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HEAT TRANSFER IN PLATE HEAT EXCHANGERS

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ABSTRACT

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This study illustrates the different types of plate heat exchangers that are commonly used in various domestic and industrial applications. The main purpose of this paper was to devise a methodology that is capable of calculating optimum number of plates in the design of a plate heat exchanger. To obtain the appropriate number of plates, typically several iterations must be made before a final acceptable design is completed, since plate amount depends on many factors such as, flow velocities, physical properties of the streams, flow channel geometry, allowable pressure drop, plate dimensions, and the gap between the plates. The methodology presented here can be used as a general guide for designing a plate heat exchanger.

To investigate the effects of relevant parameters on the thermal-hydraulic design of a plate heat exchanger, several experiments were carried out for single-phase and counter flow arrangement with two brazed plate heat exchangers by varying the flow rates and the inlet temperatures of the fluid streams. The actual heat transfer coefficients obtained based on the experiment were nearly close to the calculated values and to improve the design, a correction factor was introduced. Besides, the effect of flow channel velocity on the pressure drop inside the unit is presented.

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LIST OF SYMBOLS AND ABBREVIATIONS

A	heat transfer area of the plate [m^2]
A_c	flow cross-sectional area of a channel [m^2]
A_T	nominal heat transfer area [m^2]
b	channel spacing or gap between plates [m]
C_p	specific heat capacity [J/kg.^{o}C]
D_h	hydraulic diameter [m]
f	friction factor
ff	fouling factor [%]
g	acceleration due to gravity [m/s^2]
G	mass flux [kg/m^2.s]
h	convective heat transfer coefficient [W/m^2.^{o}C]
k	thermal conductivity of a fluid [W/m.^{o}C]
k_p	thermal conductivity of the plate [W/m.^{o}C]
L	flow length of the plate [m]
LMTD	log mean temperature difference [^{o}C]
$\dot{m}$	mass flow rate [kg/s]
n	number of channels
N	number of plates effective in heat transfer
Nu	Nusselt number, dimensionless
P	pressure [bar]
P_e	wetted perimeter of a channel [m]
Pr	Prandtl number, dimensionless
Q	heat transfer rate [W]
Re	Reynolds number, dimensionless
T	temperature [^{o}C]
u	flow channel velocity [m/s]
U	overall heat transfer coefficient [W/m^2.^{o}C]
v	velocity along a pipe [m/s]

$\dot{V}$	volume flow rate [m^3/s]
w	width of the plate [m]
ΔP	pressure drop [pa]
ΔT	temperature difference [^oC]
$\Sigma\Delta P_{NI}$	distribution pressure drop [pa]
δ	thickness of the plate [m]
μ	dynamic viscosity [kg/m.s]
ρ	density [kg/m^3]

Abbreviations and subscripts

a	acceleration
c	cold fluid stream
calc	calculated value
cont	pipe contraction
enl	pipe enlargement
f	friction
g	gravity
h	hot fluid stream
i	inlet
o	outlet
t	total

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1 INTRODUCTION

The process of transferring heat between two or more fluid streams at different temperatures is often accomplished by specially designed equipment that are commonly known as heat exchangers. There exist several types of heat exchangers based on their size, shape, and transfer mode.

This study concerns plate heat exchangers (PHEs), which are one of the most common types in practice. The main objectives of this work are to present comprehensive descriptions of PHEs, and to give a general idea of the problem field of their design, sizing, and optimization. Emphasis is given to the thermal-hydraulic design of the plate heat exchanger.

In this research, a general design methodology is presented to calculate the optimum number of plates in PHEs from a given set of fluid streams and their operating conditions, allowable pressure drops, and plate geometry with dimensions.

To investigate the effects of fluid inlet/outlet temperatures, flow rates, and plate geometry on the thermal-hydraulic design of plate heat exchangers, several experiment have been carried out for a single-phase (water/water) and counter-current flow arrangement. The actual overall heat transfer coefficients obtained based on the experimental results were compared with the overall heat transfer coefficients calculated from the program. Furthermore, the effect of flow channel velocity on the pressure drop was studied.

2 BASIC FEATURES OF PLATE HEAT EXCHANGERS

Heat exchangers are devices that are used to transfer heat between two or more fluid streams at different temperatures. They can be classified as either direct contact or indirect contact type where the media are separated by a solid wall so that they never mix. Due to the absence of a wall, direct contact heat exchangers could achieve closer approach temperatures, and the heat transfer is often accomplished with mass transfer. Here the focus is on the indirect contact heat exchangers where a plate wall separates the hot and cold fluid streams, and the heat flow between them takes place across this interface. Plate heat exchangers and shell-and-tube heat exchangers are examples of indirect contact type exchangers.

The traditional shell-and-tube heat exchangers have large hydraulic diameters and small surface area to volume ratios. This problem has led to the development of different types of high performance compact heat exchangers having a heat transfer surface area to volume ratio of above 700 m^2/m^3 on at least one of the fluid sides [1, 2]. Compact heat exchangers provide a smaller size and their specific construction features also promote enhanced thermal-hydraulic performance and increased energy efficiencies, with significant materials and operating cost savings.

A plate heat exchanger is a compact heat exchanger which provides many advantages and unique application features. These include flexible thermal sizing, easy cleaning for sustaining hygienic conditions, achievement of close approach temperatures due to their pure counter-flow operation, and enhanced heat transfer performance.

2.1 Historical background

The earliest development of PHEs was for milk pasteurization, which involved heating the milk to a certain temperature, and holding it at this temperature for a short time and then immediately cooling it. This process requires the heat transfer equipment to be thermally very efficient and, most importantly, be cleaned easily. It was difficult to meet these operational requirements in most of the early heat transfer equipment that were used for pasteurization of milk, and this led to the development of PHEs.

Plate heat exchangers were not commercially exploited until the 1920s, as Dr Richard Seligman, the founder of APV International in England, invented the first operational PHE (plate pasteurizer) in 1923. Almost a decade later, Bergedorfer Eisenwerk of Alfa Laval in Sweden (AB Separator at that time) developed a similar commercial PHE [3]. The first ever Finnish plate heat exchanger was delivered to Säteri Oy, Valkeakoski for a solution heater. This unit was manufactured in Sweden in the late 1920s.

In order to accommodate larger throughput capacities, higher working temperatures, and larger pressures, among other factors, the overall design and construction of PHEs has progressed significantly to expand its uses from the original milk pasteurization to a wide range of today's industrial applications [4].

2.2 Basic operating principle

The basic operation of a PHE is similar to any other heat exchanger, including the shell-and-tube heat exchanger, in which heat is transferred between two fluid streams through a separating wall. Here, in this case, the separating wall is a plate which is used for heat transfer and to prevent mixing of the streams.

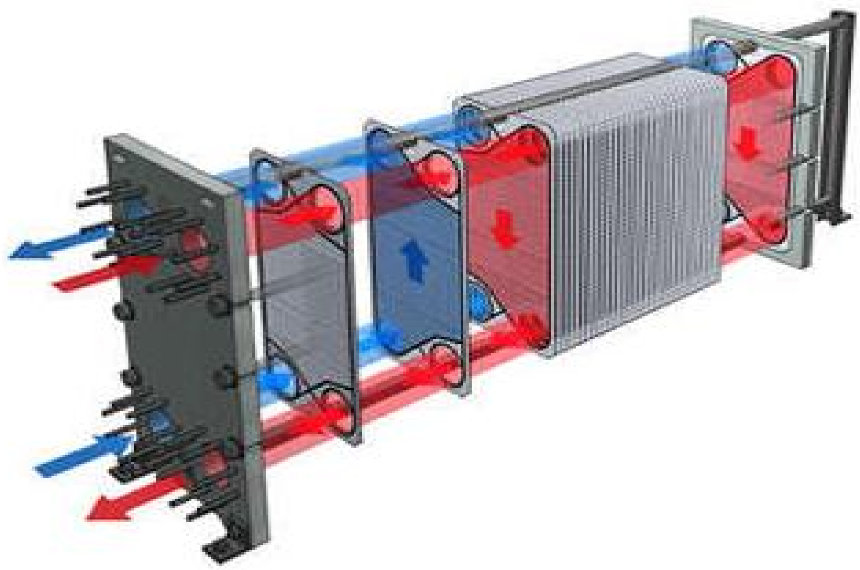

Figure 1: Operating principle of a PHE [5].

As it can be seen from Fig. 1 the hot and cold fluid streams flow into alternate channels between the corrugated plates, entering and leaving via ports at the corner of the plates. Thus, heat transfer takes place from the warm fluid through the separating plate to the colder fluid in a pure counter-current flow arrangement.

2.3 General characteristics

Due to their structural features, PHEs provide a number of advantages over the traditional shell-and-tube heat exchangers. Some of those features which are worth mentioning here include [4]:

- For comparable fluid conditions, PHEs have higher heat transfer coefficients than shell-and-tube types. This is because the plate surface corrugations readily promote enhanced heat transfer by means of several mechanisms that include promoting turbulent flows, small hydraulic diameter flow passages, and increased effective heat transfer area.
- Because of high heat transfer coefficients, PHEs usually have a much smaller thermal and physical size. For the same effective heat transfer area, the weight and

volume of PHEs are approximately only 30% and 20%, respectively, of those of shell-and-tube heat exchangers.

- Because of their true counter flow arrangements and high heat transfer coefficients, PHEs are able to operate under very close approach temperature conditions. For instance, approach temperatures of 0.3 ^{o}C in gasketed units and 0.1 ^{o}C in brazed units could be achieved. As a result, heat recovery of up to 95% and 98% are feasible in gasketed and brazed units respectively, which is a significant higher thermal performance compared to the 50% recovery for shell-and-tube heat exchangers. PHEs are therefore highly suited for use in the heat recovery from rather low-grade heat sources.
- Inspection and cleaning of gasketed PHEs can be carried out very easily as the plate-pack can be disassembled and reassembled. Gaskets can also be replaced conveniently. Moreover, these heat exchangers have a special feature of providing a great flexibility for altering their thermal sizes by simply adding or removing some plates to meet the changing heat load requirements in a process plant.
- Due to the thin channels created between the two adjacent plates, the volume of fluid contained in PHEs is small. It then enables to react with the changes in the process conditions in a short time, and it will also be easier to control.
- Plates with different surface patterns can be combined in a single PHE. Different multi-pass arrangements can also be configured. This flexibility enables better optimization of operating conditions for plate heat exchangers.
- PHEs generally have low hold-up volume and less weight; hence, their handling, transportation and installation costs are lower.
- Flow-induced vibration, noise, and erosion-corrosion due to fluid impingement on heat transfer surface are eliminated in PHEs.
- Heat loss is negligible in PHEs and no insulation is generally required. This is due to the fact that only the plate edges are exposed to the atmosphere, and the end plates do not take part in heat transfer as well.

When compared to other types of compact and non-compact heat exchangers, PHEs are very competitive for a variety of applications. However, the gasketed PHE are limited to applications only with relatively lower operating pressures and temperatures. This restriction is due to the gasket material which cannot withstand higher pressure/ temperature or the corrosiveness of the fluid, and this creates leakage problems. To overcome this disadvantage, special gasket material can be used. Moreover, several variant types of PHEs such as the brazed plate, and welded plate heat exchangers have been developed to operate at higher pressures and temperatures.

The standard gasketed plate-and-frame heat exchangers have been generally considered for operating pressures of up to 25 bar; higher pressures can also be achieved using special construction. Similarly, the maximum operating temperature of 180 ^{o}C is found in most gasketed plate heat exchangers, even though gaskets made of special materials can operate at higher temperatures [6, 7].

3 TYPES OF PLATE HEAT EXCHANGERS

On the basis of their specific structure and how the plates are attached together, several types of plate heat exchangers are available. However, the most common types are: gasketed, welded, brazed, and fusion-bonded plate heat exchangers.

3.1 Gasketed plate heat exchangers

A gasketed plate heat exchanger consists of a series of thin corrugated plates fitted with gaskets that separate the fluids. A typical gasketed plate heat exchanger is the plate-and-frame heat exchanger shown in Fig. 2. The plates come with corner parts arranged so that the two media, between which heat is to be exchanged, flow through alternate channel spaces. Appropriate design and gasketing permit a stack of plates to be held together by compression bolts joining the end plates. Gaskets prevent leakage to the outside and allow the inter-plate channels to be sealed and to direct the fluids into alternate channels, ensuring the two media never mix.

