Design and optimization of plate heat exchanger networks

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# Nomenclature

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<th>Description</th>
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<tbody>
<tr>
<td>PHE</td>
<td>plate heat exchanger</td>
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<tr>
<td>HEN</td>
<td>heat exchanger network</td>
</tr>
<tr>
<td>GPHE</td>
<td>gasket plate heat exchanger</td>
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<tr>
<td>WPHE</td>
<td>welded plate heat exchanger</td>
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<tr>
<td>LMTD</td>
<td>logarithmic mean temperature difference</td>
</tr>
<tr>
<td>NTU</td>
<td>number of transfer units</td>
</tr>
<tr>
<td>LP</td>
<td>linear programming</td>
</tr>
<tr>
<td>MINLP</td>
<td>mixed integer nonlinear programming</td>
</tr>
<tr>
<td>NLP</td>
<td>non-linear programming</td>
</tr>
<tr>
<td>MILP</td>
<td>mixed integer linear programming</td>
</tr>
<tr>
<td>ILP</td>
<td>integer linear programming</td>
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<tr>
<td>GA</td>
<td>genetic algorithm</td>
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<tr>
<td>SA</td>
<td>simulated annealing</td>
</tr>
<tr>
<td>IDE</td>
<td>integrated different evolution</td>
</tr>
<tr>
<td>CAT</td>
<td>Constant Approach Temperature</td>
</tr>
<tr>
<td>Re</td>
<td>Reynold number</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
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<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>GAMS</td>
<td>General Algebraic Modelling System</td>
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<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
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Symbols

$L_p$  plate length

$\varphi$  chevron angle

$B_p$  plate width

$d_{port}$  port diameter

$d_e$  equivalent diameter

$A_b$  plate area

$f_{ch}$  cross-section area

$\mu$  dynamic viscosity

$\lambda$  heat conductivity

$A$  heat transfer area

$N$  total number of blocks

$X$  number of passes for stream

$h$  heat transfer coefficient

$Q$  heat load

$\Delta T$  logarithmic mean temperature difference

$\rho$  stream density

$v$  stream velocity in each channel

$t$  width of channel

$R_f$  fouling resistance of streams

$g$  flow rate of the stream

$a$  cross-section area between channels

$T$  temperature
$C$  heat capacity

$R$  the ratio of flow heat capacities of streams

$A_{total}$  total heat transfer area

$f$  friction factor

$\Delta P_{height}$  pressure drop of height change

$\Delta P_{friction}$  pressure drop due to friction

$H$  height

$U$  heat transfer coefficient

$AT_{min}$  minimum approach temperature

$F_T$  correlation factor

$A_{ij}$  matrix of incident streams on exchangers

$S$  number of process and utility streams

$E$  number of heat exchanger in the existing HEN

$P$  column factor was added incidence matrix

$TT$  target temperature

$QP$  cross-pinich heat transfer

$HU$  hot utility

$CU$  cold utility

$TRC$  total retrofit cost

$BP$  cost of implementing by-pass

$AA$  cost of adding area

$UC$  total cost of utility saving
### Greek symbol

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>δ</td>
<td>inter-plate gap</td>
</tr>
<tr>
<td>β</td>
<td>chevron angle</td>
</tr>
<tr>
<td>(A_{\text{total}})</td>
<td>total area of heat exchanger</td>
</tr>
<tr>
<td>(Q^0)</td>
<td>required heat load</td>
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### Subscript

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<tr>
<td>b</td>
<td>block</td>
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<tr>
<td>h</td>
<td>hot stream</td>
</tr>
<tr>
<td>c</td>
<td>cold stream</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
<tr>
<td>1</td>
<td>inlet</td>
</tr>
<tr>
<td>2</td>
<td>outlet</td>
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<tr>
<td>max</td>
<td>maximum</td>
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Abstract

With the growth of energy consumption and the increase of greenhouse gas emissions, it is important to improve heat transfer efficiency and save energy. One of the most effective ways is to consider use of heat transfer enhancement for increasing heat recovery. Plate heat exchangers, which allow a small minimum temperature approach, are widely used in the energy-intensive process industries to enhance heat transfer coefficient. A major limitation of applying plate heat exchanges is the lack of reliable design methods to quantify the energy saving effectively. The key objective is to develop a novel method for a single plate heat exchanger design and integrate the plate heat exchangers into conventional heat exchanger network retrofits.

A new computer-aided design of two-stream multi-pass plate heat exchangers is proposed, including gasket plate heat exchangers and welded plate heat exchangers, with different plate geometries. To account for multi-pass flow arrangements, the plate heat exchanger is separated into several pure counter-current or co-current one-pass blocks. The correlations of inlet and outlet temperatures of different blocks are obtained in order to apply the logarithmic mean temperature difference (LMTD) method for thermal design in each single-pass block with known temperatures. The selection of the number of passes for streams, plate geometries and plate patterns are considered as integer variables to optimize the total area of plate heat exchanger. An MINLP model is developed in GAMS using ANTIGONE solver in order to derive the optimal solution. A case study is used to demonstrate the capability of proposed method to obtain the optimal solution with required heat load and constraints. The proposed design model can also be further applied to the complex heat exchanger network design.

Application of plate heat exchangers into the heat exchanger networks (HENs) retrofit increases the heat recovery due to their small minimum approach temperature. However, the installation cost of plate heat exchangers is relatively high. Thus, the optimization process is based on the trade-off between energy reduction and capital cost. For a fixed structure HEN, the heat recovery is limited. Structure modifications have the possibility of providing more energy saving but bring more cost at the same time. A decision on the best retrofit strategy to apply to a given HEN depends on the given retrofit objective. This thesis presents a methodology for the application of plate heat exchanger in HENs, both with fixed structure and with structure modification. The key point is to find the most beneficial location to apply plate heat exchangers and develop an algorithm to automatically identify the best modifications. The objective of the optimization process of HENs retrofit is to minimize the energy consumption and maximise the retrofit profit at the same time. Case studies highlight the benefits of the new approach. The results are compared with conventional technologies to provide the insight on the potential benefit of integrating plate heat exchanger into HENs retrofit. This work provides an adequate basis on which the decision can be made based on industrial applicability, profit, and energy saving.
Declaration

No portion of the work referred to in this thesis has been submitted in support of an application for another degree or qualification of this or any other university of other institution of learning.

Kexin Xu
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To all my friends and colleagues who have contributed in different ways to the completion of this project. Nevertheless, special thanks to Mary, Xiao, Jose, Luyi, Fei, Nikos, Meng and the rest of CPI staff and students for their help and support whenever I needed it. Special thanks to Steve Doyle who provided tremendous help with SPRINT and gave me practical suggestions on the research.
Chapter 1 Introduction

1.1 Introduction

With the growth of energy consumption and stringent environmental protection legislation, energy saving has been made a priority. One of the largest energy consumers globally are the process industries. Shell-and-tube heat exchangers have been widely used in the process industries over the past several decades (Rohsenow et al., 1998, Manglik, 2003), but it is very difficult to reduce their relative large minimum approach temperature and their total volume. Heat transfer enhancement technologies are one of the most efficient and cost-effective way to enhance heat transfer behaviour, as it usually requires low capital investment for fixed network topology, and no additional heat transfer area in existing heat exchangers (Sheikholeslami et al., 2015). However, for enhancement to be effective, at least 50% of the overall heat transfer resistance must be either the tube-side film coefficient or the shell-side coefficient. If one of the film coefficients is not controlling, applying heat transfer equipment with high-performance heat transfer equipment, such as plate heat exchangers, is a beneficial option to improve energy efficiency (Kakae et al., 2012, Kapustenko et al., 2009).

Plate heat exchangers are one of the most efficient heat transfer equipment (Gherasim et al., 2011b), which use metal plates to transfer heat between fluids. Compared with conventional shell-and-tube heat exchangers, plate heat exchangers can significantly increase energy efficiency, enhance thermal-hydraulic performance, and reduce the fuel consumption and CO₂ emissions (Sundén and Manglik, 2007, Pantzali et al., 2009).

Plate heat exchangers have many distinct advantages (Rao et al., 2002, Gherasim et al., 2009, Pandey and Nema, 2011, Luan et al., 2008). Among them, the most dramatic feature is that the minimum approach temperature in plate heat exchangers can be as low as 2°C due to high heat transfer coefficient. Besides, compared to shell-and-tube heat exchangers, plate heat exchangers have larger surface area, lower total cost, are less prone to fouling, and present a more flexible thermal design compared with conventional heat exchangers (Gherasim et al., 2011a). Due to these advantages, plate heat exchangers are widely used in various energy-intensive process industries, such as refrigeration, petrochemical plants, refineries and natural gas processing (Hesselgreaves,
However, no systematic methods are currently available for the design and optimization of heat exchanger networks involving plate heat exchangers.

![Figure 1 Schematic diagram of a simplified preheat train, modified from Coletti and Macchietto (2011)](image)

In an oil refinery, crude oil distillation is the most energy-consuming process (Nasr and Givi, 2006). In the distillation column, crude oil can be separated into hydrocarbon streams according to differences in boiling temperatures. However, in order to carry out distillation, crude oil must be heated up to 360°C to 380°C before it can be fed to the distillation column. To maximize energy savings, heat from hot side products and pump arounds is utilised to recover heat into the crude oil. This process is called the crude oil preheat train as shown in Figure 1, in which the heating operation is built up on a heat exchanger network. The final heating of crude oil is carried out in a furnace (Borges et al., 2009). By introducing preheat train, the energy demand for heating crude oil up to the desired temperature dramatically reduces, which could provide 60%-70% of the total energy demands (Gonçalves et al., 2014). However, during the operation of the preheat train, the thermal efficiency of conventional heat exchanger networks decreases due to fouling and large minimum approach temperatures. Thus, to increase energy savings and reduce fouling possibility, plate heat exchangers are taken into consideration to replace the conventional heat exchanger in crude oil preheat trains.
Plate heat exchangers, which include gasket plate heat exchangers and welded plate heat exchangers, use metal plates to transfer heat between fluids (Shah and Focke, 1988). Welded plate heat exchangers are gasket free. Between the two end plates, a completely welded plate pack is bolted in a conventional frame. The special structure of welded plate heat exchangers dramatically enhances their integrity, which makes it possible to operate in the condition of high temperature (up to 350°C) and high pressure (40 bar) (Sundén and Manglik, 2007). As for the gasket plate heat exchangers, due to the limitation of gaskets, this type of heat exchanger could only handle fluids with less than 200°C, and the pressure should be no more than 25 bar (Abu-Khader, 2012). Besides, the high turbulence generated by chevron-plate pattern in welded plate heat exchangers leads to high overall heat transfer coefficients, which are three to five times that of conventional shell-and-tube heat exchangers. Thus, considering of the operation conditions and specific structures, welded plate heat exchangers are possible to apply in crude oil preheat trains.

By applying welded plate heat exchanger into crude oil preheats trains, the greenhouse gas emission and fuel consumption can be significant reduced. Take a refinery plant with crude distillation capacity 40000 bbl/d as an example. If we apply two optimized welded plate heat exchangers to this crude oil preheat train, the annual fuel savings would be 427,000 USD, in which we assume that fuel value is 35/bbl USD. Also, the annual emission savings would be 130,000 USD, in which we assume that CO₂ value is 15/ton USD (Alfalaval, 2015). Based on the BP Statistical Review of World Energy in June 2017, the total world refinery throughputs in 2016 are 76,833 thousand barrels daily, which means the annual fuel savings would be 820,000 thousand USD and the annual emission savings would be 247,000 thousand USD if we replace two conventional shell-and-tube heat exchanger in each of the preheat trains.

However, most data are industry-owned so far and there is no published methodology to quantify the benefits of energy saving. Researchers are now focusing on investigating the effect of fouling issues (Ishiyama et al., 2009, Panchal and Huangfu, 2000, Markowski et al., 2013). Welded plate heat exchangers are less prone to fouling due to higher shear stress and turbulent flow (Abu-Khader, 2012). However, since plate heat exchangers are still regarded as a new technology in the refineries, the potential benefits of replacement need to be validated before they are widely used.
This project will develop the first truly systematic methods for design and optimization for plate heat exchangers, and apply them for the retrofit of heat exchanger network. This research is not restricted to the application of plate heat exchanger into crude oil preheat trains. The approach can be used into more complex networks. The crude oil preheat trains is only used as a case study to show the possibility to integrate plate and shell-and-tube heat exchanger into heat exchanger networks and quantify the benefit.

1.2 Objectives

Currently the challenges of applying plate heat exchangers into heat exchanger networks are: 1) lack of general design method of plate heat exchangers, including welded plate heat exchangers and gasket plate heat exchangers; 2) identifying the best heat exchangers to replace in retrofit; 3) develop the most cost effective retrofit plan. Thus, this research project will develop systematic methods for design and optimization of plate heat exchanger networks. The benefits of the proposed methodology, for example energy saving and total cost, will be quantified through different case study. This research work is focusing on tackling with three problem and the related research objectives are listed below.

Objective 1:

Develop an algorithm for optimal design of plate heat exchangers, including welded and gasket plate heat exchangers:

The heat transfer and pressure drop performance data of plate heat exchangers are industrially owned, especially for welded plate heat exchangers. However, for welded plate heat exchangers, Nusselt number and friction factor expressions can be considered to be the same as gasket plate heat exchangers (Picón-Núñez et al., 2006). Thus, this project aims to develop a general design methodology for plate heat exchangers, which includes selection of plate pattern, determination of flow arrangement and pressure drop consideration simultaneously. Therefore, the research questions for this project based on the objectives are:

Question a: How to integrate flow arrangements into the design model and how to model different flow arrangements?
Question b: How to build the design model so that it can automatically select the most cost-effective combination of plate geometries, chevron angle and flow arrangement depending on the given process data?

Objective 2:

Develop an optimization method of integrating a mixture of plate and shell-and-tube heat exchangers into heat exchanger networks without the need for topology modifications.

This objective is to develop a novel methodology for the use of plate heat exchangers in heat exchanger retrofit. Integrating plate heat exchangers into a conventional heat exchanger network can significantly reduce energy cost and enhance heat recovery. However, the cost of installing new plate heat exchangers is high. Thus, a reasonable retrofit is one that has the right balance between efficient use of existing equipment and limited amount of modifications, while maximizing energy recovery. To ensure industrial applicability, this objective aims to propose a cost-effective retrofit design of HEN without topology modifications. The research questions are:

Question a: How to identify the best heat exchangers in the existing HEN to replace? What are the criteria of the number of heat exchangers to replace?

Question b: How to deal with downstream effect after applying plate heat exchangers? How to quantify the benefit of retrofit?

Question c: How to deal with two different heat transfer technologies in one heat exchanger network, especially when it comes with two different minimum approach temperatures?

Objective 3:

Develop a novel methodology for applying plate heat exchangers into HENs retrofit with structural modifications:

Through applying plate heat exchangers in retrofit design of existing HENs, the amount of energy saving is limited without structural modifications. On one hand, structural modifications are possible to increase heat recovery; on the other hand, they are costly because of high expense on piping work and possibly civil engineering. The challenge of modification is to identify the location which can bring most retrofit profit to apply plate heat exchangers. The research questions are:
**Question a:** How to identify the location for installing plate heat exchangers?

**Question b:** How to quantify the potential benefit of applying plate heat exchangers into retrofit design of heat exchanger network? Energy saving or total retrofit cost or payback time?

**Question c:** Since the plate heat exchangers have smaller minimum approach temperature and potentially more energy saving compared with shell and tube heat exchanger, how to set up the objective of energy saving?

**Question d:** How to tackle with two different minimum approach temperatures caused by different type of heat exchangers in one heat exchanger network?

### 1.3 Thesis outline

The “journal format”, which including the published or submitted papers, is used in the organization of this PhD thesis to fulfil the requirement of The University of Manchester. In total, this thesis is report is separated into 6 chapters. The outline of this thesis and the connection between different chapters are shown in Figure 2.

Chapter 1 introduces the main motivation and challenges of develop an optimization model for multi-pass plate heat exchangers. The potential benefit of application of plate heat exchangers into a heat exchanger network is highlighted also. The three main research gaps to tackle within this research work and three related objectives are detailed.

Chapter 2 presents a review of detailed background on plate heat exchangers, and other different types of heat exchangers, the important factors that affect the heat transfer and different methodology on thermal-hydraulic design of plate heat exchangers are reviewed. The literature review on the retrofit of heat exchanger network is detailed, including three existing retrofit method and network structure analysis.

