2g_c \rho_m \tag{11}$$

where f' is an empirical Fanning friction factor ($4f'=f$ in some books); N_c is the number of tube rows crossed in one crossflow section; G is the mass velocity. The friction factor f' is determined from correlations for flow normal to a tube bank of a specified arrangement.

The total pressure drop on the shellside is the sum of pressure drop associated with each crossflow section between baffles, pressure drop associated with each window section, and pressure drop for crossflow sections on each end between the first baffle and the tubesheet. This is expressed as:

$$\Delta Pshell = [(Nb-1)\Delta pbsb + Nb\Delta pw]sl + 2\Delta pbsb(1+Nw/Nc) \tag{12}$$

where Nb is the number of baffles, sb is the pressure drop correction factor to account for bundle-to-shell bypass stream, and sl is the pressure drop correction factor to account for baffle-to-shell and tube-to-baffle, and Nw is the correction for the area occupied by tubes.

2.6.5. Plate Heat Exchangers

In taking the overall heat transfer measurements from inlet and outlet temperatures in a plate heat exchanger, the overall heat transfer coefficient U is obtained from:

$$1/U = 1/h_1 + 1/h_2 + t/k_p + R_f \qquad (13)$$

where R_f is the fouling factor, h_1 and h_2 are the film coefficients for the two fluids in the exchanger channels, t is the plate thickness, and k_p is the thermal conductivity of the plate material.

Using the usual relationships:

$$Q = F\ U\ A\ LMTD \qquad \text{and} \qquad (2b)$$
$$Q = m_h\ c_h\ \Delta T_h = m_c\ c_c\ \Delta T_c \qquad (14)$$

together with Eq. (13) it is possible to calculate the film coefficients from experimental studies. Correction factor can be obtained using the NTU correlations.

2.6.6. Channel dimensions and pressure drop

Various studies have been made of the influence of geometry in simplified channels but most treatments are empirical and based on the "hydraulic diameter" concept. The characteristics dimension De of the plate heat exchanger channel is defined as:

$$De = 4 * A_f / P \qquad (15a)$$

which can be reduced to:

$$De = 2b / \varnothing \qquad (15b)$$

where A_f is the cross-sectional area of flow, P is the wetted parameter, b is the mean plate spacing (plate pitch-plate thickness) and Ø is the ratio of the developed to the projected area of the corrugated plate. In some references (Kakac et al), an alternative definition of De = 2b is used.

The flow is now considered to be in circular tube of diameter De with the appropriate mean axial velocity, u_m, given by:

$$u_m = vf / bw \tag{16}$$

where vf is the volumetric flowrate between two plates and w is the width of plate (between gaskets).
The individual heat transfer coefficients for plate heat exchanger can be obtained from correlations similar to the expression given in Eq. (13) except that the constants are different. The follwing correlation can be used, for the prediction of heat transfer coefficients in typical plate heat exchangers:

$$Nu = fn\ (\ Re,\ Pr\ ) \tag{17a}$$

where the Nusselt number, Nu, is given by:

$$Nu = h\ De / k \tag{17b}$$

where h is the channel inside film heat transfer coefficient.
Also heat transfer can be correlated as:

$$Nu = C\ Re^n\ Pr^m \tag{17c}$$

where n = 0.65 to 0.85
m = 0.30 to 0.45

Having obtained the film coefficients, they are usually correlated in the form of Eq. (13). For example, the following correlations from studies with Newtonian fluids in a variety of pass arrangements:

a) APV HX $Nu=0.298\ Re^{0.646}Pr^{0.316}$ $150<Re<3500$ (18)

b) Alfa-Laval P13 $Nu=0.430\ Re^{0.591}Pr^{0.303}$ $200<Re<3000$ (19)

These equations reflect the changes in flow regimes observed from the pressure drop data. At high Reynolds numbers the Reynolds number exponent approaches a value of about 2/3 which is typical of most turbulent flows. However, at lower Reynolds numbers the exponent approaches a value of about 1/3 which is found for laminar tube flow under many conditions. As expected over the entire range, the exponent on the Prandtl number is about 1/3. Clearly a viscosity ratio term can be added to these equations when large temperature differences are used with liquids whose viscosities are temperature sensitive.

We can now attempt to correlate pressure drop data in the usual friction factor-Reynolds number form:

$$f = fn\ (Re) \qquad (20)$$

where the friction factor f is defined by:

$$f = \Delta p\ D_e\ /\ 2\ \rho\ u^2\ L \qquad \text{or} \qquad (21a)$$

$$f = 8\ Nu\ /\ (Re * Pr^{0.333}) \qquad (21b)$$

$$\Delta P = f\ L/D\ \rho\ u^2/\ 2 \qquad (22)$$

and Δp is the pressure drop across a plate of length L.

The pressure drop components due to gravitational force is defined as follows:

$$\Delta P = \pm\ \rho_m\ g\ L\ /\ g_c \qquad (23)$$

where + sign stands for vertical upflow (equivalent pressure drop due to the elevation change), - stands for vertical downflow (equivalent pressure rise), g is the gravitational acceleration, and L is the exchanger length.

3. DATA GATHERING

3.1. Nesjavellir Pilot Plant

The Nesjavellir Pilot Plant was built 1974 by Hitaveita Reykjavikur to carry out extensive studies in four main subjects, namely; the evaluation of output capacity and production characteristics of the geothermal reservoir, investigation of ways to dispose geothermal water from the power plant, strengthening out design assumptions for the power plant, and to carry out production studies in developing experience of power plants at high temperature areas. The last subject primarily deals with various experiments involving scaling and corrosion, and the investigation of fluidized bed heat exchanger performances. The overall diagram of the system is shown in Figure 4.1 schematically.

3.2. Process Description

Figure 4.2 shows the overall arrangement of the heat exchangers being utilized for study purposes. The plant uses two kinds of exchangers, that is, one plate heat exchanger (PHE 2) and two fluidized bed heat exchanger (ESMIL 1/2). The fluidized bed heat exchanger uses aluminum chopped wires of 3 mm in diameter as the stationary particles. This kind of development has opened new and economical ways for the utilization of high-temperature areas for district heating. From this arrangement, the cold water at about 4°C enters the first fluidized bed heat exchanger and heated up to about 23°C. Part of this fluid is heated in the plate heat exchanger and mixed again with the rest of the fluid to about 66°C at the plate heat exchanger and finally further heated at the second fluidized bed heat exchanger to attain the required temperature for consumption of about 90°C. The fluidized bed heat exchangers in this system utilizes geothermal brine with an inlet temperature of 155°C as the primary fluid while the plate heat exchanger uses steam as the heating medium. Tables 3, and 4 shows the results of the measurements taken.

The other plate heat exchanger (PHE 1), which is independent from the rest of the system, is used to test various materials in the plates. In this set-up the greater variation in flowrates can be achieved. Table 2 shows these gathered measurements.

3.3. Temperature and Flow Measurements monitoring

The overall design of the system was dictated by the type of experimental measurements desired. The goal was to produce a sufficient actual data from a pilot plant scale, with steady-state conditions existing for a sufficiently long period of time to allow certain basic measurements to be taken. Furthermore, it was the intention that these measurements and resulting analysis be of use in certain practical application, such as developing a computer program in evaluating the performance of a heat exchanger.

For determination of fouling, it is necessary to calculate the overall heat transfer coefficient as a function of time. For this purpose the heat exchangers are equipped with measuring points.

Measurements taken at Nesjavellir is accomplished by taking readings from the computer and from the thermometers and flowmeters installed at the unit. The instrumentation located throughout the system includes standard mercury-in-glass thermometers and dial-gage pressure indicators. Electronic thermometers are also installed at some points to automatically monitor the temperatures.

4. FORMULATION OF A COMPUTER PROGRAM

4.1. Objective

One of the primary objective report is to develop a computer program that could help in minimizing the burden of designing a heat exchanger for a given performance. This is an important development that is equally useful in dealing and evaluating its design and performance. The developed computer program can find the main applications in the following subjects:

1.0 Predictions of the performance of the exchanger
2.0 Development of design criterias
3.0 Determination of the effect of varying the operating variables
4.0 Demonstration of the selection method of heat exchanger type.

These four items are the main objectives and have to be tested from the gathered data and measurements.

The complete listing of the program with the corresponding outputs is presented in Appendix 1.

4.2. Description and procedure of using the program

There are two computer programs developed in this project. One deals with the plate heat exchanger and other deals with the fluidized bed heat exchangers. Both of these programs uses almost the same procedures in computing heat exchanger performance. The main difference between the two programs is that the plate heat exchanger program calculates steam to water heat transfer while the fluidized bed (Esmil) heat exchanger program calculates water to water heat transfer. This results in using different equations especially suited for the given performance. In addition, the correlations and the properties used vary accordingly. To account for the fouling factor, the equations used here are based on standard references on heat transfer and from literature surveys (Gudmundsson, 1979). All fluid properties are calculated from correlations of steam table data (Keenan et al). The developed computer programs are described as follows:

4.2.1. Fluidized Bed Heat Exchangers

The structure of the computer program is shown in Figure 5.1. This program operates by inputting the fluid and the physical data of the heat exchanger. Fluid data refers to the flowrate of the cold stream, the inlet and the target outlet temperatures of the geothermal and cold stream. Usually inlet temperatures for both fluids are known, and should also clearly specify how much of the cold fluid is to be heated to a certain temperature. From these given conditions the program automatically calculates the target heat duty, initial effectiveness, initial capacity ratio, and the initial flowrate of the geothermal fluid. Then the program calculates the flow velocities, the different dimensionless numbers in heat transfer, which are needed in order to calculate the overall heat transfer coefficient, the number of thermal units (NTU), and the calculated (real) effectiveness. After this step, the real heat duty is then calculated. It is in this portion of the program that iteration begins, when the calculated heat duty is not equal to the target heat duty. New value for mass flow of the hot fluid is then calculated and will commence with new velocities. This process is repeated until the real heat duty equals the target heat duty. When the two values are equal then it proceeds to calculate the pressure drops of both the cold and the hot stream sides. Results are then displayed after all these subroutines are completed.

4.2.2. Plate heat exchanger

The structure of this program is illustrated in Figure 5.2. Basically, the system has the same approach used in evaluating fluidized bed heat exchanger performance. As can be easily noted they differ in the considerations used in the iteration process of calculating the real heat duty. As the hot fluid is condensing steam, the model showed great sensitivity against variations of mass flow of the hot fluid and subcooling of the condensing steam. In order to avoid numerical instability, new value for mass flow of the hot fluid had to be calculated on basis of the ratio between calculated enthalpy drop on the hot side to the measured enthalpy drop. The pressure drop equations used in this program are limited only to the cold side of the fluid.

5. RESULTS

5.1. Comparisons of Data

There are several points to be noted in interpreting between the actual measurements taken and theoritical predictions. Gross predictions of the performance of heat exchangers would be expected to indicate considerable evaluation. The results of computer calculations indicated that theorical predictions are slightly higher than the measured ones. These predictions are substantiated by the testing results and comparison with results from the computer runs as illustrated in Figures 6.1-8.5. The following are the description of the plotted results between the actual and calculated measurements:

Figure 6.1 : The graph shows the comparison between measured values and calculated data. From its formation, it is evident that calculated values are slightly higher than the measured data. This data was taken from PHE 1 using the outlet temperatures of the cold fluid.

Figure 6.2 : The graph shows the same results as in the case of the outlet temperatures of the hot fluid, using the same type of heat exchanger.

Figure 6.3 : The calculated and measured heat duty in PHE 1 were compared and both results are almost identical. The points are almost situated on the same line showing identical results.

Figure 6.4 : For PHE 1, the resulting curve between the hot and cold flow rates develops into a parabolic formation. The hot flow rate increases as the cold flow rate is increased.

Figure 6.5 : The pressure drop in this curve increases as the cold flow rate increased. The resulting curve forms into a straight line as a function of flow rate squared.

Figure 7.1 : For PHE 2, the calculated data is higher than the measured values. In this case the flow rates variation of the cold fluid are not so large, so the points are closely located to each other, but show systematic differences between measurements and theory.

Figure 7.2 : The measured and calculated heat duty is also compared in this graph. The results showed similar pattern as that of the cold fluid outlet temperature in Figure 7.1.

Figures 8.1 to 8.4 : These graphs are the results of the measurements from the Esmils 1/2 (fluidized bed). As shown, the calculated results correlate well with the measured ones. For the heat duty, both results are situated on the same line showing identical results.

Figure 8.5 : For the Esmils 1/2, the pressure drop on the cold side is plotted versus the flow rates squared. Both have similar results, and show similar tendency as for the plate heat exchanger.

The comparisons between experiments and theory show rather good agreement. For the small heat exchanger (PHE 1), the measured and calculated outlet temperature agree very well except for one point and the main reason could be that in that case the condensed steam is only slightly subcooled. For the bigger plate heat exchanger (PHE 2), the calculated cold fluid outlet temperature is always about 1-2°C higher than measured. This could be explained due to consistent error in measurement of the temperature. For the Esmil heat exchangers, the outlet temperature of the hot fluid is 0 to ±0.5°C higher for Esmil 1, but are within ±0.3°C for Esmil 2. Therefore, we could say that the results of the theoritical predictions are in good agreement with the measured data within the range of measurements.

The overall heat transfer coefficient based on the tubes surface was determined by measuring the inlet and outlet temperatures of the fluids in the test section and by measuring the mass flow of the fluid.

5.2. Performance Results

Heat exchanger performance is conditioned by a wide range of variables, both in terms of geometric parameters and well characteristics. From the results of the experimental model the performance of the exchanging unit could be demonstrated in the developed computer program where the theoritical prediction differs slightly from the actual results. This resulted in a satisfactory agreement with the experimental measurements are shown in Table 5. Matching the outputs from Tables 2,3,and 4 with the computed results, one can establish a method of predicting performances with high accuracy. Using the developed program, the effect of changing the magnitude of some of the variables has been fully demonstrated.

6. DISCUSSION

The results of the measurements taken from Nesjavellir are evident that operating temperatures and flowrates are one of the main factors that provides a selection criteria and will serve as an input to the case method of heat exchanger design (see Figures 6.1-8.5). The actual measurements gathered (Tables 2,3,4) clearly implies that these factors are one of the design variables associated with a heat exchanger design. With this available information on hand, it is now the designer's decision as to how he could effectively adjust these variables within imposed constraints and come up with a design having optimum objective function.

7. CONCLUSIONS

1. Selecting heat exchangers for geothermal applications involves various performance criteria. A particular design has been subjected to certain test requirements with satisfactory results. In the case of the Nesjavellir Pilot Plant, combined plate heat exchangers and the fluidized bed heat exchangers have been proven to meet these requirements.

2. Results of the measurements with a computer program has been demonstrated and the comparison between theorical predictions with experimental measurements has been done satisfactorily. This is very essential in formulating analysis of the performance characteristics of the exchanging unit in steady and unsteady conditions. The results show in general good agreement with the measured data and has demonstrated that the theory predicts well both for plate heat exchanger and the fluidized bed heat exchangers (Esmils 1/2).

3. The computer programs developed could provide sufficient information in evaluating heat exchanger performances. This is very important in dealing with these units for geothermal applications. However, this developed method has the limitations of given type of heat exchangers and has not been tested in a multi-fluid exchanger when the performance of the other fluid, flow arrangement, and overall heat exchanger constraints are imposed. These aspects are not elaborated here since considering various constraints and geometry would result to a complicated computer programming which would be difficult to satisfy in a time available for this study.

8. ACKNOWLEDGEMENTS

The author gratefully acknowledge the following individuals and institutions who have contributed in the preparation of this report. Special thanks to the Philippine National Oil Company - Energy Development Corporation, United Nations Development Program for sponsorship of this training and to the National Energy Authority of Iceland for the hospitality and use of the computer facilities. The author also expresses his sincere gratitude to the UNU Geothermal Training Programme and staff headed by Dr. Jón Steinar Gudmundsson, and to the professor of the University of Iceland, Dr. Valdimar K. Jónsson, and Páll Valdimarsson, research assistant, for their guidance and wholehearted support in the fulfillment of this project.

NOMENCLATURE

A	:	Cross-sectional area of the heat exchanger, m^2
cp_c	:	Specific heat of cold fluid, J/kg °C
cp_h	:	Specific heat of hot fluid, J/kg °C
De	:	Outside diameter of tube, m
Di	:	Inside diameter of tube, m
Dh	:	Hydraulic diameter, m
f	:	Friction factor, dimensionless
E	:	Enthalpy, kJ/kg
H	:	Film coefficients of fluid, W/m^2 °C
k	:	Thermal conductivity, W/m °C
L	:	Length of tube, m
mc	:	Flowrate of the cold fluid, kg/sec
mh	:	Flowrate of the hot fluid, kg/sec
N	:	Number of tubes
Ne	:	Number of effective plates
Np	:	Number of passes
Nu	:	Nusselt Number, dimensionless
Pr	:	Prandtl Number, dimensionless
Q	:	Heat Duty, KW
Re	:	Reynolds Number, dimensionless
rho	:	Density of hot or cold fluid, kg/m^3
Rf	:	Fouling factor, dimensionless
Sp	:	Spacing distance between tubes, m
Sg	:	Gap of plates, m
Sgp	:	Specific gravity of particle in the fluidized bed
tp	:	Thickness of plate, m
U	:	Overall heat transfer coefficient, W/m^2 °C
w	:	Width of plate, m
Vh	:	Velocity of hot fluid, m/s
Vc	:	Velocity of cold fluid, m/s
T	:	Temperature of hot or cold fluid, °C

Greek symbols:

ϵ	:	Effectiveness of heat exchanger, dimensionless
μ	:	Dynamic viscosity, N s/ m^2
ρ	:	Density of fluid, kg/m^3
Ø	:	Ratio of the actual flow area to the projected area of the corrugated plate.

Subscripts:

i	:	Inside diameter of pipe, m
o	:	Outside diameter of pipe, m
h	:	Hot side
c	:	Cold side
1	:	Inlet of hot fluid
2	:	Outlet of hot fluid

REFERENCES

Afgan, N.H., Schlunder, E.U., (1974): "Heat Exchangers-Design and Theory Sourcebook," McGraw-Hill, New York, NY, 1974.

Drew, S.R.,(1986): "Geothermal Heat Exchanger Design," Notes presented for the engineers attending the Geothermal Institute, The University of Auckland.

Gudmundsson, J.S., Bott, T.R., (1979): "Deposition of Silica From Geothermal Waters On Heat Transfer Surfaces," Desalination Transactions, 28 (1979) pp. 124-145.

Gudmundsson, J.S.,(1987): "The Elements of Direct Uses," United Nations Institute for Training and Research, Workshop on Small Geothermal Resources, May 11-22, 1987, Pisa, Italy.

Gimenez, E.C., (1987): "Reports on Technical Feasibility Study of Agricultural Drying Using Geothermal Energy in the Philippines," Unpublished Internal Report of PNOC-EDC (E/C Geothermal Division).