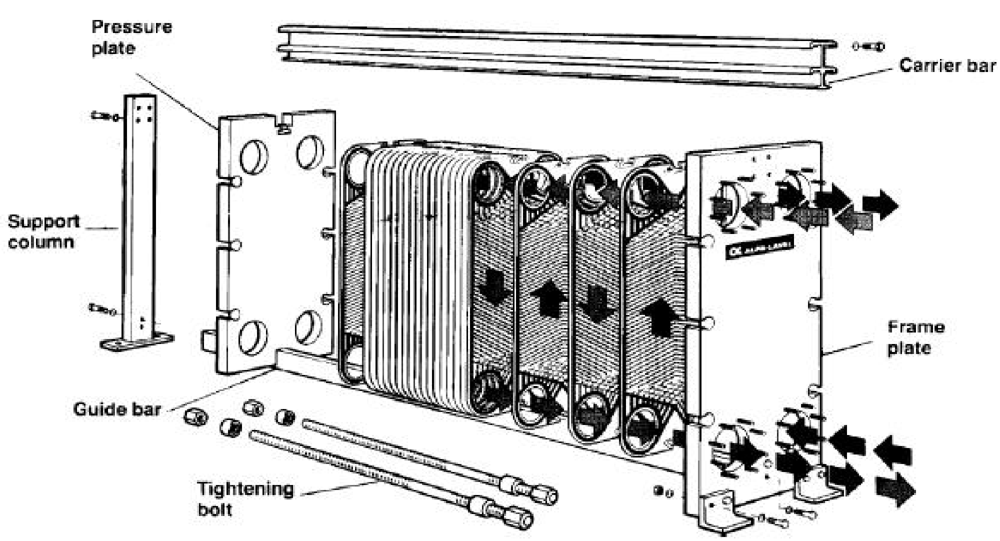

Figure 2: An exploded view of a plate-and-frame heat exchanger [8]

The operation of gasketed plate heat exchangers are constrained by the operating temperature (-40 up to 180 ^{o}C) and pressure (~25-30 bar) limits [9-11].

The special feature of this type of plate heat exchanger is that their flexible constructions admit the heat transfer plates to be removed easily for cleaning, inspection or maintenance accessibility. Moreover, heat transfer plates can also be added or rearranged to meet new process conditions.

3.2 Brazed plate heat exchangers

The brazed plate heat exchanger, as shown in Fig. 3, consists of a pack of pressed stainless steel plates brazed together, completely eliminating the use of gaskets, end frames, and bolts from the design. Instead, the plates are held together by brazing with copper under vacuum. This results in a much less complicated, lighter weight and more compact heat exchanger. Brazing of the corrugated, gasket-free plates together cause the two fluids to be directed through alternating channels between the plates. Their simple design also results in greatly reduced shipping and installation costs.

Figure 3: Typical Brazed plate heat exchanger [12].

Apart from the above features, the brazed plate heat exchangers also have exceptional strength and durability. This is due to the fact that, in addition to sealing around the periphery of the plates, the internal contact points are also brazed together at thousands of contact points in each unit which admits them to operate at higher pressures and temperatures than gasketed units.

The operating temperature of brazed heat exchangers ranges from -195 ^{o}C to 350 ^{o}C, and their maximum operating pressure is 45 bar [1, 4]. However, today's new testing methods allow brazed units to operate up to 60 bar pressure conditions [8, 17].

In terms of maintenance, the brazed plate units cannot be disassembled for cleaning or for the addition of heat transfer plates as bolted units can. If cleaning is required it can be cleaned chemically.

Brazed plate heat exchangers were originally aimed at the refrigeration /heat pump market for water-cooled evaporators and condensers. Nowadays, it is also being used for process water heating, heat recovery and district heating systems, among others. Its low cost compared to most other compact heat exchangers makes it attractive as standard equipment in plants such as chillers and air compressors [1].

3.3 Welded plate heat exchangers

This type of plate heat exchangers could be classified as semi-welded or fully-welded. Fig. 4 shows a semi-welded heat exchanger which is constructed by welding pairs of heat transfer plates (twin-plates) and assembling them in a plate-and-frame pack with gaskets only in the plate channels that handle the alternate non-corrosive fluid stream. This design is especially useful for handling relatively corrosive media, which flow in the welded twin-plate channels.

Figure 4: Semi-welded plate heat exchanger [8]

The semi-welded PHEs can withstand pressures up to 30 bar on the welded twin-plate side, and this relatively higher operating pressure extends its applications to include evaporation and condensation in the refrigeration and air-conditioning systems, among others [4].

In fully-welded heat exchangers, the hot and cold fluid streams are separated by welds and no gasket is used. Fig. 5 shows a fully welded or gasket-free PHE where a completely welded plate pack is bolted between the two end plates in a frame. This design principle ensures that the exchanger is highly resistant to pressure and temperature and also leak-tight. However, unlike the gasketed and semi-welded models, the fully-welded PHEs lose cleaning accessibility, and the flexibility of either adding or removing plates to meet varying heat load requirements. When the heat exchanger fouls, it can be cleaned chemically since mechanical cleaning is not possible.

Figure 5: Fully-welded plate heat exchanger [8]

Fully-welded PHEs are particularly attractive for applications where the heat transfer or thermal processing undergoes rapid changes in temperature or pressure. They are also intended for thermal processes that involve handling of highly aggressive or corrosive fluids. They can withstand temperatures up to 350 ^{o}C and pressures of up to 40 bar [1, 4].

The fully-welded plate and shell heat exchanger shown in Fig. 6 is manufactured by a Finnish manufacturer, Vahterus Oy [13]. The unit is constructed from a fully-welded pack of circular plates and this pack is housed within a shell (pressure vessel). The construction of this unit combines the best features of plate heat exchangers as well as shell-and-tube heat exchangers offering a durable, compact and gasket free heat exchanger capable of operating at high temperature and high pressure conditions. Based on its application areas where to be used, the fully-welded plate and shell heat exchanger has several versions. For instance, the plate and shell openable type renders the flexibility of use by allowing the fully-welded plate pack to be completely withdrawn from the shell for inspection or cleaning. And, this allows the unit to be used in slightly fouling applications.

Figure 6: Fully-welded plate and shell heat exchanger [13]

3.4 Fusion-bonded plate heat exchangers

The fusion-bonded plate heat exchanger (model AlfaNova) is the newest type of heat exchanger available only from Alfa Laval [8]. This unit is made of 100 % stainless steel with the components fused together using a patented technology (unique active diffusion bonding) that enables the unit to operate at high temperature and pressure limits, and wider application areas where to be used.

Figure 7: Fusion-bonded plate heat exchanger [8]

Fusion-bonded plate heat exchangers give higher mechanical and fatigue resistance than conventional brazed units. Moreover, the fully stainless-steel construction makes it to resist corrosion and withstand temperatures of up to 550 ^{o}C. The fusion-bonded plate heat exchanger can be seen in Fig. 7.

4 CONSTRUCTION AND OPERATION OF PLATE HEAT EXCHANGERS

As briefly described in the previous section, a plate heat exchanger is constructed from a thin, rectangular, pressed sheet metal plates that are attached together by means of brazing, welding or using frames clamped with bolts in gasketed types. The plates have circular ports at each corner in which the fluids enter and exit. The heat transfer plates have corrugations that enable the fluid inside the flow channel to induce turbulent flow which in turn increases the heat transfer between the hot and cold fluid streams.

4.1 Corrugated plate patterns

The thermal-hydraulic performance of plate heat exchangers is strongly influenced by the plate surface corrugation patterns in the plate pack they are fitted with. Heat transfer plates are normally produced by stamping specially designed corrugations on the surface of thin metallic sheets [7]. The corrugated plates used in plate heat exchangers can be manufactured from any metal or alloy that can be pressed, cold formed or welded.

When the plates are assembled in a stack, the corrugations on the adjoining plates form interrupted flow passages, and these inter-corrugation flow paths promote enhanced convective heat transfer coefficients and decreased fouling characteristics. The corrugations also increase the effective surface area for heat transfer as well as plate rigidity, and the multiple metal-to-metal contact points between adjacent plates lend greater mechanical support to the stack.

Heat transfer plates can be produced in many different sizes, shapes and corrugation patterns. However, to achieve efficient heat transfer and to be competitive in the market, each plate pattern must undergo extensive research on technical and commercial aspects.

According to the study by Shah and Focke [14] more than 60 different plate-surface corrugation patterns have been developed worldwide during the past century. Among these, Fig. 8 represents the most commonly used corrugation patterns that include: the washboard, herringbone, chevron, protrusions and depressions, washboard with secondary corrugations, and oblique washboard [4].

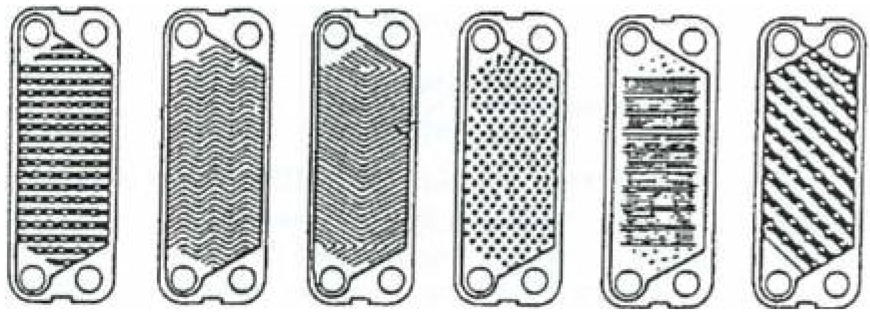
(a) (b) (c) (d) (e) (f)

Figure 8: Typical categories of different plate-surface corrugation patterns [14]:
(a) washboard, (b) herringbone, or zig-zag, (c) chevron, (d) protrusions and depressions, (e) washboard with secondary corrugations, and (f) oblique washboard.

Among others types, the corrugation patterns of chevron plates have generally been the most successful design offered by the majority of PHE manufacturers [8, 17-21]. Consequently, the application of chevron type plates has been growing considerably since the last decade [7, 15, 16].

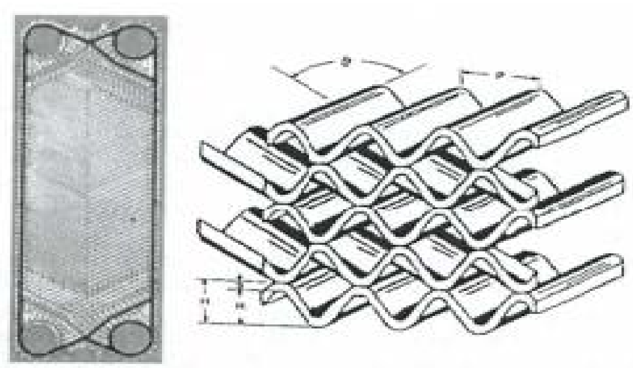

Figure 9: Corrugation pattern of a chevron-type plate with its inter-plate flow channel [20, 22].

Typical features of chevron-type surface pattern and inter-plate cross-corrugated flow channels are shown in Fig. 9. The corrugations are pressed to the depth of the plate spacing. This means that the two adjacent plates will have numerous contact points and will produce a more turbulent flow and rigid structure as well. Moreover, the flow passages enhance the heat transfer coefficient by increasing the effective surface area, disrupting boundary layers, and promoting swirl flow [15, 23].

For adjacent plates the corrugation angles are placed in opposite directions. The angle and depth of the corrugations together with the dimensions of the plates determine the thermal and hydraulic properties of the plate heat exchanger [21].

4.2 Plate material

The most commonly used plate metal in plate heat exchangers is stainless steel, although other materials such as titanium, nickel, incoloy, hastelloy, and tantalum can be used as well [1, 4].