Chapter 3 illustrates a new computer-aided methodology of design and optimization of multi-pass plate heat exchangers, which includes gasket plate heat exchangers and welded plate heat exchangers. This chapter is the basis of Chapters 4 and 5, since this optimized model is applied to the heat exchanger network retrofit. A MINLP model is formulated based on the selection of best geometries and number of passes of streams to optimize the area of plate heat exchangers.

Chapter 4 is focused on the application of plate heat exchangers into heat exchanger network retrofit with a fixed network structure. To deal with Objective 2, this chapter
tackles the challenge of integration of plate heat exchanger into conventional heat exchanger networks. A method of dealing with two different minimum approach temperatures is presented. The most sensitive heat exchanger to be placed is identified, and potential benefit is quantified.

Chapter 5 presents a novel step-by-step methodology of applying plate heat exchanger into retrofit of HENs based on topology changes. A new network pinch methodology is used to identify the best retrofit strategy and best location to apply modifications. The objective is to achieve the energy target with minimum modification steps and minimum total retrofit cost. A distinctive feature of the new proposed method is the ability to consider different minimum approach temperatures for the different types of exchangers used in the network within an optimization framework. To quantify the potential benefit of applying PHEs into HEN retrofit with structure modifications, the retrofit results are compared with the results allowing adding shell-and-tube heat exchangers into HEN retrofit by following the same guidelines.

Chapter 6 presents the main findings of this research work, and the limitations and their corresponding possible improvements are highlighted. The recommendations for future work are discussed at the end of this Chapter.
Chapter 1: Introduction
Background, Motivation, Research

Chapter 2: Literature Survey
Review of previous work on plate heat exchangers design and HEN

Chapter 3: Single plate heat exchanger
Publication 1: A new computer-aided optimization-based method for the design of single multi-pass plate heat exchangers

Chapter 4: Fixed structural
Publication 2: Application of plate heat exchangers into heat exchanger network retrofit without structure modifications

Chapter 5: Structural Modifications
Publication 3: New guidelines for the application of structural changes in heat exchanger network

Chapter 7: Conclusions and Future Work
Summary of key findings from the thesis and future work

Figure 2 Thesis outline
Chapter 2  Literature Survey

Over the past several decades, there has been an increasing concern about the energy saving and carbon emissions. One of the effective ways to reduce the energy consumption is to increase the heat recovery in energy systems. In conventional heat exchanger networks, shell and tube heat exchangers are commonly used. There has been an increasing emphasis on using more effective technologies to improve heat recovery in heat exchanger networks, which is main objective in this research. Heat transfer enhancement is one of the main techniques due to its relatively simple piping work and low capital cost. However, for enhancement to be effective, one of the heat transfer coefficients must be controlling. Besides, the energy saving is limited by applying heat transfer enhancement. Under this circumstance, heat transfer equipment with higher energy-efficiency heat exchangers is considered. Therefore, at the beginning of this chapter, the existing common-used types of heat exchangers and their corresponding advantages are reviewed. Among them, the notable features of plate heat exchangers are stressed. In addition, the factors that affect heat transfer in plate heat exchanger are reviewed. This is followed by the previous optimization methodologies of plate heat exchanger design. To integrate plate heat exchangers into retrofit design of heat exchanger networks, the existing methods of HEN retrofit are reviewed. Pinch analysis, mathematical programming, and hybrid methods which are combinations of the other two methods, are the three main methodologies for HENs retrofit. Ideally, the best retrofit strategy should have the fewest steps of modifications and lowest capital cost. To achieve this goal, plate heat exchangers can be considered to be integrated into heat exchanger networks as they have relatively high heat transfer coefficients. Keeping the existing structure of the network in the retrofit design it is possible reduce the piping work and economic cost. But the energy saving is limited due to the restriction of the fixed structure. The second part of the literature survey detailed the reason caused the redundant use of energy and reviewed various methodologies of the existing methodologies of the retrofit of HENs.

2.1 Existing types of heat exchanger

A heat exchanger is a device that can transfer heat between two or more fluids at different temperatures. The fluids may be in direct contact or be separated by a solid
wall to prevent mixing (Shah and Sekulic, 2003). Heat exchangers have been applied in a variety of industries, such as chemical, food, electronics, environmental engineering, refrigeration, and air-conditioning. There are plenty of heat exchangers, which could be divided into several classifications according to different criteria (Hewitt et al., 1994). However, in this part, only three types of heat exchangers most widely used are introduced in detail: shell-and-tube heat exchangers, plate heat exchangers, and plate-fin heat exchangers.

2.1.1 Shell and tube heat exchangers

This type of exchangers consists of round tubes, which are mounted in a cylindrical shell with the tubes parallel to the shell. One fluid flows inside the tubes, the other one flows over the outside of the tube so that it can transfer heat or absorb heat in this way (Shah and Sekulic, 2003). Due to the shape of shell and tube heat exchangers, they are robust. So typically shell and tube heat exchangers are used for high-pressure applications, normally, the temperature allowance is higher than 260 °C and the pressure allowance is greater than 30 bar (Saunders, 1988).

The conventional shell and tube heat exchanger is the most common used type of heat exchanger in crude oil preheat trains (Ishiyama et al., 2010). Although the researchers found some enhancement devices to improve its performance (Bergles, 1998), it still has a large volume, relatively large minimum approach temperature and a fouling tendency.

2.1.2 Plate heat exchanger

This type of heat exchanger consists of a number of metal plates, and fluids transfer heat by flowing between the plates. Since the fluids spread out over the plate, the fluids could be exposed to larger surface area than conventional heat exchangers. Plate heat exchangers, including gasket plate heat exchangers and welded plate heat exchangers, are one of the most efficient types of heat transfer equipment. The main advantages of plate heat exchanger are as follows:

1. The heat transfer surface area is easily changed because of the flexible plate size, flow arrangements and plate patterns in gasket designs.
2. The heat transfer coefficient is much higher than shell-and-tube heat exchanger due to the corrugations of plates. By introducing swirl, reducing hydraulic diameter,
increasing heat transfer area, the heat transfer coefficient of plate heat exchanger is significantly enhanced.

3. Owing to the high heat transfer coefficient, and the pure counter-flow arrangement, the surface area is dramatically reduced. For a given duty, the total volume of plate heat exchanger is twice or three times smaller than that of shell-and-tube heat exchanger.

4. Due to the high shear rate, and high turbulence, the fouling tendency could be minimized, which could be reduced by up to 10% of that of traditional shell-and-tube heat exchanger.

5. The minimum approach temperature is dramatically reduced owing to the high heat transfer coefficient and true counter-flow arrangement. It could be as low as 1°C. As a result, 90% heat recovery could be achieved. However, only 50% heat recovery could be obtained by shell-and-tube heat exchangers.

6. It is possible for plate heat exchangers to heat or cold down several fluids in a single plate heat exchanger by introducing intermediate divider sections between the plates.

7. In a single plate heat exchanger, plates with different pattern could be combined. Selection of different flow arrangement could also be considered. The flexibility makes it possible to do a better optimization of a plate heat exchanger.
In general, the main advantages of plate heat exchanger are all listed and the comparison of the main operation conditions of plate heat exchanger and shell-and-tube is stated in Table 1.

However, the main limitation of plate heat exchangers is that they can only be applied to process with relatively low pressure and temperature. This is because the material of gasket could not bear high temperature or some corrosive fluids. In the most applications of plate heat exchanger, the upper bound temperature is about 160°C, though some gaskets with special material could withstand temperature up to 400°C (Wadkar, 1998, Manglik and Muley, 1993). In order to eliminate this limitation, several different types of plate heat exchangers have been developed. Among them, welded plate heat exchanger (WPHE) is the main type of plate heat exchangers.

2.1.2.1 Gasket plate heat exchanger

<table>
<thead>
<tr>
<th></th>
<th>Plate heat exchanger</th>
<th>S&amp;T heat exchanger</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature cross</td>
<td>Impossible</td>
<td>Possible</td>
</tr>
<tr>
<td>Approach $\Delta T$</td>
<td>$1^\circ C$</td>
<td>$5^\circ C$</td>
</tr>
<tr>
<td>Multiple duty</td>
<td>Possible</td>
<td>Impossible</td>
</tr>
<tr>
<td>Piping connections</td>
<td>From one direction</td>
<td>From several directions</td>
</tr>
<tr>
<td>Heat transfer ratio</td>
<td>3-5</td>
<td>1</td>
</tr>
<tr>
<td>Operating weight ratio</td>
<td>1</td>
<td>3-10</td>
</tr>
<tr>
<td>Hold-up volume</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>Space ratio</td>
<td>1</td>
<td>2-5</td>
</tr>
<tr>
<td>Leakage detection</td>
<td>Easy to detect</td>
<td>Difficult to detect</td>
</tr>
<tr>
<td>Access for inspection</td>
<td>On each side of plate</td>
<td>Limited</td>
</tr>
<tr>
<td>Disassembly time</td>
<td>15 min</td>
<td>60-90 min</td>
</tr>
<tr>
<td>Fouling ratio</td>
<td>0.1-0.25</td>
<td>1</td>
</tr>
</tbody>
</table>
In this type of heat exchanger, gaskets seal the edge of the plates and the plates are held together in a frame, as shown in Figure 3. Normally, there is a fixed end cover with connecting ports and a removable end cover in the frame. The upper and bottom carrying bars are used to guarantee the alignment of the suspended plates. Long bolts can be used to clamp together plates with a fixed end cover. In this way, the gaskets are compressed and a seal is formed.

In each plate, there are four ports in the corner, which provide the access to the flow passages on either of the plates. The corner ports are arrayed to form distribution headers when the plates are assembled so that the two fluids could go through the ports. When the two fluids go through different plates, the heat could be transferred. This type of plate heat exchanger is relatively flexible due to its structure, and the plates could be easily added or removed.

![Figure 3 Configuration of gasket plate heat exchanger (Alfalaval)](image)

### 2.1.2.2 Welded plate heat exchangers

Welded plate heat exchangers are built around a pack of corrugated heat transfer plates, welded alternately to form channels as shown in Figure 4. By eradicating the gaskets, the integrity of heat exchanger is dramatically improved and the upper limit of temperature and pressure is increased. However, at the same time the welded plate heat
exchanger loses the flexibility of increasing or decreasing the surface area to satisfy the changing heat load.

Welded plate heat exchangers are applied to the process that involves highly aggressive fluids. The upper bound of temperature of this type of heat exchanger is around 350°C, and it can withstand pressure lower up to 40 bar. In addition, it has high leakage protection. Welded plate heat exchangers are used in oil and gas production, refineries, pharmaceutical industry, specialty chemicals, and hydrocarbon process industry (Wang and Sunden, 2003).

However, except for these two main types of plate heat exchanger, there are still others, such as brazed plate heat exchanger, and wide-gap plate heat exchangers.

A summary of operating conditions of different types of plate heat exchanger is listed in the Table 2, which can be regarded as selection guidance. The data are derived from the manufacturer Alfa Laval. Considering the temperature restriction, welded plate heat exchanger is considered to apply in the crude oil preheat trains.
Table 2. A selection guide of PHEs for different conditions (Sundén and Manglik, 2007)

<table>
<thead>
<tr>
<th>Operating conditions</th>
<th>Gasketed</th>
<th>Brazed</th>
<th>Semi-welded</th>
<th>Fully welded</th>
<th>Wide gap</th>
<th>Double wall</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure [bar]</td>
<td>2.5</td>
<td>3</td>
<td>2.5</td>
<td>4.0</td>
<td>0.9</td>
<td>2.5</td>
</tr>
<tr>
<td>Temperature [°C]</td>
<td>30-200</td>
<td>-195-225</td>
<td>-30-200</td>
<td>-200-350</td>
<td>30-200</td>
<td>30-200</td>
</tr>
</tbody>
</table>

**Type of service applicability**

<table>
<thead>
<tr>
<th>Type of fluid media</th>
<th>Liquid-liquid</th>
<th>Gas-liquid</th>
<th>Gas-gas</th>
<th>Condensation</th>
<th>Evaporation</th>
<th>Nature of type of fluid media</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid-liquid</td>
<td>E</td>
<td>E</td>
<td>E</td>
<td>E</td>
<td>E</td>
<td>M</td>
</tr>
<tr>
<td>Gas-liquid</td>
<td>G-P</td>
<td>E</td>
<td>E-P</td>
<td>E-P</td>
<td>G-P</td>
<td>M</td>
</tr>
<tr>
<td>Gas-gas</td>
<td>M-P</td>
<td>G-P</td>
<td>G-P</td>
<td>G-P</td>
<td>M-P</td>
<td>M</td>
</tr>
<tr>
<td>Condensation</td>
<td>G-P</td>
<td>E</td>
<td>E-M</td>
<td>E-M</td>
<td>E-M</td>
<td>M</td>
</tr>
<tr>
<td>Evaporation</td>
<td>G-P</td>
<td>E</td>
<td>E-M</td>
<td>E-M</td>
<td>E-M</td>
<td>M</td>
</tr>
</tbody>
</table>

**Nature of type of fluid media**

| Corrosive/aggressive         | M-P           | M-P        | E        | E            | M          | M                             |
| Viscous                      | E             | M-P        | E        | G-M          | E          | E                             |
| Heat sensitive               | E             | E          | E        | E            | E          | E                             |
| Hostile reaction             | M-P           | M          | G        | G            | M-P        | E                             |
| Fibrous                      | P             | P          | P        | P            | E          | P                             |
| Slurries                     | M-P           | P          | M        | M            | G          | M                             |
| Fouling                      | G-M           | M          | M        | M            | G          | M                             |

**Maintenance flexibility**

| Mechanical cleaning          | B             | N          | O        | N            | B          | B                             |
| Repair                       | B             | N          | O        | N            | B          | B                             |
| Modifications                | B             | N          | O        | N            | B          | B                             |

*Notes: E-excellent, G-good, M-moderate, P-poor, B-both sides of plate, O-one side of plate, N-no side of plate.*


2.1.3 Plate-fin heat exchangers

This type of heat exchanger is a kind of compact heat exchanger that transfers heat through fin chambers and plates. By adding fins to heat exchanger, the heat transfer surface could be increased. Heat can be transferred to fluid through parting sheet. The relatively small channels increase the turbulent flow of fluids. Heat transfer coefficients are relatively higher due to the higher thermal conductivity of the fins.

There are mainly four types of fins: plain, which refer to simple straight-finned triangular or rectangular design; herringbone, where the fins are placed sideways to provide a zigzag path; serrated and perforated which refer to cuts and perforations in the fins to augment flow distribution and improve heat transfer.

Compared to other types of heat exchanger, plate-fin heat exchangers have some significant advantages as follows.

1. Heat recovery is enhanced due to the lower minimum temperature approach, which could be as low as 1°C.
2. The volume of plate-fin heat exchangers can be much smaller, owing to the large surface area.
3. It provides the possibility of transfer heat between three fluids or more, which could reduce the number of heat exchangers.
4. It has a high degree of flexibility since it can operate with any combination of gas, liquid, and two-phase fluids.
5. Fins increase the structural integrity of the heat exchanger and make it possible for heat exchangers to withstand higher pressures.

Due to the compact size, lightweight characteristics and small temperature difference of plate-fin heat exchangers, this type of heat exchanger is widely used in the aerospace industry and cryogenics (Shah and Sekulic, 2003). However, there are some drawbacks of plate-fin heat exchanger. First, it is more likely to suffer from fouling issue due to its small flow channels. Second, the cost of maintenance and manufacture is high. More importantly, the operating temperature is relatively low, which cannot satisfy the process temperature requirements.
2.2 Factors affecting heat transfer

When optimizing a plate heat exchanger, several factors can affect heat transfer. In this section, chevron-plate geometry and flow arrangements are introduced to state how they influence heat transfer behaviour. In addition, the previous optimal design methods of plate heat exchanger are reviewed.

![Typical plate corrugation patterns](image)

**Figure 5** Typical plate corrugation patterns: (a) washboard, (b) zig-zag, (c) chevron, (d) protrusions and depressions, (e) washboard with secondary corrugations, (f) oblique washboard (Sundén and Manglik, 2007)

2.2.1 Chevron-plate geometry

There are a numbers of plate type configurations, such as washboard, chevron, and zigzag to choose when designing a plate heat exchangers, as shown in Figure 5. However, among the 60 different types of plate pattern, the chevron-plate has proved to be the most efficient one and is widely used in plate heat exchanger over the decades (Kakac et al., 2012).
Almost 20 types of chevron plates with different sizes and chevron angles are available. Different plate geometries can lead to different heat transfer behaviour and pressure drop. Thus, to optimize a single welded plate heat exchanger, chevron-plate geometry should be considered.