Gimenez, E.C., (1987): "Reports on Technical Feasibility Study of Heat Exchanger Selection for Geothermal Application in the Philippines," Unpublished Internal Report of PNOC-EDC (E/C Geothermal Division).

Gimenez, E.C., (1987): "Fish Farming Using Geothermal Energy, the Philippine Experience," Unpublished Internal Report of PNOC-EDC (E/C Geothermal Division).

Harrison, R., Mortimer N.D., (1985): "Handbook on the Economics of Low Enthalpy Geothermal Energy Developments," Sunderland Polytechnic Energy Workshop, Vol. 2 Part III.

Holman, J.P., (1976): "Heat Transfer," McGraw-Hill, New York, NY, 4th edition, 1976.

Jónsson, V.K., (1987): Personal communications.

Kakac, S., Bergles, A.E., Mayinger, F., (1981): "Heat Exchangers: Thermal-Hydraulic Fundamentals and Design". McGraw-Hill, New York, NY, 1981.

Kakac, S., Shah, R.K., Bergles, A.E., (1983): "Low Reynolds Number Flow Heat Exchangers," Hemisphere Publishing, New York, NY, 1983.

Klaren, D.G., Maaskant, B., (1982): "Fluid Bed Heat Exchanger Principle and Non-fouling Performance on Iceland Geothermal Brines," Pacific Geothermal Conference 1982, Auckland, New Zealand.

Klaren, D.G., (1981): "Fluid Bed Heat Exchangers - A New Approach In Severe Fouling Heat Transfer," Esmil Research, Ijmuiden, The Netherlands, 1981.

Michaelides, E.E., (1981): "Thermodynamic Properties of Geothermal Fluids," Geothermal Resources Council, Transactions, Vol. 5, October 1981.

Serghides, T.K., (1984): "Estimate Friction Factor Accurately," Kerr-McGee Corp., Chemical Engineering March 5, 1984.

Tolentino, B.S., (1985): "Lectures of Geothermal Energy in the Philippines," UNU Geothermal Training Programme Report No. 12.

Valdimarsson, P. (1987): Personal communications.

APPENDIX 1. Equations for properties of water and steam

The properties of steam appear in steam tables (Keenan et al). Correlation of these properties were derived in order to make numerical computations easier. The following correlations were fitted to data from Keenan et al tables for the values of these properties at saturated conditions.

$$hs = 2472.048 + 2.72064^{T} - 93.85e^{-4}T^{2} + 33.695e^{-6}T^{3} - 7.451e^{-8}T^{4} \text{ , kJ/kg} \tag{1}$$

where T is in °C. The maximum error in this equation is less than 2% up to temperatures of 320°C.

The specific heat of the hot and cold fluid is:

$$cp = 4.205-(1.555E-03*T)+(2.7929E-05*T^{2})-(1.586E-07*T^{3}) + (5.0E-10*T^{4}) \tag{2}$$

The enthalpy is found by integration:

$$h = \int_{0}^{T} c_{p} dT \tag{3}$$

Thermal conductivity is expressed as:

$$k = 0.569+(0.001833*T)-(7.0E-06*T^{2})-(T^{3}*2.78E-09)+(T^{4}*1.4953E-11) \tag{4}$$

The dynamic viscosity is expressed as:

$$\mu = ((1697.58)-(46.1926T)+(0.69629T^{2})-(T^{3}*5.819E-03)+(24.8E-06*T^{4})-(41.9E-09*T^{5}))/10^{6} \tag{5}$$

The densities are expressed as follows:

$$\rho_{c} = 1003.408-(0.196973*T)-(T^{2}*0.002524983) \tag{6}$$

APPENDIX 2. Heat Transfer Performance of a Fluidized Bed Heat Exchanger

In the correlation used for the wall to liquid heat transfer coefficient of a fluidized bed, the superficial velocity of the liquid in the tubes is an important parameter. The way this velocity $U_{l,s}$ is calculated in this presented equation:

$$U_{l,s} = U_i \, \alpha^{\,n} \tag{7}$$

where:

α = porosity in the fluidized bed

n = empirical factor = 2.4

$U_i = U_\infty \, 10^{-dp/Di}$

and dp is the particle diameter and U_∞ is the terminal velocity of one single particle falling in a stagnant liquid with which the tube would normally be filled and which must behave as an infinite fluidum. For spherical particles, the terminal velocity can be determined from the equations:

$$CD_\infty Re^2 dp_\infty = 4/3 \; dp^3 \rho_l g(\rho_s - \rho_l)/nl^2 \tag{8}$$

In the above equations the symbols have the following meaning: CD_∞ = drag coefficient = 0.44, and nl = dynamic viscosity of the liquid.

For the wall to liquid heat transfer coefficient of a fluidized bed heat exchanger, it has been found by Klaren that the correlation of Ruckenstein showed a good fit with his experimental results obtained over the past years. The relevant part of this correlation reads as follows:

$$Nu = 0.067 \; Pr^{0.33} \; Re_{dp}^{-0.237} \; Ar^{0.522} \tag{9}$$

In the above equations the dimensionless numbers are composed as follows:

Nusselt number, $Nu = dp\, \beta_l / \gamma_l$ (10)

Prandtl number, $Pr = n_l\, c_l / \gamma_l$ (11)

Reynolds number, $Re_{dp} = \rho_l\, U_{l,s}\, dp / n_l$ (12)

Archimedes number, $Ar = gdp^3 \rho_l\, (\rho_s - \rho_l) / n_l^2$ (13)

where β_l = wall to liquid heat transfer coefficient and γ_l = heat conductivity of the liquid.

The friction factor equations used in pressure drop calculations are defined as follows:

$$f = [(A-(B-A)^2/C-2B+A]^{-2} \tag{14}$$

where:

$A = -2.0 \log [kmd/Di/3.7 + 12/Re]$ (15)

$B = -2.0 \log [kmd/Di/3.7 + 2.51A/Re]$ (16)

$C = -2.0 \log [kmd/Di/3.7 + 2.51B/Re]$ (17)

Di = inside diameter if the tube

kmd= roughness factor

This explicit friction factor equation according to Serghides is valid for transitional and turbulent flow (Re>2,100) at any relative roughness (kmd/Di).