The selection of plate material is primarily determined by fluid compatibility and heat duty. However, different manufacturers use different plate materials based on their specific design, application, and cost preferences. Special applications may require special plate material but generally they can be classified into the following groups [11, 24, 25]:

- Stainless steel: EN 1.4301, EN 1.4401, EN 1.4550; or it includes alloys such as, EN 1.4571, EN 1.4547, EN 1.4539, EN 1.4434, etc.
- Nickel alloys, which include: EN 1.4539, EN 1.0402, EN 2.4061, EN 2.4605, EN 1.4562, EN 1.4563, EN 2.4858, EN 2.4606, EN 2.4856, etc.
- Titanium and titanium alloys, which include grades ASTM Gr1, Gr2, and Gr11
- Other metal/metal alloys and non-metals, such as graphite, tantalum, etc.

Based on the type of fluid and plate material compatibility, Table 1 can be used as a general and simplified selection guide for different application areas.

Table 1: Plate material selection guide for typical fluid media [24, 26]

Material	Fluid
Stainless steel	Water, cooling tower water, dilute chloride solutions, copper sulfate solutions, food products, pharmaceutical media, brews, etc
Nickel	Caustic (50-70%) solutions
Incoloy	Hydrogen gas/water vapour with mercury carryovers, and acids ($\leq$ 70 ^{o}C)
Hastelloy	Sulfuric and nitric acids
Titanium	Sea or brackish water, dilute acids ($\leq$ 70 ^{o}C), chlorine solutions, and chlorinated brines
Titanium-palladium alloys	Dilute nitric and sulfuric acids (10% concentration and $\leq$ 70 ^{o}C)

The thermal conductivity of the plate wall is also an important consideration for the thermal-hydraulic design of a plate heat exchanger. Therefore, in this case, plate materials with higher thermal conductivity are preferred.

4.3 Gasket material

Gaskets are typically one-piece moulded elastomers that are used in gasketed and semi-welded plate heat exchangers. The performance of plate-and-frame heat exchangers such as their safety and lifetime leakage-proof reliability is highly dependent on the gaskets used and their compatibility with the process fluid [27].

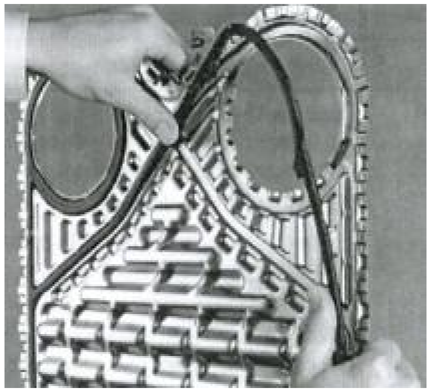

Figure 10: A typical gasket setting on a plate surface [20]

As it can be seen in Fig. 10, the gaskets are placed in the peripheral grooves on the corrugated plate surface to prevent intermixing of the media and leakage to the outside.

Gaskets are generally made from a variety of elastic and formable materials, such as rubber and its different forms. The selection and manufacture of gaskets depends on the basis of their specific material characteristics that include fluid compatibility, operating temperature, and pressure conditions.

The most commonly used gasket materials in plate heat exchangers include: Nitrile Butadiene (NBR), Ethylene Propylene Rubber (EPDM), Fluorocarbon Rubber (FPM), and fluoro-elastomer (VITON) [4].

4.4 Operation and selection

The basic principle of most plate heat exchanger variants is similar to other heat exchangers such as shell-and-tube types. However, they differ in their sizes, hold-up volume, and operating conditions including flow rates and approach temperature difference of the two fluid streams. In principle, a single-pass plate heat exchanger is a pure counter-flow exchanger, which along with its relative compactness and enhanced convective characteristics allows for a very small approach temperature operation.

Even though gasketed plate heat exchangers are relatively more compact and provide enhanced thermal performance than shell-and-tube heat exchangers, their selection, application, and operation are highly influenced by the operating temperature and pressure limits. In the selection of gasketed plate heat exchangers, gasket detention and plate deformation due to high pressures of the fluid streams have to be considered.

There are several other factors that influence operating constraints and the following may be considered during the selection of plate heat exchangers [4]:

- Complex inter-plate flow channels usually impair high shear rates and shear-sensitive media may thus be prone to degradation.
- Possibility of flow maldistribution in handling highly viscous fluids or very low flow rates.
- Plate manufacturing governs the overall size of the plate heat exchanger (based on size of presses for stamping them), which in turn restricts applications requiring very high flow rates as the pressure drop becomes excessive.
- High pressure drop also makes them unsuitable for air cooling, gas-to-gas heat exchange, and low operating pressure consideration applications.

Despite the above factors, plate heat exchangers offer viable alternative features than most types of other heat exchangers. This is because, their compact construction, ease of cleaning, flexibility of altering the thermal size, pure counter-current flow operation, and enhanced thermal-hydraulic performance can compensate for any limitations in most cases.

Due to the existence of many different types of plate heat exchangers and numerous manufacturers, it is difficult to precisely summarize the operating limits. However, to give a general overview, a summary of typical operating ranges for gasketed and brazed plate heat exchangers is given in Table 2. This is based on the product lists from the following manufacturers: Alfa Laval [8], SWEP [17], Tranter [18], and APV [19].

Table 2: Typical operating range of gasketed and brazed plate heat exchangers [4]

	Gasketed PHEs	Brazed PHEs
Maximum operating pressure	25bar (30 bar with special Construction)	60 bar
Maximum operating temperature	160 ^oC (200 ^oC with special gaskets)	350 ^oC
Maximum flow rate	3600 m^3/h	140 m^3/h
Heat transfer coefficient	Up to 7500 W/ (m^2 ^oC)	Up to 11000 W/ (m^2 ^oC)
Heat transfer area	0.1 – 2200 m^2	0.02 – 80 m^2
Maximum connection size	450 mm	100 mm
Approach temperature difference	As low as 0.3 ^oC	As low as 0.1 ^oC
Heat recovery	As high as 95 %	As high as 98 %
Pressure drop		Up to 100 kpa per m channel length
Number of plates	Up to 1200	
Port size	Up to 435 mm	
Plate thickness	0.4 – 1.2 mm	0.3 – 0.4 mm
Plate size	0.4 – 3.5 m length	0.2 – 3.5 m length
Plate spacing	2.2 – 16.0 mm	1.5 – 3.0 mm

5 PLATE HEAT EXCHANGERS AS COMPACT HEAT EXCHANGERS

A compact heat exchanger is characterized by having a comparatively large amount of surface area in a given volume, compared to the traditional types, most specifically the traditional shell-and-tube heat exchanger. This special feature of compact heat exchangers results in reduced size and weight, and frequently reduced cost and improved performance.

The most widely available compact heat exchanger types in the process industries include: plate-and-frame heat exchangers, brazed plate heat exchangers, welded plate heat exchangers, spiral heat exchangers, plate-fin heat exchangers, the Marbond heat exchangers, printed-circuit heat exchangers, and compact shell-and-tube heat exchangers.

As the plate heat exchanger is one type of compact heat exchangers, it shares the following advantages and limitations of a typical compact heat exchanger.

5.1 Benefits of compact heat exchangers

Compact heat exchangers offer a number of benefits over the traditional shell-and-tube heat exchangers. Their principal advantages are described below [1]:

- Most compact heat exchangers give improved energy efficiency compared to shell-and-tube units. For instance, the plate-and-frame heat exchangers are now able to achieve 98 % efficiency.
- Compact heat exchangers can achieve closer approach temperatures which allow a greater amount of energy contained in one stream to be transferred to the other stream. This in turn leads to better heat recovery, and more efficient evaporators and condensers.
- Due to their reduced size and weight, the transportation and installation cost of compact heat exchangers are relatively lower. Additionally, less pipe-work, and reduced foundation is needed.
- To increase their effectiveness, several of the compact heat exchanger can be configured for multi-pass and multi-stream applications.

- Compact heat exchangers allow tighter temperature control. This is often beneficial when dealing with heat-sensitive materials and can lead to improved product quality and consistency.
- The lower fluid hold-up required by compact heat exchangers, compared to traditional units, can lead to safer operating conditions and also allows them to react more quickly to changes in plant conditions.

5.2 Limitations of compact heat exchangers

The main concern of potential users of compact heat exchangers is fouling. The perceived limitation of compact heat exchangers is the perception that those with small passages are likely to foul. However, some types of compact heat exchangers, such as spiral or plate-and-frame heat exchangers are designed specifically to handle fouled streams. This is because their flexible construction allows them to dismantle and clean the heat transfer surfaces. Moreover, plate heat exchangers normally have lower fouling rates which are achieved as a result of the turbulence created by their corrugated plate patterns. The other limitation of compact heat exchangers is that they are only suitable to applications with relatively low fluid pressure and temperature conditions.

5.3 Typical applications of plate heat exchangers

Compact heat exchangers are becoming increasingly important in many industrial processes, both as contributors to increased energy efficiency, and more recently as the basis for novel 'intensified' unit operations [1].

The use of compact heat exchangers also facilitate the repackaging of air conditioning and refrigeration equipment; for instance, the much reduced volume of brazed plate heat exchangers compared to their shell-and-tube counterparts effectively enabling a new approach to be made to the modular design of liquid chillers.

Plate heat exchangers, with their relative compactness and enhanced thermal-hydraulic performance, provide an additional attractive feature of flexibility in altering their thermal size by easily adding or removing the heat transfer plates.

Plate heat exchangers are universally used for efficient heating, cooling, heat recovery, condensation and evaporation in a multitude of applications. Their broad applications cover refrigeration and air-conditioning /HVAC/, heat pump systems, energy production, marine power systems, food and beverage processing, pharmaceutical and biotechnological industry, and petrochemical systems, among others.

6 BASIC DESIGN METHODS OF PLATE HEAT EXCHANGERS

The thermal-hydraulic design of plate heat exchangers is essentially similar to the general methodology employed for designing any other type of exchanger. The major design considerations may include:

- process/design or problem specifications
- thermal and hydraulic design
- mechanical/structural design, and operation and maintenance constraints
- manufacturing considerations and cost
- trade-off factors and system-based optimization

The process/ design specification provides all the necessary information to optimally design the exchanger for a particular application. This includes type of exchanger construction and material used, types of fluids and their flow arrangement, heat load, and pressure drop constraints. The mechanical design includes essential aspects of the mechanical or structural integrity of the exchanger under both steady-state and transient operating conditions. Manufacturing evaluations and cost estimates have to be made so that appropriate trade-offs can be considered in order to perform a system-based optimization.

6.1 Thermal-hydraulic design

The thermal-hydraulic design of a heat exchanger involves the quantitative evaluation of heat transfer, pressure drop, and sizing/rating of the exchanger.

The two fundamental problem specifications in the thermal-hydraulic design of heat exchangers are those of rating and sizing. The rating problem is concerned with the determination of the heat load and the fluid outlet temperatures for prescribed fluid flow rates, inlet temperatures, and allowable pressure drops of each side for an existing plate heat exchanger, where the heat transfer surface area, flow arrangement, and flow passage dimensions are specified. The sizing problem, on the other hand, requires the determination of construction type, flow arrangement, needed surface area (or size of exchanger) for a given set of fluid streams and their operating conditions (inlet/outlet temperatures and flow rates), the specified heat load, and pressure drop constraints [9].