The basic geometry of the chevron plate is shown in Figure 6, and it can be seen clearly that the main parameters of the single plate are plate length for heat transfer ($L_P$), chevron angle ($\phi$), plate width for heat transfer ($B_p$). Among them, chevron angle is an important factor that dramatically influences heat transfer.

![Figure 6](image)

**Figure 6** Basic geometry of chevron plate (Shah et al., 1990)

There are three different corrugation plates with different chevron angle as shown in Figure 7. The H-type channel has relatively large heat transfer and pressure drop, for which the chevron angle is about 60°. However, the chevron angle of L-type channel is 30°, which have lower intensity of heat transfer and hydraulic resistance. M-channels combine H-type channels and L-type channels so that make it possible for this type channel to make intermediate heat transfer.

![Figure 7](image)

**Figure 7** Three types of channels: (a) L type channel, (b) H type channel, (c) M type channel (Arsenyeva et al., 2011)
2.2.2 Flow Arrangement

In plate heat exchangers, there are three types of generalized flow arrangements to achieve heat transfer between fluids, as shown in Figure 8 as shown: (a) co-current flow (b) counter-current flow (c) multi-pass flow arrangement.

![Diagram of flow arrangements](image)

**Figure 8** Different flow arrangement: (a) co-current flow, (b) counter-current flow, (c) multi-pass flow
Co-current flow and counter-current flow are the two basic patterns. However, counter-current is more widely used in industry than co-current flow owing to its relatively high heat transfer efficiency.

Single pass is that pattern when both of cold and hot streams flow in a single direction through all the channels. Similarly, a multi-pass arrangement is the pattern when any of cold or hot streams flow in more than more direction. For the multi-pass arrangements, the fluids would stay for more time and have a longer distance in welded plate heat exchangers, which may lead to better heat transfer. The number of passes of each stream can be different, especially when the pressure drop on one side needs to minimized or the ratio of the flow rate is relatively high, then different numbers of passes are considered.

However, among the numerous multi-pass arrangements, only a few arrangements are applied in practice in industry. Consideration of pressure drop constraint, normally the number of passes are not exceeded four.

2.2.3 Fouling

Fouling phenomena significantly decreases the thermal effectiveness to heat flow (Webb and Kim, 1994), which is a key factor that affects the application of plate heat exchanger into chemical processes (Abu-Khader, 2012). An insulting layer created by deposits over the surface of plate heat exchangers dramatically narrows the flow area, which leads to the increase of hydraulic and thermal disturbances (Sundén and Manglik, 2007). As a result, an additional cost for energy lost and cleaning expenses is added to industrial processes. Thus, to maintain the production efficiency, deposits that causes fouling must be removed by cleaning procedures.

2.3 Basic design method of a single PHE

The methodology employed for design of plate heat exchangers is generally the same as the thermal-hydraulic design methods for other types of heat exchangers (Shah et al., 1988, Raju and Bansal, 1983). There are five considerations when designing a plate heat exchanger as follows:

1. Process specifications
2. Thermal-hydraulic design
3. Operation and maintenance constraints, mechanical design
4. Manufacturing consideration
5. Trade-off consideration and optimization

In this part, the general thermal-hydraulic design methods are stated in detail. Sizing and rating problems are the two fundamental problems in heat exchangers design. The rating problem is to derive the outlet temperature of fluids and the total heat load with a given geometry of heat exchanger and inlet temperature of fluids. However, for a sizing problem, the total heat load and both of inlet and outlet temperature of fluids are given. The aim of this type of problem is to determine the geometry of heat exchange, total area, and flow arrangement. (Shah, 1983, Thonon et al., 1995)

There are two main methods to perform thermal design for plate heat exchanger, which include logarithmic mean temperature difference (LMTD) method and $\varepsilon$-number of transfer units (NTU). Shah and Mueller (1985) summarized other methodologies.

2.3.1 Logarithmic mean temperature difference (LMTD) method

The logarithmic mean temperature difference is a logarithmic average of the temperature difference between the hot and cold fluids at each end of the heat exchanger, which is used to determine the temperature driving force for heat transfer in heat exchangers. The larger the LMTD, the more heat is transferred. The LMTD is defined by the logarithmic mean as follows:

\[
LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}
\]  

(1)

In the above equation, $\Delta T_1$ represents the temperature difference between the two fluids at inlet or outlet. $\Delta T_2$ is the temperature difference between the other end. LMTD is used to calculate total transferred heat in heat exchanger:

\[
Q = U \times A \times LMTD
\]

(2)

Where $Q$ is the total heat load, W; $U$ is the heat transfer coefficient, W/(K·m$^2$), $A$ is the surface area of heat exchanger, m$^2$.

LMTD could be derived by the following method. Select an element of area in a heat exchanger, the heat transfer flux could be:

\[
dQ = U \times dA \times \Delta T = U \times dA \times (T_h - T_c)
\]

(3)
Assume that the heat transfer coefficient is constant; the above equation can be used across the entire surface area. On the other hand, by employing the energy balance equations, the heat transfer flux could be:

\[ dQ = C_h \times dT_h = C_c \times dT_c \]  

(4)

Where \( C_h \) and \( C_c \) are the heat capacity of hot and cold fluids respectively, \( \text{J/(kg}\cdot\text{K}) \).

Combining Equations (3) and (4), we can get:

\[ d(T_h - T_c) = dQ \left( \frac{1}{C_c} - \frac{1}{C_h} \right) \]  

(5)

Then we obtain:

\[ \frac{d(T_h - T_c)}{T_h - T_c} = U \left( \frac{1}{C_c} - \frac{1}{C_h} \right) dA \]  

(6)

For co-current flow, the integrated equation for the entire heat exchanger is:

\[ T_{h,0} - T_{c,0} = (T_{h,i} - T_{c,i}) \exp \left( UA \left( \frac{1}{C_c} + \frac{1}{C_h} \right) \right) \]  

(7)

Where \( T_{h,0} \) and \( T_{c,0} \) are the outlet temperature of the hot stream and cold stream, K; \( T_{h,i} \) and \( T_{c,i} \) are the inlet temperatures of hot stream and cold streams, respectively.

Similarly, for counter-current flow, we can obtain:

\[ T_{h,0} - T_{c,i} = (T_{h,i} - T_{c,c}) \exp \left( UA \left( \frac{1}{C_c} - \frac{1}{C_h} \right) \right) \]  

(8)

By combining Equations (4), (7) and (8), we obtain:

\[ Q = UA \frac{(T_{h,i} - T_{c,i}) - (T_{h,0} - T_{c,0})}{\ln(T_{h,i} - T_{c,i})/T_{h,0} - T_{c,0}} = UA \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \]  

(9)

This equation can only be used for single-pass plate heat exchangers. As for the multi-pass flow arrangements, it becomes more complex when it comes to the mean temperature. A correction factor \( F \) is introduced to correct LMTD in counter-flow.

\[ Q = U \times A \times F \times \Delta T L M T D \]  

(10)

Where \( F \) is the ratio between true mean temperature difference and LMTD, it is fixed by the flow arrangement, the ratio between the heat capacity of two streams \( R \) and the heat transfer effectiveness \( \varepsilon \).
Chapter 2

Literature Survey

\[ F = f(R, \varepsilon, \text{flow arrangement}) \]  \hspace{1cm} (11)

\[ \varepsilon = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} \] \hspace{1cm} (12)

\[ R = \frac{C_c}{C_h} \] \hspace{1cm} (13)

However, there is no specified method to calculate the correction factor for different flow arrangements. The traditional method is to create corresponding graphs for selected flow arrangement by analytical and numerical methods.

2.3.2 The \( \varepsilon - \text{NTU method} \)

Although LMTD method is widely used in designing welded plate heat exchangers, this method is more suitable with conditions where the inlet and outlet temperatures of streams are given. The NTU method is used to calculate the rate of heat transfer in heat exchangers when there is not sufficient information to calculate LMTD. This method not only solves the design problem but also the rating problem of welded plate heat exchangers.

We define the ratio between actual heat transfer and the maximum transfer heat that could be achieved in the heat exchanger as heat transfer effectiveness \( \varepsilon \) as follows:

\[ \varepsilon = \frac{Q}{Q_{\max}} \] \hspace{1cm} (14)

\[ Q_{\max} = C_{\min}(T_{h,i} - T_{c,i}) \] \hspace{1cm} (15)

Where \( C_{\min} \) is the minimum heat capacity between \( C_c \) and \( C_h \), \( J/(\text{kg} \cdot \text{K}) \). Combining Equations (14) and (15), we can obtain:

\[ Q = \varepsilon C_{\min}(T_{h,i} - T_{c,i}) \] \hspace{1cm} (16)

At the same time, the heat transfer effectiveness \( \varepsilon \) can also be expressed by the following function:

\[ \varepsilon = f(\text{NTU}, R, \text{flow arrangement}) \] \hspace{1cm} (17)

Where NTU is defined as:

\[ \text{NTU} = \frac{U A}{C_{\min}} \] \hspace{1cm} (18)
In the equation above, $U$ is the overall heat transfer coefficient, W/(m$^2$·K); $A$ is the total transfer area, m$^2$.

$$R = \frac{c_{\text{min}}}{c_{\text{max}}}$$  \hspace{1cm} (19)

Where $c_{\text{min}}$ and $c_{\text{max}}$ are the minimum and maximum heat capacities of two streams, J/(kg·K).

By rewriting the Equation (8), we can obtain:

$$T_{h,0} - T_{c,i} = (T_{h,i} - T_{c,o})e^{\frac{UA}{c_{\text{min}}} \left( \frac{C_{\text{min}}}{C_{c}} + \frac{C_{\text{min}}}{C_{h}} \right)}$$  \hspace{1cm} (20)

$$Q = C_{h}(T_{h,i} - T_{h,0}) = C_{c}(T_{c,o} - T_{c,i})$$  \hspace{1cm} (21)

By combining Equations (16), (20) and (21), the heat transfer effectiveness $\varepsilon$ for co-current flow can be restated as:

$$\varepsilon = \frac{1 - \exp\left[ -\left(1 + \frac{c_{\text{min}}}{c_{\text{max}}} \frac{UA}{c_{\text{min}}} \right) \right]}{\frac{c_{\text{min}}}{c_{\text{max}}} + 1}$$  \hspace{1cm} (22)

Which can be simplified as:

$$\varepsilon = \frac{1 - \exp\left[ -(1 + R)\text{NTU} \right]}{1 + R}$$  \hspace{1cm} (23)

Similarly, the heat transfer effectiveness $\varepsilon$ for counter-current flow can be written as:

$$\varepsilon = \frac{1 - \exp\left[ -\text{NTU}(1 - R) \right]}{1 - R \cdot \exp\left[ -\text{NTU}(1 - R) \right]}$$  \hspace{1cm} (24)

The methodology of applying $\varepsilon$ - NTU method to multi-pass flow arrangement will be detailed in the next chapter.

### 2.4 Previous literature review of plate heat exchanger design

Focke (1986) tried to find the minimal heat transfer area in 1996, which was among the earliest attempts of seeking the appropriate plate pattern. Cooper (1983) and Shah (1988) used logarithmic mean temperature difference (LMTD) and $\varepsilon$ - NTU method to set up a thermal-hydraulic model of plate heat exchangers. They took plate patterns with different geometries and pressure drop into consideration by employing trial iterations.
These traditional methods made great progress for plate heat exchanger design, but they are time-consuming and neglect the effect of flow arrangement.

Wang and Sunden (2003) utilized a series of different correlations to predict the performance of plates according to the specified chevron angle and Reynolds number. When optimizing the total cost of plate heat exchangers, the conditions that include pressure drop and without pressure drop are considered. Similarly, researchers proposed different methods to investigate the performance of chevron-type plates in plate heat exchangers (Martin, 1996, Tsai et al., 2009, Muley and Manglik, 1997, Muley and Manglik, 1999, Muley et al., 1999, Heggs et al., 1997). However, these methods could only apply to plate heat exchangers with single pass arrangements. The multi-pass flow arrangement still fails to be included.

Mehrabian (2009) presented a design methodology for plate heat exchangers, which could be applied manually. Heat transfer and hydraulic resistance correlations have been estimated. Similarly, Picon-Nunez et al. (2006) proposed a graphical representation design method to select multi-pass flow arrangement. Picon-Nunez et al. (2010) further developed this method by proposing parameter plots to select plate patterns (Pinto and Gut, 2002, Gut and Pinto, 2003, Gut and Pinto, 2004, Gut et al., 2004). However, this method is too time-consuming and fails to consider the effect of chevron angle on pressure drop and heat transfer coefficient.

By employing existing performance data, plates can be rearranged to provide an insight in operating of a two-stream plate heat exchanger (Wright and Heggs, 2002a, Wright and Heggs, 2002b). Though adjusting existing plates, the plate heat exchanger can fulfill the requirement of process conditions. New correlations between Nusselt number and friction factor were proposed by Kanaris et al. (2009), which were developed by using CFD software to model the flow in PHE channels. However, there are still some practical restrictions that have not been considered.

Currently, in the practical application of plate heat exchangers, it is more likely to combine different types of plates with several chevron angles in one heat exchanger in order to achieve better heat transfer behavior. The earliest attempt was done by Marriot (1977), he proposed a design method of one pass flow arrangement with different angles of plates. Considering pipiing and maintenance costs, a one-pass flow arrangement has some merits. However, multi-pass flow arrangements are more efficient for satisfying the requirement of heat load.
With relative high velocities of streams, higher heat transfer coefficients could be achieved in multi-pass flow arrangements. Zinger et al. (1988) and Kumar (1984) suggested the LMTD correction factor to deal with multi-pass arrangements. Pignotti et al. (1992) developed a methodology to analysis complex flow arrangements in two-stream recuperators, in which the overall heat transfer coefficient and fluid properties were taken as constant. The correlations for temperature effectiveness, and NTU for multi-pass arrangement in plate heat exchangers, have been proposed by Kandlikar et al. (1989), which cover the number of passes up to four.

With the rapid development of computers, researchers started to seek more time saving tools to solve this problem. Tovazhnyansky et al. (1992) set up a series of equations of algebraic functions to calculate the different heat transfer coefficient in each pass. Then Arsenyeva et al. (2009) further improved this method by considering the effect of mixed plates. Then, Arsenyeva et al. (2011) and Arsenyeva et al. (2013) set up a more complete model for the optimal design a single plate heat exchangers. This not only took into account different channel arrangements, but also considered various flow arrangements. However, it failed to consider fouling issues.

Najafi et al. (2010) used a genetic algorithm to develop a multi-objective optimization for plate heat exchangers. According to the different demands, users could select the optimal solutions based on their requirement. Mota et al. (2014) and Qiao et al. (2013) proposed different optimization methods for considering the effect of flow arrangement and phase change. But these methods are time-consuming and have not addressed the effect of fouling.

Fouling has a complicated behavior, the main factors that affect fouling in plate heat exchangers have been reviewed (Petermeier et al., 2002, Zettler et al., 2005). How fouling influences flow distribution with variety of plate patterns and the effect of fouling issues on flow maldistribution between channels has been investigated (Bossan et al., 1995).

Due to the highly turbulent flow in a plate heat exchanger, it is less prone to fouling than a shell and tube heat exchanger. Besides, the velocity profile of a plate heat exchanger is more uniform. Thus, low velocity zones, which are most vulnerable to fouling, are eliminated.

Fouling in the hot end of the preheat train mainly relates to high wall temperature and low shear stress. The shear stress is relatively high in welded plate heat exchangers due
to turbulence in the flow. However, it is difficult to quantify the potential improvements. Thus, this project will focus on develop a systematic optimization method of integrating a mixture of plate heat exchangers and shell-and-tube heat exchangers into heat exchanger networks and quantify the potential benefit in terms of energy saving through analysis case studies.

2.5 Background on network pinch

For an existing heat exchanger network, pinching the network shows all the possibilities to reduce energy consumption by utilizing the degrees of freedom for a certain network structure. The degree of freedom is the total number of utility paths and loops in the existing HEN. The open path that connects hot and cold utilities through process exchangers is called utility path and the closed path from one exchanger and return back to the same exchanger is called a loop.

Figure 9 showed as an example of an existing heat exchanger network. The heat recovery of this HEN is 200 MW with minimum approach temperature $\Delta T_{\text{min}}$ of 20°C, and Composite Curves show that $\Delta T_{\text{min}}$ is 22.5 °C to achieve the same amount of heat transfer.