APPENDIX 3. Program listing

```
PROGRAM FBHX (input, output) ;

CONST
     pi = 3.14159 ;
     aa = 1003.408 ;     { constants in properties correlations }
     bb = 0.196973 ;                  {     - " -     }
     cc = 0.00252498 ;                {     - " -     }
     ll = 4.205 ;                     {     - " -     }
     mm = 0.001555 ;                  {     - " -     }
     nn = 0.000027929 ;               {     - " -     }
     oo = 0.0000001586 ;              {     - " -     }
     pp = 0.0000000005 ;              {     - " -     }
     Krough = 0.00005 ;     { friction coefficient }

VAR
   mc,      { cold fluid mass flow }
   tc1,     { cold fluid outlet temperature }
   tc2,     { target outlet temperature of outlet fluid }
   mh,      { hot fluid mass flow }
   th1,     { hot fluid inlet temperature }
   th2,     { hot fluid outlet temperature }
   A,       { heat exchanging area }
   U,       { overall heat transfer coefficient for clean hx }
   rf,      { fouling factor }
   di,      { inside diameter of tubes }
   de,      { external diameter of tubes }
   N,       { number of tubes }
   L,       { length of tubes }
   Np,      { number of passes }
   Sp,      { spacing of tubes }
   dp,      { diameter of particles }
   Sgp,     { specific gravity of particles }
   LMTDt,   { target log mean temperature difference }
   LMTD,    { log mean temperature difference }
   Th2T,    { target hot fluid outlet temperature }
   NTU,     { number of heat transfer units }
   NTUf,    { target number of heat transfer units }
   kt,      { tube material conductivity }
   Eps,     { epsilon }
   C,       { heat capacity flow ratio }
   Cmin,    { heat capacity flow of minimum fluid }
   Nb,      { number of baffles }
   Tc1T,    { target outlet temperature of cold fluid }
   qTar,    { target heat duty based on cold fluid }
   AFlowTubes, { exchanging area of tubes }
   AFlowShell, { exchanging area of shell }
   Ud,      { overall heat transfer coefficient for fouled hx }
   Ar,      { archimedes number }
   Vh,      { velocity of hot fluid }
   Vc,      { velocity of cold fluid }
   REc,     { reynolds number of cold fluid }
   REh,     { reynolds number of hot fluid }
   PRc,     { prandtl number of cold fluid }
```

```
   PRh,    { prandtl number of hot fluid }
   NUc,    { nusselt number of cold fluid }
   NUh,    { nusselt number of hot fluid }
   Hc,     { heat transfer coefficient on cold side }
   Hh,     { heat transfer coefficient on hot side }
   q,      { heat duty }
   Ds,     { diameter of shell }
   hb,     { height of baffle }
   Nt,     { number of tubes excluded due to baffle's opening }
   Ab,     { area of the baffle }
   Uz,     { superficial velocity of the liquid in tubes }
   fcold,  { factor for computing cold side pressure drop }
   fhot,   { factor for computing hot side pressure drop }
   dpcold,{ cold side pressure drop }
   dphot,  { hot side pressure drop }
   Se,     { pressure drop correction factor }
   Sb,     { pressure drop correction factor }
   Nw,     { pressure drop constant }
   Nc,     { area occupied by tubes }
   Refh,   { Reynolds number at FBHX }
   IIA,    { friction factor coefficient }
   IIB,    { friction factor coefficient }
   IIC,    { friction factor coefficient }
   kmd,    { roughness factor }
   Flbed,  { friction factor at FBHX }
   dpw,    { pressure drop coefficient }
   RhoT,   { density at reference temperature }

   Ul,x,y,th,tc,poro,ao,fcold : real ;
   dummy : char ;
   skra : text ;
   i : integer ;


FUNCTION POWER (x,y:real) : real ;
   BEGIN
        Power:=exp (y*ln(x));
   END;


FUNCTION rho(t:real) :real ;    { specific weight of water }
   BEGIN
     rho:=aa-bb*t-cc*t*t ;
   END ;


FUNCTION my(t:real) :real ;     { viscosity of water }
  BEGIN

my:=(1697.58-46.1926*t+0.69629*t*t-5.819e-3*t*t*t+24.8e-6*t*t*t*t-4
e-9*t*t*t*t*t)*1e-6 ;
  END ;
```

```
FUNCTION k(t:real) :real ;      { conductivity of water }

  BEGIN
  k:=0.569+1.833e-3*t-7.299e-6*t*t-0.278e-9*t*t*t+0.01495e-9*t*t*t*
  END ;


FUNCTION h(t:real) :real ;      { enthalpy of water }
  BEGIN
   h:=(ll-(mm*t)/2+(nn*t*t)/3-(oo*t*t*t)/4+(pp*t*t*t*t)/5)*t;
 END ;


FUNCTION cp(t:real) :real ;     { heat capacity of water }
  BEGIN
   cp:=(ll-(mm*t)+(nn*t*t)-(oo*t*t*t)+(pp*t*t*t*t))*1000 ;
  END ;



Procedure FluidData ;      { input routine for fluid data }
BEGIN

     Writeln('FLUIDIZED BED HEAT EXCHANGER') ;
     Writeln('Note - secondary fluid is on the shell side') ;
     Writeln('Counterflow') ;


{data for esmil 1 :}
mc:=12.06;            { cold mass flow [ kg/s ] }
tc2:=5.2;               { cold fluid inlet temperature [ C ] }
th1:=91.0;              { hot fluid inlet temperature [ C ] }
Th:=Th1 ;               { hot fluid reference temperature [ C ] }
Poro:=0.7 ;             { porosity of particles }
{ data for esmil 2 :}
{mc:=12.165;
tc2:=66.6;
tc1T:=90.8;
th1:=158.7;
Th2T:=91.2 ;
Th:=Th1 ;
Tc:=Tc2 ;}

END;


Procedure HEData ;
BEGIN
     {Writeln('enter di:') ;
     Readln(di) ;
     Writeln('enter de:') ;
     Readln(de) ;
     Writeln('enter N:') ;
     Readln(N) ;
     Writeln('enter L:') ;
     Readln(L) ;
```

```
      Writeln('enter Np:') ;
      Readln(Np) ;
      Writeln('enter number of baffles on shell side - Nb:') ;
      Readln(Nb) ;
      Writeln('enter spacing of tubes - Sp:') ;
      Readln(Sp) ;
      Writeln('enter conductivity of tubes - kt:') ;
      Readln(kt) ;
      Writeln('enter fouling factor - rf:') ;
      Readln(rf) ;
      Writeln('enter diameter of particle - dp:') ;
      Readln(dp) ;
      Writeln('enter specific density of particle - Sgp:') ;
      Readln(Sgp) ;
      Writeln(' ') ;}
      di:=0.035;        { inside diameter of tubes [ m ] }
      de:=0.038;        { outside diameter of tubes [ m ] }
      N:=18;            { number of tubes [ - ] }
      L:=10.7;          { length of tubes [ m ] }
      Np:=1;            { number of passes [ - ] }
      Nb:=11;           { number of baffles [ - ] }
      Sp:=0.042;        { spacing of tubes [ m ] }
      kt:=53;           { conductivity of tube material [ W/(m*C)/W ]
      dp:=0.0045 ;      { diameter of particle [ m ] }
      Sgp:=2.7;         { specific gravity of particle [ - ] }
      Se:=1 ;
      Sb:=1 ;
      Nw:=1 ;
      Ds:=0.3 ;
      Nt:=3 ;
      hb:=0.225 ;

END;


Procedure  InitialValues ;
BEGIN
  LMTDt:= ((Th1-Tc1T)-(Th2T-tc2))/(ln((Th1-Tc1T)/(Th2T-tc2))) ;
  A:=n*pi*l*di ;                        { heat exchanging area }
  Ao:=a*de/di ; Writeln(a,ao);     { outside heat exchanging area }
  AFlowTubes:=di*di*pi/4.0*N/Np ;{ heat exchanging area of tubes }
  AFlowShell:=(Int(Sqrt(N)))*(Sp-de)*L/(Nb+1) ;
                                  { heat exchanging area of shell }
  qTar:=mc*(h(tc1t)-h(tc2)) ;          { target heat duty [ KW ] }
  C:=0.99 ;                             { target Cp flow ratio }
  Cmin:=mc*cp(tc) ;    { cold side (minimum) heat capacity flow }
  mh:=mc*cp(tc)/cp(th)/C ;  { hot mass flow initial guess }
  Eps:=(Tc1T-tc2)/(th1-tc2) ;          { target effectiveness }

END;

Procedure Velocities ;
BEGIN
   Vh:=mh/rho(th)/AFlowTubes ;              { hot side velocity }
   Vc:=mc/rho(Tc)/AFLowShell ;              { cold side velocity }
```

```
   Ul:=Power(poro,2.4)*Power(10,-dp/di)
       *sqrt(3.0303*dp*9.81*(Sgp-1)) ;    { particle velocity }
 END;

Procedure Numbers ;
BEGIN
     REh:=Ul*dp*rho(th)/my(th) ;       { hot side reynolds number }
     REc:=Vc*de*rho(Tc)/my(Tc) ;       { cold side reynolds number }
     PRh:=cp(th)*my(th)/k(th) ;        { hot side prandtl number }
     PRc:=cp(Tc)*my(Tc)/k(Tc) ;        { cold side prandtl number }

  If REc<1000 then                     { xxx correlation }
     Nuc:=(0.43+0.5*Power(REc,0.5))*Power(PRc,0.38)
  Else
     Nuc:=0.25*Power(REc,0.6)*Power(PRc,0.38) ; { nusselt number on
                                             cold side calculated }
     Ar:=9.81*dp*dp*dp*rho(Tc)*rho(Tc)*(Sgp-1)/(my(th)*my(th)) ;
                                 { archimedes number }
     Nuh:=0.067*Power(REh,-0.237)*Power(PRh,0.33)*Power(Ar,0.522) ;
                    { nusselt number on hot side calculated }
     Hc:=NUc*k(Tc)/de ; { heat transfer coefficient on cold side
                         calculated }
     Hh:=NUh*k(th)/dp ;  { heat transfer coefficient on hot side
                         calculated }

END;


Procedure NTUcalc ;
BEGIN
     U:=1/(1/Hh + A*ln(de/di)/(2*pi*kt*L*N) + di/(de*Hc)) ;
          { overall heat transfer coefficient calculated }
     Ud:=1/(rf+1/U) ;    { for fouled heat exchanger }
     NTU:=Ud*A/Cmin;     { number of heat transfer units calculated

END;



Procedure Effectiveness ;
BEGIN
     Eps:=(1-exp(-NTU*(1-C)))/(1-C*exp(-NTU*(1-C))) ;

     tc1:=tc2+Eps*(th1-tc2) ;  { actual cold outlet temperature
calculated }
     If (mc*cp(tc2)<mh*cp(th1)) then
     BEGIN
       tc1:=tc2+Eps*(th1-tc2) ;
       q:=mc*(h(tc1)-h(tc2)) ; { actual heat duty on cold side
calculated }
       th2:=th1-(q*1000/(mh*cp(th))) ;{ actual outlet temperature
                                       calculated }
     end
   Else
     BEGIN
       th2:=th1-Eps*(th1-tc2) ;
```

```
        q:=mh*(h(th1)-h(th2)) ;
        tc1:=tc2+(q*1000)/(mc*cp(tc)) ;
      end;
END;

Procedure DataOut ;
BEGIN
     Writeln('FLUIDIZED BED HEAT EXCHANGER');
     Writeln('PERFORMANCE DATA') ;
     Writeln(' ') ;
     Writeln('qTar = ',qTar:10:2) ;
     Writeln('   q = ',q:10:2) ;
     Writeln('  mc = ',mc:10:2) ;
     Writeln(' tc1 = ',tc1:10:2) ;
     Writeln(' tc2 = ',tc2:10:2) ;
     Writeln('Tc1T = ',Tc1T:10:2) ;
     Writeln('  mh = ',mh:10:2) ;
     Writeln(' th1 = ',th1:10:2) ;
     Writeln(' th2 = ',th2:10:2) ;
     Writeln(' Th2T=',Th2T:10:2) ;
     Writeln(' Eps = ',Eps:10:2) ;
     Writeln(' NTU = ',NTU:10:2) ;
     Writeln('  Ud = ',Ud:10:2) ;
     Writeln('LMTD = ',LMTD:10:2) ;
     Writeln('   C = ',C:10:2) ;
     Writeln('Cmin = ',Cmin:10:2) ;
     Writeln('  Vh = ',Vh:10:2) ;
     Writeln('  Ul = ',Ul:10:2) ;
     Writeln('  Vc = ',Vc:10:2) ;
     GotoXY(40,5)  ;  Write('   kmd =',kmd:10:4) ;
     GotoXY(40,6)  ;  Write('  Refh =',Refh:10:4) ;
     GotoXY(40,7)  ;  Write('   dpw =',dpw:10:4) ;
     GotoXY(40,8)  ;  Write('  RhoT =',RhoT:10:4) ;
     GotoXY(40,9)  ;  Write(' Flbed =',Flbed:10:4) ;
     GotoXY(40,10) ;  Write(' fcold =',fcold:10:4) ;
     GotoXY(40,11) ;  Write('dpcold =',dpcold/1e5:10:4) ;
     GotoXY(40,12) ;  Write('  fhot =',fhot:10:4) ;
     GotoXY(40,13) ;  Write(' dphot =',dphot/1e5:10:4) ;
     GotoXY(40,14) ;  Write('   REh = ',REh:10:4) ;
     GotoXY(40,15) ;  Write('   REc = ',REc:10:4) ;
     GotoXY(40,16) ;  Write('   PRc = ',PRc:10:4) ;
     GotoXY(40,17) ;  Write('   PRh = ',PRh:10:4) ;
     GotoXY(40,18) ;  Write('   NUc = ',NUc:10:4) ;
     GotoXY(40,19) ;  Write('   NUh = ',NUh:10:4) ;
     GotoXY(40,20) ;  Write('    Ar = ', Ar:10:4) ;
     GotoXY(40,21) ;  Write('    Hc = ',Hc:10:4) ;
     GotoXY(40,22) ;  Write('    Hh = ',Hh:10:4) ;
      GotoXY(40,24) ; IF mc*cp(tc)<mh*cp(th) THEN Write('cold is
minimum') ELSE Write('hot is minimum');
{     Writeln('a = ',a) ;
     Writeln('rf = ',rf) ;
     Writeln('q2 = ',q2) ;
     Writeln('di = ',di) ;
     Writeln('de = ',de) ;
     Writeln('n = ',n) ;
     Writeln('l = ',l) ;
```

```
      Writeln('np = ',np) ;
      Writeln('Nb = ',Nb) ;
      Writeln('Sp = ',Sp) ;
      Writeln('kt = ',kt) ;
      Writeln('dp = ',dp) ;
      Writeln('Sgp = ',Sgp) ;
      Writeln('AFlowTubes = ',AFLowTubes) ;
      Writeln('AFlowShell = ',AFlowShell) ;
      Writeln('Ud=', Ud) ;
      Writeln('NTUf = ',NTUf) ;
      Writeln('LMTDt =',LMTDt) ;
      Writeln('Eps1= ', Eps1) ;
      Writeln('my(tc) = ',my(tc)) ;
      Writeln('my(th) = ',my(th)) ;
      Writeln('ro(th) = ',ro(th)) ;
      Writeln('ro(tc) = ',ro(tc)) ;
      Writeln('cpc = ',cp(tc)) ;
      Writeln('cph = ',cp(th)) ;
      Writeln('k(tc1) = ',k(tc)) ;
      Writeln('k(th1) = ',k(th)) ;
      Writeln('h(tc1t) = ',h(tc1t)) ;
      Writeln('h(tc2) = ',h(tc2)) ;
      Writeln('h(tc1t) = ',h(tc1t)) ;
      Writeln('Tc1T = ',Tc1T) ;
      Writeln('tc1 = ',tc1) ;
      Writeln('mh = ',mh) ;
      Writeln('mhi=',mhi) ;
      Writeln('th1=',th1) ;
      Writeln('Th2T= ', Th2T) ;
      Writeln('Th2= ',Th2) ;
      Writeln('qtar=',qtar) ;
      Writeln('q1=',q1) ;
      Writeln('q2=',q2) ;
      Writeln('LMTDt=',LMTDt) ;
      Writeln('LMTD=',LMTD) ;
      Writeln('NTU =', NTU) ;
      Writeln('NTUf = ',NTUf) ;
      Writeln(' ') ;}

END;

Procedure Deltap ;
BEGIN

kmd:=Krough/di ;
Refh:=Vh*di*rho(th)/my(th) ;
IIA := -2*ln(kmd/3.7 + 12/Refh)/ln(10) ;
IIB := -2*ln(kmd/3.7 + 2.51*IIA/Refh)/ln(10) ;
IIC := -2*ln(kmd/3.7 + 2.51*IIB/Refh)/ln(10) ;

fhot:= 1/(IIA - (IIB-IIA)*(IIB-IIA)/(IIC-2*IIB+IIA))
         /(IIA - (IIB-IIA)*(IIB-IIA)/(IIC-2*IIB+IIA)) ;

dpw := fhot*L/di*rho(th)*Vh*Vh/2 ;
RhoT:= rho(th)*(poro+(1-poro)*Sgp) ;
Flbed:=1.5*9.81*di*(Sgp-1)/Vh/Vh ;
```

```
dphot:=dpw*(1+81*(1-poro)*Power(Flbed,1.5)) + RhoT*L*9.81 ;
Ab:=((pi/4*Ds)*(Ds-hb))-((Nt*pi/4)*de*de) ;
fcold:=0.174+0.32*Sp/de/(Power((Sp/de-1),(0.43+1.13*de/Sp)))*
       Power(Rec,-0.15) ;
Uz:=Vc*mc/(Ab*Rho(tc)) ;
Nc:=Int(Sqrt(N)) ;
dpcold:=((Nb-1)*fcold*Nc*Rho(tc)*Vc*Vc*Sb/2+Nb*(2+0.6*Nw)*Rho(tc)*
         Uz*Uz/2)*Se+fcold*Nc*Rho(tc)*Vc*Vc*Sb*(1+Nw/Nc);
     Writeln( ' ' ) ;
     Writeln( ' ' ) ;
     Writeln('fhot = ',fhot) ;
     Writeln('dphot = ',dphot/1e5) ;
     Writeln('fcold = ',fcold) ;
     Writeln('dpcold = ',dpcold/1e5) ;

END;

BEGIN
     Assign(skra,'FBHX.OUT') ;
     Rewrite(skra);
     FluidData ;
     HEData ;
     InitialValues ;

     i:= 1 ;

Repeat
  begin
           Velocities ;
           Numbers ;
           NTUcalc ;
           Effectiveness ;
           mh:=mh*qtar/q ;
           tc:=(tc2+tc1)/2 ;
           th:=(th1+th2)/2 ;
           C:=mh*cp(th)/(mc*cp(tc));
           If c>1 then c:=1/c ;
           LMTD:=((Th1-tc1)-(Th2-tc2))/(ln((Th1-tc1)/(Th2-tc2))) ;
            DataOut ;
{           Readln(dummy);}
   end;
until (abs(tc1-Tc1T)<0.01);

If (abs(tc1-Tc1T)>0.01) then writeln('no convergence');
Deltap ;

           NTUf:=(Tc1-Tc2)/LMTD ;
           DataOut;
           Close(skra) ;


END.
```

```
PROGRAM PHE (input, output) ;

CONST
     pi = 3.14159 ;
     aa = 1003.408 ;       { constants in properties correlations }
     bb = 0.196973 ;                 {     - " -     }
     cc = 0.00252498 ;               {     - " -     }
     ll = 4.205 ;                    {     - " -     }
     mm = 0.001555 ;                 {     - " -     }
     nn = 0.000027929 ;              {     - " -     }
     oo = 0.0000001586 ;             {     - " -     }
     pp = 0.0000000005 ;             {     - " -     }
     qq = 0.001885 ;                 {     - " -     }
     rr = 1.74598 ;                  {     - " -     }
     ss = 0.000012453 ;              {     - " -     }
     tt = 0.0045137 ;                {     - " -     }

VAR
   mc,      {  cold fluid mass flow }
   Tc1,     {  cold fluid outlet temperature }
   Tc1T,    {  target outlet temperature of outlet fluid }
   Tc2,     {  cold fluid inlet temperature }
   mh,      {  hot fluid mass flow }
   Th1,     {  hot fluid inlet temperature }
   Th2,     {  hot fluid outlet temperature }
   Th2T,    {  target outlet temperature for hot fluid }
   A,       {  heat exchanging area  }
   U,       {  overall heat transfer coefficient for clean hx  }
   rf,      {  fouling factor  }
   L,       {  plate length }
   w,       {  plate width }
   Sp,      {  spacing of plates }
   dh,      {  hydraulic diameter of flow channels }
   korr,    {  area correction factor due corrugations of plate }
   tp,      {  thickness of plate }
   kt,      {  plate material conductivity }
   Np,      {  number of plates }
   qh,      {  heat transferred from hot fluid }
   hout,    {  enthalpy of outgoing hot fluid ( condensate) }
   NTU,     {  number of transfer units }
   Eps,     {  epsilon }
   C,       {  heat capacity flow ratio }
   Cmin,    {  heat capacity flow of minimum fluid }
   qTar,    {  target heat duty based on cold fluid }
   Ud,      {  overall heat transfer coefficient for fouled hx  }
   REc,     {  reynolds number of cold fluid }
   PRc,     {  prandtl number of cold fluid}
   NUc,     {  nusselt number of cold fluid }
   Hc,      {  heat transfer coefficient on cold side }
   Hh,      {  heat transfer coefficient on hot side }
   q,       {  heat duty }
   Th,      {  temperature for evaluation fo properties on hot side }
   Tc,      {  temperature for evaluation fo properties on cold side
   Vc,      {  velocity on the cold side }
   fcold,   {  friction factor on the cold side }
   dpcold,{    pressure drop on the cold side }
```

```
 x,y : real ;
dummy : char ;
skra : text ;
i : integer ;

FUNCTION Power (x,y:real) : real ;
BEGIN
     Power:=exp (y*ln(x));
END;

FUNCTION Rho(t:real) :real ;    { specific weight of water }
BEGIN
     Rho:=aa - bb*t - cc*t*t ;
END ;

FUNCTION My(t:real) :real ;     { viscosity of water }
BEGIN
     My:=(1697.58 - 46.1926*t + 0.69629*t*t - 5.819e-3*t*t*t
         + 24.8e-6*t*t*t*t - 41.9e-9*t*t*t*t*t)*1e-6 ;
END ;

FUNCTION K(t:real) :real ;      { conductivity of water }
BEGIN
     K:=0.569 + 1.833e-3*t - 7.299e-6*t*t - 0.278e-9*t*t*t
        + 0.01495e-9*t*t*t*t ;
END ;

FUNCTION H(t:real) :real ;      { enthalpy of water }
BEGIN
     H:=(ll - (mm*t)/2 + (nn*t*t)/3 - (oo*t*t*t)/4 +
        (pp*t*t*t*t)/5)*t;
END ;

FUNCTION Hs(t:real) :real ;     { enthalpy of steam }
BEGIN
     Hs:=2472.048 + 2.72064*t - 93.850e-4*t*t + 33.695e-6*t*t*t
         - 7.451e-8*t*t*t*t ;
END ;

FUNCTION Cp(t:real) :real ;      { heat capacity of water }
BEGIN
     Cp:=(ll - mm*t + nn*t*t - oo*t*t*t + pp*t*t*t*t)*1000 ;
END ;

Procedure FluidData ;   { input routine for fluid data }
BEGIN

  Writeln('PLATE HEAT EXCHANGER') ;
  Writeln('Counterflow with condensing steam as hot fluid') ;

  {data for plate heat exchanger}
        mc:=7.69;       { cold mass flow [ kg/s ] }
       Tc2:=23.3;     { cold fluid inlet temperature [ C ] }
       Th1:=120;    { hot fluid inlet temperature [ C ] }

END;
```

```
Procedure HEData ;      { input routine for hx data }
BEGIN

  Np:=52;               { number of plates [ - ], if odd then Np+1
  L:=1.4;               { plate lenght [ m ] }
  Sp:=0.007;            { spacing of plates [ m ] }
  w:=0.64 ;             { width of plates [ m ] }
  kt:=16.3;             { conductivity of plate material
                         [ W/(m*C)/W ] }
  tp:=0.0074 ;          { thickness of plates [ m ] }

END;

Procedure  InitialValues ;
BEGIN

  A:=Np*w*l*korr ;       { heat exchanging area}
  dh:=2.0*Sp/korr ;      { cold side passage hydraulic diameter }
  qTar:=mc*(H(Tc1t)-H(Tc2)) ; { target heat duty based
                                on cold side }
  mh:=qTar/(Hs(Th1)-H(Th2T)) ;{ hot mass flow inital guess no
                                subcooling }
  Eps:=(Tc1T-Tc2)/(Th1-Tc2) ; { target effectiveness }
  Tc:=(Tc1T+Tc2)/2 ;     { cold side properties reference
                           temperature }
  Th:=Th1 ;  { hot side properties reference temperature }
  Cmin:=mc*Cp(Tc) ; { cold side (minimum) heat capacity flow }
  C:=mc*Cp(Tc)*(Th1-Th2t)/qTar/1000 ;{ target Cp flow ratio }

END;

Procedure Numbers ; { routine for calculation of heat transfer
                      coefficients}
BEGIN

  REc:=4*mc/My(Tc)/(w+sp)/Np ; { cold side reynolds number }
  PRc:=Cp(Tc)*My(Tc)/K(Tc)  ;   { cold side prandtl number }

  IF REc<150 THEN               { xxx correlation }
        Nuc:=0.406*Power(REc,0.425)*Power(PRc,0.35)
  ELSE
        Nuc:=0.298*Power(REc,0.646)*Power(PRc,0.316) ;{APV HX}
       { Nuc:=0.430*Power(REc,0.591)*Power(PRc,0.303) ;}
       {Alfa Laval P13}

  Hc:=NUc*K(Tc)/dh ;  { heat transfer coefficient on cold side
calculated }
  Hh:=10000 ;{ constant coefficient assumed on hot side for
             condensing steam }

END;
```

```
Procedure NTUcalc ;
BEGIN

  U:=1/(1/Hh + tp/kt + 1/Hc) ; { overall heat transfer coefficie
                                 calculated }
  Ud:=1/(rf+1/U) ;              { for fouled heat exchanger }
  NTU:=Ud*A/Cmin;       { numbers of transfer units calculated }
END;

Procedure Deltap ;
BEGIN
  fcold:= 8*Nuc/(REc*Power(PRc,0.333)) ;
  Vc:=2*mc/rho(tc)/Np/(Sp*w) ;
  dpcold:=fcold*L/Dh*1/2*rho(tc)*Vc*Vc ;
Writeln('nu= ',nuc,'re= ',rec,'pr= ',prc) ;
Writeln('vc= ',vc) ;
END;

Procedure Effectiveness ; { calculation of effectiveness }
BEGIN

  Eps:=(1-exp(-NTU*(1-C)))/(1-C*exp(-NTU*(1-C))) ;
  Tc1:=Tc2+Eps*(Th1-Tc2) ; { actual cold outlet temp calculated
  q:=mc*(H(Tc1)-H(Tc2)) ;  { actual q on cold side calculated }
  hout:=Hs(Th1)-q/mh;      { actual outlet enthalpy on hot side
                             calculated }
  IF (hout<H(Tc2)) THEN hout:=H(Tc2) ; { lower bound of enthalpy
                 set to prevent numerical instability }
END;

Procedure DataOut ;
BEGIN
 Clrscr;
  Writeln('PLATE HEAT EXCHANGER');
  Writeln('PERFORMANCE DATA') ;
  Writeln(' ') ;
  Writeln('   Target heat duty : ',qTar:10:1) ;
  Writeln('Cold side heat duty : ',q:10:1) ;
  Writeln(' Hot side heat duty : ',qh:10:1) ;
  Writeln('                Cold mass flow : ',mc:5:4) ;
  Writeln('        Cold inlet temperature : ',Tc2:5:4) ;
  Writeln('       Cold outlet temperature : ',Tc1:5:4) ;
  Writeln('Target cold outlet temperature : ',Tc1T:5:4) ;
  Writeln('                Hot mass flow : ',mh:5:4) ;
  Writeln('        Hot inlet temperature : ',Th1:5:4) ;
  Writeln('       Hot outlet temperature : ',Th2:5:4) ;
  Writeln('Target hot outlet temperature : ',Th2T:5:4) ;
  Writeln(' Friction factor on cold side : ',fcold:5:4) ;
  Writeln('    Presure drop on cold side : ',dpcold:5:4) ;
  Writeln('Eps = ',Eps) ;
  Writeln('NTU = ',NTU) ;
  Writeln('Ud = ',Ud) ;
  Writeln('Hc = ',Hc) ;
  Writeln('C = ',C) ;
  Writeln('Cmin = ',Cmin) ;
END;
```

```
{ main program begins }

BEGIN
     FluidData ;
     HEData ;
     InitialValues ;
{ iteration starts }
     Repeat
     begin
           Numbers ;
           NTUcalc ;
           Effectiveness ;
           IF hout<H(Th1) THEN
                          BEGIN
                             Th2:=Th1-(H(Th1)-hout)/Cp(Th)*1000.0 ;
                          END
                      ELSE
                        BEGIN
                          Th2:=Th1 ;
                        END ;
           mh:=mh*(Hs(Th1)-hout)/(Hs(Th1)-H(Th2T)) ;
           IF (mh>qTar/(Hs(Th1)-H(Th1))) THEN mh:=qTar/
               (Hs(Th1)-H(Th1));
           Qh:=mh*(Hs(Th1)-hout) ;
           Tc:=(Tc2+Tc1)/2 ;
           Th:=(Th1+Th2)/2 ;
           C:=mc*Cp(Tc)*(Th1-Th2T)/Qh/1000 ;
           IF c>1 THEN c:=0.99 ;
           DataOut ;
    Deltap ;
           Readln(dummy);
        end;
until (abs(Th2-Th2T)<0.01);
END.
```