6.2 Flow arrangement and distribution

The arrangement of the hot and cold fluid flows relative to each other is important for how efficiently the heat transfer area of the heat exchanger can be used to transfer the required heat load. In plate heat exchangers, the following three different flow arrangements for the hot and cold fluid streams are generally encountered:

- Parallel-flow arrangement with two fluid streams flowing in the same direction (Fig. 11)
- Counter-flow arrangement with two fluid streams flowing in opposite directions (Fig. 12); and
- Multi-pass arrangement where the path of at least one fluid stream is reversed through the flow length two or more times (Fig. 13)

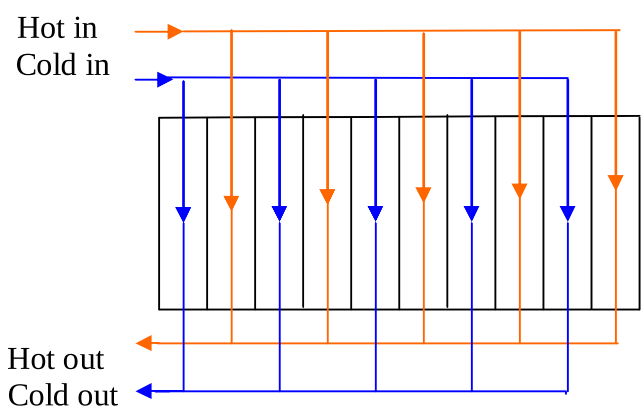

Figure 11: Parallel-flow arrangement in a two-fluid PHE

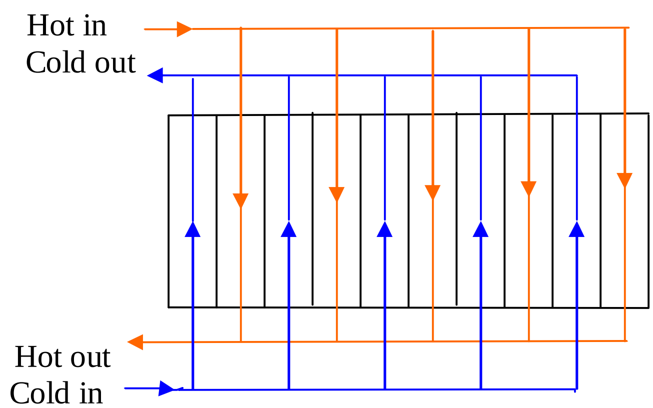

Figure 12: Counter-flow arrangement in a two-fluid PHE

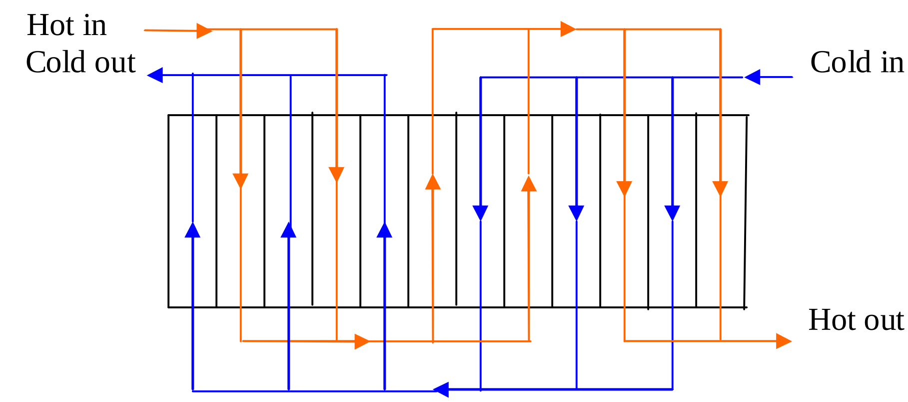

Figure 13: A typical multi-pass arrangement in a two-fluid PHE

The temperature of the fluid streams in plate heat exchangers generally vary along their flow path because of the flow distribution and temperature gradient variations across the plates. Figures 14 and 15 show the temperature variations in parallel- and counter-flow arrangements for single-phase flows of two fluid streams. In Figures 14 and 15, A is the heat transfer area, $T_{h,\,i}$ and $T_{h,\,o}$ are the inlet and outlet temperatures of the hot stream, and similarly $T_{c,\,i}$ and $T_{c,\,o}$ are the inlet and outlet of the cold stream respectively.

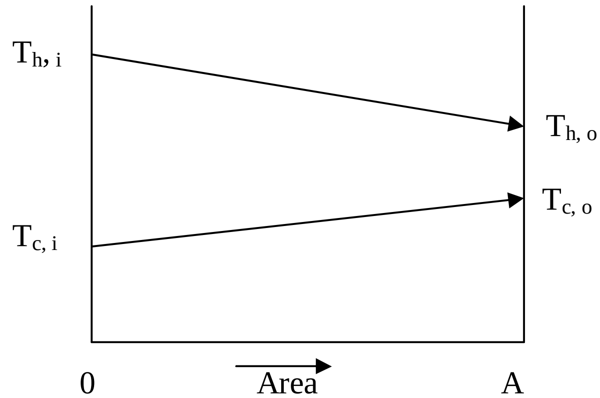

Figure 14: Temperature distribution in parallel-flow arrangement

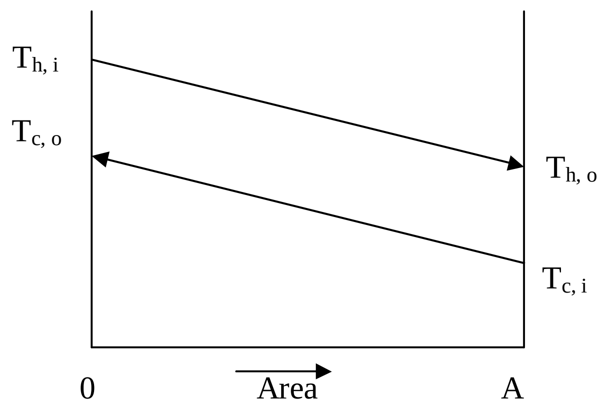

Figure 15: Temperature distribution in counter-flow arrangement

In parallel-flow arrangement, the final temperature of the cold fluid stream is always lower than the outlet hot fluid stream temperature. In the limiting case of an infinitely large area, the two would be equal. However, in a counter-flow arrangement, because of a favourable temperature gradient, the final cold fluid stream temperature may exceed the outlet temperature of the hot fluid stream. This thermodynamic advantage of a counter-flow arrangement results in a smaller surface area requirement for a given heat load when compared with a parallel-flow arrangement. Hence, only counter-flow arrangements will be considered here.

6.3 Energy balance and design equations

The total heat transfer rate between the counter flows in plate heat exchangers can be calculated from one of the fluid streams as follows:

$$Q = \dot{m}_h \, C_{p,h} \, \Delta T_h \tag{1}$$

Where: $\dot{m}_h$ is the mass flow rate of the hot stream (water), $C_{p,h}$ is the specific heat capacity of hot water, and ΔT_h is the temperature difference between the inlet and outlet of the hot stream ($T_{h,i} - T_{h,o}$).

According to the 1st law of thermodynamics, the total heat transfer rate in both the hot and cold fluid streams is equal. Therefore, the mass flow rate of the cold side $\dot{m}_c$ could be obtained from the ratio of the total heat rate to the specific heat capacity multiplied by the temperature difference between the inlet and outlet of the cold stream.

$$\dot{m}_c = \frac{Q}{C_{p,c} \Delta T_c} \tag{2}$$

Where: $C_{p,c}$ is the specific heat capacity of cold water, and ΔT_c is the temperature difference between the inlet and outlet of the cold stream ($T_{c,o} - T_{c,i}$).

The logarithmic mean temperature difference LMTD can be obtained from the basic counter flow LMTD equation:

$$\text{LMTD} = \frac{\left(T_{h,i} - T_{c,o}\right) - \left(T_{h,o} - T_{c,i}\right)}{\ln\left(\frac{\left(T_{h,i} - T_{c,o}\right)}{\left(T_{h,o} - T_{c,i}\right)}\right)} \tag{3}$$

The overall heat transfer coefficient U can be calculated from the basic heat transfer equation:

$$U = \frac{Q}{A_T \, LMTD} \tag{4}$$

Here, as Shah and Focke suggested [28], A_T is the nominal heat transfer surface area which is obtained by multiplying the projected area of a single plate A (A = L w) with the total number of effective plates in heat transfer.

$A_T = A\ N$ (5)

Eqn. (4) is used when the total heat transfer area is known. However, for the rating or sizing problem, the overall heat transfer coefficient has to be estimated from the two-fluid operating conditions.

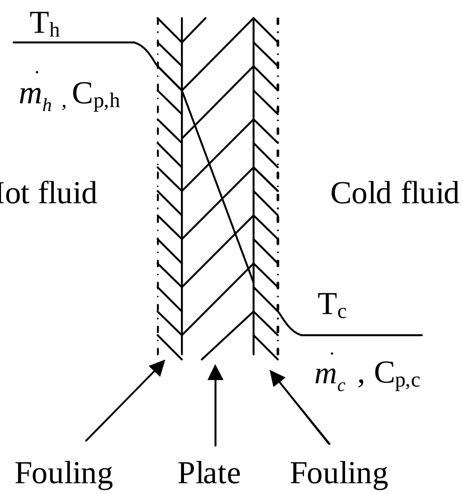

Figure 16: Elements of the overall thermal resistance of a single-pass two-fluid PHE.

As Fig. 16 shows, the overall heat transfer coefficient in a PHE is a function of the convective heat transfer coefficients in the two fluid streams, their fouling resistances, and the thermal resistance due to conduction through plate thickness. The approach used to calculate the overall heat transfer coefficient is expressed below.

The hydraulic diameter D_h is expressed as the ratio of four times the flow channel cross-sectional area to the wetted perimeter. Shah and Wanniarachchi [29] suggested to use two times the channel spacing as the hydraulic diameter for plate heat exchangers. This has come from the fact that compared to the width (w) the channel spacing (b) is negligible, hence, $b + w \cong w$.

$$D_h = \frac{4A}{P_e} = \frac{4[bw]}{2[b+w]} = 2b \tag{6}$$

The number of channels for the hot (n_h) and cold (n_c) sides in plate heat exchangers could be calculated respectively as:

$n_h = \frac{N-2}{2}$ (7)

$n_c = \frac{N}{2}$ (8)

The flow cross-sectional area for a single channel A_c is equal to the plate spacing times the width of the plate:

$A_c = b\ w$ (9)

The mass flux G, and the flow velocity u, in a single channel can be expressed as:

$G = \frac{\dot{m}}{A_c n}$ (10)

$u = \frac{G}{\rho}$ (11)

The dimensionless numbers Re, Pr, and Nu for the single-phase flow streams in the design of PHEs could be obtained from the following equations [30]:

$Re = \frac{\rho u D_h}{\mu}$ (12)

$Pr = \frac{C_p \mu}{k}$ (13)

$Nu = \frac{h D_h}{k} = 0.37\ Re^{0.67}\ Pr^{0.33}$ (14)

Where: μ is the dynamic viscosity, h is the convective heat transfer coefficient, k is the thermal conductivity, and ρ is the density of the fluid.

$h = \frac{Nuk}{D_h}$ (15)

Thus, the overall heat transfer coefficient can be estimated from the following equation:

$U = \frac{1}{\left(\frac{1}{h_h} + \frac{1}{h_c} + \frac{\delta}{k_p}\right)\left(1 + \frac{ff}{100}\right)}$ (16)

Where: δ is the plate thickness, k_p is thermal conductivity of plate material (stainless steel), and ff is the fouling factor in percentage.

The total pressure drop ΔP_t in a plate heat exchanger consists of several friction and head loss elements and can be expressed as [4]:

$$\Delta P_t = \Delta P_f + \Delta P_g + \Delta P_a + \Sigma \Delta P_{Ni} \tag{17}$$

Where: ΔP_f is the frictional pressure drop or shear loss, ΔP_g is the pressure drop due to gravity, ΔP_a is the flow acceleration pressure drop, and $\Sigma \Delta P_{Ni}$ is the sum of all other pressure losses due to inlet/outlet flow distribution and includes the pressure drop in ports and manifolds.

The frictional pressure drop for single-phase flow applications [31] is usually calculated as:

$$\Delta P_f = 2f \left[\frac{L}{D_h}\right]\left[\frac{G^2}{\rho}\right] \tag{18}$$

Where L is the plate flow length between ports, G is mass velocity or flux inside a channel, ρ fluid density, and f is the Fanning friction factor.
The Fanning equation is expressed as [30]:

$$f = \frac{2.5}{\mathrm{Re}^{0.3}} \tag{19}$$

The gravitational pressure drop for single-phase flows in a vertical channel can be calculated from:

$$\Delta P_g = \pm \rho g L \tag{20}$$

Here the '+' sign is for vertical up flow and the '-' sign is for vertical down flow. And, g is the gravitational acceleration.

The pressure drop due to flow acceleration ΔP_a is usually negligible for single-phase flows.