![Figure 9 An existing heat exchanger network (Asante and Zhu, 1997)](image)

The only one degree of freedom in this existing heat exchanger network is the utility path as shown in the bubble region of Figure 9a. To maintain the existing structure of the network, the only possible solution to save energy consumption is to exploit this utility path, which starts from the heater (H) and ends up at the cooler (C). To determine the amount of energy saving that can be achieved, the $\Delta T_{\text{min}}$ is reduced to 0°C and the heat recovery is enhanced by 20MW shown in Figure 10a. It can be clearly seen from the Figure 10b that the targeting maximum energy recovery is still larger than the one
obtained from Figure 10a. The main reason that limits the heat recovery is because the structure of existing HEN itself.

**Figure 10** Maximum energy recovery of existing HEN

The heat exchanger match, which restricts the heat recovery, is called as the pinching match. Figure 11a shows the pinching match in the network, and the point that cold and hot streams touch is referred as network pinch. Generally, the network pinch temperature is derived when the minimum approach temperature is set as 10°C or 20°C. However, we assume $\Delta T_{\text{min}}$ as 0°C at this stage for the sake of illustration only.

**Figure 11** Pinching match and network pinch

Topology changes are the only option to solve the limited energy saving problem caused by the network pinch. As shown in Figure 12, there is a trade-off between heat recovery and adding extra heat transfer area when it comes with structure modification in the retrofit design of an existing HEN. Adding a new heat exchanger and stream splitting, resequencing and repiping are regarded as the main ways to change topology.
in terms of retrofit HEN. Among them, repiping, which refers to moving of the heat exchanger to another location, is not considered in this research since the implementation of repiping is impractical in many situations. For example, the pressure rating of equipment and the material of construction might not be suitable for other streams. The detailed background on how these methods carry this out is shown as follows:

Figure 12 Trade-off after each modification

1. Adding new heat exchanger: the reduces heat load for the existing match and adjusts the duty below or above the pinch by creating a loop or utility path (shown in Figure 13). Figure 13 shows an example of adding a new heat exchanger to creat a loop to explore greater heat recovery. If the new heat exchanger is inserted next to the pinching match, the new match with the new heat exchanger make the pinching heat exchanger to no longer exist. Then more heat load can be removed to the pinching match and more energy recovery is achieved through exploiting the utility path. Adding the new heat exchanger to create a utility path is only considered when there is a potential that the utility exchanger to be able to increase heat load and no violation of constraints for the network.
2. Splitting: when the number of pinching matches in the existing HENs are more than one, especially when the pinching matches are located next to each other, as shown in Figure 14, stream splitting is considered as an efficient way to achieve energy saving. If there is more than one stream with adjacent pinched exchangers, the one with the highest duty is selected. The selected pinched exchangers should be located on a variable utility path or loop in order to create an avenue to increase heat transfer and the splitting ratio of the split stream needs to be further optimized.
Figure 15 The details of resequencing to overcome the network pinch

3. Resequencing: this refers to relocating of the exchanger in the existing HEN, which eases the heat transfer of the pinching match through transferring heat from below to above the network pinch. By introducing a resequence, the pinching match can increase heat load, thus to further increase the energy saving. As shown in Figure 15, the constraints eased by changing the sequence of the pinching match so that the pinched exchanger is able to take up extra heat transfer with the utilization of utility path.

2.6 Existing methodologies for the HENs retrofit

This part reviews background of network pinch and different methodologies of HEN retrofit, including pinch analysis, mathematical programming, and hybrid methods.

2.6.1 Pinch analysis

Pinch analysis aims to minimize the energy consumption in the heat exchanger networks. This is achieved in pinch analysis by reducing cross-pinch heat transfer. Through the calculation of the thermodynamic behavior of the network, the composite curves, grand composite curve and grid diagrams are obtained to provide an insight into the pinch point and potential energy target. The pinch point is defined as the closet point between the hot and cold composite curves with pinch temperatures for hot stream and cold stream respectively. There are two basic principles for pinch analysis method to achieve the maximum heat recovery, which have been stated by Shenoy (1995) and Gundersen (2013). No cross-pinch heat transfer is allowed and cold or hot utilities are not allowed to locate above or below the pinch respectively. Targeting and design stages
are the two main stages of pinch analysis. The maximum heat recovery and minimum energy consumption are identified at the targeting stage, and the strategies and step-by-step procedures to achieving the target energy saving based on the principles of pinch analysis is specified at the design stage.

In 1986, the concept of pinch analysis was first introduced to the retrofit design of heat exchanger networks by Linnhoff and Tjoe (1986). The pinch design method was used to eliminate the cross-pinch heat transfer by rearranging heat exchangers transferring heat across the pinch and adding new area to the heat exchangers. However, this methodology is based on user interaction, and it relies heavily on users experience. Thus, this methodology is not suitable for complex heat exchanger networks. A linear model (Shokoya, 1992), which used an area matrix to reflect the detailed area distribution in the heat exchanger network, is set up to achieve the energy target. Vertical heat transfer between hot and cold sides is assumed at the first stage. A deviation area matrix is defined as the target area minus the existing area matrices at the second stage. So far, the retrofit target focused on the area, but a realistic retrofit should link the modifications with their economic cost.

A cost-based retrofit design of HENs was proposed in 1993 (Carlsson et al.), which involved the cost of maintenance, heat exchanger area, and piping. Criss-cross heat transfer is allowed in this methodology. A computer-aided optimization model is formulated to find the optimal matches with target heat recovery at certain minimum approach temperature. The proposed methodology is not suitable for large-scale applications since a large amount of work and computation time is required to obtain the optimal solution.

Due to the large scale and complexity of heat exchanger networks, researchers started to seek a methodology to simplify the HEN before solving. Path analysis, which was proposed by Reisen et al. (1995), was used to decompose the existing heat exchanger network and evaluate the subnetwork. By applying path analysis, the entire network is decomposed to the subnetworks, which significantly reduces the complexities of the problem (Sreepathi and Rangaiah, 2014). In 1998, Reisen et al.(1998) improved the existing method by introducing zones for finding the most sensitive part in the HEN to retrofit. The novel method specified the targeting locations to apply topology change and could achieve more energy saving.
A graphical methodology proposed by Nordman and Berntsson (2001) was introduced for the retrofit design of HENs. A novel pinch curve, which stressed the cross-pinching heat transfer and the trade-off between heat transfer area and energy target, was developed to obtain a cost-effective retrofit design. This work was further studied by Nordman and Berntsson (2009) to investigate different scenarios of all the possible retrofit solutions based on economic considerations. The result of all the possible solutions was that when replacing the heaters/coolers closer to the pinch, a higher economic benefit was achieved.

Li and Chang (2010) proposed a general guideline based on the pinch retrofit method. This method was used to identify the heat exchangers that limit the heat recovery and eliminate each cross-pinching match. The modified pinch method achieved the energy target by dividing the heat loads into two parts and matching the new streams on the same side of pinch point. Although checking the feasibility of the new match secures the retrofit design is reasonable, the retrofit solutions might cause uneconomic issue.

### 2.6.2 Optimization methods

Optimization methods use mathematical programming models to solve the retrofit problem based on different objectives. The key point of applying optimization method into the HEN retrofit problem is to find an accurate method to represent the problem and determine the optimal solution. The optimization method exploits either a deterministic approach or stochastic approach. Deterministic approaches are normally achieved by Non-linear programming (NLP), mixed integer linear programming (MILP) or mixed integer non-linear programming (MINLP). The stochastic approach is based on genetic algorithms (GA) or simulated annealing (SA).

#### 2.6.2.1 Deterministic approach

Optimization methods were first applied to heat exchanger retrofit by Yee and Grossmann (1986) to set up a MILP model. This MILP model was used to predict the minimum topology change in an existing HEN by maximizing the using of heat transfer equipment and re-matching the existing heat exchangers units. Since this methodology is based on the existing structure of heat exchanger network, the structure of the retrofit network is similar as the original HEN.

A novel two-stage methodology developed by Ciric and Floudas (1989) aimed to minimize the retrofit cost of the existing HENs to achieve a certain amount of heat
recovery. An MILP model was formulated in the first stage depending on the given data of HENs. The retrofit process allows for repiping, adding new heat exchangers, adding extra heat transfer area and relocating the existing heat exchangers. The second stage is to generate superstructures that show all the possible HEN structures by analysing the results from the first stage. The MILP model is simplified to an NLP model. This work was further extended by Ciric and Floudas (1990). The main achievement in the new proposed method was that it combined the two-stage approach to one stage to achieve simultaneously optimization.

A two stage retrofit methodology for HENs, which contains pre-screening and optimization stages, was proposed by Yee and Grossmann (1991). At the pre-screening stage, the optimal economic cost at different levels of heat recovery was to be determined. The economic cost included utility cost, cost of adding additional heat transfer area, and cost of topology change. The utility cost was estimated from the model developed by Papouli as and Grossmann (1983). The additional heat transfer area was determined by the method proposed by Townsend and Linnhoff (1984). The model proposed by Yee and Grossman (1991) for topology change was used for calculating the economic cost of topology change. The optimization stage optimizes the minimum number of new heat transfer equipment requirements to obtain the optimal cost. An MINLP model is built to select the optimal solution from all the possible superstructure retrofit designs.

In 1999, a novel methodology, which involves screening and optimization stage for the retrofit design of HENs, was proposed (Briones and Kokossis). At the first stage, the integer variables were introduced to identify structure modifications and the area of heat exchanger and heat transfer requirement were set as continuous variables. The decisions on how many new heat exchangers and how many structure modifications were required based on objective of the model (e.g. minimum economic cost, minimum total area). The optimization stage uses the results from the screening stage to set up a hyper-target and obtain the retrofit solution. The hyper-target approach used solution streams instead of employing energy-area curves. Compared to the former methodology, this approach effectively reduced the heat transfer area and the retrofit expense and the relevant results are shown in the literature (Saboo et al., 1986, Ahmad and Patela, 1987, Carlsson et al., 1993, Ciric and Floudas, 1989).
A two-stage retrofit approach of HENs, which involved developing a Constant Approach Temperature (CAT) model and an MINLP model to find the optimal solutions, was developed by Ma et al. (2000). One of the dramatic features of the CAT model was that it linearized the calculation of heat transfer area as the minimum approach temperature for all the heat exchangers were fixed. The CAT model, which was based on the approach proposed by Yee and Grossman (1991) can be simplified as a MILP model and less computation time is required. It is more likely to obtain the global solution as the area calculations are linearized. Since the model fails to consider the exchanger area in the first step, an MINLP model is formulated at the optimization stage to account for the area. To solve the difficulty of finding the global optimal solution, an initial guess was provided based on the result from the first stage. Structure change, heat recovery and area of exchangers were considered simultaneously. The computation effort was significantly reduced by applying the proposed method.

Various types of heat exchangers, including conventional shell and tube heat exchangers, gasket plate heat exchangers and double pipe heat exchangers, were considered in the retrofit design of HENs by developing MINLP model, which was proposed by Sorsak and Kravanja (2004). The main significance of this methodology was that it compared the different types of heat exchangers with only using single type of double pipe exchanger model in the superstructure. The results showed that the feasibility of the retrofit design relies heavily on the selection of heat exchanger type.

Ponce-Ortega et al. (2008b) developed a novel methodology to take into the consideration of the non-constant process conditions experienced in different operating condition. A MINLP model was proposed for the retrofit design of HENs based on the superstructure model developed by Yee and Grossmann (1991). Before this, most of the research neglected the change of operation conditions and regarded them as constant. Stream were assumed to have constant temperatures during the process (Ponce-Ortega et al., 2008a).

Nguyen et al. (2010) developed a temperature interval approach for the retrofit of HENs by optimizing network topology and area of heat exchanger simultaneously. This approach is based on the model proposed by Barbaro and Bagajevicz (2005). The hot and cold streams were separated into numbers of temperature intervals, and the heat transfer in each small interval could be obtained. The scenarios of allowing for relocating of heat exchangers or not were both considered.
A MILP iteration model proposed by Pan et al. (2013) was developed for the retrofit design of HENs. They applied a pinching match to the retrofit of the network superstructure by two optimization stages. The matches with small heat duty or area were eliminated after the optimization process in the first stage. In the second stage, the best topology derived from the first stage was further optimized to obtain the minimal economic cost of retrofit. Through simplification of the MINLP problem to MILP, the proposed method significantly reduced the computation effort.

The significant advantage of deterministic optimization was that it became possible to apply it to different types of retrofit problem with flexibility to the various scenarios. However, due to the complexity of the non-linear model, it is relatively difficult to obtain the global optimal and solutions are sensitive to the initial points.

2.6.2.2 Stochastic-based approach

Stochastic-based approaches involve simulate annealing (SA), integrated different evolution (IDE) and genetic algorithms (GA). Simulated annealing was applied to retrofit design of HENs in 1996 (Nielsen et al.). The model was based on the approach proposed by Dolan et al. (1989). The optimization process including selecting the heat exchanger type, and thermal behaviour based on the constraints. Hydraulic resistance was considered in the research investigated by Nielson et al. (1996).

A novel automatic two-loop methodology, using SA for the optimization of the network, was developed by Athier et al. (1998). In the outer loop, the topology change of HEN was optimized by SA, subject to process constraints. The economic cost of adding new heat exchangers, repiping and relocating of heat exchangers were included. In the inner loop, a NLP model was formulated to optimize adding extra area of heat exchanger.

Rodriguez. (2005) proposed a method for the retrofit of HENs by using SA to formulate an optimization model, which involved integer and continuous variables. The structure modifications were defined as integer variables and heat transfer duty and area were defined as continuous variables. The results were not accurate as the model was simplified.

A two-level approach making use of the GA algorithm was presented by Bochenek and Jezowski (2006) for the retrofit design of HENs. A structure matrix was formulated which contained multiple variables to present the key features of the network at the first level. In the first stage, the optimization process was achieved by a GA algorithm. The
location of the splitters and heat exchanger details were obtained. These results were used to determine the heat transfer area of exchanger and the split ratio of the splitters at the second level. However, the complexity of the model required prolonged optimization effort, regardless of the size of the heat exchanger network.

Razaei and Shafiei (2009) proposed an approach to integrate GA into nonlinear programming and integer linear programming (ILP) into the retrofit design of HENs to overcome the difficulty of handling with continuous variables in GA. The objective function of this method was to minimize the retrofit expense and the continuous variables were heat transfer duty, split ratio of the splitters and hot and cold stream temperatures. These variables were optimized by the NLP model. Topology change was determined by building a GA model to obtain the node representation of the heat exchanger locations. At the end, the objective function was solved by an MILP model.

Zhang and Rangaiah (2013) combined the IDE model proposed by Zhang and Rangaiah (2011), and topology representation developed by Jezowski et al. (2007) together with model presented in 2010 (Bochenek and Jeżowski) for the retrofit design of HENs. This method allowed it possible to deal with the different types of variables simultaneously.

A hybrid GA algorithm was developed by Liu et al. (2014) for the HEN retrofit. A MINLP model was formulated to minimize the total retrofit cost, including the utility cost and modification cost. The optimization procedure was divided into six steps based on the superstructure, fully using of the existing network structure. The analysis of the exiting HENs to select the correct retrofit sequence was produced at the first stage. After determining the sequence of exchangers at the second stage, the locations of the existing exchangers were represented at the third stage. Through analysing the heat transfer area based on the hybrid genetic approach, the existing heat exchangers were relocated in the fourth stage. The final network structure was obtained through revising and conducting a genetic algorithm approach for the last two stages.

Compared to deterministic method, stochastic optimization method shows more possibility to avoid local optimal solutions, but it requires more computation effort because of the nature of the optimization process.

The main advantage of mathematical programming is that it is flexible to the retrofit target and process data as it converts the optimization problem to the automated mathematical model. However, it is difficult to apply mathematical programming methods to large scale retrofit problems since it is challenging to formulate the
mathematical model accurately. The optimization process normally is achieved by one step or multi-step. To obtain the optimal solution of retrofit problem, one step optimization is preferred. For the multi-step optimization model, although the computation time is possible to reduce, the effect on accuracy of results cannot be ignored.

2.6.3 Hybird Methodology

Hybird approaches combine mathematical programming methods and pinch analysis methods while keeping their advantages. Hybird methods can automatically obtain the optimal solution and keep user interaction, thus it is possible to achieve the effective retrofit solution with less computation time.

Briones and Kokossis (1996) applied a hybrid methodology into the retrofit design of HENs, which included three stages in total. In the targeting stage, a MILP model was formulated to find the possible solutions within the required targets of area and modifications. The results obtained from the first stage were used in the second stage to find the optimal solution for the exchanger area and topology changes based on the pinch analysis method. In the third stage, a NLP model was formulated and a superstructure derived based on the retrofit result to minimize the capital cost. However, it is not the best retrofit solution if only energy saving is taken into consideration.