APPENDIX 4. Computer outputs

```
FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =    2330.94
  mc =      12.06
 tc1 =      51.40
 tc2 =       5.20
Tc1T =      23.10
  mh =       4.71
 th1 =      91.00
 th2 =      45.42
 Th2T=     36.30
 Eps =       0.54
 NTU =       1.16
  Ud =    2773.29
LMTD =      39.90
   C =       0.39
Cmin =   50623.63
  Vh =       0.73
  Ul =       0.15
  Vc =       0.84
   kmd =    0.0000  Refh =    0.0000   dpw =    0.0000  RhoT =    0.0000 Flbed =    0.0001 fcold =    0.0000dpcold =-9522963[illegible]
t =    0.0000 dphot =-638855511910.0000   REh =  2081.7403   REc = 21771.9267   PRc =   10.7[illegible]87   PRh =    1.9560   NUc =  [illegible]
NUh =   77.3775    Ar = 15465978.0890    Hc =  3752.9612    Hh = 11626.8369FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =    1097.59
  mc =      12.06
 tc1 =      26.97
 tc2 =       5.20
Tc1T =      23.10
  mh =       3.88
 th1 =      91.00
 th2 =      35.40
 Th2T=     36.30
 Eps =       0.65
 NTU =       1.24
  Ud =    2953.07
LMTD =      45.01
   C =       0.32
Cmin =   50623.63
  Vh =       0.28
  Ul =       0.15
  Vc =       0.85
   kmd =    0.0000  Refh =    0.0000   dpw =    0.0000  RhoT =    0.0000 Flbed =    0.0001 fcold =    0.0000dpcold =-9522963863.4
t =    0.0000 dphot =-638855511910.0000   REh =  1599.7337   REc = 38645.9691   PRc =    5.6492   PRh =    2.5308   NUc =  272
NUh =   67.5245    Ar = 8765109.2003    Hc =  4416.4640    Hh =  9908.1451FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =     891.24
  mc =      12.06
 tc1 =      22.85
 tc2 =       5.20
Tc1T =      23.10
  mh =       3.94
 th1 =      91.00
 th2 =      36.20
 Th2T=     36.30
 Eps =       0.64
 NTU =       1.16
  Ud =    2777.69
LMTD =      47.16
   C =       0.33
Cmin =   50623.63
  Vh =       0.23
  Ul =       0.15
  Vc =       0.85
```

```
NUh =    65.[illegible]    Ar = [illegible]    Hc =  4007.42[illegible]    Hh =  9577.5099FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =     899.38
  mc =      12.06
 tc1 =      23.01
 tc2 =       5.20
TclT =      23.10
  mh =       3.96
 th1 =      91.00
 th2 =      36.49
Th2T=     36.30
 Eps =       0.64
 NTU =       1.15
  Ud =    2754.86
LMTD =      47.29
   C =       0.33
Cmin =   50623.63
  Vh =       0.23
  Ul =       0.15
  Vc =       0.85
   kmd =     0.0000  Refh =     0.0000   dpw =     0.0000  RhoT =     0.0000 Fibed =     0.0001 fcold =     0.0000dpcold =-9522963869.400
t =     0.0000 dphot =-638855511910.0000   REh =  1500.5796   REc = 27418.7858   PRc =      8.2706   PRh =      2.8158   NUc =   256.7
NUh =    65.9202    Ar = 7802898.0673    Hc =  4007.7922    Hh =  9613.0151FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =     904.18
  mc =      12.06
 tc1 =      23.10
 tc2 =       5.20
TclT =      23.10
  mh =       3.96
 th1 =      91.00
 th2 =      36.50
Th2T=     36.30
 Eps =       0.64
 NTU =       1.15
  Ud =    2756.71
LMTD =      47.26
   C =       0.33
Cmin =   50623.63
  Vh =       0.23
  Ul =       0.15
  Vc =       0.85
   kmd =     0.0000  Refh =     0.0000   dpw =     0.0000  RhoT =     0.0000 Flbed =     0.0001 fcold =     0.0000dpcold =-9522963869.400
t =     0.0000 dphot =-638855511910.0000   REh =  1509.5609   REc = 27472.6074   PRc =      8.2526   PRh =      2.8095   NUc =   256.7
NUh =    65.9806    Ar = 7834710.7460    Hc =  4010.0230    Hh =  9623.8084FLUIDIZED BED HEAT EXCHANGER
PERFORMANCE DATA
qTar =     904.24
   q =     904.18
  mc =      12.06
 tc1 =      23.10
 tc2 =       5.20
TclT =      23.10
  mh =       3.96
 th1 =      91.00
 th2 =      36.50
Th2T=     36.30
 Eps =       0.64
 NTU =       1.15
  Ud =    2756.71
LMTD =      47.26
   C =       0.33
Cmin =   50623.63
  Vh =       0.23
  Ul =       0.15
  Vc =       0.85
   kmd =     0.0014  Refb =18170.7484   dpw =  238.4928  RhoT = 1480.6914 Flbed =    16.0998 fcold =     2.1822dpcold =     0.4628  fho
 0.0293 dphot =     5.3004   REh =  1509.5609   REc = 27472.6074   PRc =      8.2526   PRh =      2.8095   NUc =   256.7934   NUh =
06    Ar = 7834710.7460    Hc =  4010.0230    Hh =  9623.8084
```

```
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.9
 Hot side heat duty :        561.9
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0238
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.1501
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1423
    Presure drop on cold side : 698.0520
Eps =    5.3481564656E-01
NTU =    8.5353269504E-01
Ud =   2.2544785064E+03
hout =    4.0714435746E+02
C =   3.3423546842E-01
Cmin =    8.1649408388E+03
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.7
 Hot side heat duty :        561.7
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0084
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.2823
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1420
    Presure drop on cold side : 696.4401
Eps =    5.3469534325E-01
NTU =    8.5342313959E-01
Ud =   2.2541891321E+03
hout =    4.0766729245E+02
C =   3.3431081464E-01
Cmin =    8.1649408388E+03
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.7
 Hot side heat duty :        561.7
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0066
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.2970
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1421
    Presure drop on cold side : 696.7490
Eps =    5.3468192582E-01
NTU =    8.5341093217E-01
Ud =   2.2541568881E+03
hout =    4.0772561530E+02
C =   3.3431922011E-01
Cmin =    8.1649408388E+03
```

```
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.7
 Hot side heat duty :        561.7
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0064
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.2986
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1421
    Presure drop on cold side : 696.7834
Eps =    5.3468042911E-01
NTU =    8.5340957058E-01
Ud =   2.2541532916E+03
hout =    4.0773212121E+02
C =   3.3432015776E-01
Cmin =    8.1649408388E+03
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.7
 Hot side heat duty :        561.7
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0064
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.2988
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1421
    Presure drop on cold side : 696.7872
Eps =    5.3468026215E-01
NTU =    8.5340941869E-01
Ud =   2.2541528904E+03
hout =    4.0773284696E+02
C =   3.3432026236E-01
Cmin =    8.1649408388E+03
PLATE HEAT EXCHANGER
PERFORMANCE DATA
   Target heat duty :        523.2
Cold side heat duty :        561.7
 Hot side heat duty :        561.7
                Cold mass flow : 1.9540
        Cold inlet temperature : 6.3000
       Cold outlet temperature : 75.0064
Target cold outlet temperature : 70.3000
                Hot mass flow : 0.2423
        Hot inlet temperature : 134.8000
       Hot outlet temperature : 97.2988
Target hot outlet temperature : 111.8000
 Friction factor on cold side : 0.1421
    Presure drop on cold side : 696.7877
Eps =    5.3468024353E-01
NTU =    8.5340940175E-01
Ud =   2.2541528457E+03
hout =    4.0773292791E+02
C =   3.3432027403E-01
Cmin =    8.1649408388E+03
```

Table 1.a Effectiveness-NTU formulae

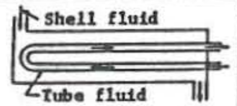
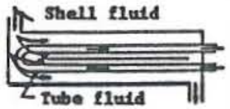
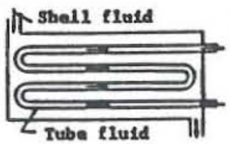
Flow Arrangement	ε-NTU Formulas	ε-NTU Formulas for $C^* = 1$	Asymptotic Value of ε when $NTU \to \infty$
 1-2 shell-and-tube exchanger; shell fluid mixed; TEMA E shell	$\varepsilon = \dfrac{2}{(1+C^*) + (1+C^{*2})^{1/2}\coth(\Gamma/2)}$ where $\Gamma = NTU[1+C^{*2}]^{1/2}$ $\coth(\Gamma/2) = (1+e^{-\Gamma})/(1-e^{-\Gamma})$	$\varepsilon = \dfrac{2}{2+\sqrt{2}\coth(\Gamma/2)}$ where $\Gamma = \sqrt{2}\,NTU$	$\varepsilon = \dfrac{2}{(1+C^*) + (1+C^{*2})^{1/2}}$
 1-2 shell-and-tube exchanger; shell fluid unmixed; TEMA E shell	If $C_{min} = C_{tube}$, $C_{max} = C_{shell}$, $\varepsilon = 1 - \dfrac{2C^*-1}{2C^*+1}\left[\dfrac{2C^*+\exp(-NTU(C^*+1/2))}{2C^*-\exp(-NTU(C^*-1/2))}\right]$ If $C_{min} = C_{shell}$, $C_{max} = C_{tube}$ $\varepsilon = \dfrac{1}{C^*} - \dfrac{2-C^*}{C^*(2+C^*)}\left[\dfrac{2+C^*\exp(-NTU(1+C^*/2))}{2-C^*\exp(-NTU(1-C^*/2))}\right]$	If $C_{min} = C_{tube}$ and $C^* = 1/2$ $\varepsilon = 1 - \dfrac{1+e^{-NTU}}{2+NTU}$ If $C^* = 1$ $\varepsilon = 1 - \dfrac{1}{3}\left[\dfrac{2+\exp(-\frac{3}{2}NTU)}{2-\exp(-\frac{1}{2}NTU)}\right]$	If $C_{min} = C_{tube}$ $\varepsilon = \begin{cases} 2/(1+2C^*) & \text{for } C^* \geq 0.5 \\ 1 & \text{for } C^* < 0.5 \end{cases}$ If $C_{min} = C_{shell}$ $\varepsilon = \dfrac{2}{2+C^*}$
	If $C_{min} = C_{tube}$ and $C_{max} = C_{shell}$ $\varepsilon = 4/\Big[2(1+C^*) + (1+4C^{*2})^{1/2}\coth(\Gamma/4) + \tanh(NTU/4)\Big]$ where $\Gamma = NTU(1+4C^{*2})^{1/2}$	$\varepsilon = \dfrac{4}{4+\sqrt{5}\coth(\Gamma/4)+\tanh(NTU/4)}$ where $\Gamma = \sqrt{5}\,NTU$	If $C_{min} = C_{tube}$ $\varepsilon = \dfrac{4}{2(1+C^*)+(1+4C^{*2})^{1/2}+1}$
1-4 shell-and-tube exchanger; shell fluid mixed; TEMA E shell	If $C_{min} = C_{shell}$ and $C_{max} = C_{tube}$ $\varepsilon = 4/\Big[2(1+C^*) + (4+C^{*2})^{1/2}\coth(\Gamma'/4) + C^*\tanh(NTU\cdot C^*/4)\Big]$ where $\Gamma' = NTU(4+C^{*2})^{1/2}$	$\varepsilon = \dfrac{4}{4+\sqrt{5}\coth(\Gamma'/4)+\tanh(NTU/4)}$ where $\Gamma' = \sqrt{5}\,NTU$	If $C_{min} = C_{shell}$ $\varepsilon = \dfrac{4}{2(1+C^*)+(4+C^{*2})^{1/2}+C^*}$

Table 1.b Effectiveness-NTU formulae

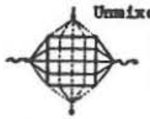
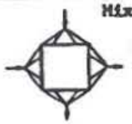
Flow Arrangement	ε-NTU Formulas	ε-NTU Formulas for $C^* = 1$	Asymptotic Value of ε when $NTU \to \infty$
 Counterflow	$\varepsilon = \frac{1 - \exp[-NTU(1-C^*)]}{1 - C^*\exp[-NTU(1-C^*)]}$	$\varepsilon = \frac{NTU}{1 + NTU}$	$\varepsilon = 1$ for all C^*
 Parallel Flow	$\varepsilon = \frac{1 - \exp[-NTU(1+C^*)]}{1 + C^*}$	$\varepsilon = \frac{1}{2}\left[1 - \exp(-2NTU)\right]$	$\varepsilon = \frac{1}{1 + C^*}$
 Unmixed fluid, Unmixed fluid. Crossflow, both fluids unmixed	$\varepsilon = 1 - \exp[-(1+C^*)NTU]\Big[I_o(2NTU\sqrt{C^*}) + \sqrt{C^*}\, I_1(2NTU\sqrt{C^*}) - \frac{1 - C^*}{C^*}\sum_{n=2}^{\infty} C^{*n/2}\, I_n(2NTU\sqrt{C^*})\Big]$	$\varepsilon = 1 - [I_o(2NTU) + I_1(2NTU)]e^{-2NTU}$	$\varepsilon = 1$ for all C^*
 Unmixed fluid, Mixed fluid. Crossflow, one fluid mixed, other unmixed	For C_{min} mixed, C_{max} unmixed, $\varepsilon = 1-\exp[-(1-\exp(-NTU\cdot C^*))/C^*]$	$\varepsilon = 1 - \exp[-(1 - \exp(-NTU))]$	For C_{min} mixed, $\varepsilon = 1-\exp(-1/C^*)$
	For C_{max} mixed, C_{min} unmixed, $\varepsilon = \frac{1}{C^*}\left[1-\exp(-C^*[1-\exp(-NTU)])\right]$	$\varepsilon = 1 - \exp[-(1 - \exp(-NTU))]$	For C_{max} mixed, $\varepsilon = [1-\exp(-C^*)]/C^*$
 Mixed fluid, Mixed fluid. Crossflow, both fluids mixed	$\varepsilon = \frac{1}{\frac{1}{1 - \exp(-NTU)} + \frac{C^*}{1 - \exp(-NTU\cdot C^*)} - \frac{1}{NTU}}$	$\varepsilon = \frac{1}{\frac{2}{1 - \exp(-NTU)} - \frac{1}{NTU}}$	$\varepsilon = \frac{1}{1 + C^*}$

Table 1.c Effectiveness-NTU formulae

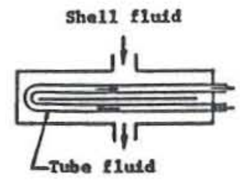
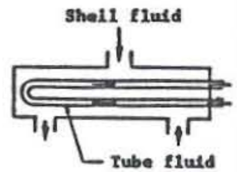
<table>
<tr><th>Flow Arrangement</th><th>ε-NTU Formulas</th><th>ε-NTU Formulas for $C^* = 1$</th><th>Asymptotic Value of ε when NTU $\to \infty$</th></tr>
<tr><td rowspan="2">1-2 split flow exchanger; shell fluid mixed; TEMA G shell</td>
<td>If $C_{min} = C_{tube}$, $C_{max} = C_{shell}$
$\varepsilon = \frac{(1+G+2C^*G) + (2C^*+1)De^{-\alpha} - e^{-\alpha}}{(1+G+2C^*G) + 2C^*(1-D) + 2C^*De^{-\alpha}}$
where $D = \frac{1-e^{-\alpha}}{2C^*+1}$, $G = \frac{1-e^{-\beta}}{2C^*-1} \xrightarrow[C^*=0.5]{} NTU_t/2$
$\alpha = \frac{1}{4} NTU(2C^*+1)$, $\beta = \frac{1}{2} NTU(2C^*-1)$</td>
<td>$\varepsilon = \frac{4-e^{-NTU/2}-e^{-3NTU/2}}{4-3e^{-NTU/2}+\frac{2}{3}\left(2+2e^{-3NTU/4}\right)-e^{-3NTU/2}}$</td>
<td>For $C^* > 1/2$
$\varepsilon = \frac{2C^*+1}{2C^{*2}+C^*+1}$
For $C^* \le 1/2$
$\varepsilon = 1$.</td></tr>
<tr><td>If $C_{min} = C_{shell}$, $C_{max} = C_{tube}$, use the above formula with C^* replaced by $1/C^*$, NTU replaced by NTU$\cdot C^*$, and ε replaced by εC^*.</td>
<td>The same as above formula.</td>
<td>For $C^* < 2$
$\varepsilon = \frac{C^*+2}{C^{*2}+C^*+2}$
For $C^* \ge 2$, $\varepsilon = 1/C^*$</td></tr>
<tr><td rowspan="2">1-2 divided flow exchanger; shell fluid mixed; TEMA J shell</td>
<td>If $C_{min} = C_{tube}$, $C_{max} = C_{shell}$
$\varepsilon = \frac{2}{1 + 2C^*\phi'}$
where $\phi' = 1 + \gamma\left(\frac{1+\phi}{1-\phi}\right) - 2\gamma\left\{\frac{\gamma\phi + (1-\phi)e^{-NTUC^*(\gamma-1)/2}}{(1-\phi)^2 + \gamma(1-\phi^2)}\right\}$
$\phi = \exp(-NTU\, C^*\gamma)$,
$\gamma = (1+4C^{*2})^{1/2}/2C^*$</td>
<td>$\varepsilon = \frac{2}{1 + 2\phi'}$
where $\phi' = 1 + \gamma\left(\frac{1+\phi}{1-\phi}\right) - 2\gamma\left\{\frac{\gamma\phi + (1-\phi)e^{-NTU(\gamma-1)/2}}{(1-\phi)^2 + \gamma(1-\phi^2)}\right\}$
$\phi = \exp(-NTU\, \gamma)$, $\gamma = \sqrt{5}/2$</td>
<td>If $C_{min} = C_{tube}$
$\varepsilon = \frac{2}{1+2C^*+(1+4C^{*2})^{1/2}}$</td></tr>
<tr><td>If $C_{min} = C_{shell}$, $C_{max} = C_{tube}$, use the above formula with C^* replaced by $1/C^*$, NTU replaced by NTU C^*, and ε replaced by εC^*.</td>
<td>The same as above formula.</td>
<td>If $C_{min} = C_{shell}$
$\varepsilon = \frac{2}{C^* + 2 + (C^{*2} + 4)^{1/2}}$</td></tr>
</table>

Table 1.d Effectiveness-NTU formulae

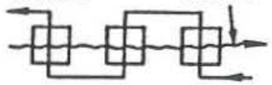
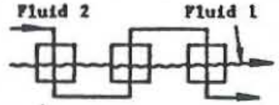
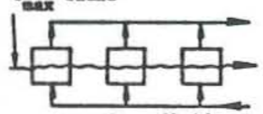
Flow Arrangement	ε-ε_p Formulas	Special Cases of $C^* = 1$ and 0	Number of Passes $n \to \infty$
 n-pass exchanger (n=3 as shown), both fluid streams parallel, overall counter-flow arrangement, fluids mixed between passes, each pass having the same ε_p.	$\varepsilon = \dfrac{[(1-\varepsilon_p C^*)/(1-\varepsilon_p)]^n - 1}{[(1-\varepsilon_p C^*)/(1-\varepsilon_p)]^n - C^*}$ Note $C^* = C^*_p$, and $NTU = nNTU_p$ $\varepsilon_p = \dfrac{[(1-\varepsilon C^*)/(1-\varepsilon)]^{1/n} - 1}{[(1-\varepsilon C^*)/(1-\varepsilon)]^{1/n} - C^*}$	$\varepsilon = \begin{cases} \dfrac{n\varepsilon_p}{1+(n-1)\varepsilon_p} & \text{for } C^* = 1 \\ 1-(1-\varepsilon_p)^n & \text{for } C^* = 0 \end{cases}$ $\varepsilon_p = \begin{cases} \dfrac{\varepsilon}{n-(n-1)\varepsilon} & \text{for } C^* = 1 \\ 1-(1-\varepsilon)^{1/n} & \text{for } C^* = 0 \end{cases}$	$\varepsilon = \varepsilon_{counterflow}$ $\varepsilon_p \to 0$
 n-pass exchanger (n=3 as shown), both fluid streams parallel, overall parallel flow arrangement, fluids mixed between passes, each pass having the same ε_p.	$\varepsilon = \dfrac{1}{1+C^*}[1 - \{1 - (1+C^*)\varepsilon_p\}^n]$ Note $C^* = C^*_p$, and $NTU = nNTU_p$ $\varepsilon_p = \dfrac{1}{1+C^*}\left[1 - \{1 - (1+C^*)\varepsilon\}^{1/n}\right]$	$\varepsilon = \begin{cases} \frac{1}{2}[1-(1-2\varepsilon_p)^n] & \text{for } C^* = 1 \\ 1-(1-\varepsilon_p)^n & \text{for } C^* = 0 \end{cases}$ $\varepsilon_p = \begin{cases} \frac{1}{2}[1-(1-2\varepsilon)^{1/n}] & \text{for } C^* = 1 \\ 1-(1-\varepsilon)^{1/n} & \text{for } C^* = 0 \end{cases}$	$\varepsilon = \varepsilon_{parallel\ flow}$ $\varepsilon_p \to 0$
 n-pass exchanger (n=3 as shown), C_{min} stream in parallel, C_{max} stream in series, series fluid mixed between passes, each pass having the same ε_p.	Parallel stream as C_{min} stream, $\varepsilon = \dfrac{1}{C^*}\left[1 - \left(1 - \dfrac{C^*\varepsilon_p}{n}\right)^n\right]$ Note $C^* = nC^*_p$, and $NTU = NTU_p$ $\varepsilon_p = \dfrac{1}{C^*_p}\left[1 - (1 - nC^*_p\varepsilon)^{1/n}\right]$	$\varepsilon = \begin{cases} 1-(1-\varepsilon_p/n)^n & \text{for } C^* = 1 \\ \varepsilon_p & \text{for } C^* = 0 \end{cases}$ $\varepsilon_p = \begin{cases} n[1-(1-\varepsilon)^{1/n}] & \text{for } C^* = 1 \\ \varepsilon & \text{for } C^* = 0 \end{cases}$	$\varepsilon = \frac{1}{C^*}[1-\exp(-C^*\varepsilon_p)]$ $= \frac{1}{C^*}\left[1-\exp(-C^* \times [1-\exp(-NTU)])\right]$ $\varepsilon_p = 1-\exp(-NTU_p)$ Note $NTU = NTU_p$