The additional flow distribution pressure drops are estimated in accordance with Shah and Focke [28] as follows:

$$\Sigma \Delta P_{Ni} = \frac{1.5 G^2}{2\rho} \tag{21}$$

In addition to the above losses, the frictional pressure losses at the inlet/outlet pipe connections due to sudden enlargement or contraction in the cross-section of the pipes should also be considered and could be respectively expressed as [32]:

$$\Delta P_{enl} = \frac{\rho [v_1 - v_2]^2}{2} \tag{22}$$

$$\Delta P_{cont} = \frac{\rho [v_2 - v_1]^2}{2} \tag{23}$$

Where, v_1 and v_2 are the velocities of the fluid inside the smaller and larger pipes respectively.

6.4 Design methodology for calculating plate amount

To calculate the number of plates in plate heat exchangers, the plate dimensions as well as the hot and cold fluid streams with their inlet and outlet temperatures should be given. The flow rate of at least one of the fluid streams is normally known. However, the flow rate of the second stream can be calculated from the heat transfer rate equation. To obtain the appropriate number of plates, typically several iterations must be made before a final acceptable design is completed. This is due to the fact that the plate amount depends on many factors such as, flow channel velocities, physical properties of the fluid streams, flow channel geometry, allowable pressure drops, plate spacing, plate thickness, plate size, and plate material, among others.

In this research work, a systematic design methodology was devised to calculate the number of plates. This can be seen from the schematic diagram outlined in Fig. 17.

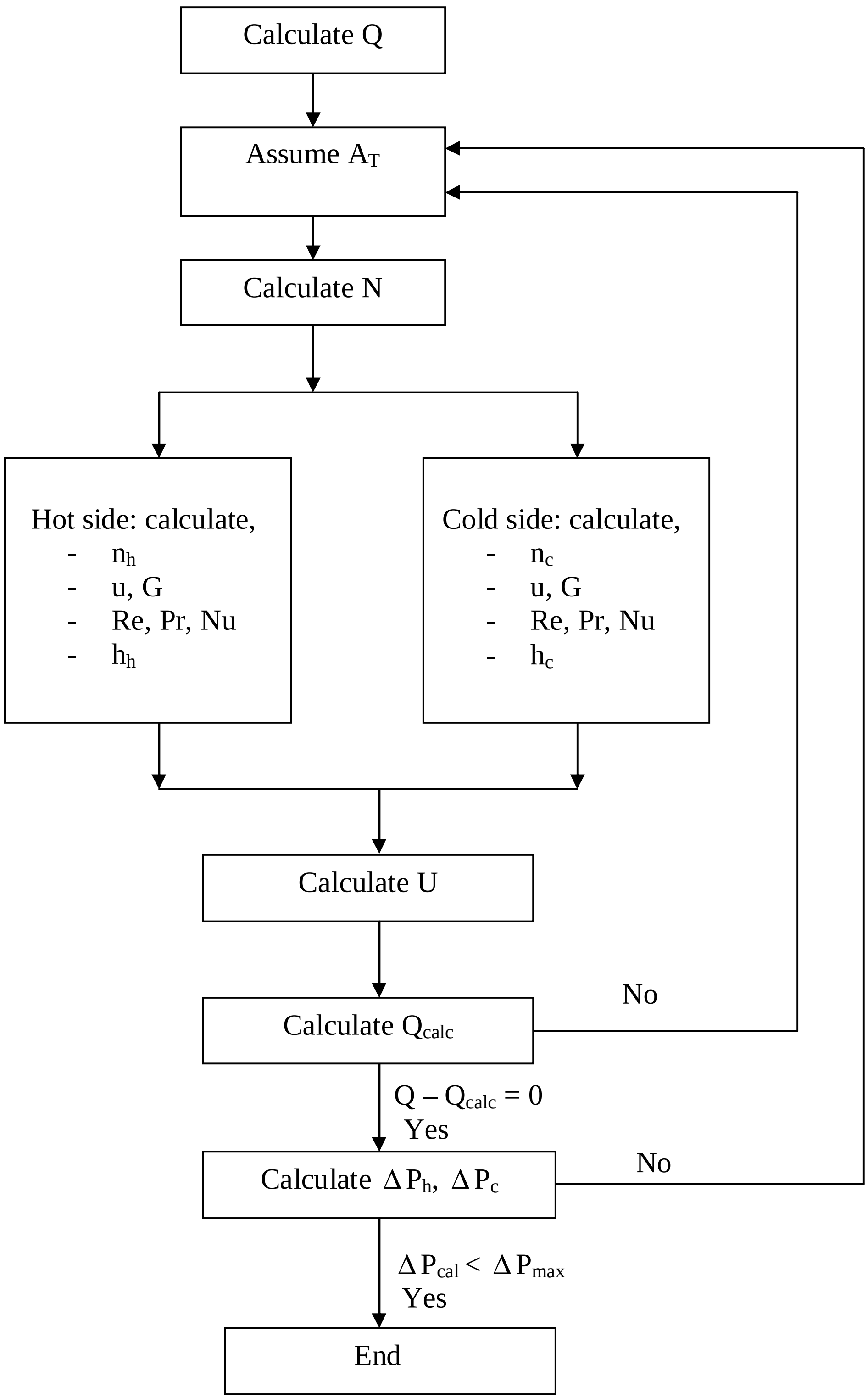

Figure 17: Simplified schematic diagram of the design methodology

The principle of the methodology is based on assuming the total heat transfer area for the plate heat exchanger and calculating the overall heat transfer coefficient from Eq. (16). Then, the appropriate number of plates can be obtained by equating the heat transfer rates obtained from Eq. (1) and Eq. (4).

The maximum pressure drop was considered as one of the design specifications. If the calculated pressure drop becomes higher than the maximum allowable value, additional plates have to be added by increasing the margin.

7 EXPERIMENTAL SET-UP AND PROCEDURE

The experimental system established here to study the effect of flow rate and inlet/outlet temperatures on the thermal-hydraulic design of plate heat exchangers is schematically shown in Fig. 18.

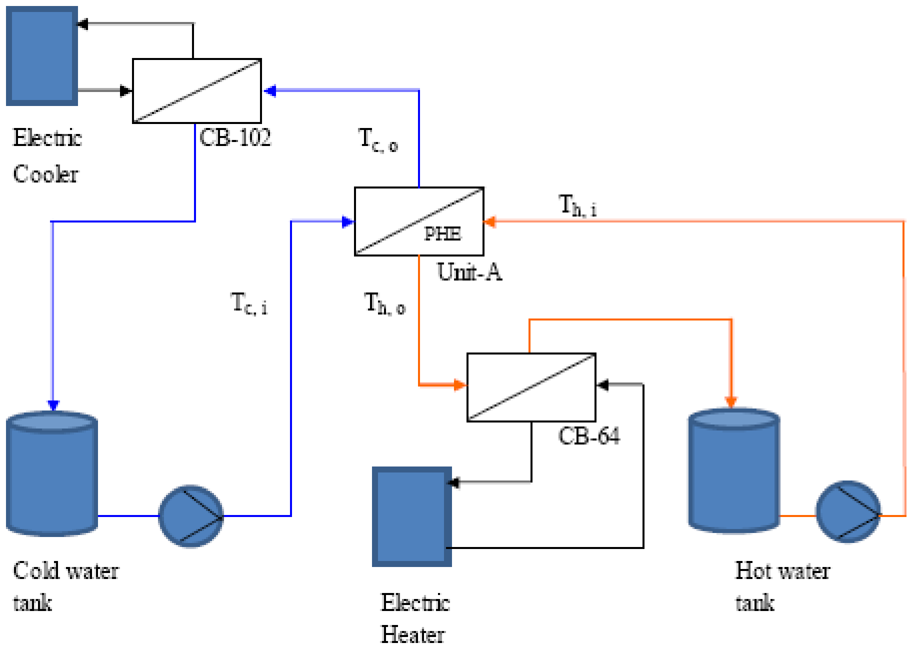

Figure 18: Simplified schematic diagram for the lab ring arrangement

As it can be seen in the diagram the configuration comprises the following main components:

- Three Loval's brazed plate heat exchangers, where unit-A is the test unit, CB-64 and CB-102 are the intermediate units. The flow inside the unit is counter-current i.e., hot water flows downwards and cold water flows upwards.
- Two water tanks, the first is cold water tank and the second is hot water tank.
- Two pumps were used. One for the hot water and the other for the cold water.
- Two thermostats as electric heater and electric cooler. The electric heater was used to further increase the temperature of the outlet hot water leaving the test unit by exchanging the heat in the CB-64 unit. Similarly, the electric cooler was used to reduce the temperature of the outlet cold water leaving the test unit and this was achieved in the CB-102 unit.

The procedures followed in the experiment are briefly explained as follows. Initially, the system was kept at 50 ^{o}C for the hot side and 10 ^{o}C for the cold side. The maximum temperature inside the hot water tank was achieved by filling the tank with hot tap water and then by using the electric heater to raise the temperature to 50 ^{o}C. Similarly, the cold water tank was filled with cold tap water and the electric cooler was used to further reduce the temperature to the required level of 10 ^{o}C. Once these temperatures had been achieved, both streams were allowed to circulate and after about three minutes steady-state was observed. Finally, data was collected with different temperature mix points until thermal balance was reached in both tanks. Several experiments were carried out, and the temperature of the water inside the tanks were adjusted to the above values at the beginning of every experiment, and by changing the flow rates data was collected with different temperature mix points.

8 RESULTS AND DISCUSSION

In this chapter, the results of the experimental and calculated overall heat transfer coefficients are presented. The dependence of pressure drop on the fluid velocity inside the plate channel can also be seen at the end of this section.

Two brazed plate heat exchangers (unit-A and unit-B) with different plate amounts were tested and data of their corresponding inlet and outlet temperatures, and flow rates were collected.

The experimental overall heat transfer coefficient, U_{exp}, is the actual value obtained using Eq. (4). Here the total area of the plate is given, i.e., the number of plates of the unit and the area of a single plate are known. However, the calculated overall heat transfer coefficient, U_{prog}, was obtained from Eq. (16) using the simulation program. Its methodology is represented in Fig. 17.

The results of the experimental and calculated overall heat transfer coefficients are shown in Tables 3 – 9.

Table 3: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2.\,^oC$	$W/m^2.\,^oC$
10.00	40.33	47.10	14.93	460	1930.06	1915.74
12.30	33.57	38.13	15.93	471	1940.42	1865.54
14.30	32.87	36.87	17.37	471	1977.37	1877.89
16.30	32.30	35.70	18.93	474	2005.85	1894.88
18.30	31.77	34.63	20.50	473	2010.31	1904.15
20.37	31.20	33.50	22.10	474	2042.81	1920.28
22.40	30.90	32.80	23.73	474	2035.55	1944.22
24.30	31.00	32.47	25.30	476	2116.81	1970.13
26.40	31.00	32.00	27.03	477	2239.99	1998.59
28.00	30.63	31.20	28.33	477	2358.91	2015.23
29.20	30.00	30.20	29.27	478	2716.32	2061.91

Table 4: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.30	34.40	40.80	15.40	907	3044.61	2847.39
12.37	33.07	38.70	16.60	897	3063.23	2838.98
14.60	32.87	37.70	18.33	903	3109.42	2867.92
16.30	32.70	37.10	19.63	900	3098.00	2885.27
18.40	32.50	36.27	21.23	900	3124.91	2909.30
20.50	32.23	35.37	22.83	895	3128.00	2922.82
22.20	31.80	34.33	24.10	893	3141.99	2928.58
24.40	31.20	32.93	25.73	893	3198.03	2938.29
26.40	30.60	31.60	27.17	892	3397.71	2945.59
28.20	30.10	30.50	28.50	891	3878.99	2954.70
29.20	30.00	30.20	29.30	888	4191.59	3022.83

Table 5: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.20	38.60	47.13	18.20	1774	4700.09	4423.92
12.67	37.07	44.43	19.50	1780	4734.67	4434.07
14.33	36.40	43.07	20.50	1779	4734.67	4442.92
16.50	35.27	40.87	21.80	1781	4716.56	4440.03
18.60	34.20	38.90	23.00	1785	4721.49	4455.25
20.60	33.17	36.90	24.10	1782	4775.89	4451.58
22.40	32.27	35.23	25.13	1782	4789.40	4462.62
24.50	31.53	33.67	26.43	1782	4801.38	4486.05
26.20	31.10	32.60	27.53	1782	4837.05	4509.27
28.30	30.60	31.30	28.90	1778	4977.22	4536.27