Asante and Zhu (1996) (1997) developed a methodology to minimize the steps of topology changes to achieve the target heat transfer in the retrofit design of HENs by applying a hybrid method. The locations that limit heat recovery and the structure change were identified by the network pinch in the diagnosis stage. The objective of the optimization stage was to maximize the heat recovery through rebalancing heat load among heat exchangers after each modification. The optimal solution can be selected with minimum retrofit cost.

Varbanov and Klemes (2000) combined pinch analysis and mathematical programming into the retrofit design of HENs to derive various solutions of structure modification for the existing HENs. Smith et al. (2010) further modified the network pinch approach and combined topology changes energy-capital trade-off optimization to a single step. This method considers the temperature dependency of the thermal properties since this gives a true representation of the interaction in the network during retrofit.
Bakhtiari and Bedard (2013) developed a method considering of some practical situations (e.g. stream splitting and segmentation) based on the approach proposed by Asante and Zhu (1997). In the topology change stage, the target was to achieve the maximum heat recovery. In the cost optimization stage, the objective was to minimize using extra heat transfer area. The cost-energy target was possible with this methodology.

Hybrid methods can deal with large-scale retrofit problems. This is because the method is sequential. The utility consumption can significantly reduce due to using of degrees of freedom of the network. However, extra area is added into the existing heat exchanger to keep the balance of the network. The key drawback of hybrid method is that instead of taking economic cost into the first consideration, heat recovery is always put into the first place for the retrofit target, which may leads to high retrofit cost.
Chapter 3

Publication 1: Automated design of two stream multi-pass plate heat exchangers

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Automated design of two stream multi-pass plate heat exchangers

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Abstract

Gasket and welded plate heat exchangers (PHEs) have a significant potential to improve energy efficiency in the process industries. However, to realise the full potential to achieve such energy savings requires a systematic approach to screen the many options available, which is currently lacking a reliable optimization approach for PHEs with different types of flow arrangement and plate geometries. This work presents a novel approach for the optimal design of two-stream multi-pass plate heat exchangers, including gasket and welded plate heat exchangers, with different plate geometries and flow configurations. This paper presents a new method to obtain the optimal global solution with minimum total transfer area. An MINLP mathematical model is developed to select the best combination of the flow pass configuration using available commercial plate geometries constrained to be within practical design considerations. Two case studies are used to demonstrate the proposed method for both gasket and welded PHEs. Results show that better design with reduced heat transfer area by 10.71% is obtained compared with previously proposed approaches.

Keywords: Plate heat exchanger, Optimization, design, MINLP mathematical model

Highlights:

- A design and optimization methodology for two different types of plate heat exchangers (PHEs), including gasket and welded plate heat exchangers, is proposed
- Plate pattern and flow arrangement selection are addressed
- Flow arrangement is integrated into the design by applying LMTD method
-The total heat transfer area can be considerably decreased through the proposed method

1. Introduction

Increasing energy consumption and CO₂ emissions have created an urgent necessity to improve energy efficiency in the process industries [1-3]. Effective exploitation of high-performance heat transfer equipment can make a major contribution to this goal [4]. Conventional shell-and-tube heat exchangers are the most commonly used type of heat transfer equipment [5, 6]. Although the methodologies for the design of a shell-and-tube heat exchanger are relatively mature, their inherent design features mean that they are largely restricted for use with relatively large temperature differences, restricting heat recovery potential. They also have a high capital cost and large physical size, restricting opportunities for retrofit of existing systems [6]. Additionally, shell-and-tube heat exchangers can be readily affected by fouling deposition, especially in high-temperature applications, such as crude oil pre-heating [7, 8]. By contrast, plate heat exchangers (PHEs) can significantly increase heat recovery through exploitation of small temperature differences, improve thermal-hydraulic behavior, and reduce energy consumption and greenhouse emissions [6, 9].

There are many forms of corrugated plate and channel arrangements for PHEs [10]. Fluids stay longer in PHEs when compared with shell-and-tube heat exchangers, since hot and cold streams flow over the entire plates [11]. For a given duty, the total volume and weight of PHEs are three times smaller than those of shell-and-tube heat exchangers [2]. High heat transfer coefficient in plate heat exchangers and their more efficient flow arrangements allow minimum approach temperatures in PHEs to be as low as 2 °C [6]. Higher heat recovery efficiency, smaller footprint, easier to deal with fouling mitigation and lower capital cost are the main advantages over shell-and-tube heat exchangers [6, 12]. Gasket plate heat exchangers (GPHEs) and welded plate heat exchangers (WPHEs) are the two main types of PHEs [6]. They have different structures as shown in Figure 1 and Figure 2. For GPHEs, the number of plates can be adjusted by adding or removing plates, which gives a flexible thermal design to meet the process requirement [6]. However, the gaskets restrict the temperature and pressure to be below 200°C and 25 bar [4]. Compared with GPHEs, the integrity of WPHEs is significantly enhanced, since plates are welded together. Consequently, WPHEs can tolerate higher temperatures (up to 350°C) and higher pressures (up to 40 bar),
exceeding the gasket limitations [4]. Owing to the unique structure of WPHEs, they are less likely to suffer from leakage issues and are more suitable for conditions of rapid-change [13]. Based on these distinct features, PHEs are commonly applied in food, petrochemical plants and other energy-intensive process industries [14]. To increase their potential to the full, it is important to enhance the thermal-hydraulic behaviour of PHEs to reduce the capital cost further and further increase energy recovery.

Fig. 1. The structure of a gasket plate heat exchanger [1]

Fig. 2. The structure of welded plate heat exchanger (Compabloc) [1]

The heat transfer behaviour of PHEs is significantly affected by plate geometry, plate type and flow arrangement, which needs to be systematically optimized. Figure 3 shows the detailed corrugations of several commonly used plates. In the recent years, it has been demonstrated that chevron-plates are the most energy-efficient plate type amongst the over 60 different types of plates, and commonly used by manufacturers of PHEs [15, 16]. Researchers have conducted numerous experimental and numerical studies [17-23] on how the chevron angle affects heat transfer behavior. Different chevron angles have different Reynolds numbers and friction factors. However, in the process industries only a small number of fixed chevron angle have been applied, as shown in Figure 4. In this
work, the three most commonly-used channels are used. The H type channel, where the chevron angle of which is 60°, has the best heat transfer behavior and the largest resistance to flow because of high turbulence intensities and large velocities [21]. The L type channel has a chevron angle of 30°. The small chevron angle leads to lower heat transfer coefficient and pressure drop. M type channels, where the hydraulic resistance and heat transfer performance are between the other two, combines the L and H type channels. Thus, the optimization process should account for the trade-off between heat transfer coefficient and hydraulic resistance.

The flow arrangement with the best heat transfer behavior needs to be selected depending on the required design criteria. Although piping and maintenance expenses are relatively low for a single-pass flow arrangement, higher heat transfer coefficients are achieved in multi-pass flow arrangements, since fluids stay for a longer time [24].

One of the most dramatic advantages of PHEs is their flexibility to satisfy the required process conditions by choosing different plate types, plate geometries and flow arrangement [25]. However, these numerous choices increase the design complexity and the difficulty of searching for the optimal design arrangement. Determination of flow arrangement (pass arrangement for cold and hot streams), plate type and geometry selection, and design constraint considerations (for example pressure drop) are the three key points for the optimum design of PHEs. Most design optimization methods of PHEs are industrially owned, and few are in the open literature.

To develop the thermal design of PHEs, the logarithmic mean temperature difference (LMTD) and ε-NTU methodologies are the most commonly used approaches [6]. Cooper [26] and Shah [27] used a trial and error method to test different geometries and find the best design solution by employing ε-NTU and LMTD approaches. However,
these methodologies take a large amount of time and do not include flow arrangement selection. Wang and Sunder [28] presented an optimization design method for PHEs, which adjusted all the possible plate patterns on the plate surface to maximize the pressure drop utilization. Again, the optimization process failed to consider different flow arrangements might obtain a better result for the design with certain constraints. A screening method was proposed by Gut and Pinto [29]. The objective of this method was to screen out inferior results to overcome the limitations of the mixed nonlinear programming (MINLP) problem that was formulated. The objective of this approach was to minimize the total area with consideration of number of passes, feed location, and total number of plates as variables. But plate pattern selection was not considered. Najafi and Najafi [30] developed an optimization design method of PHEs with multiple objectives by minimizing hydraulic resistance and maximizing total area of PHE simultaneously. This presented challenges to obtain a globally optimal design solution and needs a considerable computation time. Picon-Nunez [31] applied an optimal plate heat exchanger model to heat recovery. The $\epsilon$-NTU method was employed to select flow arrangements and evaluate the temperature correction factor. Most of the design methods for a single PHE fail to consider all the factors that affect heat transfer behavior.

Multi-pass arrangements enhance the heat transfer performance because of relatively high flow velocities in channels. But, unsymmetrical passes of hot and cold streams may diminish effective temperature differences. Initially, most of the studies [32-34] used the LMTD correction factor to solve this problem. A closed-form formula for two-fluid heat recovery was proposed by Pignotti and Shah [35] for the analysis of complex flow arrangements. This method was further improved by Pignotti and Tamborenea [36] by introducing the computer-aided method for calculating the thermal effectiveness of arbitrary flow arrangements by employing a matrix algorithm. The formulas for up to four passes for different flow arrangements relating to the thermal design of PHEs were proposed by Kandlikar and Shah [37]. The details of traditional thermal design methods, including these formulas, are available in plate heat exchanger design handbooks [5].

With the development of computation technologies, Tovazshnyansky et al. [38] proposed a computational method to address different heat transfer arrangements based on blocks of algebraic equations. Arsenyeva et al. [39] further improved this approach. However, for practical application, the geometry of plates cannot be an
arbitrary value and plates should be selected from available commercial plates, as assumed. A method of design a multi-pass PHE was proposed by Arsenyeva et al. [3], which includes the selection of plate type by using the $\epsilon$-NTU method in the thermal design. The plate geometry data collected from manufacturers can be used to build a mathematical model to evaluate heat transfer behavior. However, this method requires a large amount of computation time and is better to be applied as a rating problem. Traditionally, since the entering flow rates, stream temperature data are given in a sizing problem, the LMTD method is possibly a better option compared to the $\epsilon$-NTU method to simplify the design process [40]. The $\epsilon$-NTU method might be a favorable option to solve the rating problem, where details of geometries and the size of heat exchangers are fully specified [2, 5, 6].

To decrease computation effort, the LMTD approach is applied in the thermal-hydraulic design process, which integrated multi-pass flow arrangement, flow geometry, plate type and chevron angle into automated optimization work. Currently, most of the design methods for PHEs are focused on GPHEs, and only a very few design approaches of WPHE available. This is because the thermal design of WPHEs is more complex than that of GPHEs [41]. To overcome the shortcomings of previous work, this paper presents a systematic automated methodology for the design and optimization of multi-pass plate heat exchangers, including both GPHEs and WPHEs. The objective is to automatically derive the optimal solution (assumed to be minimum heat transfer area) from various flow arrangements and available sets of commercial plates within the required duty and pressure drop allowance. Two case studies are presented to apply the new design approach for GPHEs and WPHEs.

2. Methodology

The objective of this methodology is to optimize a plate heat exchanger by identifying a global solution from different types of flow arrangement, various plate types, and chevron angle for a specified process requirement. The main assumption is that there is no maldistribution between channels and no phase transition. Another important assumption is that fluid properties are constant.

2.1 Detailed design methodology of PHEs

The step by step methodology of PHEs design is detailed as follows, including how to select the number of passes for the hot and cold streams, plate geometries, and develop the model for thermal-hydraulic performance.

2.1.1 Flow arrangement selection
The flow arrangements is between gasket plate heat exchangers and welded plate heat exchangers must be distinguished due to their specialized structures. In general, the types of different overall flow arrangement for gasket plate heat exchangers include counter-current flow and co-current flow. For a single-pass flow arrangement, pure counter-current flow has higher thermal efficiency compared with the co-current flow. For a multi-pass flow arrangement, although increasing the number of passes enhances the heat transfer coefficient, it increases the complexity of the thermal design. For welded plate heat exchangers, cross-flow and counter-current flow arrangements are achieved locally and globally. Compablocs are one of the most commonly used WPHEs, and the heat transfer efficiency of Compabloc can be as high as five times that of conventional shell-and-tube heat exchangers. Based on information from the manufacturer of Compabloc (Alfa Laval), the LMTD correction factor for Compabloc is close to 1 (0.9-0.98). The number of passes in each stream is optimised by installing baffles into the plate pack. The details of the baffle assembly and flow arrangement are shown in Figure 5 and Figure 6.

For a specified pressure drop, the required heat load can be achieved by modifying the flow pass arrangement. The flexible number of plates gives more freedom to the flow arrangement to reach the process requirement [42]. Multi-pass arrangements dramatically increase the heat transfer coefficient, since a large number of passes allows for a greater contact area between the fluids. The pressure drop is increased by increasing the number of passes of streams. Thus, the proposed methodology aims to select the best flow arrangement with the highest heat transfer coefficient within the allowed pressure drop.
As mentioned above, the thermal design of PHEs is more complex when multi-pass flow arrangements are considered. An approach to this problem will now be developed. To obtain a minimum heat transfer area, the optimization process is not straightforward. The key point for calculating overall heat transfer coefficient is to separate multi-pass the PHE into parallel blocks, which are assumed to be single-pass PHEs. Pure co-current or counter-current flow arrangement is achieved in each block, and the traditional thermal design method can be further applied to each block.

Consider an Xh-Xc multi-pass plate heat exchanger, in which Xh and Xc represent the number of passes for the hot side and cold side respectively, the total number of blocks N is calculated as Xh·Xc. Figure 7 illustrates that an example of a multi-pass PHE separated into six single-pass blocks with three passes on the hot side and two passes on the cold side. Among the six parallel blocks, it can be seen from Figure 7 that Block 1,2,4 are counter-current flow, and Block 3,5,6 are co-current flow. The logarithmic mean temperature difference (LMTD) method is then applied to each single PHE to obtain thermal-hydraulic performance of the entire heat exchanger. Setting up linear expressions of inlet and outlet temperatures of process fluids among blocks is one of the most important steps in the thermal design.

For practical applications, the number of passes for each stream cannot exceed four. In combination, the total number of different types of flow arrangement is 16. To reduce the number of integer variables and further decrease computation time, each of these 16 different flow arrangements is treated as an independent optimization process. To compare different flow arrangements, enumeration technology is applied to select the best flow arrangement with the minimum total area of PHE.

### 2.1.2 Plate pattern selection

As the most efficient type of plate [43], the chevron plate is employed for the further optimization process. The structure of a chevron plate is shown in Figure 8. The plate
effective length $L_p$, plate width $W$, port diameter $d_{port}$, chevron angle $\beta$ are the most important parameters in a chevron-plate design that influence energy efficiency.

![Fig. 8. Structure of chevron plate](image)

Although key geometries of the plate can be set as continuous variables, when designing a chevron plate, the number of available sets of plates is limited in practical application. Only the available plate geometries from industrial manufactures are considered for the further optimization process. The proposed method will automatically choose the best-standardized plate size with the highest heat transfer coefficient from Alfa Laval options. Table 1 summarises the most widely used plate types for PHEs with detailed geometries, including plate width, plate length, equivalent diameter, and other important plate parameters. Since the plate geometries are integers, an MINLP model is formulated to select optimal plate type. This increases the difficulty of searching for the global optimum plate pattern and the optimization workload.

**Table 1.** The detailed geometries of five different plate types [3]

<table>
<thead>
<tr>
<th>Plate type</th>
<th>$\delta$, mm</th>
<th>$d_e$, mm</th>
<th>$W$, mm</th>
<th>$A_b$, $m^2$</th>
<th>$d_{port}$, mm</th>
<th>$f_{ch} \times 10^3$, mm</th>
<th>$L_p$, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>M3</td>
<td>2.4</td>
<td>4.8</td>
<td>100</td>
<td>0.03</td>
<td>36</td>
<td>0.240</td>
<td>320</td>
</tr>
<tr>
<td>M6</td>
<td>2.0</td>
<td>4.0</td>
<td>216</td>
<td>0.15</td>
<td>50</td>
<td>0.432</td>
<td>694</td>
</tr>
<tr>
<td>M6M</td>
<td>3.0</td>
<td>6.0</td>
<td>210</td>
<td>0.14</td>
<td>50</td>
<td>0.630</td>
<td>666</td>
</tr>
<tr>
<td>M10B</td>
<td>2.5</td>
<td>5.0</td>
<td>334</td>
<td>0.24</td>
<td>100</td>
<td>0.835</td>
<td>719</td>
</tr>
<tr>
<td>M15B</td>
<td>2.5</td>
<td>5.0</td>
<td>449</td>
<td>0.62</td>
<td>150</td>
<td>1.123</td>
<td>1381</td>
</tr>
</tbody>
</table>

* $\delta$ is the gap between plates, $d_e$ is the equivalent diameter, $W$ is the width of plate, $A_b$ is the area of single plate, $f_{ch}$ is the cross-section area, $L_p$ is the effective length.