Table 1.e Effectiveness-NTU formulae

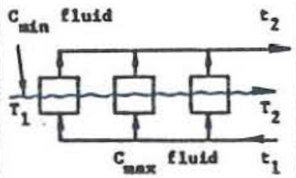
Flow Arrangement	ε-ε_p Formulas	Special Cases of $C^* = 1$ and 0	Number of Passes $n \to \infty$
	Parallel stream as C_{max} stream, and $C_{max}/n < C_{min}$ $\varepsilon = \frac{T_2 - T_1}{t_1 - T_1} = 1 - \left(1 - \frac{\varepsilon_p}{nC^*}\right)^n$ Note $C^* = 1/nC_p^*$, $NTU = NTU_p/C^*$ $\varepsilon_p = \frac{(t_1 - t_2)_p}{\Delta_{o,p}} = \frac{1}{C_p^*}\left[1-(1-\varepsilon)^{1/n}\right]$	$\varepsilon = \begin{cases} 1 - (1-\varepsilon_p)^n & \text{for } C_p^* = 1 = \frac{1}{nC^*} \\ 1 - e^{-NTU} & \text{for } n \to \infty,\ C^* = 0 \end{cases}$ $\varepsilon_p = \begin{cases} 1 - (1-\varepsilon)^{1/n} & \text{for } C_p^* = 1 = \frac{1}{nC^*} \\ 1 & \text{for } n \to \infty,\ C_p^* = 0 \end{cases}$	$\varepsilon = 1-\exp(-\varepsilon_p/C^*)$ $= 1-\exp\left[-\frac{1}{C^*} \times \left\{1-\exp(-NTU \cdot C^*)\right\}\right]$ $\varepsilon_p = 1-\exp(-NTU_p)$
n-pass exchanger (n=3 as shown), C_{max} stream in parallel, C_{min} stream in series, series fluid mixed between passes, each pass having the same ε_p.	Parallel stream as C_{max} stream, and $C_{max}/n > C_{min}$ $\varepsilon = 1 - (1-\varepsilon_p)^n$ Note $C^* = C_p^*/n$, and $NTU = nNTU_p$, $\varepsilon_p = 1 - (1-\varepsilon)^{1/n}$	$C^* = 1$ only when $n = 1$ $\varepsilon = \begin{cases} 1 - (1-\varepsilon_p)^n & \text{for } C_p^* = 1 = \frac{C^*}{n} \\ 1 - \exp(-NTU) & \text{for } C^* = 0 \end{cases}$ $\varepsilon_p = \begin{cases} 1 - (1-\varepsilon)^{1/n} & \text{for } C_p^* = 1 \\ 1 - \exp(-NTU_p) & \text{for } C_p^* = 0 \end{cases}$	$n \to \infty$ only when $C^* = 0$ $\varepsilon = 1 - \exp(-NTU)$ $\varepsilon_p = 1-\exp(-NTU_p)$

Table 2. Measurements from PHE 1 (Figure 4.3)

Run	Th1 (°C)	Th2 (°C)	Tc1 (°C)	Tc2 (°C)	mc (kg/s)
1	148	143	131	6.5	0.5
2	144	137.5	118.9	6.5	0.73
3	139	126	100.0	6.5	1.24
4	137.1	118.5	83.8	6.4	1.57
5	136.2	116	80.2	7.3	1.69
6	134.8	111.2	70.4	6.2	1.98
7	134.8	111.8	70.3	6.3	2.0

where Th1 = inlet temperature of the hot fluid
Th2 = outlet temperature of the hot fluid
Tc1 = outlet temperature of the cold fluid
Tc2 = inlet temperature of the cold fluid
mc = flowrate of the secondary fluid

Table 3. Measurements from PHE 2 (Figure 4.4)

Run	Th1 (°C)	Th2 (°C)	Tc1 (°C)	Tc2 (°C)	mc (kg/s)
1	120	93	87.7	22.1	8.17
2	120	95	91	23.3	7.69
3	120	98	93.4	23.4	7.44
4	120	98	93.4	23.4	7.40
5	120	96	92.9	23.1	7.58
6	120	95	92.0	23.3	7.667
7	120	95.5	91.8	23.2	7.70

where Th1 = inlet temperature of the hot fluid
Th2 = outlet temperature of the hot fluid
Tc1 = outlet temperature of the cold fluid
Tc2 = inlet temperature of the cold fluid
mc = flowrate of the secondary fluid

Table 4.1 Measurements from ESMIL 1 (Figure 2.2)

Run	Th1 (°C)	Th2 (°C)	Tc1 (°C)	Tc2 (°C)	mc (kg/s)
1	91.0	36.3	23.1	5.2	12.06
2	91.0	36.2	23.3	5.1	12.07
3	91.4	36.6	23.4	5.5	12.07
4	91.2	35.9	23.4	5.3	11.97
5	91.1	36.6	23.1	5.4	12.16
6	91.1	35.9	23.4	5.1	12.16
7	91.0	36.7	23.2	5.4	12.16

Table 4.2 Measurements from Esmil 2 (Figure 2.2)

Run	Th1 (°C)	Th2 (°C)	Tc1 (°C)	Tc2 (°C)	mc (kg/s)
1	159.1	91.0	90.0	66.4	12.06
2	158.3	91.0	90.1	66.5	12.07
3	159.4	91.4	90.5	66.7	12.07
4	158.7	91.2	90.2	66.6	11.97
5	158.7	91.1	90.2	66.4	12.16
6	158.2	91.1	90.2	66.6	12.16
7	158.7	91.0	90.1	66.4	12.16

where Th1 = inlet temperature of the hot fluid
Th2 = outlet temperature of the hot fluid
Tc1 = outlet temperature of the cold fluid
Tc2 = inlet temperature of the cold fluid
mc = flowrate of the secondary fluid

Table 5. Calculated Results

PHE 1

Q,meas (KW)	Q,cal (KW)	Tc1,cal (°C)	Th2,cal (°C)	mh (k/s)	ΔPc KPa	mc^2 $(k/s)^2$
261.7	269.7	134.8	132.8	0.123	0.129	0.25
344.4	346.0	119.4	137.5	0.160	0.248	0.53
485.9	448.9	93.0	126.0	0.204	0.630	1.54
508.6	485.9	80.4	118.8	0.218	0.962	2.47
515.6	491.5	76.8	116.0	0.220	1.095	2.86
531.8	541.7	68.3	111.2	0.228	1.462	3.92
535.5	517.4	68.14	111.8	0.229	1.487	4.00

PHE 2

2204.6	2271.4	92.1	98.0	0.989	40.44	58.2
2179.9	2251.0	91.0	95.0	0.975	41.04	59.1
2181.1	2229.3	93.4	98.0	0.971	38.70	55.3
2169.4	2221.9	93.4	98.0	0.968	38.35	54.7
2215.7	2245.0	92.9	96.0	0.974	40.04	57.4
2205.7	2246.9	92.0	95.0	0.974	40.83	58.8
2211.9	2259.7	91.8	95.5	0.980	41.13	59.3

ESMIL 1

q,meas (KW)	q,cal (KW)	mh (k/s)	Tc1,cal (°C)	Th2,cal (°C)	ΔPc (Bar)	ΔPh (Bar)
904.24	904.18	3.96	23.1	36.50	0.46	5.3
920.15	920.07	4.03	23.3	36.51	0.46	5.2
904.93	904.87	3.95	23.4	36.78	0.46	5.3
907.49	907.41	3.96	23.4	36.50	0.45	5.3
901.90	901.85	3.96	23.1	36.75	0.47	5.3
932.48	932.40	4.09	23.4	36.70	0.47	5.2
906.99	906.93	3.99	23.2	36.74	0.47	5.3

ESMIL 2

1194.67	1194.88	4.11	90.0	91.02	0.42	4.4
1195.68	1196.31	4.16	90.1	90.98	0.42	4.4
1205.89	1206.34	4.15	90.5	91.36	0.42	4.4
1185.80	1186.28	4.10	90.2	90.96	0.41	4.4
1215.31	1215.94	4.22	90.2	91.17	0.43	4.4
1205.12	1205.87	4.22	90.2	91.19	0.43	4.4
1210.19	1210.71	4.2	90.1	91.15	0.43	4.4

where:

Tc1,cal = calculated outlet temp. of the cold fluid
Th2,cal = calculated outlet temp. of the hot fluid
q,cal = calculated heat duty
q,meas = target heat duty
mh = flow rate of the hot fluid
mc = flow rate of the cold fluid
mc^2 = flow rate of the cold fluid (squared)
ΔPc = pressure drop on the cold side
ΔPh = pressure drop on the hot side