Table 6: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
12.40	40.37	47.90	22.10	2466	5617.82	5342.75
14.50	38.03	44.43	22.63	2471	5637.11	5315.63
16.63	37.10	42.60	23.67	2468	5666.62	5317.63
18.60	36.03	40.77	24.60	2470	5653.73	5328.30
20.60	35.23	39.20	25.60	2478	5710.78	5353.53
24.50	33.77	36.30	27.63	2481	5772.88	5394.51

Table 7: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.30	34.20	39.50	14.20	903	1748.01	1697.55
12.40	32.97	37.40	15.63	904	1809.36	1699.88
14.37	32.40	36.37	17.17	905	1810.83	1714.44
16.20	32.20	35.70	18.53	903	1883.04	1728.33
18.60	32.00	35.00	20.47	904	1918.77	1753.34
20.40	31.80	34.37	22.03	901	1880.79	1761.83
22.30	31.63	33.67	23.60	901	1929.33	1773.81
24.60	31.13	32.53	25.50	904	1961.18	1789.65
26.30	29.90	30.70	26.80	903	1926.81	1791.61
27.10	29.33	29.80	27.33	902	2316.83	1802.20

Table 8: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2. ^oC$	$W/m^2. ^oC$
10.47	42.53	50.43	16.57	1749	2970.95	2757.11
12.43	40.63	47.70	17.73	1735	2956.26	2737.47
14.60	37.97	44.00	18.90	1724	2953.64	2712.70
16.70	36.40	41.53	20.30	1720	2951.94	2707.79
18.37	35.40	39.87	21.37	1713	3002.25	2708.55
20.30	34.30	38.00	22.73	1712	3022.39	2713.66
22.40	33.17	36.00	24.30	1710	2993.29	2712.53
24.20	32.33	34.53	25.57	1704	3042.70	2724.74
26.10	31.80	33.13	27.13	1790	3195.83	2782.61
28.50	31.20	31.80	29.00	1775	3164.46	2773.47
29.57	30.60	30.80	29.70	1777	4200.94	2802.56

Table 9: Comparison of experimental and calculated overall heat transfer coefficients

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h	$W/m^2. ^oC$	$W/m^2. ^oC$
12.50	40.77	47.93	18.80	2459	3722.05	3385.81
14.40	37.50	43.40	19.60	2458	3685.94	3337.34
16.50	36.03	41.10	20.80	2463	3735.40	3347.28
18.50	35.10	39.30	22.17	2469	3759.22	3351.22
20.60	34.13	37.60	23.60	2462	3727.46	3356.83
22.20	33.47	36.40	24.70	2465	3717.48	3371.17
24.40	32.63	34.73	26.17	2456	3803.50	3376.88
26.20	31.90	33.33	27.40	2462	3887.17	3391.06
28.30	30.90	31.50	28.80	2457	4223.89	3391.39

As the data in the above tables show, the calculated overall heat transfer coefficients obtained from the program were slightly lower than the actual experimental values. These small discrepancies could be due to the fact that, in reality the flow velocities inside the channels cannot be equal. Besides, getting a reliable data was not easy when the inlet and outlet temperatures of the hot and cold streams become very close to each other. Therefore, it is crucial to make some correction so as to have a more reasonable and optimal design of the plate heat exchanger.

A correction factor, k-value, was introduced to the program to adjust the overall heat transfer coefficient. This k-value was observed to be a function of the input parameters. Hence, the newly calculated overall heat transfer coefficient can be expressed as the overall heat transfer coefficient calculated from Eq. (16) multiplied by the correction factor.

Tables 10 – 16 show the experimantal heat transfer coefficients, the k-values, and the final overall heat transfer coefficient after the correction has been made. Similarly, Figures 19 – 25 also represent their corresponding plots to show how this k-value was useful to improve the design.

Table 10: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h		$W/m^2.\,^oC$	$W/m^2.\,^oC$
10.00	40.33	47.10	14.93	460	1.0347	1930.06	1982.27
12.30	33.57	38.13	15.93	471	1.0452	1940.42	1949.85
14.30	32.87	36.87	17.37	471	1.0491	1977.37	1970.05
16.30	32.30	35.70	18.93	474	1.0531	2005.85	1995.44
18.30	31.77	34.63	20.50	473	1.0569	2010.31	2012.43
20.37	31.20	33.50	22.10	474	1.0613	2042.81	2037.93
22.40	30.90	32.80	23.73	474	1.0649	2035.55	2070.32
24.30	31.00	32.47	25.30	476	1.0685	2116.81	2105.04
26.40	31.00	32.00	27.03	477	1.1403	2239.99	2278.90
28.00	30.63	31.20	28.33	477	1.1775	2358.91	2372.94
29.20	30.00	30.20	29.27	478	1.3448	2716.32	2772.88

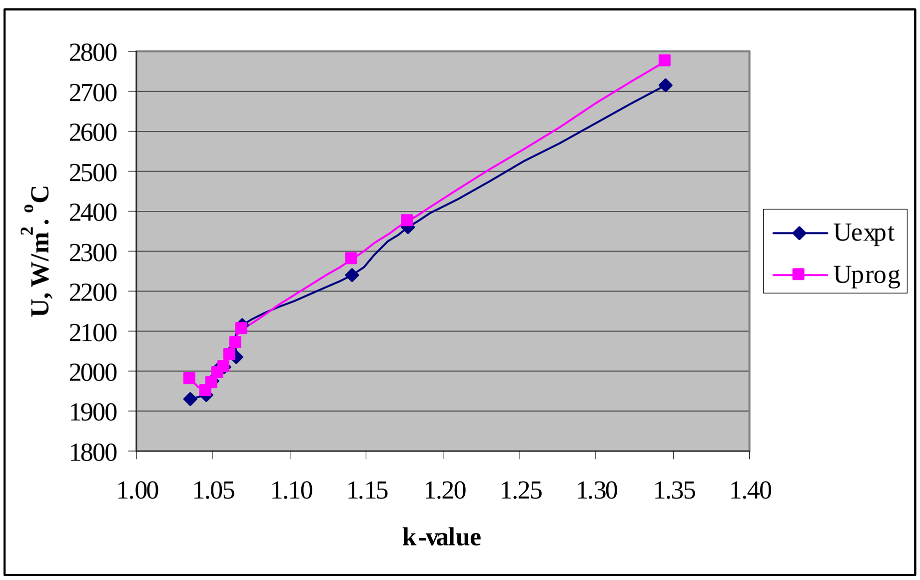

Figure 19: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h \approx$ 470 l/h)

Table 11: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h		$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.30	34.40	40.80	15.40	907	1.0537	3044.61	3000.21
12.37	33.07	38.70	16.60	897	1.0582	3063.23	3004.10
14.60	32.87	37.70	18.33	903	1.0626	3109.42	3047.50
16.30	32.70	37.10	19.63	900	1.0655	3098.00	3074.16
18.40	32.50	36.27	21.23	900	1.0697	3124.91	3112.22
20.50	32.23	35.37	22.83	895	1.0742	3128.00	3139.56
22.20	31.80	34.33	24.10	893	1.0786	3141.99	3158.65
24.40	31.20	32.93	25.73	893	1.0849	3198.03	3187.69
26.40	30.60	31.60	27.17	892	1.1092	3397.71	3267.36
28.20	30.10	30.50	28.50	891	1.2421	3878.99	3670.07
29.20	30.00	30.20	29.30	888	1.3875	4191.59	4194.04

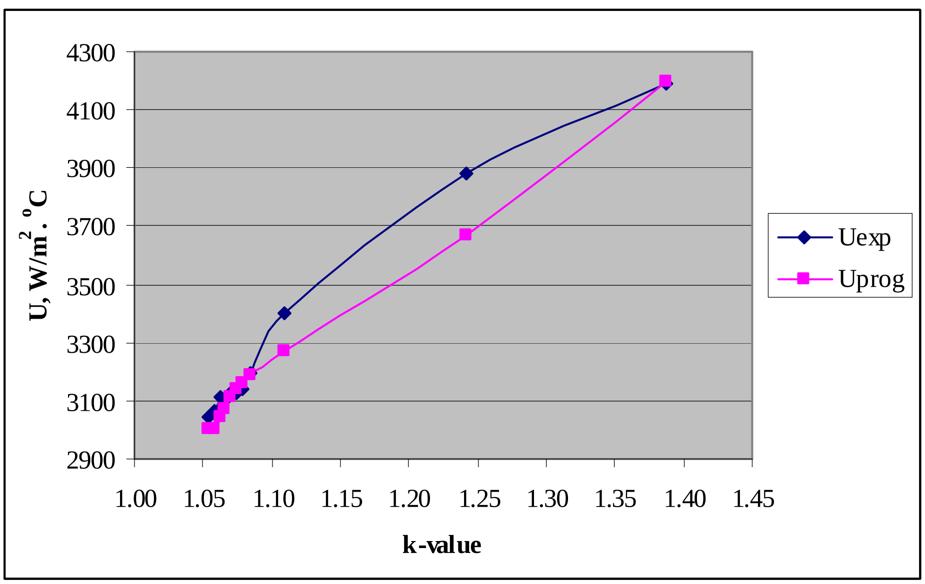

Figure 20: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h \approx$ 900 l/h)

Table 12: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
°C	°C	°C	°C	l / h		W/m². °C	W/m². °C
10.20	38.60	47.13	18.20	1774	1.0633	4700.09	4703.90
12.67	37.07	44.43	19.50	1780	1.0686	4734.67	4738.13
14.33	36.40	43.07	20.50	1779	1.0722	4734.67	4763.64
16.50	35.27	40.87	21.80	1781	1.0782	4716.56	4787.31
18.60	34.20	38.90	23.00	1785	1.0847	4721.49	4832.64
20.60	33.17	36.90	24.10	1782	1.0922	4775.89	4861.90
22.40	32.27	35.23	25.13	1782	1.0990	4789.40	4904.59
24.50	31.53	33.67	26.43	1782	1.1069	4801.38	4965.56
26.20	31.10	32.60	27.53	1782	1.1133	4837.05	5020.36
28.30	30.60	31.30	28.90	1778	1.1193	4977.22	5077.59

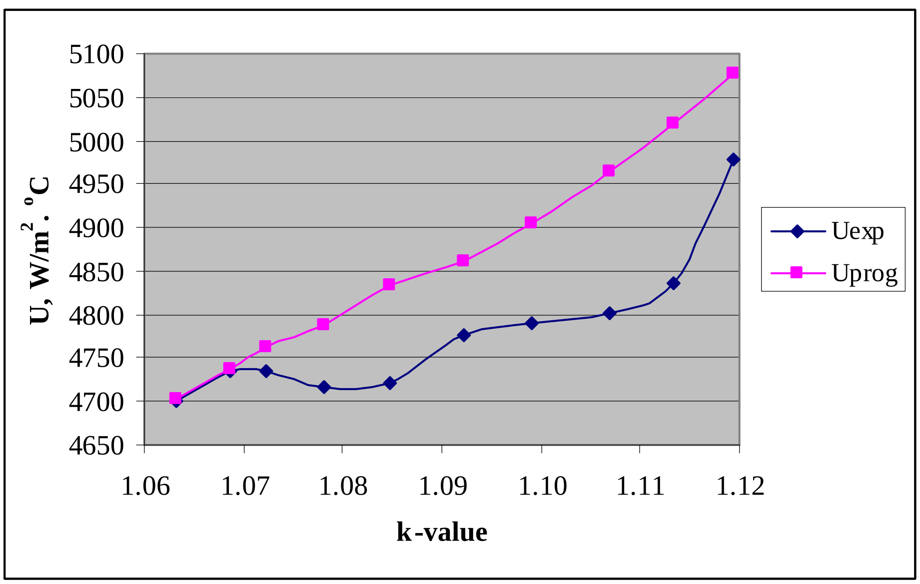

Figure 21: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h \approx$ 1800 l/h)