As mentioned above, heat transfer behavior is sensitive to chevron angles. Thus, the detailed design methodology includes three types of channels with different chevron angle and directly links them with the Nusselt number, as shown in empirical Equation 1 [3].

$$
Nu = m \cdot Re^s \cdot Pr^{0.4} \cdot \left( \frac{\mu}{\mu_w} \right)^{0.14}
$$

(1)
The parameters \( m \) and \( s \) change with different plate type and channel type. Table 2 lists the values of the two parameters under different situations, which are from Alfa Laval. \( Pr \) is the Prandtl number, \( \mu \) is viscosity the bulk fluid and \( \mu_w \) is the viscosity at the wall.

Counting chevron angles and plate types presents 15 different combinations in total. To derive an optimal solution from those combinations automatically, we introduce binary variable \( \text{sec}(PT, CA) \) to select plate type and chevron angle, where \( PT \) and \( CA \) are the abbreviations of plate type and chevron angle. To select the optimal combination with maximum heat transfer coefficient and minimum surface area of PHE, the following equation is obtained:

\[
\sum \text{sec}(PT, CA) = 1
\]  

(2)

In this equation, the value was 1 if selected and 0 otherwise.

<table>
<thead>
<tr>
<th>Plate type</th>
<th>M3</th>
<th>M6</th>
<th>M6M</th>
<th>M10B</th>
<th>M15B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel type</td>
<td>H</td>
<td>L</td>
<td>M</td>
<td>H</td>
<td>L</td>
</tr>
<tr>
<td>( m )</td>
<td>0.265</td>
<td>0.12</td>
<td>0.18</td>
<td>0.25</td>
<td>0.12</td>
</tr>
<tr>
<td>( s )</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
</tr>
</tbody>
</table>

For the welded plate heat exchanger, the plate pattern selection is slightly different. According to Alfa Laval, the chevron angle used in Compobloc is a 45° angle with a 5 mm pressing depth. Thus, only the M type channel can be used in WPHE design.

2.1.3 Thermal-hydraulic model

As mentioned previously, this proposed approach applies the LMTD method in term of the thermal design. The general equations to calculate surface area \( A \) of PHEs is [5]:

\[
A = \frac{Q}{U \Delta T_{LM}}
\]  

(3)

For multi-pass PHEs, \( A \) is also related to the number of plates in each block, which can be expressed as:

\[
A = nA_b
\]

Where \( Q \) is the total heat transfer load, \( U \) is the overall heat transfer coefficient, \( \Delta T_{LM} \) is logarithmic mean temperature difference, \( n \) is number of blocks.

Considering the fouling issues, \( U \) is expressed as:

\[
U = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2} + \frac{1}{\frac{\rho_w}{\lambda_w} R_f}}
\]  

(4)
In which \( h_i \) is the heat transfer coefficient of stream \( i \); \( \delta_w \) is the thickness of the wall; \( \lambda_w \) is thermal conductivity of the wall material; \( R_f \) is the total fouling resistances.

The empirical correlation in terms of Nusselt number is stated in Eq. (1), where the Nusselt number is defined as:

\[
Nu = \frac{h \cdot d_e}{\lambda} \tag{5}
\]

\( \lambda \) represents the thermal conductivity of the fluid, \( d_e \) represents the equivalent diameter of the inter-plate channel, where the channel is defined as the space between the two plates.

\[
d_e = \frac{4t \cdot \delta}{2(t + \delta)} \approx 2\delta \tag{6}
\]

In which \( \delta \) is the gap between plates, \( t \) is the width of the channel.

The Reynolds number is calculated as:

\[
Re = \frac{v \cdot d_e \cdot \rho}{\mu} \tag{7}
\]

Where \( v \) is the velocity of the fluid, \( \rho \) is the density of fluid:

\[
v = \frac{x}{a \cdot \rho} \tag{8}
\]

Where \( a \) is the cross-section area between channels; \( x \) is the flowrate of the process fluid.

The Prandtl number is:

\[
Pr = CP \cdot \mu / \lambda \tag{9}
\]

Where \( c \) is the specific heat capacity of the fluid.

For counter-current flow, Kotjabasakis and Linhoff [44] described how to simulate a heat exchanger for certain process data, as shown in the following equations.

\[
Q_H = CP_H(T_{H1} - T_{H2}) \tag{10}
\]

\[
Q_C = CP_C(T_{C2} - T_{C1}) \tag{11}
\]

\[
Q_H = Q_C = UA \Delta T_{LM} = UA \frac{(T_{H1} - T_{C2}) - (T_{H2} - T_{C1})}{\ln(T_{H1} - T_{C2})} \tag{12}
\]

Combining Eq.(10) – Eq.(12), the correlations between temperatures of fluids can be derived and stated in Eq.(13) and Eq.(14). The derivation is detailed in the literature [45].

\[
T_{H2} = \frac{(R-1)T_{H1} + R(x-1)T_{C1}}{(RX-1)} \tag{13}
\]
\[ T_{C2} = \frac{(x-1)T_{H1} + (1-Rx)T_{C1}}{(Rx-1)} \] (14)

\[ R = \frac{C_{Pc}}{C_{Ph}} \] (15)

\[ x = \exp \left[ \frac{[UA(R-1)]}{C_{Pc}} \right] \] (16)

Where \( C_{Ph} \) is the heat capacity flowrate for the hot stream, \( C_{Pc} \) is the heat capacity flow rate of the cold stream, \( T_{H1} \) and \( T_{H2} \) are the inlet and outlet temperatures of the hot stream, \( T_{C1} \) and \( T_{C2} \) are the inlet and outlet temperatures of cold stream respectively. \( R \) represents the ratio of flow heat capacity of cold and hot streams. Eq. (17) and Eq. (18) list correlations among inlet and outlet temperatures of the hot and cold side. These equations are derived by following the same procedure as mentioned before.

\[ T_{H2} = \frac{R+x}{Rx+x} T_{H1} + \frac{Rx-R}{Rx+x} T_{C1} \] (17)

\[ T_{C2} = \frac{x-1}{xR+1} T_{H1} + \frac{Rx+1}{Rx+x} T_{C1} \] (18)

\[ x = \exp \left[ \frac{[UA(R-1)]}{C_{Pc}} \right] \] (19)

By combining Equation (1) - (19), the heat transfer area \( A \) for single-pass PHEs co-current or counter-current flow can be derived for different plate geometries.

For multi-pass PHEs, the key point of the design is to find the relationships among temperatures in the adjacent blocks by applying the LMTD method. In each single block, by employing the above equations the temperatures of streams in any block is linked with the temperatures of streams in its neighboring blocks. In the case that several streams with different temperatures enters the same block, the temperature can be regarded as the average temperature of these streams. Therefore, a systematic linear relation of temperatures in the different blocks can be set up. Since the entire PHE is separated into \( N \) blocks, the total area of PHE \( A_{total} \) is equal to \( N \cdot A \).

Pressure drop performance is significantly affected by the plate geometries and the number of passes of each stream. Thus, the hydraulic and thermal performance of plate heat exchangers needs to be included in the optimal design simultaneously. To achieve better heat transfer performance, the pressure drop is preferred to be maximized within the allowance.

The pressure drop is expressed as:

\[ \Delta P_{\text{friction}} = \frac{4f L_p}{2d_h \rho A_b^5} \] (20)
In which \( f \) represents the Fanning friction factor, and the expression for \( f \) is shown in the Eq. (21) [46], \( L_p \) is plate length for pressure drop. The chevron angle and intermediate variable \( f_1, f_0 \) affect the Fanning friction factor. The intermediate variable is determined by the Reynolds number.

\[
\frac{1}{f} = \frac{\cos \beta}{\sqrt{0.045 \tan \beta + 0.09 \sin \beta + f_0 / \cos \beta}} + \frac{1-\cos \beta}{\sqrt{3.8f_1}} 
\]  

(21)

When \( \text{Re} \gg 2000 \),

\[
f_0 = \frac{16}{\text{Re}} \quad f_1 = \frac{149}{\text{Re}} + 0.9625
\]

When \( \text{Re} < 2000 \),

\[
f_0 = (1.56 \ln \text{Re} - 3)^2 \quad f_1 = \frac{9.75}{\text{Re}^{0.289}}
\]

To account for the different expressions of \( f_0 \) and \( f_1 \), a binary variable is introduced to define the different range of Reynolds number.

The other part of the pressure drop comes from the height change, which is defined as:

\[
\Delta P_{\text{height}} = \rho g H 
\]  

(22)

Where \( \rho \) is the density of the fluid, \( g \) is the acceleration due to gravity, \( H \) is the equivalent height of fluid passing through. However, since the height of a PHE is comparably small, \( \Delta P_{\text{friction}} \) can be regarded as the total pressure drop of PHEs.

Because it has been assumed that there is no maldistribution of flow in different channels, the pressure drop in the channels is equal. Thus, the total pressure drop is expressed as:

\[
\Delta P = N \Delta P_{\text{pass}} 
\]  

(23)

### 2.2 Optimization model of plate heat exchangers

For the optimization process, the optimal solution should not only meet the process requirements but also consider the economics [47]. As shown in Eq. (24) [48], there is a direct relation between capital cost and total area of PHE. The objective of this proposed method is to minimize the heat transfer area in a multi-pass plate heat exchanger.

\[
\text{Total cost} = 2.5 * 2070 * A_{\text{total}}^{0.85} 
\]  

(24)

To obtain the optimal solution for a single multi-pass PHE, a mixed-integer non-linear programming (MINLP) problem is formulated and solved with the ANTIGONE solver.
in GAMS. ANTIGONE is one of the most widely used global solvers for MINLP optimization [49]. To minimize the total area of a PHE, the general problem is shown in Eq.(25). Among all the variables, the integer variables include the total number of plates in each block, the number of passes for the cold and hot side, plate geometries and chevron angle. There are two constraints. One constraint is that the total heat load should be not less than the required heat load, since the area of the plate is an integer. The other one is that pressure drop should be maintained in the allowable range.

\[ \text{Minimize: } A_{\text{PHE}} = f(X_h, X_c, \beta, n, \delta, d_e, A_B, L_p) \]

Subject to \[ Q \geq Q^0 \]

\[ \Delta P \leq \Delta P_{\text{max}} \] (25)

Where \( A_{\text{PHE}} \) is the total area of the PHE, \( \beta \) is the chevron angle, \( n \) is the number of plates in each block, \( \delta \) is the gap between plates, \( d_e \) is the equivalent diameter, \( A_B \) is the area of each single plate, \( Q^0 \) is process heat transfer requirement, and \( \Delta P_{\text{max}} \) is the maximum pressure drop allowance.

The overall algorithm of the new optimization method for the design of single multipass plate heat exchangers is shown in Figure 9. After extracting the specific operating conditions and physical properties from the required design process, the first step is to select a certain flow arrangement \( k \). Also, the local minimum heat exchanger area can

---

**Fig. 3.** Overall design algorithm of plate heat exchangers
be derived from automatically selecting the best configurations from all available plate geometries for a specific pressure drop allowance. This process is repeated for other different types of flow arrangement and the global optimum result with minimum heat transfer area of the PHE is obtained by comparing all local optimum results.

3. Case study

3.1 Case study 1: optimal design of gasket plate heat exchanger

3.1.1 Verification of the new design methodology

A case from Arsenyeva [3] is used to verify and test the accuracy of the new optimization-based design method for gasket plate heat exchangers. The stream properties data are summarised in Table 3. The cold side distillery wash fluid needs to be heated from 28°C to 90.5 °C by hot water with the inlet temperature of 95°C. The $\Delta P_{\text{max}}$ for both sides is 1.0 bar.

<table>
<thead>
<tr>
<th>Table 3. The stream properties of cold side and hot side</th>
<th>Cold side</th>
<th>Hot side</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>5</td>
<td>15</td>
<td>m³/h</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>28</td>
<td>95</td>
<td>°C</td>
</tr>
<tr>
<td>Target temperature</td>
<td>89</td>
<td>79.36</td>
<td>°C</td>
</tr>
<tr>
<td>Density</td>
<td>978.4</td>
<td>960</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Viscosity</td>
<td>15.03</td>
<td>0.297</td>
<td>cP</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.66</td>
<td>0.68</td>
<td>W/m K</td>
</tr>
<tr>
<td>Heat capacity</td>
<td>3.18</td>
<td>4.21</td>
<td>kJ/kg K</td>
</tr>
</tbody>
</table>

The key point of verification is to fix plate patterns as M6M and keep the numbers of chevron angle the same as the literature [3]. The comparison of results between the basic design method [3] and new proposed method with fixed plate pattern are given in Table 4. When applying the proposed design methodology, the results imply the minimum heat transfer area of PHE is 5.04 m² with 38 plates in M channel and the number of passes for the hot stream and cold streams is 2 and 4 respectively. It is concluded that the results are consistent with the literature [3] for a fixed plate pattern. Having validated the new proposed design model, it will further applied in the optimization process of a single multi-pass PHE design.
Table 4. The comparison of fixed pattern design between literature [3] and proposed method

<table>
<thead>
<tr>
<th>Flow arrangement</th>
<th>1-1</th>
<th>2-1</th>
<th>3-1</th>
<th>4-1</th>
<th>1-2</th>
<th>2-2</th>
<th>3-2</th>
<th>4-2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate type</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
<td>M6M</td>
</tr>
<tr>
<td>Basic design:</td>
<td>7.56</td>
<td>32.34</td>
<td>21.7</td>
<td>35.48</td>
<td>9.8</td>
<td>6.58</td>
<td>8.26</td>
<td>8.68</td>
</tr>
<tr>
<td>Total area (m²)</td>
<td>7.56</td>
<td>32.34</td>
<td>21.7</td>
<td>35.48</td>
<td>9.8</td>
<td>6.58</td>
<td>8.26</td>
<td>8.68</td>
</tr>
<tr>
<td>New design:</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total area (m²)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3.1.2 The computer-aided optimization process of a multi-pass GPHE design

The same case study as Section 3.1.1 is used to apply the new optimization design method of a single multi-pass GPHE. The optimization process is executed in GAMS with the ANTIGONE solver. The computation time is only 1.5 minutes for the optimization process for a particular flow arrangement. However, it takes more than 10 minutes to derive the solution in the published literature [3].

Table 5. The optimization results of a multi-pass GPHE design

<table>
<thead>
<tr>
<th>Flow arrangement</th>
<th>1-1</th>
<th>2-1</th>
<th>3-1</th>
<th>4-1</th>
<th>1-2</th>
<th>2-2</th>
<th>3-2</th>
<th>4-2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate type</td>
<td>M6</td>
<td>M10B</td>
<td>M6</td>
<td>M15B</td>
<td>M10B</td>
<td>M6</td>
<td>M10B</td>
<td>M6M</td>
</tr>
<tr>
<td>Number of plates</td>
<td>60</td>
<td>136</td>
<td>135</td>
<td>36</td>
<td>34</td>
<td>44</td>
<td>30</td>
<td>64</td>
</tr>
<tr>
<td>Total area (m²)</td>
<td>8.76</td>
<td>32.64</td>
<td>21.00</td>
<td>22.32</td>
<td>8.09</td>
<td>6.60</td>
<td>7.20</td>
<td>8.96</td>
</tr>
<tr>
<td>Flow arrangement</td>
<td></td>
<td></td>
<td>1-3</td>
<td>2-3</td>
<td>3-3</td>
<td>4-3</td>
<td>1-4</td>
<td>2-4</td>
</tr>
<tr>
<td>Plate type</td>
<td>M6M</td>
<td>M6</td>
<td>M6M</td>
<td>M10B</td>
<td>M10B</td>
<td>M6</td>
<td>M6</td>
<td>M10B</td>
</tr>
<tr>
<td>Number of plates</td>
<td>39</td>
<td>30</td>
<td>39</td>
<td>24</td>
<td>84</td>
<td>32</td>
<td>36</td>
<td>32</td>
</tr>
<tr>
<td>Total area (m²)</td>
<td>5.46</td>
<td>4.50</td>
<td>5.46</td>
<td>5.76</td>
<td>19.6</td>
<td>4.80</td>
<td>5.4</td>
<td>7.68</td>
</tr>
</tbody>
</table>

Table 5 shows the optimization results for design of a GPHE under different flow arrangements. The results imply that the flow arrangement 2-3 gives the minimum heat transfer area of 4.5 m² within the required heat load and pressure drop allowance. The
optimal solution is achieved by using 30 M6 type of plates. By comparison, Arsenyeva et al. [3] obtained the minimum area of 5.04 m² with 38 plates to satisfy the same process requirement. The heat transfer area can be reduced by 10.71% by the proposed method. The reason why there is a difference between the proposed method and literature is due to the round-up process to find the minimum number of plates in each block. Differences between the heat transfer equations, between the LMTD and $\epsilon$-NTU methods for thermal design for a multi-pass gasket PHE and the effect of locations of the inlet fluids on process behavior are other possible reasons that differences in the differences. Therefore, the optimization solutions obtained within the required process constraints illustrate the capability of the proposed method to give a feasible global result with higher energy efficiency. The proposed method significantly decreases the computation time from several hours to 20 minutes for the entire optimization work of plate heat exchanger design.