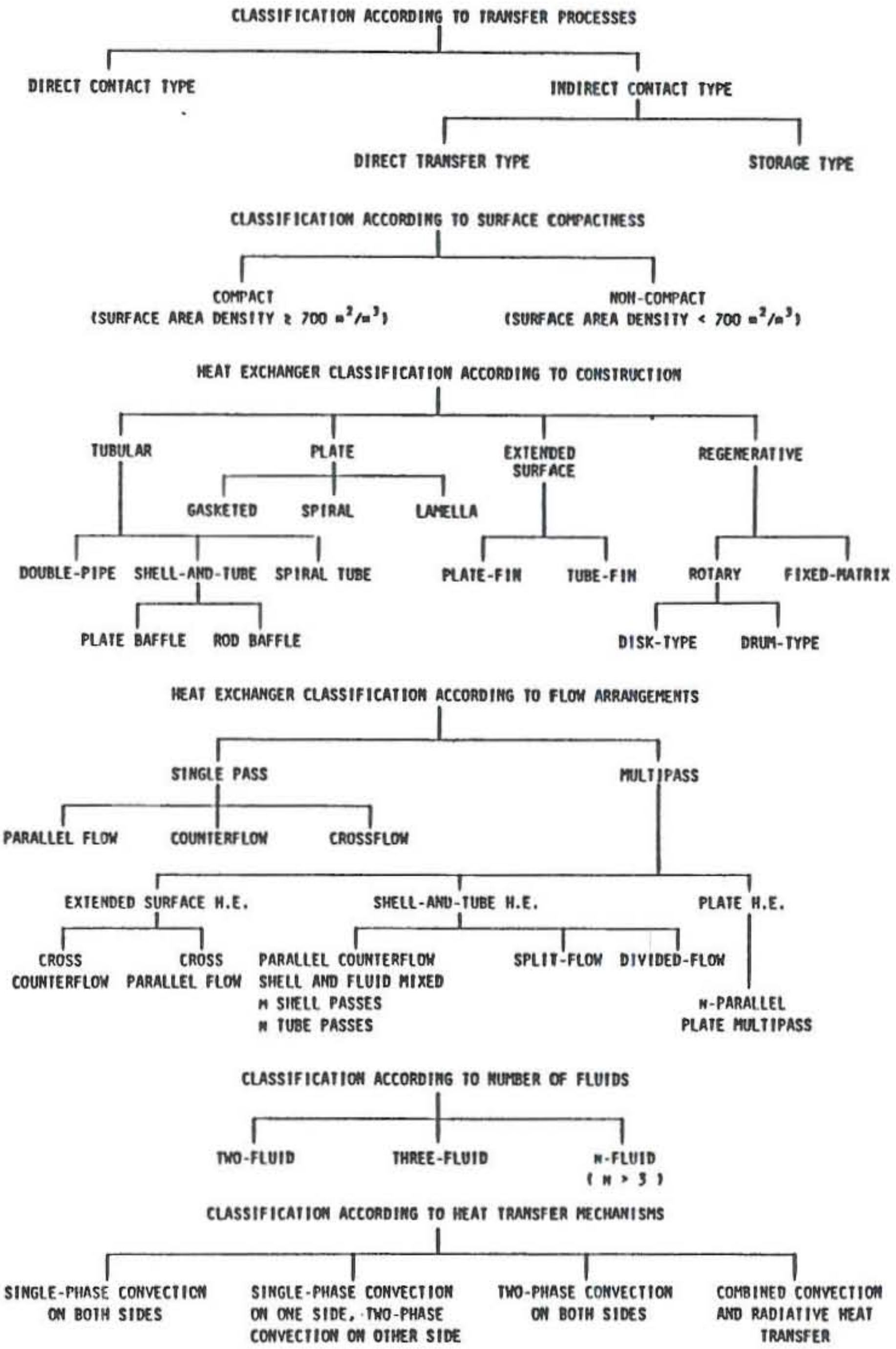


Figure 1. Classification of Heat Exchangers

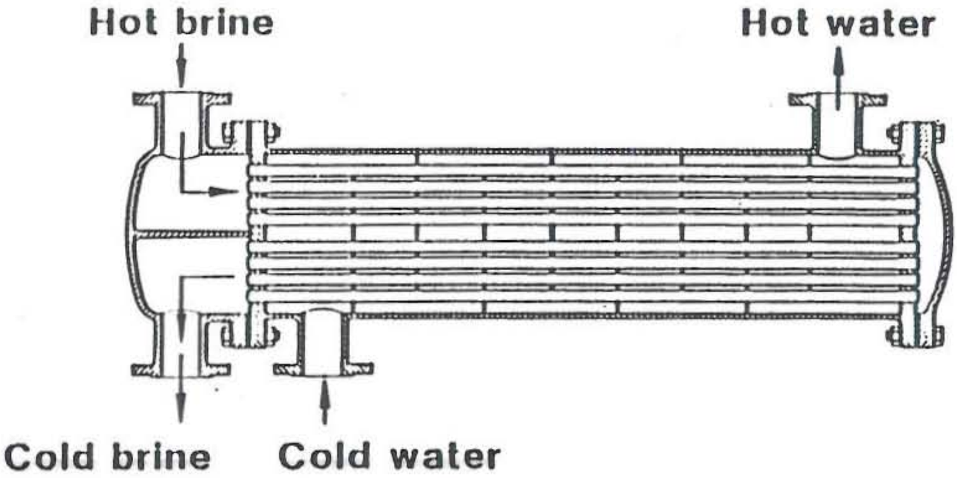


Figure 2.1 Shell and Tube Heat Exchanger

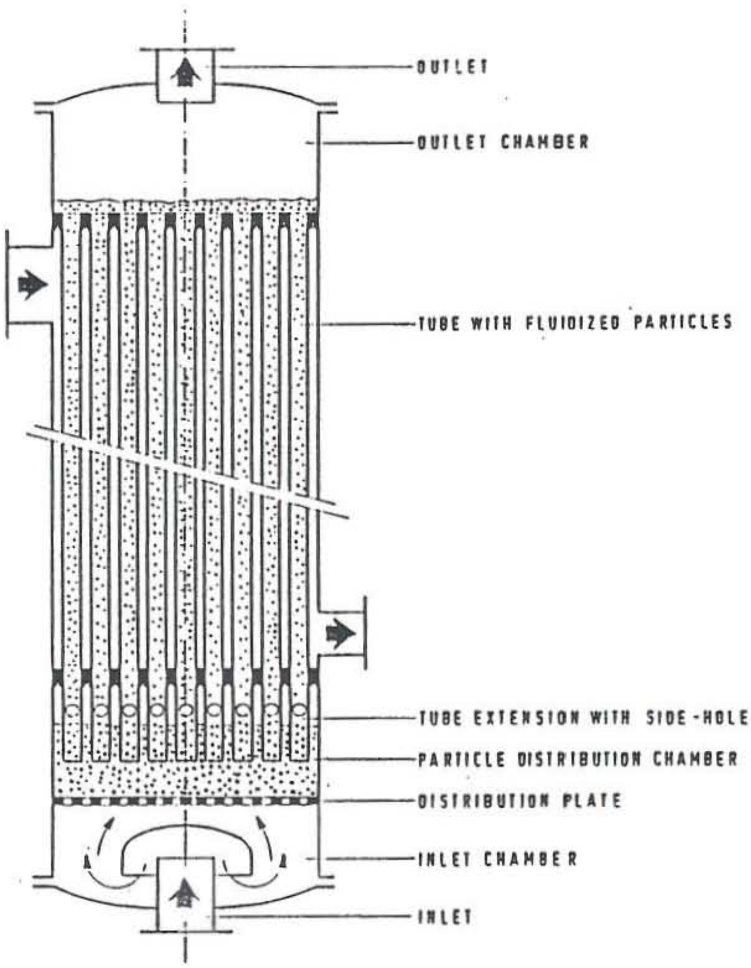


Figure 2.2 Fluidized Bed Heat Exchanger

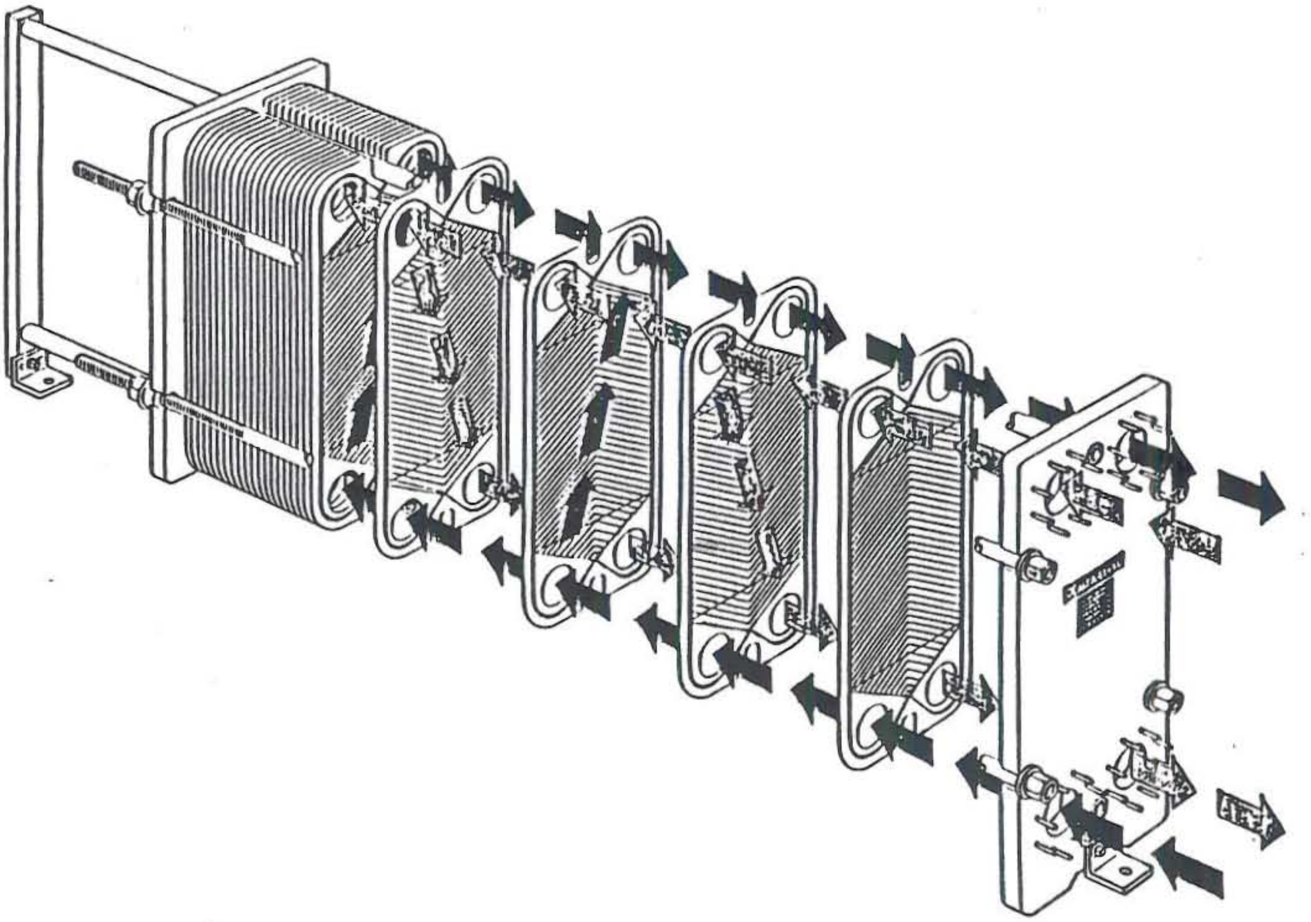


Figure 2.3 Plate Heat Exchanger

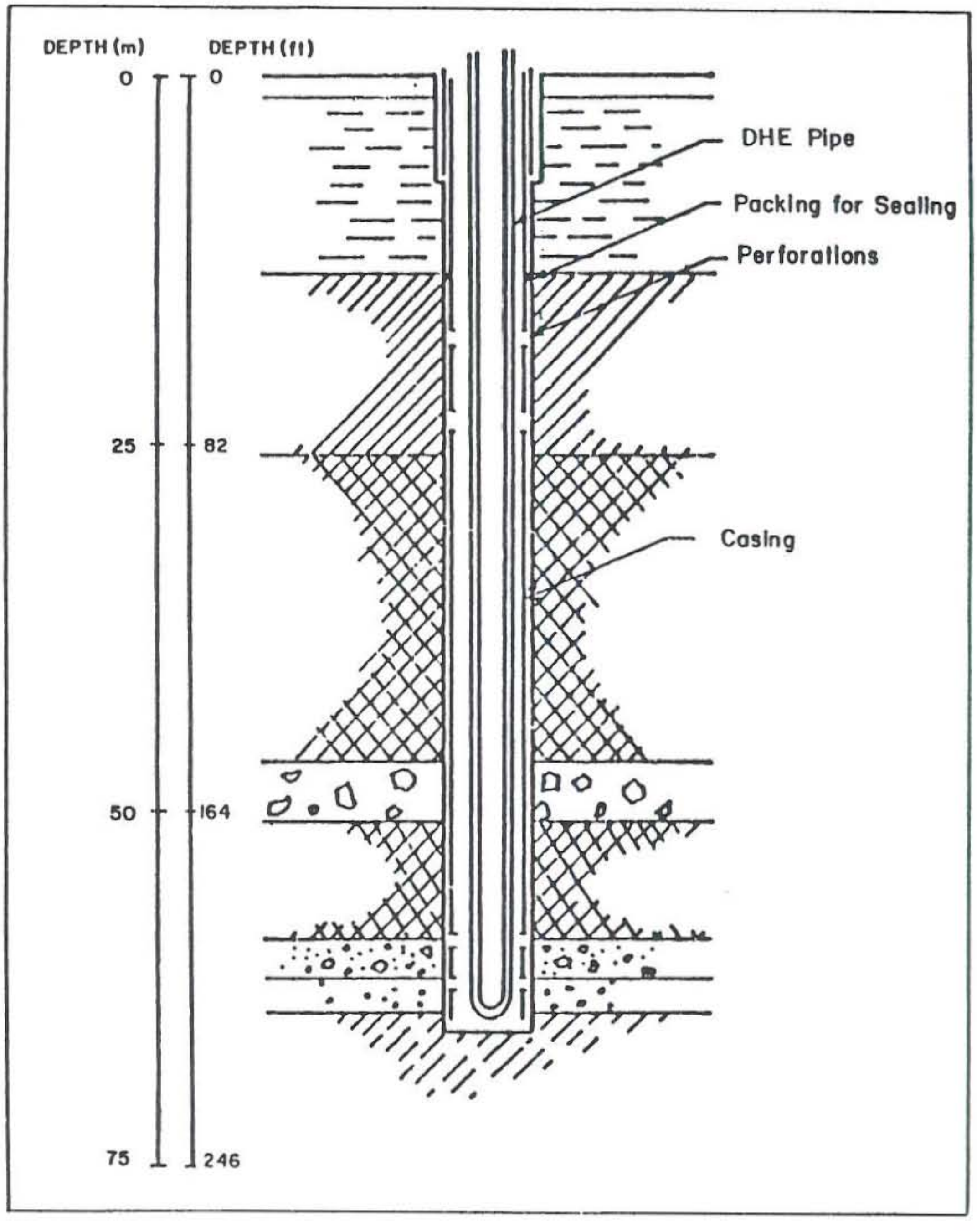


Figure 2.4 Downhole Heat Exchanger

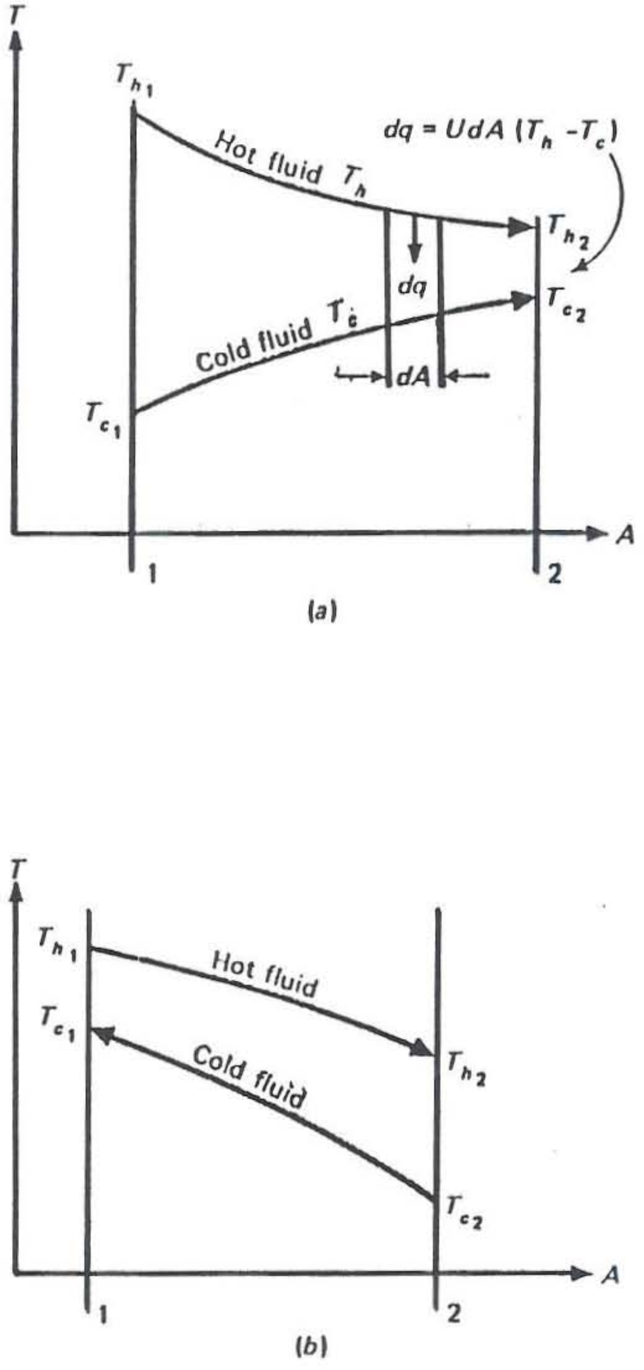


Figure 3. Parallel and Counterflow Profile

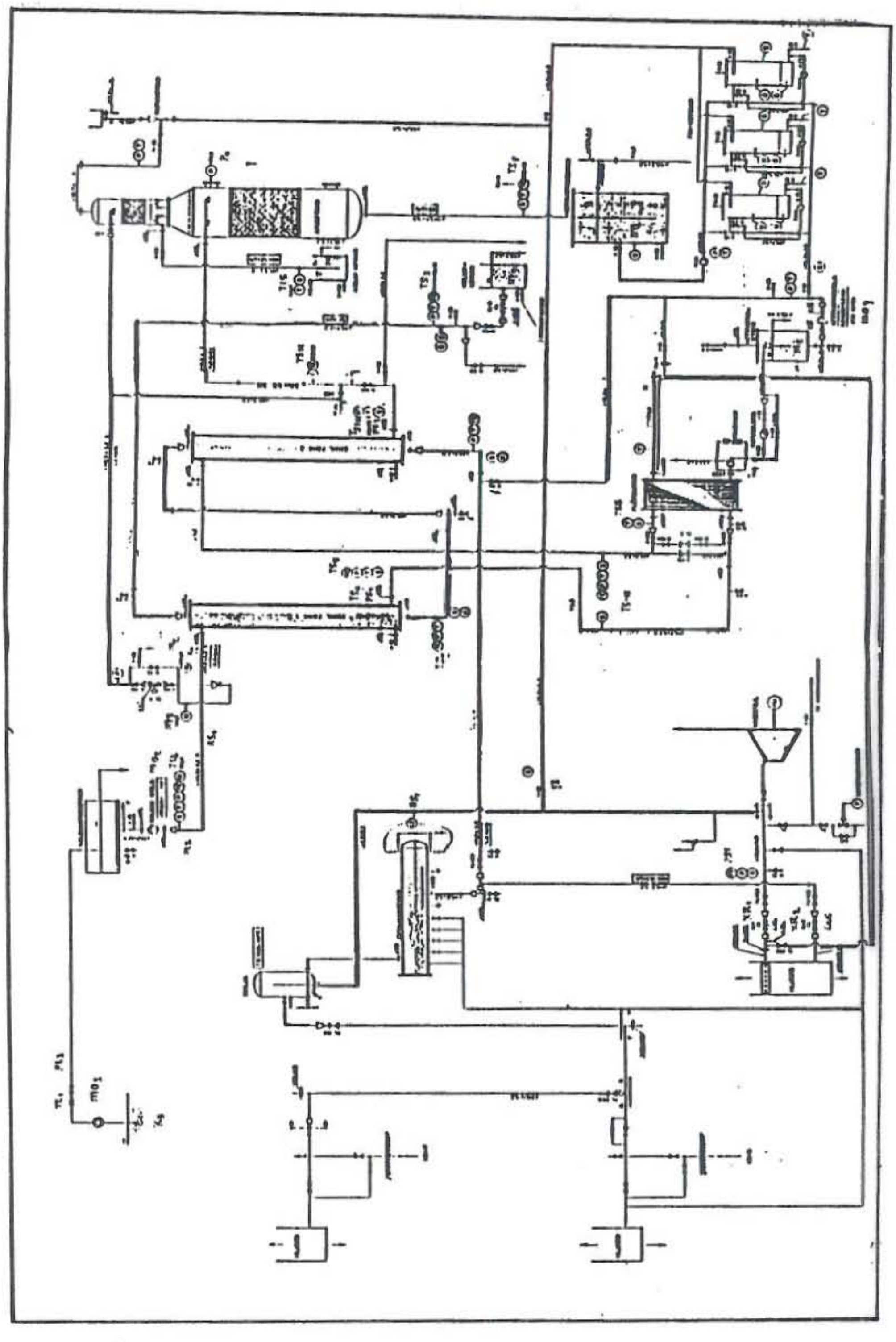


Figure 4.1 The Nesjavellir Pilot Plant

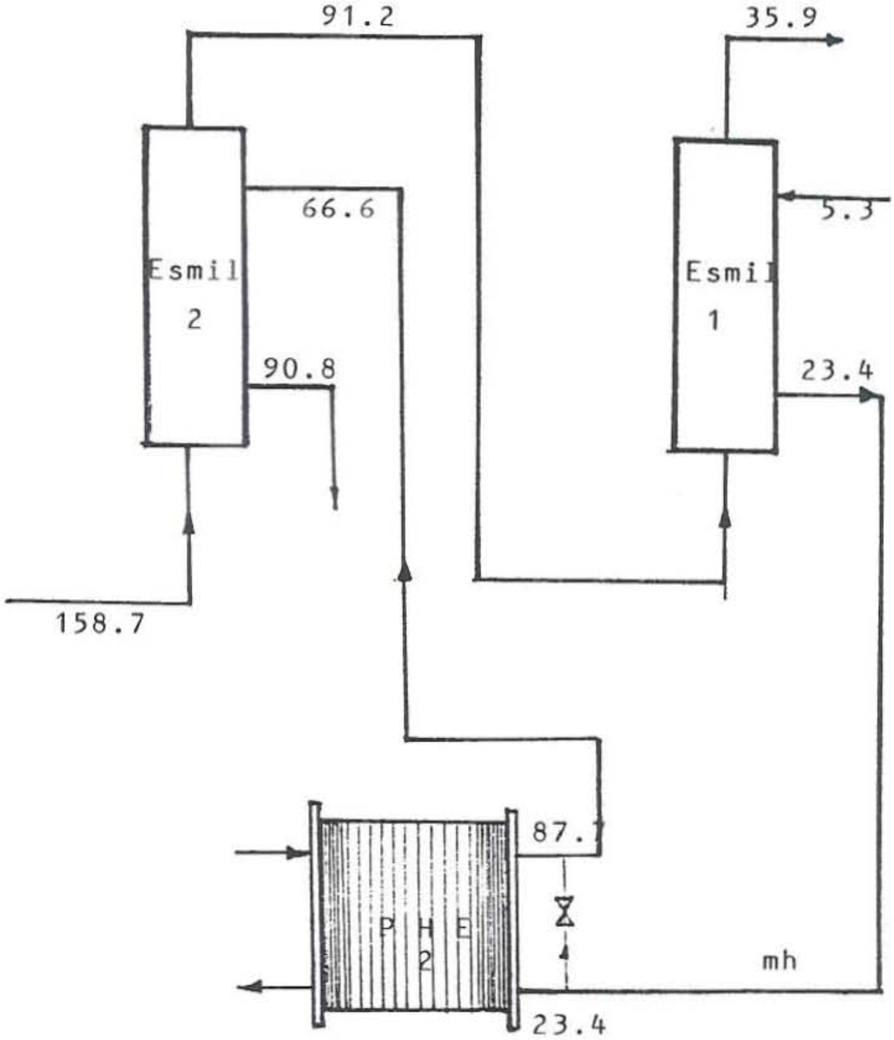


Figure 4.2 Heat Exchanger Arrangements

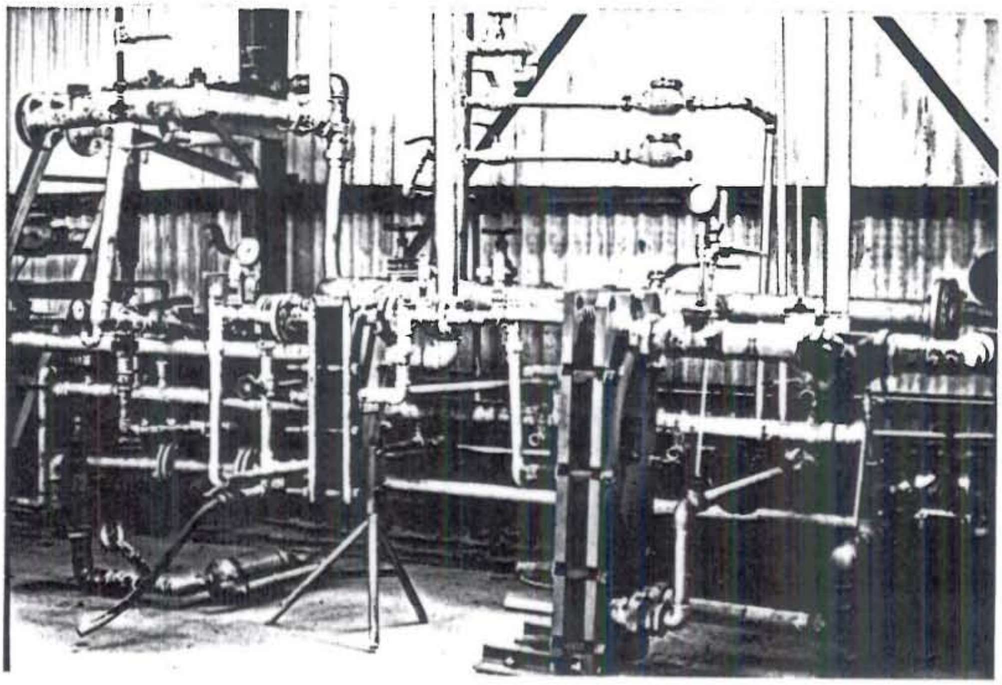