Table 13: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
°C	°C	°C	°C	l / h		W/m^2. °C	W/m^2. °C
12.40	40.37	47.90	22.10	2466	1.0654	5617.82	5692.16
14.50	38.03	44.43	22.63	2471	1.0720	5637.11	5698.57
16.63	37.10	42.60	23.67	2468	1.0781	5666.62	5733.17
18.60	36.03	40.77	24.60	2470	1.0845	5653.73	5778.37
20.60	35.23	39.20	25.60	2478	1.0915	5710.78	5843.32
24.50	33.77	36.30	27.63	2481	1.1067	5772.88	5970.15

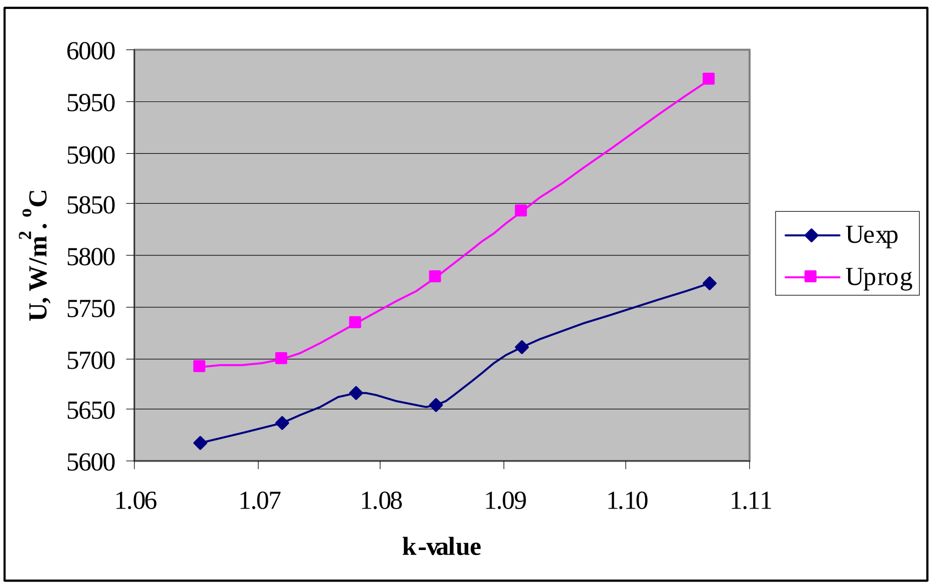

Figure 22: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h$ ≈ 2450 l/h)

Table 14: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h		$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.30	34.20	39.50	14.20	903	1.0605	1748.01	1800.31
12.40	32.97	37.40	15.63	904	1.0659	1809.36	1811.95
14.37	32.40	36.37	17.17	905	1.0694	1810.83	1833.37
16.20	32.20	35.70	18.53	903	1.0731	1883.04	1854.72
18.60	32.00	35.00	20.47	904	1.0773	1918.77	1888.88
20.40	31.80	34.37	22.03	901	1.0801	1880.79	1902.97
22.30	31.63	33.67	23.60	901	1.0840	1929.33	1922.89
24.60	31.13	32.53	25.50	904	1.0892	1961.18	1949.22
26.30	29.90	30.70	26.80	903	1.1252	1926.81	2015.87
27.10	29.33	29.80	27.33	902	1.2501	2316.83	2253.02

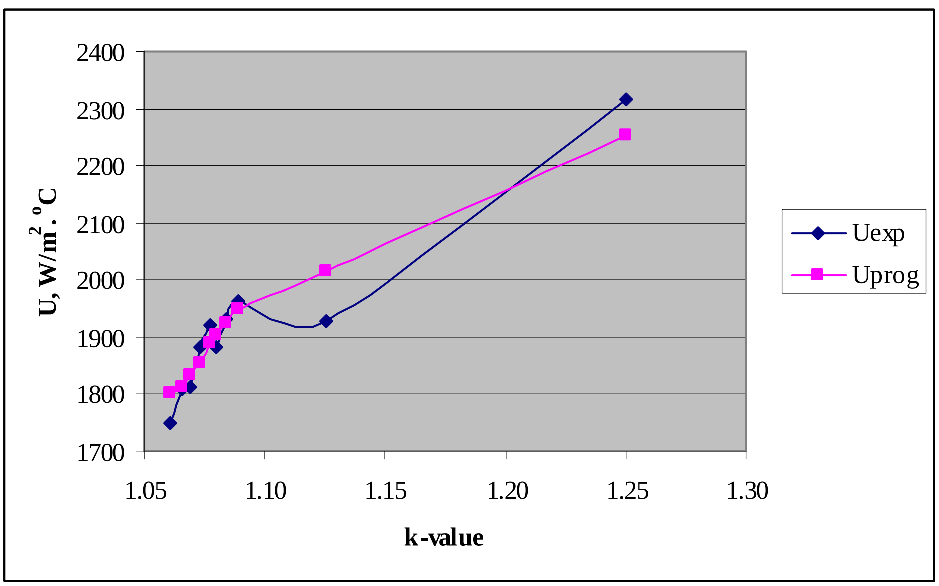

Figure 23: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h \approx$ 900 l/h)

Table 15: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h		$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
10.47	42.53	50.43	16.57	1749	1.0688	2970.95	2946.76
12.43	40.63	47.70	17.73	1735	1.0731	2956.26	2937.55
14.60	37.97	44.00	18.90	1724	1.0795	2953.64	2928.42
16.70	36.40	41.53	20.30	1720	1.0853	2951.94	2938.76
18.37	35.40	39.87	21.37	1713	1.0903	3002.25	2953.05
20.30	34.30	38.00	22.73	1712	1.0960	3022.39	2974.27
22.40	33.17	36.00	24.30	1710	1.1024	2993.29	2990.40
24.20	32.33	34.53	25.57	1704	1.1081	3042.70	3019.37
26.10	31.80	33.13	27.13	1790	1.1161	3195.83	3105.66
28.50	31.20	31.80	29.00	1775	1.1765	3164.46	3262.91
29.57	30.60	30.80	29.70	1777	1.5142	4200.94	4243.51

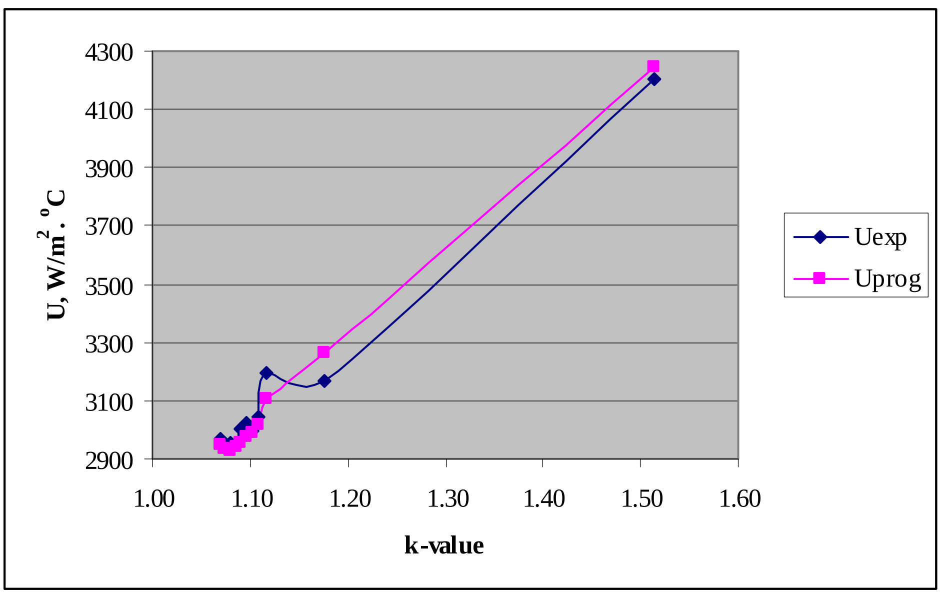

Figure 24: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h$ ≈ 1800 l/h)

Table 16: Improved U_{prog}, U_{exp}, and the correction factor

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	k-value	U_{exp}	U_{prog}
^oC	^oC	^oC	^oC	l / h		$W/m^2.\ ^oC$	$W/m^2.\ ^oC$
12.50	40.77	47.93	18.80	2459	1.0751	3722.05	3639.97
14.40	37.50	43.40	19.60	2458	1.0830	3685.94	3614.33
16.50	36.03	41.10	20.80	2463	1.0897	3735.40	3647.67
18.50	35.10	39.30	22.17	2469	1.0961	3759.22	3673.38
20.60	34.13	37.60	23.60	2462	1.1026	3727.46	3701.32
22.20	33.47	36.40	24.70	2465	1.1078	3717.48	3734.51
24.40	32.63	34.73	26.17	2456	1.1160	3803.50	3768.47
26.20	31.90	33.33	27.40	2462	1.1229	3887.17	3807.95
28.30	30.90	31.50	28.80	2457	1.2184	4223.89	4132.05

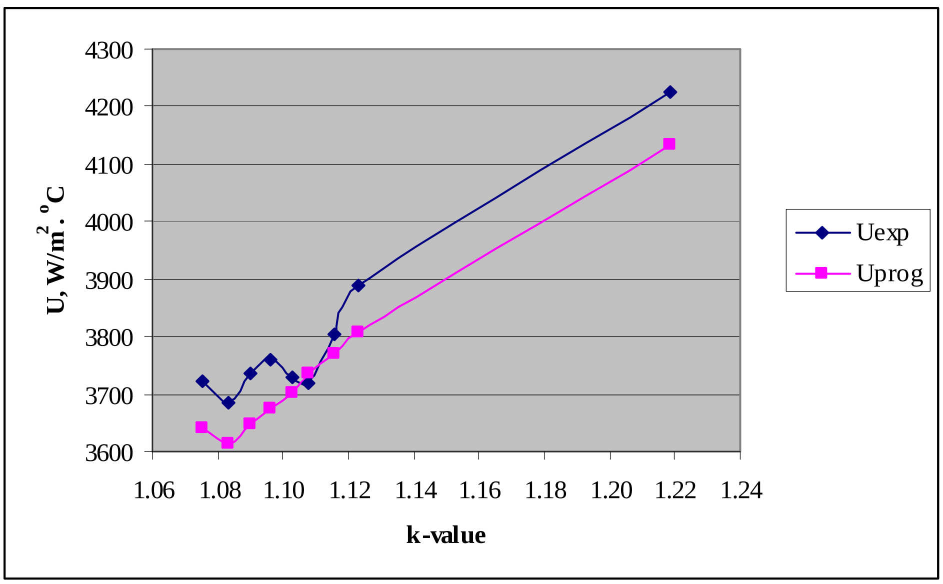

Figure 25: Plot of correction factor vs overall heat transfer coeffients ($\dot{V}_h \approx$ 2450 l/h)

In order to check the thermal efficiency, the number of plates were calculated using the program for the given input pareameters and plate dimensions. These values were compared with the actual amount of plates used during the experiment, and their deviation are presented in Tables 17 – 19. As the tables show, more than 75% of the results give reliable estimates with below 5 % deviation.