3.2 Case study 2: optimization design of welded plate heat exchanger

This case study from the literature [50] was studied to provide insights into the application of the proposed method into a welded plate heat exchanger design. A pump around stream and crude oil from the hot end of the pre-heat train are the two process streams in this case. The physical properties data of crude side and hot side are listed in Table 6.

<table>
<thead>
<tr>
<th></th>
<th>Crude side</th>
<th>Hot side</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>36.4</td>
<td>64.5</td>
<td>kg/s</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>172</td>
<td>306</td>
<td>°C</td>
</tr>
<tr>
<td>Target temperature</td>
<td>260</td>
<td>260</td>
<td>°C</td>
</tr>
<tr>
<td>Density</td>
<td>700</td>
<td>733</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Viscosity</td>
<td>0.44</td>
<td>0.12</td>
<td>cP</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.1</td>
<td>0.12</td>
<td>W/m K</td>
</tr>
<tr>
<td>Heat capacity</td>
<td>2610</td>
<td>2820</td>
<td>J/kg K</td>
</tr>
</tbody>
</table>

The optimized area of a multi-pass welded plate heat exchanger under a different flow arrangement with optimal plate type are listed in Table 7. The optimization results show that the minimum area is 25.92m², which is achieved by a X1=2 and X2=3 flow arrangement. The total number of plates is 108, and the corresponding plate type is M10B with the spacing of plates $\delta = 2.5$mm.
The second-best solution is achieved at $X_1=3$, $X_2=2$ with M6 type plates. The spacing of plates is $\delta = 2\text{mm}$, which is smaller than the spacing of M10B type of plate. For practical considerations, the optimal solution depends on the size of the area, piping arrangement, the volume of the instalment. Thus, design engineers can choose among the best solutions based on the different requirements.

**Table 7.** The optimization results of a multi-pass welded plate heat exchanger design

<table>
<thead>
<tr>
<th>Flow arrangement</th>
<th>1-1</th>
<th>2-1</th>
<th>3-1</th>
<th>4-1</th>
<th>1-2</th>
<th>2-2</th>
<th>3-2</th>
<th>4-2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate type</td>
<td>M15B</td>
<td>M10B</td>
<td>M15B</td>
<td>M15B</td>
<td>M6</td>
<td>M10B</td>
<td>M6</td>
<td>M6M</td>
</tr>
<tr>
<td>Number of plates</td>
<td>58</td>
<td>542</td>
<td>174</td>
<td>188</td>
<td>378</td>
<td>136</td>
<td>210</td>
<td>288</td>
</tr>
<tr>
<td>Total area (m$^2$)</td>
<td>35.96</td>
<td>130.08</td>
<td>107.88</td>
<td>116.56</td>
<td>56.7</td>
<td>32.64</td>
<td>31.5</td>
<td>40.32</td>
</tr>
<tr>
<td>Flow arrangement</td>
<td>1-3</td>
<td>2-3</td>
<td>3-3</td>
<td>4-3</td>
<td>1-4</td>
<td>2-4</td>
<td>3-4</td>
<td>4-4</td>
</tr>
<tr>
<td>Plate type</td>
<td>M10B</td>
<td>M10B</td>
<td>M6M</td>
<td>M6</td>
<td>M10B</td>
<td>M6M</td>
<td>M15B</td>
<td>M10B</td>
</tr>
<tr>
<td>Number of plates</td>
<td>258</td>
<td>108</td>
<td>575</td>
<td>480</td>
<td>200</td>
<td>256</td>
<td>72</td>
<td>208</td>
</tr>
<tr>
<td>Total area (m$^2$)</td>
<td>61.92</td>
<td>25.92</td>
<td>80.5</td>
<td>72</td>
<td>48</td>
<td>35.84</td>
<td>44.64</td>
<td>49.92</td>
</tr>
</tbody>
</table>

The flow rate of process fluids are two important indicators that can test whether the design is robust or not [45]. In the current design methodology, the flow rates of the cold side and hot side are fixed. To investigate the influence of flow rate on model output, sensitivity analysis is an effective method [10]. Thus, a sensitivity analysis is carried out by adjusting the flow rates for both hot side and cold side. In the meantime, keeping the flow arrangement as $X_1=1$ and $X_2=1$ reduces the complexity of the model and saves computation time. The results of a sensitivity analysis by varying the flowrate of the hot and cold side in the range of $\pm 50\%$ are listed in Table 8 and Table 9. The results show that when the flow rate of the hot stream decreases from 51.6 kg/s to 45.15 kg/s, the plate type changes from M15B to M6. According to the analysis results, the parameter has a significant impact on the result. Thus, it can be concluded that the optimal design is robust to $\pm 20\%$ flow of the hot stream but much more robust for the flow of the cold stream.
Table 8. Sensitivity analysis of the flow rate of hot stream

<table>
<thead>
<tr>
<th>Percentage</th>
<th>-50%</th>
<th>-40%</th>
<th>-30%</th>
<th>-20%</th>
<th>-10%</th>
<th>0</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>32.25</td>
<td>38.7</td>
<td>45.15</td>
<td>51.6</td>
<td>58.05</td>
<td>64.5</td>
<td>70.95</td>
<td>77.4</td>
<td>83.85</td>
<td>90.3</td>
<td>96.75</td>
</tr>
<tr>
<td>Plate type</td>
<td>M6</td>
<td>M6</td>
<td>M6</td>
<td>M15</td>
<td>M15B</td>
<td>M15B</td>
<td>M15B</td>
<td>M15B</td>
<td>M15B</td>
<td>M15B</td>
<td>M15B</td>
</tr>
<tr>
<td>Number of plates</td>
<td>90</td>
<td>115</td>
<td>159</td>
<td>203</td>
<td>54</td>
<td>58</td>
<td>63</td>
<td>112</td>
<td>250</td>
<td>581</td>
<td>1800</td>
</tr>
<tr>
<td>Total area (m²)</td>
<td>13.5</td>
<td>17.25</td>
<td>22.5</td>
<td>30.45</td>
<td>33.48</td>
<td>35.96</td>
<td>39.06</td>
<td>69.44</td>
<td>155</td>
<td>360.22</td>
<td>1116</td>
</tr>
</tbody>
</table>

Table 9. Sensitivity analysis of the flow rate of cold stream

<table>
<thead>
<tr>
<th>Percentage</th>
<th>-50%</th>
<th>-40%</th>
<th>-30%</th>
<th>-20%</th>
<th>-10%</th>
<th>0</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>50%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>18.2</td>
<td>21.84</td>
<td>25.48</td>
<td>29.12</td>
<td>32.76</td>
<td>36.4</td>
<td>40.04</td>
<td>43.68</td>
<td>47.32</td>
<td>50.96</td>
<td>54.6</td>
</tr>
<tr>
<td>Number of plates</td>
<td>90</td>
<td>115</td>
<td>159</td>
<td>203</td>
<td>54</td>
<td>58</td>
<td>63</td>
<td>112</td>
<td>250</td>
<td>581</td>
<td>1800</td>
</tr>
<tr>
<td>Total area (m²)</td>
<td>13.5</td>
<td>17.25</td>
<td>22.5</td>
<td>30.45</td>
<td>33.48</td>
<td>35.96</td>
<td>39.06</td>
<td>69.44</td>
<td>155</td>
<td>360.2</td>
<td>1116</td>
</tr>
</tbody>
</table>

4. Conclusions

A new optimization methodology is proposed to design multi-pass plate heat exchangers. The main features that determine plate heat exchanger thermal and hydraulic performance are considered simultaneously by employing discontinuous expression of Nusselt number and friction factor. The basic plate geometries and the number of plates are set as integer variables for minimization of total heat transfer area. The flow arrangement selection is integrated to the thermal-hydraulic model by using the enumeration method, which simplified the optimization workload and saved computation time. Thus, an MINLP model has been created in GAMS using the ANTIGONE solver by considering standardised plate size and the unique heat transfer and pressure drop performance for different plate patterns.

Two case studies from the literature show the success of applying the proposed design methodology to the gasket and welded plate heat exchangers. Minimum heat exchanger area can be achieved for a given heat load. The case studies demonstrate that heat transfer area can be reduced by 10.71% compared with previously published solutions.
Nomenclature

Abbreviation

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>PHE</td>
<td>plate heat exchanger</td>
</tr>
<tr>
<td>GPHE</td>
<td>gasket plate heat exchanger</td>
</tr>
<tr>
<td>WPHE</td>
<td>welded plate heat exchanger</td>
</tr>
<tr>
<td>LMTD</td>
<td>logarithmic mean temperature difference</td>
</tr>
<tr>
<td>NTU</td>
<td>number of transfer units</td>
</tr>
<tr>
<td>MINLP</td>
<td>mixed integer nonlinear programming</td>
</tr>
<tr>
<td>Re</td>
<td>Reynold number</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
</tbody>
</table>

Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_p )</td>
<td>plate length</td>
</tr>
<tr>
<td>( W )</td>
<td>plate width</td>
</tr>
<tr>
<td>( d_{port} )</td>
<td>port diameter</td>
</tr>
<tr>
<td>( d_e )</td>
<td>equivalent diameter</td>
</tr>
<tr>
<td>( A_b )</td>
<td>plate area</td>
</tr>
<tr>
<td>( f_{ch} )</td>
<td>cross-section area</td>
</tr>
<tr>
<td>( \mu )</td>
<td>dynamic viscosity</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>heat conductivity</td>
</tr>
<tr>
<td>( A )</td>
<td>heat transfer area</td>
</tr>
<tr>
<td>( N )</td>
<td>total number of blocks</td>
</tr>
<tr>
<td>( X )</td>
<td>number of passes for stream</td>
</tr>
<tr>
<td>( h )</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>( Q )</td>
<td>heat load</td>
</tr>
<tr>
<td>( \Delta T_{LM} )</td>
<td>logarithmic mean temperature difference</td>
</tr>
<tr>
<td>( \rho )</td>
<td>stream density</td>
</tr>
<tr>
<td>( v )</td>
<td>stream velocity in each channel</td>
</tr>
<tr>
<td>( t )</td>
<td>width of channel</td>
</tr>
<tr>
<td>( R_f )</td>
<td>fouling resistance of streams</td>
</tr>
<tr>
<td>( g )</td>
<td>flow rate of the stream</td>
</tr>
<tr>
<td>( a )</td>
<td>cross-section area between channels</td>
</tr>
<tr>
<td>( T )</td>
<td>temperature</td>
</tr>
<tr>
<td>( CP )</td>
<td>heat capacity</td>
</tr>
<tr>
<td>( R )</td>
<td>the ratio of flow heat capacities of streams</td>
</tr>
</tbody>
</table>
Chapter 3

Publication 1

$A_{total}$ total heat transfer area
\( f \) friction factor
\( \Delta P_{\text{height}} \) pressure drop of height change
\( \Delta P_{\text{friction}} \) pressure drop due to friction
\( H \) height
\( U \) heat transfer coefficient

**Greek symbol**
\( \delta \) inter-plate gap
\( \beta \) chevron angle
\( A_{\text{total}} \) total area of heat exchanger
\( Q^{0} \) required heat load

**Subscript**
\( b \) block
\( h \) hot stream
\( c \) cold stream
\( w \) wall
\( 1 \) inlet
\( 2 \) outlet
\( \text{max} \) maximum

5. Reference


Chapter 4

Publication 2: Application of plate heat exchangers in heat exchanger network retrofit without structure modification

(Xu, K., Akpomiemie, M. O., Smith, R., Application of plate heat exchangers into heat exchanger network retrofit without structure modification. Energy, submitted)
Chapter 4

Application of plate heat exchangers in heat exchanger network retrofit without structural modifications

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Abstract

Heat exchanger networks (HENs) retrofit have received a great deal of attention both in academic and industrial fields to achieve greater energy savings. Heat transfer enhancement techniques have been used in HEN retrofit through improving thermal performance of shell and tube heat exchangers. Compared with conventional methods such as topology modifications and additional surface area, heat transfer enhancement can significantly reduce the retrofit cost. However, one of the film coefficients must be controlling in order to use enhancement techniques economically. Also, heat recovery is limited with enhancement since it is restricted by the geometry of the existing heat exchangers. Plate heat exchangers are one of the most efficient types of heat transfer equipment. Applying plate heat exchanger into the retrofit of HENs is considered as an alternative option to increase heat transfer coefficient and energy saving in the absence of structural modifications. Due to the high installation cost of plate heat exchangers, a systematic optimization it is important to balance the trade-off between energy saving and economic cost. This paper proposes a novel approach of applying plate heat exchanger in HENs retrofit with a fixed network structure. The objective is to maximize the retrofit profit while meeting the heat transfer requirement and keeping the existing heat transfer area of heat exchangers. A case study highlights the benefits of the new approach.

Keywords: Plate heat exchanger, heat exchanger network, optimization, energy consumption

Highlights:
- Allowing small $\Delta T_{\text{min}}$ in the heat exchanger network retrofit
A cost-effective method is proposed for network retrofit using plate heat exchangers

Energy consumption is reduced by applying plate heat exchangers

1. Introduction

With the growth of energy consumption and the increase of CO₂ emissions, it is now more important than ever to improve heat transfer efficiency. HEN retrofit plays an important role of improving energy efficiency in the process industries since it is a relative cost-effective compared with a new design. There are two main methods to achieve heat recovery in a retrofit design, involving adding heat transfer area and topology (structural) modifications. Implementing these conventional methods in practice generally leads to high capital cost retrofit resulting from the amount of pipework and maintenance time [1].

Heat transfer enhancement can be applied in HENs retrofit as a cost-effective technology since it can directly increase the heat transfer rate of shell and tube heat exchangers to avoid complex topology modifications and civil engineering work. Shell-side enhancement and tube-side enhancement are the two main heat transfer enhancement techniques. However, for heat transfer enhancement, the enhancement level of heat transfer coefficient is restricted by the geometry of the existing heat exchanger. Therefore, increasing heat recovery is further limited due to the constraint of the exchanger geometry. Besides, in order to use enhancement technique efficiently, typically as least 50% of overall heat transfer resistance of either tube-side film coefficient or shell-side heat transfer coefficient must be from one of the film resistances, which means that applying heat transfer enhancement requires the controlling side of the exchanger to be addressed.

Rather than use heat transfer enhancement, other types of high-efficiency heat exchangers can be considered for HEN retrofit. Plate heat exchangers, including gasket plate heat exchangers and welded plate heat exchangers, are amongst the most high-efficiency heat transfer devices. The hot and cold streams flow through alternate channels between plates and transfer heat through the metal plates. Compared with conventional shell and tube heat exchangers, plate heat exchangers are able to dramatically improve energy efficiency with minimum approach temperature as low as 2 °C, leading to reduced utility consumption and CO₂ emissions. Owing to the relatively small hydraulic diameter, the turbulence flow enhances the local convective heat
transfer coefficient. For a given duty, the surface area of a plate heat exchanger is significantly smaller than the area of a conventional shell and tube heat exchanger.

Based on these features, this work aims to show the insight of integrating plate heat exchangers and shell and tube heat exchangers into HENs retrofit and improving the energy performance of an existing heat exchanger network.