Figure 4.3 Plate Heat Exchanger 1

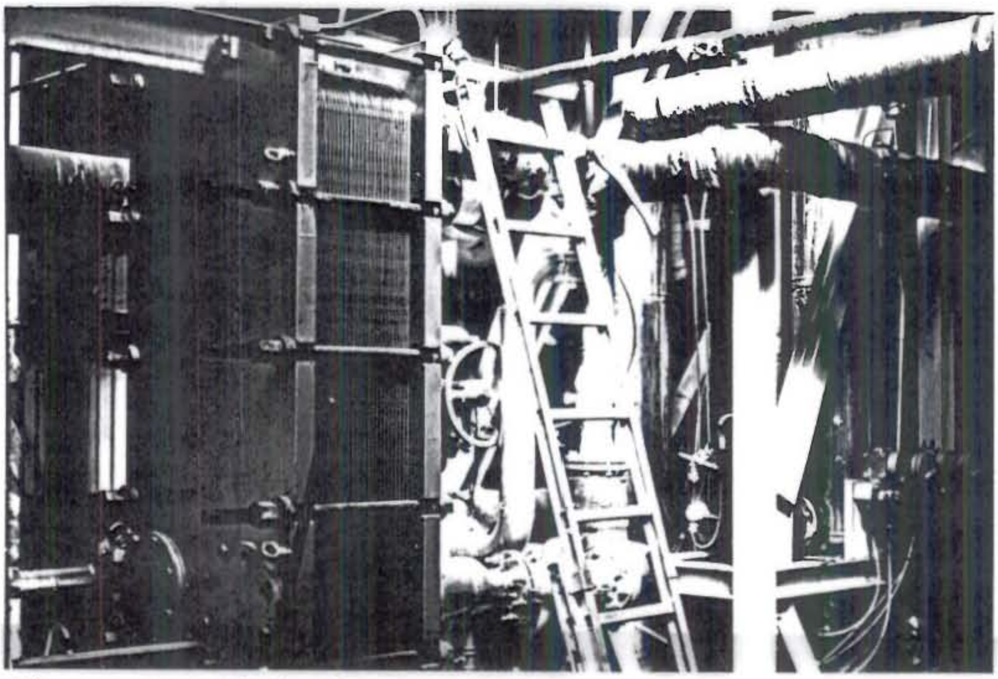


Figure 4.4 Plate heat Exchanger 2

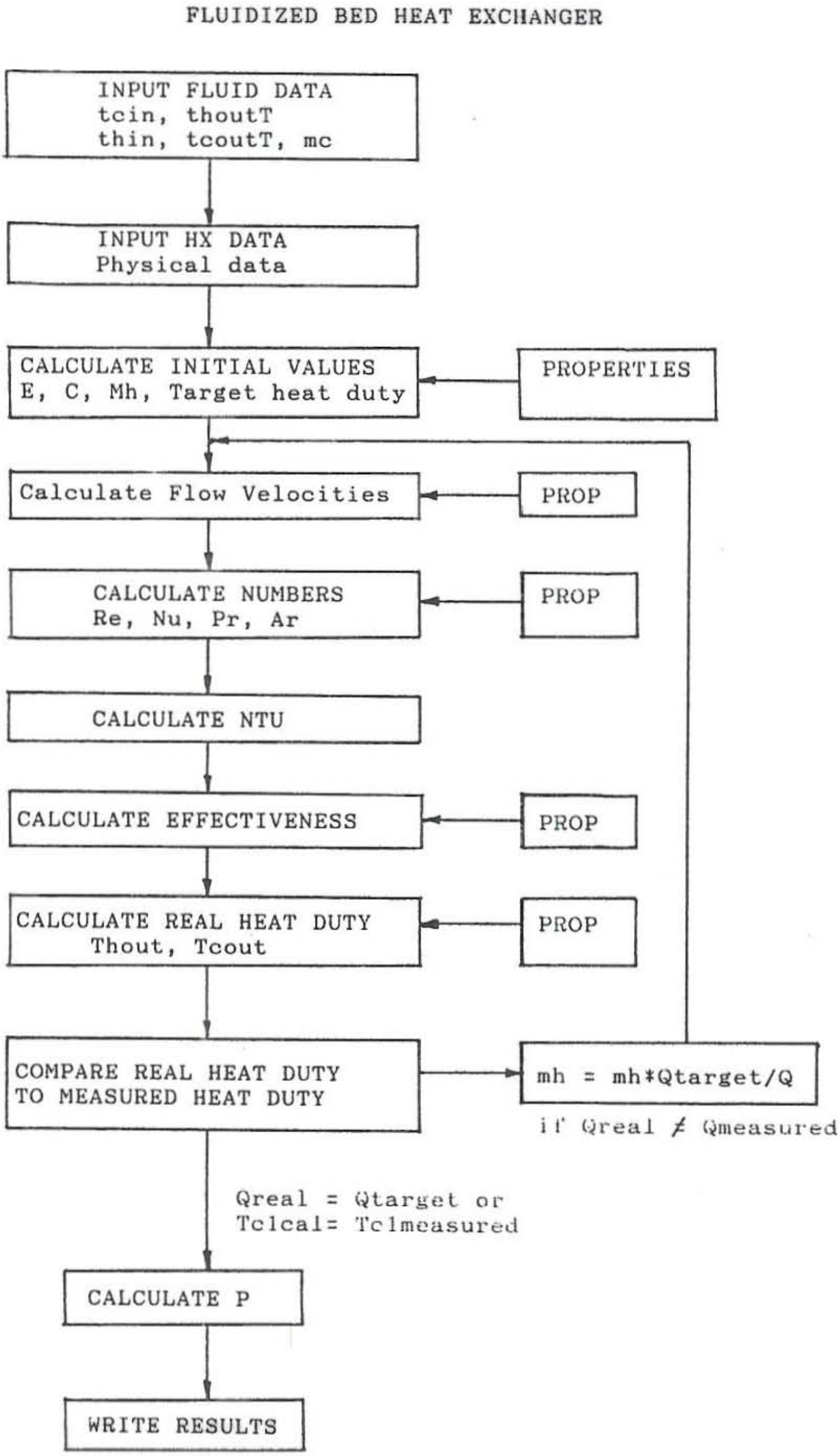


Figure 5.1 Structure of Computer Program, FBHX

PLATE HEAT EXCHANGER

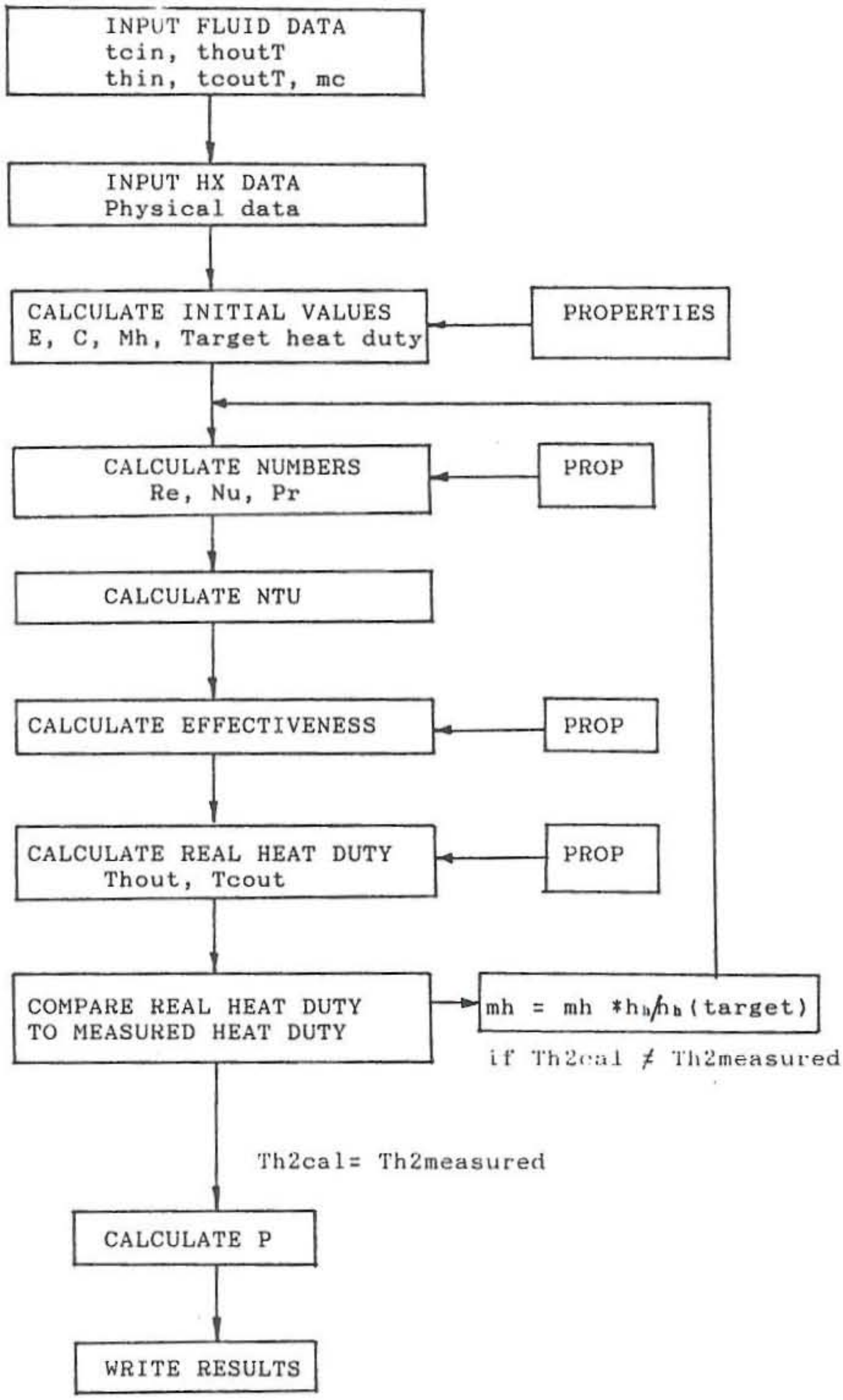


Figure 5.2 Structure of Computer Program, PHE

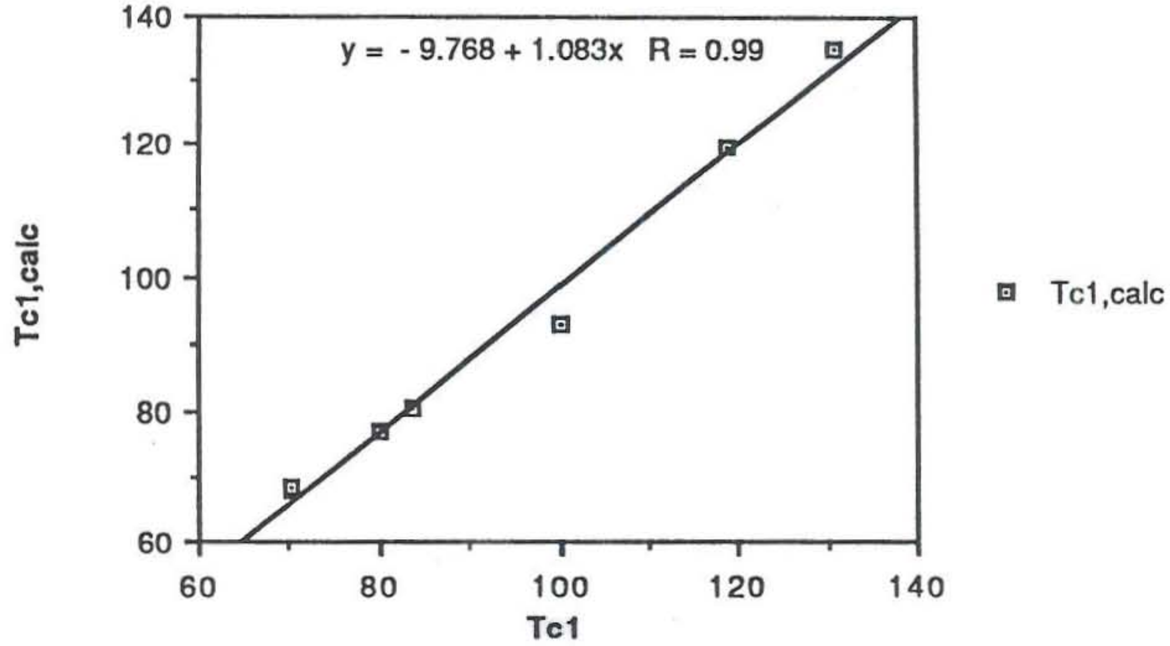


Figure 6.1 PHE 1, Tc1 measured vs Tc1 calculated

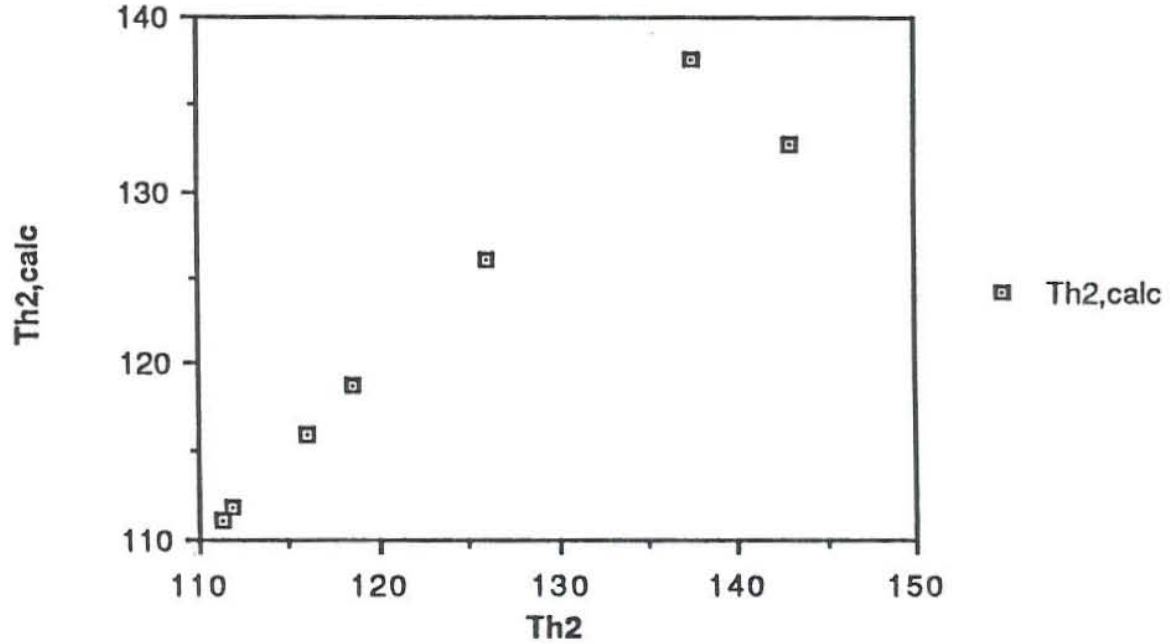


Figure 6.2 PHE 1, Th2 measured vs Th2 calculated

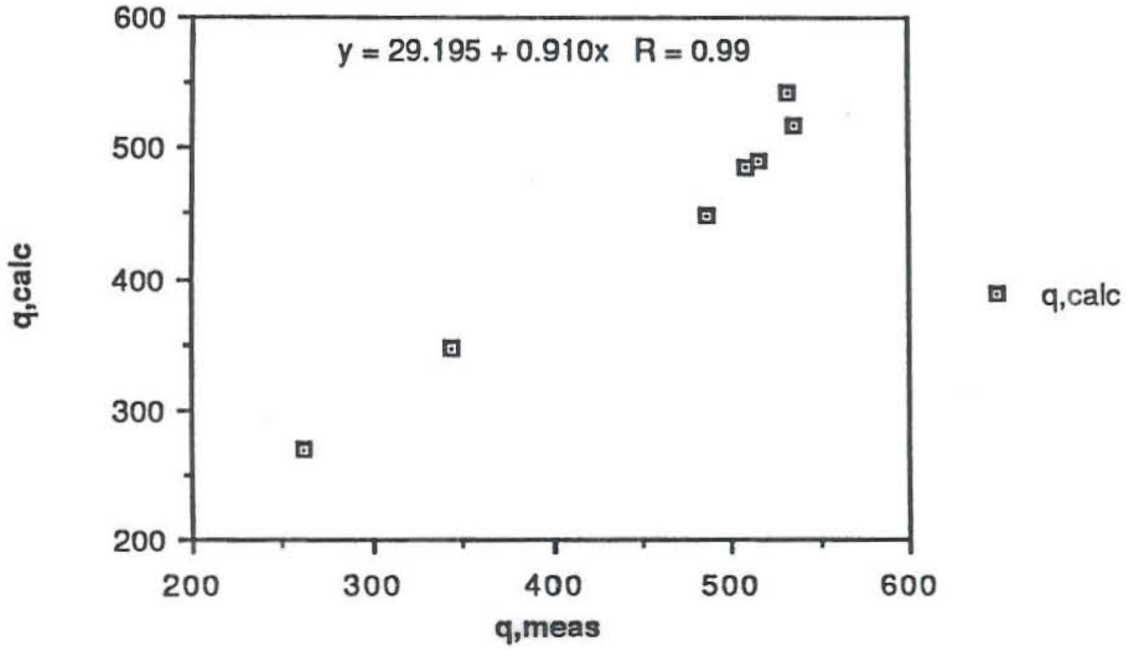


Figure 6.3 PHE 1, Q measured vs Q calculated

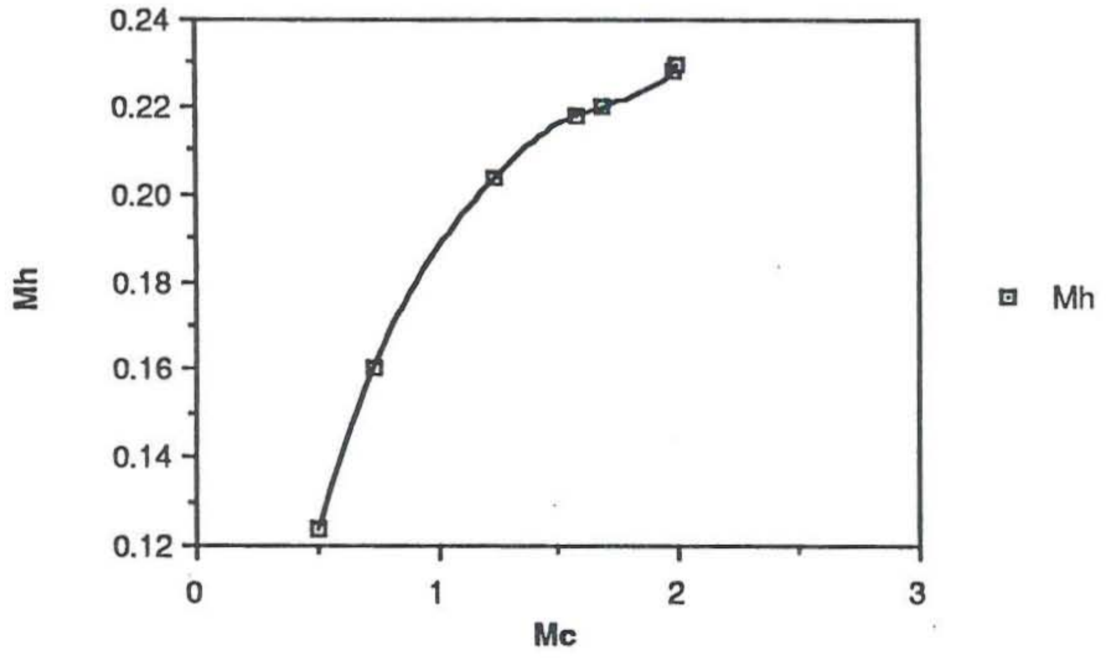


Figure 6.4 PHE 1, Mc vs Mh Curve

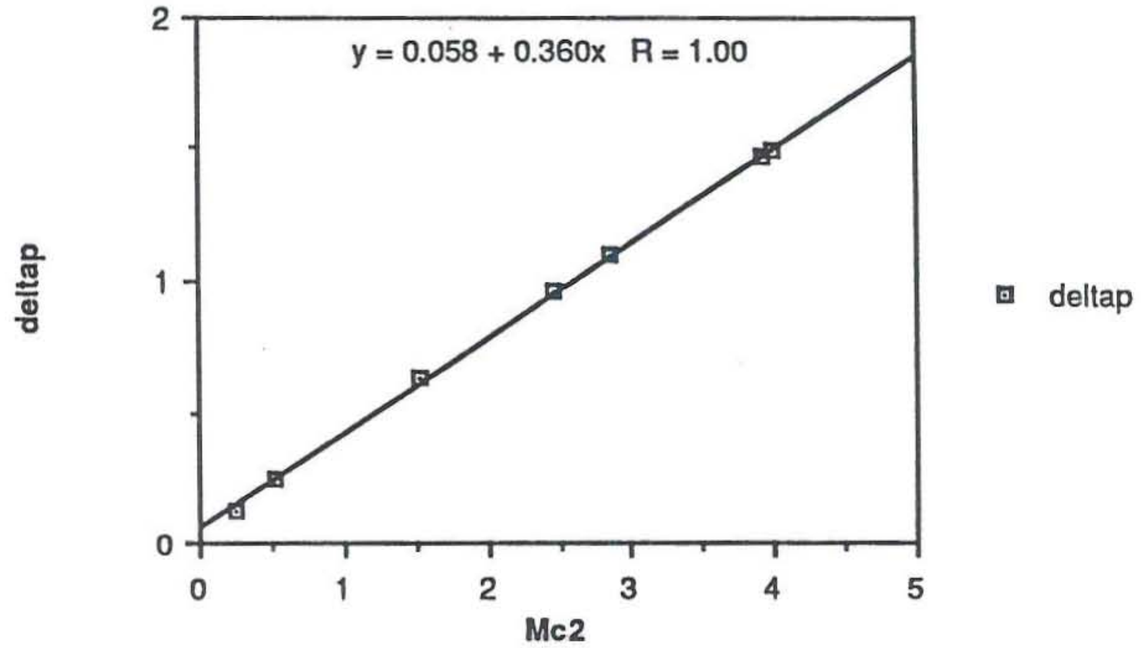