Table 17: Deviation between experimental and calculated plate amount for unit-A

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	Deviation
°C	°C	°C	°C	l / h	%
10.00	40.33	47.10	14.93	460	-8.45
12.30	33.57	38.13	15.93	471	-1.48
14.30	32.87	36.87	17.37	471	1.11
16.30	32.30	35.70	18.93	474	1.56
18.30	31.77	34.63	20.50	473	-0.33
20.37	31.20	33.50	22.10	474	0.74
22.40	30.90	32.80	23.73	474	-5.26
24.30	31.00	32.47	25.30	476	1.68
26.40	31.00	32.00	27.03	477	-5.36
28.00	30.63	31.20	28.33	477	-1.82
29.20	30.00	30.20	29.27	478	-6.48
10.30	34.40	40.80	15.40	907	4.19
12.37	33.07	38.70	16.60	897	5.51
14.60	32.87	37.70	18.33	903	5.70
16.30	32.70	37.10	19.63	900	2.24
18.40	32.50	36.27	21.23	900	1.19
20.50	32.23	35.37	22.83	895	-1.10
22.20	31.80	34.33	24.10	893	-1.57
24.40	31.20	32.93	25.73	893	0.95
26.40	30.60	31.60	27.17	892	10.75
28.20	30.10	30.50	28.50	891	14.80
29.20	30.00	30.20	29.30	888	-0.13

Table 18: Deviation between experimental and calculated plate amount for unit-A

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	Deviation
^oC	^oC	^oC	^oC	l / h	%
10.20	38.60	47.13	18.20	1774	-0.21
12.67	37.07	44.43	19.50	1780	-0.21
14.33	36.40	43.07	20.50	1779	-1.65
16.50	35.27	40.87	21.80	1781	-4.12
18.60	34.20	38.90	23.00	1785	-6.52
20.60	33.17	36.90	24.10	1782	-4.99
22.40	32.27	35.23	25.13	1782	-6.67
24.50	31.53	33.67	26.43	1782	-9.54
26.20	31.10	32.60	27.53	1782	-10.60
28.30	30.60	31.30	28.90	1778	-5.59
12.40	40.37	47.90	22.10	2466	-3.49
14.50	38.03	44.43	22.63	2471	-2.87
16.63	37.10	42.60	23.67	2468	-3.09
18.60	36.03	40.77	24.60	2470	-5.87
20.60	35.23	39.20	25.60	2478	-6.19
24.50	33.77	36.30	27.63	2481	-9.19

Table 19: Deviation between experimental and calculated plate amount for unit-B

$T_{c,i}$	$T_{c,o}$	$T_{h,i}$	$T_{h,o}$	$\dot{V}_h$	Deviation
^oC	^oC	^oC	^oC	l / h	%
10.30	34.20	39.50	14.20	903	-9.01
12.40	32.97	37.40	15.63	904	-0.42
14.37	32.40	36.37	17.17	905	-3.69
16.20	32.20	35.70	18.53	903	4.34
18.60	32.00	35.00	20.47	904	4.48
20.40	31.80	34.37	22.03	901	-3.48
22.30	31.63	33.67	23.60	901	0.97
24.60	31.13	32.53	25.50	904	1.78
26.30	29.90	30.70	26.80	903	-14.06
27.10	29.33	29.80	27.33	902	7.80
10.47	42.53	50.43	16.57	1749	2.26
12.43	40.63	47.70	17.73	1735	1.76
14.60	37.97	44.00	18.90	1724	2.38
16.70	36.40	41.53	20.30	1720	1.25
18.37	35.40	39.87	21.37	1713	4.53
20.30	34.30	38.00	22.73	1712	4.41
22.40	33.17	36.00	24.30	1710	0.27
24.20	32.33	34.53	25.57	1704	2.15
26.10	31.80	33.13	27.13	1790	7.70
28.50	31.20	31.80	29.00	1775	-8.85
29.57	30.60	30.80	29.70	1777	-2.75
12.50	40.77	47.93	18.80	2459	5.92
14.40	37.50	43.40	19.60	2458	5.23
16.50	36.03	41.10	20.80	2463	6.31
18.50	35.10	39.30	22.17	2469	6.12
20.60	34.13	37.60	23.60	2462	1.91
22.20	33.47	36.40	24.70	2465	-1.25
24.40	32.63	34.73	26.17	2456	2.49
26.20	31.90	33.33	27.40	2462	5.47
28.30	30.90	31.50	28.80	2457	5.92

The dependence of channel velocity on the pressure drop inside plate heat exchangers can be seen from Figures 26, 27, and 28. For the cold fluid stream, the pressure drops inside unit-A and unit-B were compared as represented in Fig. 26. These values were close to each other and their average was taken as the final value and this can be seen in Fig. 27. Similary, the hot side pressure drop is shown in Fig. 28.

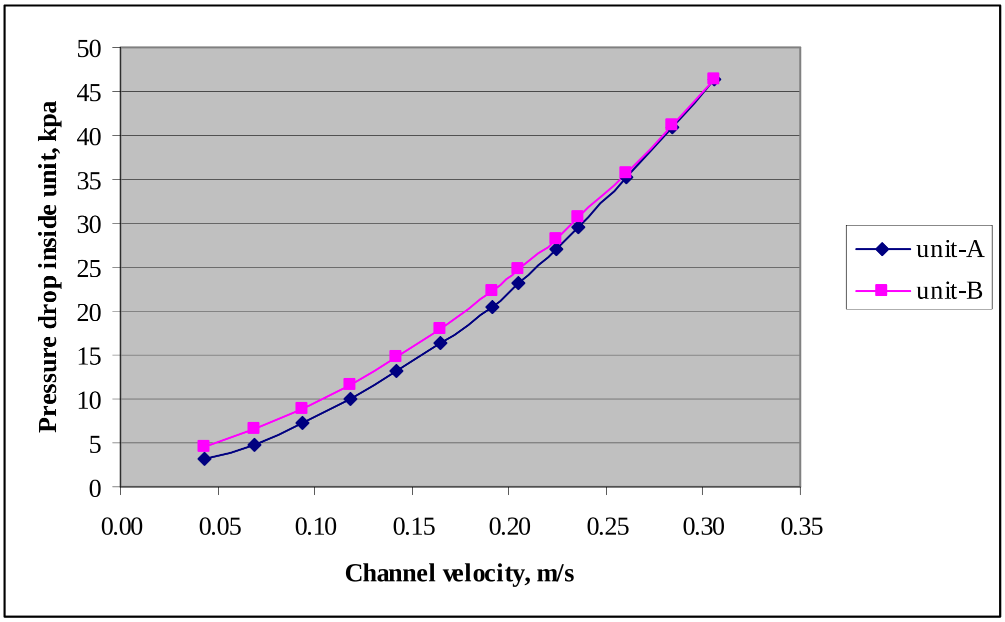

Figure 26: Channel velocity vs pressure drop of the cold stream

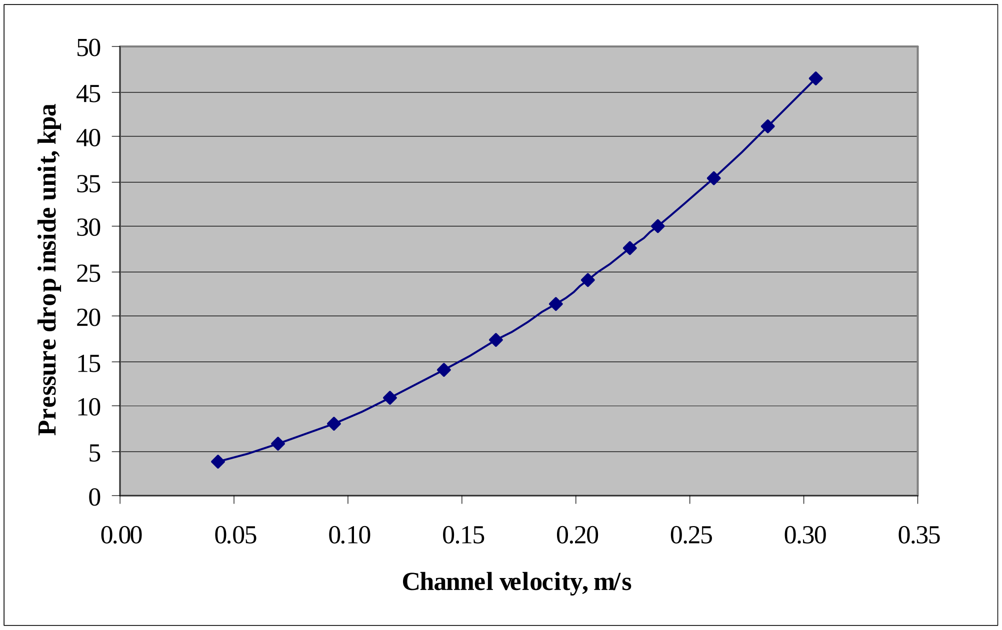

Figure 27: Channel velocity vs pressure drop of the cold stream (R^2 = 1)

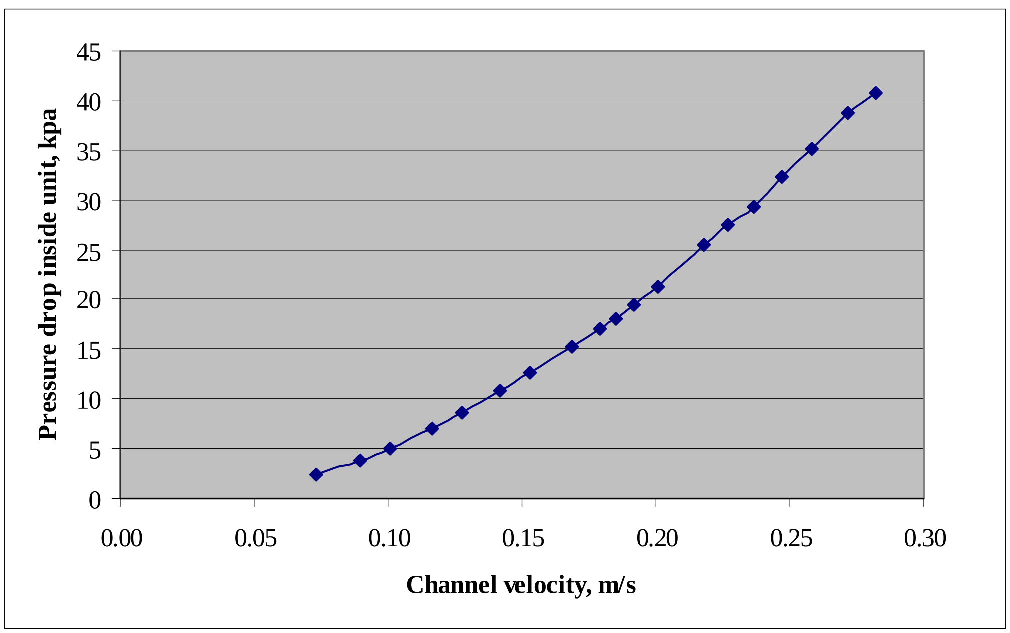

Figure 28: Channel velocity vs pressure drop of the hot stream (R^2 = 0.9997)

9 CONCLUSIONS

The optimum design of plate heat exchangers is mostly accomplished based on their thermal-hydraulic performance, which represents the relationship between the heat transfer, pressure drop, and heat exchanger area. In this research work, a general methodology is devised to calculate the amount of plates in plate heat exchangers from a prescribed set of fluid streams and their operating conditions, allowable pressure drop, and plate dimensions.

Several experiments were carried out to investigate the effects of the relevant input parameters on the thermal hydraulic performance of the plate heat exchanger. Two brazed plate heat exchangers with different plate numbers were tested to collect reliable data and to study the influence of maldistribution as well. As the results show, the actual overall heat transfer coefficients obtained based on the experiment were slightly higher than the overall heat transfer coefficients calculated from the simulation program. These small discrepancies could be due to the fact that, in reality the flow velocities inside the channels cannot be equal. To have more reasonable design, a correction factor which is a function of the input parameters was introduced to the calculated overall heat transfer coefficient.

Furthermore, the maximum allowable pressure drop of the plate heat exchanger was considered as one of the design specifications and this pressure drop is mostly higher than the calculated value. However, in some cases, if the calculated pressure drop is greater than the maximum allowable pressure drop, additional plates has to be added by increasing the margin.

It is also worth concluding that the methodology developed in this research is highly flexible and can help to investigate the influence of the plate sizing and pattern, plate spacing and plate thickness on the thermal-hydraulic design of the plate heat exchanger. The most important factor is the physical dimensions of the plate together with corrugation forms at distribution and heat transfer area. Each change needs test evaluation before new model can be launched to sales market.

In this work, brazed plate units were considered, however, the devised methodology can be applied to other types of plate heat exchangers having similar configurations such as gasketed, welded or fusion-bonded plate heat exchangers.

Plate heat exchangers have progressed significantly since they were invented and this development will certainly continue to further expand their industrial applications. However, in order to achieve this, there are still some challenges related to their construction and performance. The construction of plate heat exchangers includes the development of new plate units by using new materials. The importance with the new development and material is to increase the operating pressure and temperature of the plate heat exchanger. Similarly, the new materials are important to reduce the threat of corrosion and to permit additional working fluids.

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