Figure 6.5 PHE 1, Mc^2 vs Pressure drop relationship

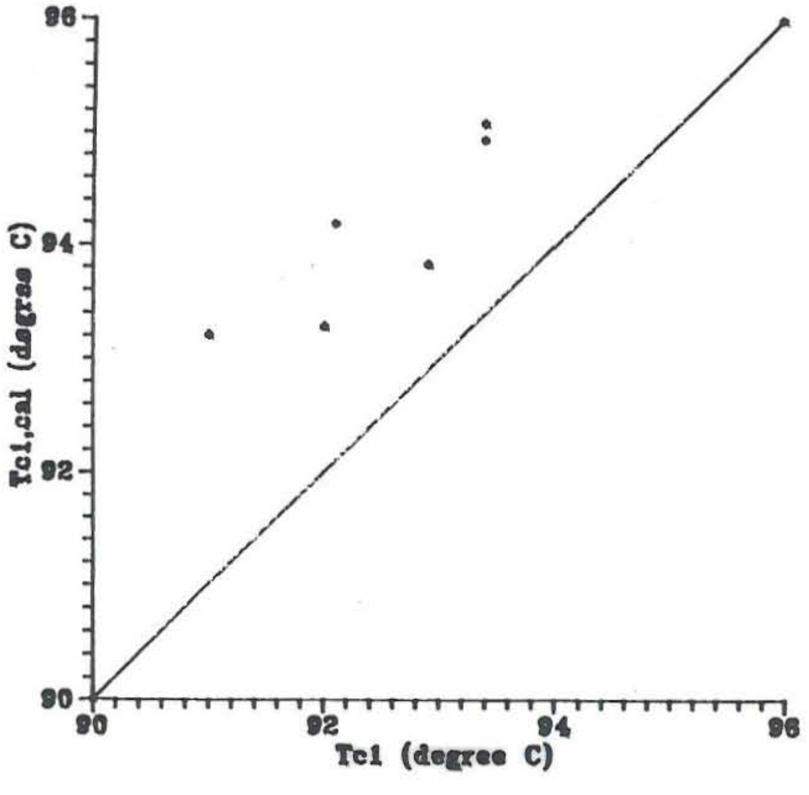


Figure 7.1 PHE 2, Tc1 measured vs Tc1 calculated

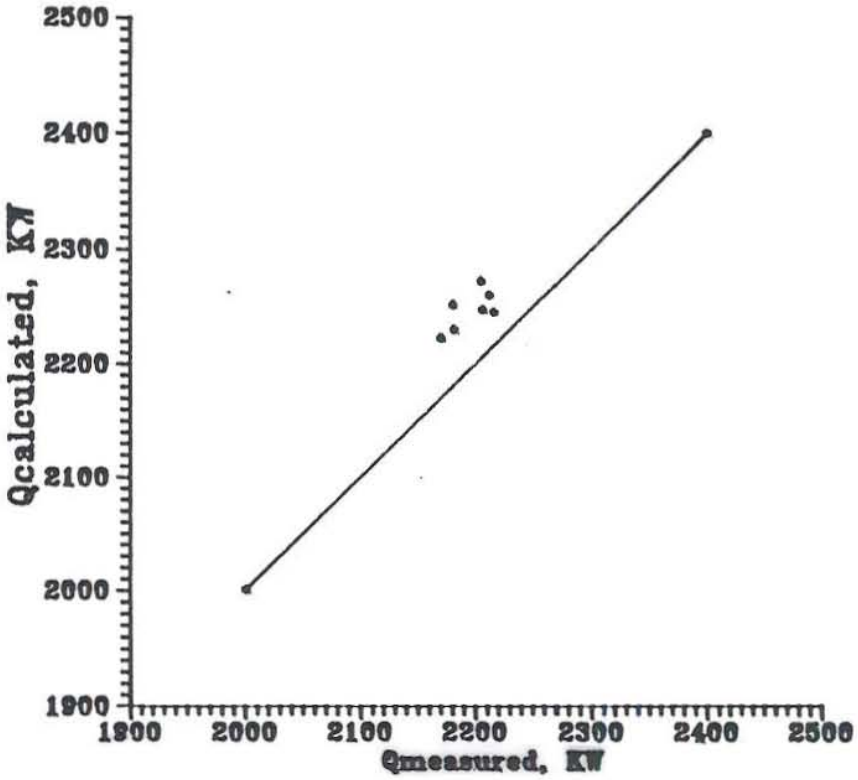


Figure 7.2 PHE 2, Q measured vs Q calculated

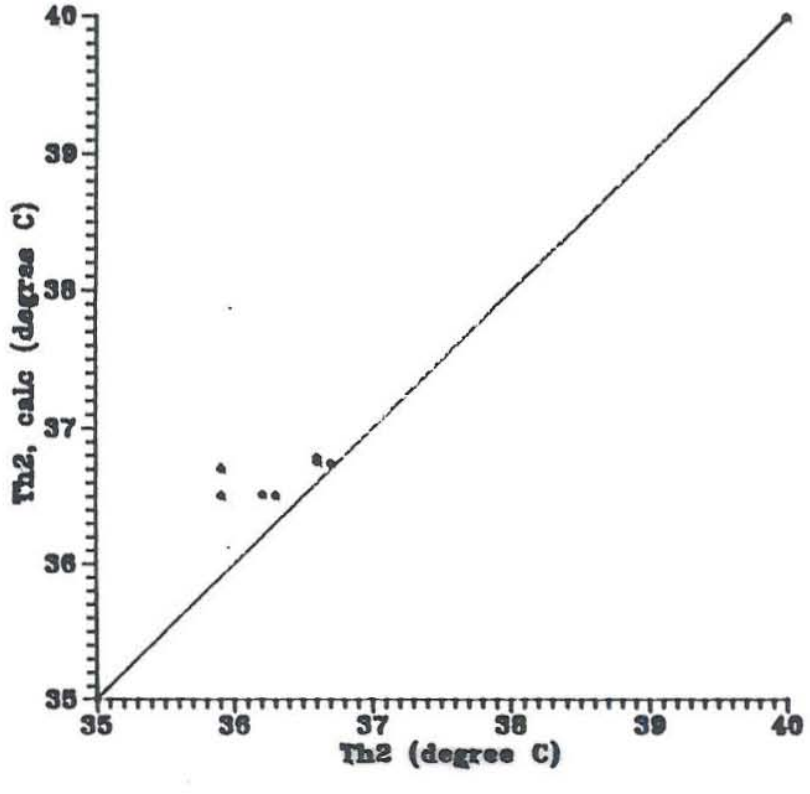


Figure 8.1 Esmil 1, Th2 calculated vs Th2 measured

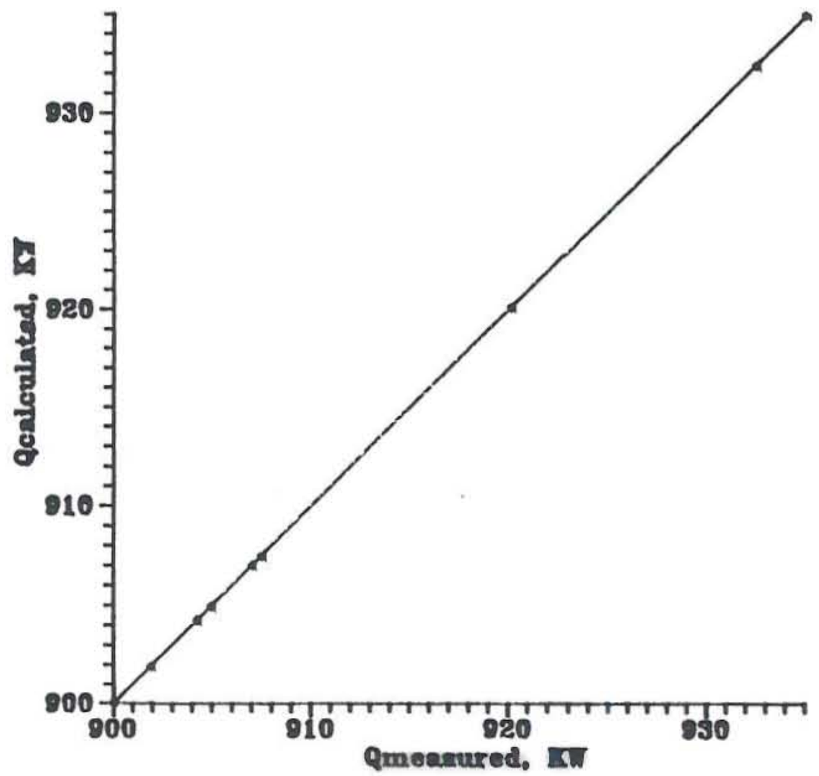


Figure 8.2 Esmil 1, Q measured vs Q calculated

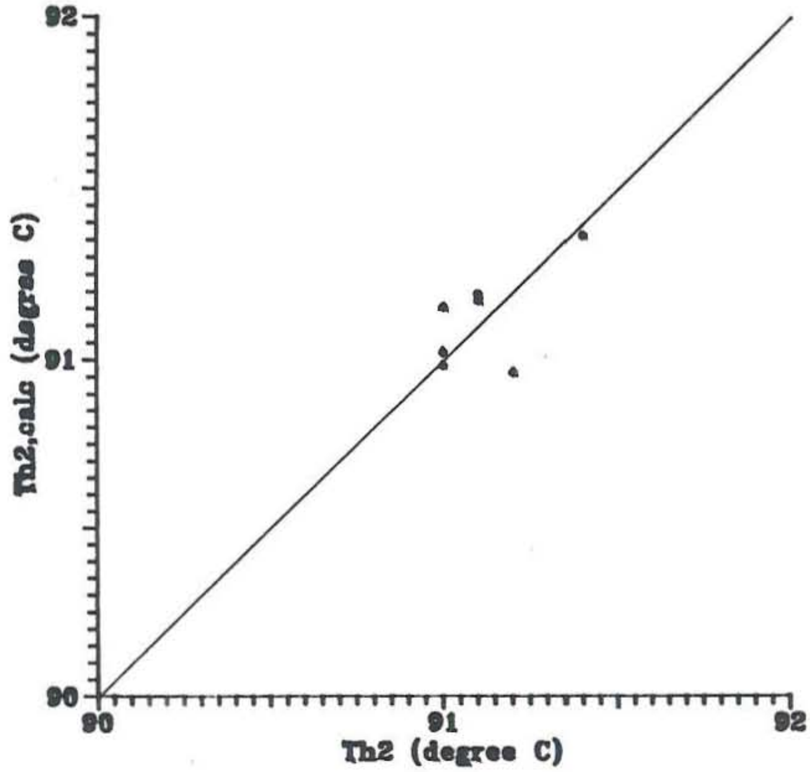


Figure 8.3 Esmil 2, Th2 calculated vs Th2 measured

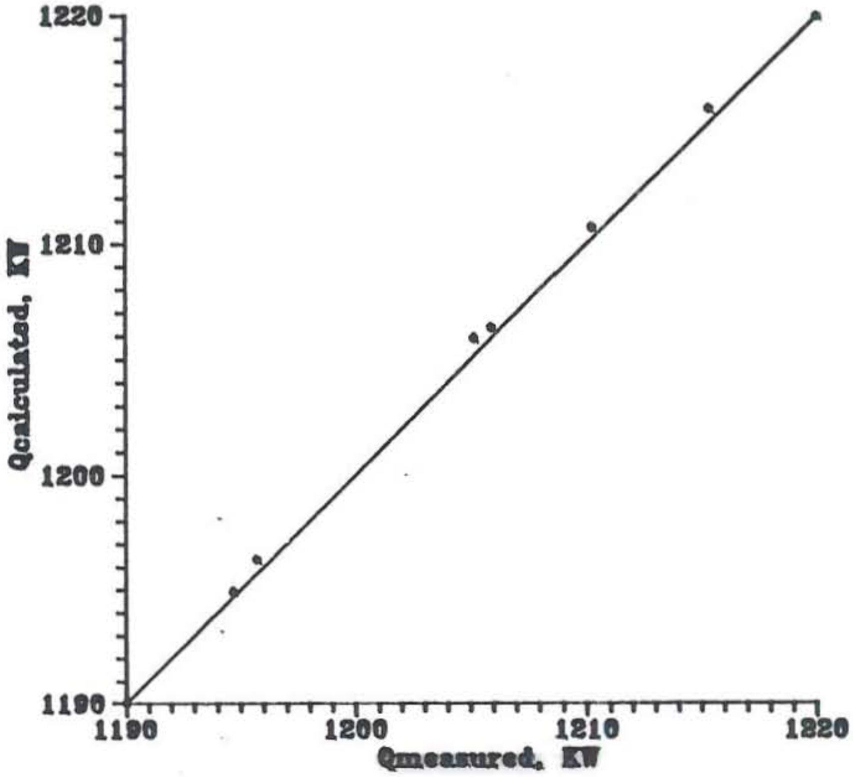


Figure 8.4 Esmil 2, Q measured vs Q calculated

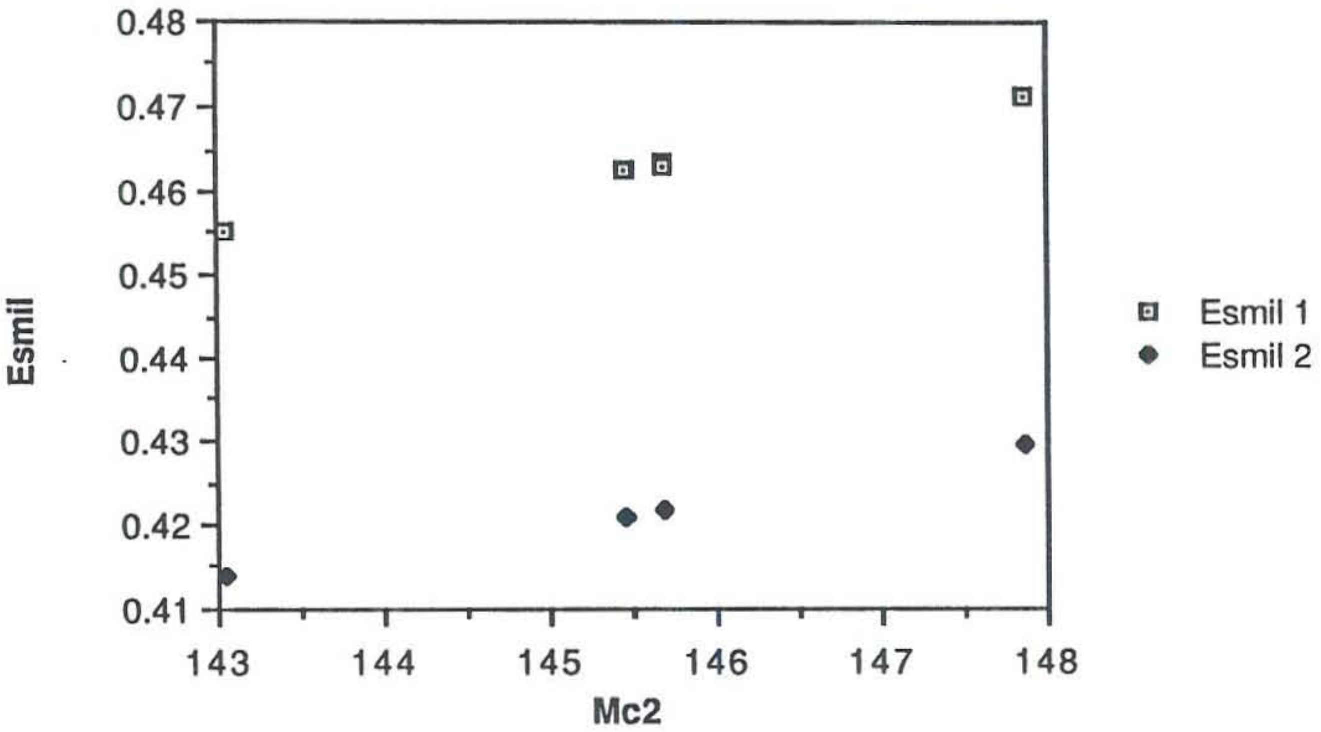


Figure 8.5 Esmil 1/2, Pressure drop vs flow rate relationship (cold side)