Established techniques and methodologies for the retrofit of HENs focus on modifying the existing HENs to obtain energy savings and overcome network bottlenecks. There are three main retrofit methods for existing HENs, including Pinch analysis, mathematical programming and hybrid methods. There are several reports in the literature dedicated to the retrofit of HENs based on these methods. Sreepathi and Rangaiah[2] presented detailed review of existing methods for HEN retrofit. Smith [3] introduces the fundamentals of energy targets, capital and total cost targets, and network design of HENs. Pinch Analysis for retrofit makes use of a targeting stage for estimating the maximum energy recovery of a network, and re-design stage to disconnect and reconnect the cross-pinch exchangers to obey the pinch decomposition. Tjoe and Linnhoff [4] first introduced a retrofit method based on Pinch Analysis. The objective of the work was to eliminate heat transfer across the pinch. The cost of the retrofit design of existing HEN has been further considered by Carlsson et al. [5]. Gadalla [6] presented a graphical method, in which the potential modifications were identified to achieve maximum heat recovery. However, Pinch Analysis requires an expert user for its application in retrofit and fails to identify the number of heat exchangers to modify and the appropriate placement for the additional heat transfer area requirement. Also, Pinch Analysis requires too many changes in a single step, which makes it fundamentally not suited to retrofit. Instead of accepting the features that already exist, it tries to convert the existing network into an ideal grass root design in a single step.

To achieve completely automatic optimization, the retrofit design of HEN is formulated to a mathematical optimization model. To identify the most cost-effective design from many possible retrofit solutions embedded in a superstructure, Grossman [7] formulated a mixed integer non-linear programming (MINLP) model for the retrofit design of an existing HEN. Ciric and Floudas [8] presented a superstructure approach for retrofit of HENs. First, the paring of all possible matches and heat exchangers were evaluated and decisions regarding selecting matches, reassigning heat exchangers, adding new heat exchangers and piping work streams made. Based on the results obtained from the first
stage, the optimization of the superstructure design was developed containing all structural features of an existing HEN to remove all unnecessary features and minimize the cost. How to obtain a local optimum solution is the main challenge to solve the MINLP problem. Many methods have been proposed to decompose the retrofit problem. A constant approach temperature model was presented by Ma et al. [9] to linearize the area calculations. Yee and Grossman [10] developed a two-stage retrofit method. An MINLP model was developed in the pre-screening stage, and a NLP model was built in the optimization stage. Stochastic optimization, which is proposed by Dolan [11], was another method to decompose the retrofit problem. One of the significant advantages of mathematical programming is that both of the total cost and energy saving for the retrofit can be considered in the optimization process at the same time [12]. Compared to the Pinch Analysis method, mathematical programming allows the retrofit design of HENs to be automated and to consider more variables. However, the mathematical programming method is very sensitive to the initial points and requires relatively long computation times.

A hybrid method for the retrofit design of HEN was proposed by Asante and Zhu [13], which is referred to as the Network Pinch Approach. The method was separated into stages: diagnosis stage and optimization stage. In the first stage, they assumed a minimum approach temperature and identified the possible structural modification based on an MILP model with maximum heat recovery. The optimization stage makes use of an NLP model to optimize the capital-energy trade-off of the structural modifications determined in the first stage. The sequential approach enables the automation of the design procedure, while maintaining user interaction. The Network Pinch Approach was modified by Smith et al. [14] with consideration of structural modifications and capital-energy optimization in a single step. To consider some practical features, Bakhtiari and Bedard [15] further improved the approach by using a modified Network Pinch approach to increase the possibility of identifying cost-effective design solutions. Structural modifications are the only possible method to overcome the Network Pinch in order to achieve greater energy saving. Resequencing heat exchangers, stream splitting, and adding new heat exchangers are the three main methods used for improving energy saving. The main limitation of Network Pinch Approach is that only one change at a time can be made in the network with the automated sequence. It is difficult to identify the best modification series in the retrofit design of an existing HEN. So, it is possible that the retrofit selection option in early stages prevents obtaining the optimal solution in the final stage.
In recent years, there has been an increased amount of work applying heat transfer enhancement techniques in HEN retrofit. Adding inserts and fins in the existing heat exchangers, which can increase the heat transfer coefficient and effective geometrical area, are the two main techniques in heat transfer enhancement. Heat transfer enhancement increases heat recovery and can also reduce energy consumption. Additionally, compared with conventional retrofit strategies it is a relatively low cost retrofit option without modifying topologies. Pan et al. [16] proposed an optimization method which focused on the systematic implementation of heat transfer enhancement in retrofit without topology modifications. The exact value of log mean temperature difference, correlation factor $F_T$ and multiple passes are considered in the optimization procedure. However, this method is restricted to small-scale problems. A novel MILP based iterative method was proposed by Pan et al. [17] to retrofit HEN with the help of heat transfer enhancement. Although this method overcame the drawbacks of existing design methods, it failed to provide an insight into how to find the best heat exchangers to enhance. In addition, the uncertainty caused by simplification was not considered. To solve these problems, a heuristic-based method has been proposed by Wang et al. [18]. This work identified the amount of energy saving and positions of the candidate heat exchanger to enhance based on sensitivity analysis. However, this work failed to guarantee the feasibility of the required heat transfer enhancement. Jiang et al. [19] extended the approach to consider accurate modelling of the chosen enhancement technique to ensure an accurate representation of proposed energy savings.

Akpomiemie and Smith [1] developed a new model to apply heat transfer enhancement through extending the both methodologies to consider the downstream effects on the network. Nevertheless, to use heat transfer enhancement effectively, one of the film coefficients must be controlling. Furthermore, although heat transfer enhancement techniques can be performed during the normal shutdown time, the potential risk of damaging the heat exchangers when adding inserts or fins is not negligible. Therefore, to break the bottleneck of existing exchanger geometries, this research proposes a novel retrofit approach by applying plate heat exchangers into HENs to increase the energy efficiency and heat recovery. Both physical insights and algorithms are presented in this work.

The aim of this work is to develop a cost-effective retrofit method for HENs to maximize retrofit profit by strategically replacing conventional heat exchangers with plate heat exchangers and allowing different global and local approach temperature in
the network with a fixed structure. This work proposes the concept of integrating two different types of heat exchangers into HEN retrofit with different minimum approach temperatures. The proposed method specifies how to identify the most sensitive heat exchanger to be replaced and how to optimize the cost of plate heat exchangers and rebalance the network after replacement in order to tackle the downstream effect and meet the required heat transfer. A case study is used to illustrate the proposed methods and highlight their potential retrofit benefits by making a comparison between various retrofit options.

2. Application of plate heat exchangers in HEN retrofit with a fixed structure

In this section, the methodology for the application of plate heat exchanger for HEN retrofit is demonstrated in detail. The maximum heat recovery of a HEN depends on the minimum approach temperature. Once the minimum approach temperature is fixed, the minimum utility consumption is fixed correspondingly according to pinch analysis. The minimum approach temperature for conventional shell and tube heat exchangers is generally 10°C or 20°C, depending in the process technology. For plate heat exchangers this can be decreased to 5°C. Integrating plate heat exchangers into the HEN retrofit strategy of traditional HENs, requires two different minimum approach temperatures for the different equipment, increases the potential heat recovery and energy saving. Figures 1 and 2 present an example of the composite curves of a HEN at different approach temperature (20 °C and 5 °C) to explain how the minimum approach temperature relates to energy recovery. It can be clearly seen from the graph that both the hot utility and cold utility consumption significant decrease at 5°C compared to say 20°C. The heat exchangers near the pinch point are identified as the candidate exchangers to be replaced due to their relatively low minimum approach temperature. However, this figures only shows some guides about potential heat recovery and the approximate region to choose the candidate heat exchanger with the most appropriate benefit in terms of energy saving. The following part will introduce the step-by-step procedures of integrating a plate heat exchanger network with conventional shell and tube heat exchangers into retrofit design, which involves different approach temperature.

This work proposes a method to deal with two different minimum approach temperature caused by different types of heat exchangers. The minimum approach temperature of conventional shell and tube heat exchangers should be higher than the existing minimum approach temperature in the network. The minimum approach temperature of
plate heat exchanger should be higher than 5°C. Thus, two constraints are introduced as the Equation (1) and (2).

\[
\Delta T_i \geq Existing \Delta T_{min} \quad i \in S&THX \\
\Delta T_j \geq 5°C \quad j \in PHE
\]

![Figure 1. Composite curves at 20°C](image1)

![Figure 2. Composite curves at 5°C](image2)

For the retrofit of HENs using plate heat exchangers, the steps include, identifying the best candidate heat exchanger require to replace, optimization of the new plate heat exchanger and HEN simultaneously and setting up a non-linear model to rebalance network. The details of these steps are presented in the following sections.

2.1 Identification of candidate heat exchangers to replace

Candidate heat exchangers are first identified by their location on utility paths. Then sensitivity analysis is carried out to determine the rank of identified exchangers.

2.1.1 Identification of utility paths [1]

A utility path is a connection between two utilities through process heat exchangers. Based on an existing HEN, the exchangers to be replaced must be on a utility path so that heat loads can be shifted along this utility path, which can lead to a cost-effective retrofit without structural modifications and additional surface area [20]. This work
proposes the use of plate heat exchangers in place of shell and tube heat exchangers in existing HENs. In addition, utility paths are used to rebalance the HENs after making modifications since the changes to the HEN impacts the performance of heat exchangers around the network.

For a simple network, inspection is used to identify the utility paths. However, for a complex HEN, an Incidence Matrix Approach is carried out to identify utility paths. The rows in the Incidence Matrix represent all streams in the network and the columns are exchangers. S represents the number of process and utility streams and E represents the number of heat exchanger in the existing HEN. The Incidence Matrix Aij describes how the streams are incident on exchangers.

\[
A_{ij} = \begin{cases} 
+1 & \text{if heat exchanger } j \text{ removes heat from stream } i \\
-1 & \text{if heat exchanger } j \text{ supplies heat to stream } i \\
0 & \text{if heat exchanger } j \text{ is not incident on stream } i 
\end{cases}
\]

A column vector P is added to the incidence matrix. Values of vector P are all zero except the entries for hot and cold utility, which are +1 and -1 respectively. The key point of the Incidence Matrix method is that all hot utilities need to be integrated into one stream and cold utilities also follow the same role.

An example from Li and Chang [21] is shown in Figure 3 to provide an insight into how to generate the Incidence Matrix and how to identify the utility paths by this method. According to Figure 3, cold utilities C1 and C2 need to be regarded as one stream, which means different cold utilities used be considered as one stream. Thus, there are 7 streams and 7 heat exchangers in total.

![Figure 3. Example HEN](image)

Following this method, the initial incidence matrix is formulated, and the values are listed in Table 1.
Table 1. Initial incidence matrix

<table>
<thead>
<tr>
<th>S/E</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>C1</th>
<th>C2</th>
<th>H</th>
<th>P</th>
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<tbody>
<tr>
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<td>+1</td>
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</tr>
<tr>
<td>H2</td>
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<td>0</td>
<td>+1</td>
<td>0</td>
<td>+1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>H3</td>
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<td>0</td>
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<td>0</td>
</tr>
<tr>
<td>C1</td>
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<td>-1</td>
<td>0</td>
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<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>C2</td>
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<td>0</td>
<td>-1</td>
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<td>0</td>
<td>-1</td>
<td>0</td>
</tr>
<tr>
<td>HU</td>
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<td>0</td>
<td>0</td>
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<td>+1</td>
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<tr>
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<td>0</td>
<td>0</td>
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<td>0</td>
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</tr>
</tbody>
</table>

If the sum values of each column are all equal to zero, then the matrix is linearly dependent. As it is clearly shown in Table 1, the initial incidence matrix is linearly dependent. Then, the initial incidence matrix is reduced by making row operations in order to generate an independent matrix. This is to ensure that all entries below the first non-zero entry of each row in the matrix are zero. In Table 1, the first non-zero entry of the first row is Exchanger 3. In Column 3, the first non-zero entry value below row 1 is in Row 4. Thus, to guarantee this value to be zero, row 1 is added to Row 4. This procedure is repeated for all rows until all entries below the first non-zero entry of each row are zero. And the obtained independent matrix and corresponding values are listed in Table 2.

Table 2. Independent incidence matrix

<table>
<thead>
<tr>
<th>S/E</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>C1</th>
<th>C2</th>
<th>H</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>0</td>
<td>0</td>
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<td>0</td>
<td>+1</td>
<td>0</td>
<td>0</td>
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</tr>
<tr>
<td>H2</td>
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<td>+1</td>
<td>0</td>
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<tr>
<td>H3</td>
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<td>0</td>
<td>0</td>
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<td>C1</td>
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<td>0</td>
<td>+1</td>
<td>0</td>
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<tr>
<td>C2</td>
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<td>+1</td>
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<tr>
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<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

The column vector P links two utilities with process stream exchangers. The column vector P is equated to a linear combination of columns in the incidence matrix after the linearly independent matrix is formulated. The linear combination obeys the enthalpy
balance, which implies the amount of heat load added to a heater must be subtracted from an exchanger in the path, added to the next exchanger in the path, and so on and finally added to a cooler in the path. For the existing HEN in Figure 3, the two utility paths are shown in Figure 4 and 5.

<table>
<thead>
<tr>
<th>S/E</th>
<th>H</th>
<th>4</th>
<th>C2</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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<td>0</td>
<td>0</td>
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</tr>
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<tr>
<td>CU</td>
<td>0</td>
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</tr>
</tbody>
</table>

Figure 4. Utility path: H-4-C2

<table>
<thead>
<tr>
<th>S/E</th>
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<th>2</th>
<th>3</th>
<th>C1</th>
<th>P</th>
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<tbody>
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<td>+1</td>
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</tr>
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<td>0</td>
<td>0</td>
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</tr>
<tr>
<td>3</td>
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<td>-1</td>
<td>0</td>
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</tr>
<tr>
<td>4</td>
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<tr>
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<td>0</td>
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<td>0</td>
</tr>
<tr>
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<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>+1</td>
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<td>CU</td>
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<td>0</td>
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<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 5. Utility path: H-1-2-3-C1

2.1.2. Sensitivity analysis

In this section, sensitive analysis is applied in order to determine the most appropriate heat exchangers that require to be replaced by plate heat exchangers so that the energy saving among others can be led. It also makes the selection of the sequence of replacement easier. The only information required in the sensitivity analysis is the base case stream data and the HEN structure. Kotjabasakis and Linnhoff [22] were the first to propose sensitivity analysis. This method is based on the equation:

\[ Q = UA\Delta T_{LM}F_T \]  

In which Q is the total heat load, W; U is the overall heat transfer coefficient, W/(K·m²); A is the area of the heat exchanger, m²; \( \Delta T_{LM} \) is the logarithmic mean temperature difference, K; \( F_T \) is the correction factor. The aim of sensitivity analysis is to increase the inlet temperature of the key utility, which is identified as the one that has the largest energy consumption, by adjusting the heat transfer coefficient, surface area.
and correction factor of candidate heat exchangers. Wang [23] developed a mathematical model to identify the high sensitivity heat exchangers, which is identified as the one that can bring the greatest increase in the inlet temperature of the key utility exchanger. This study concluded that the amount of energy saving largely depended on the gap between the heat exchanger and the key utility. Easy implementation and short computation time are the main advantages of sensitivity analysis.

2.2 Application of plate heat exchangers

The best exchangers identified are replaced with plate heat exchangers. The plate heat exchanger design model used in this paper is from Xu et al. [24], which considers effects of multi-pass flow arrangement, plate geometries and chevron angle on heat transfer behaviour. Then, the exchanger sizing model is formulated to obtain the maximum heat transfer coefficient, implemented by General Algebraic Modelling System (GAMS). The input process stream data are derived from candidate heat exchangers.

To achieve the same amount of heat recovery, the larger heat transfer coefficient, the smaller heat exchanger area and capital cost will be. Since there is a negative correlation between plate heat exchanger heat transfer coefficient and capital cost, the minimal capital cost can be further obtained. The generalized algorithm of optimization of the heat transfer coefficient is shown in the Figure 6.
Figure. 6. Optimization algorithm of design a single plate heat exchanger

2.3 Optimization of HEN

Application of plate heat exchangers to replace the conventional heat exchangers in the existing HEN can affect the performance of the other heat exchangers. The heat transfer coefficient of the new plate heat exchanger is significantly higher, which leads an increase of the inlet temperature of the next heat exchanger in the hot stream and a decrease of the inlet temperature of the next heat exchanger in cold stream respectively. These changes can further affect the other components. To rebalance the HEN, the heat loads of heat exchangers need to be relocated and shifted through the utility paths. Thus, an optimization method to retrofit an existing HEN without topology modification is proposed. Furthermore, the capital cost of plate heat exchanger is still an important component for total network cost calculation, even if the application of plate heat exchangers can lead to significant energy saving.

Therefore, there is a trade-off between energy saving and total retrofit cost. In this optimization model, the objective function is to maximize retrofit profit, which is the difference between the energy savings and total cost of the retrofit. The total cost of retrofit consists of the implementation cost of the bypass, the capital cost of plate heat exchangers, the cost of additional heat transfer area of heat exchangers which are located on the utility path. The profit from energy saving can be obtained from the total amount of utility saving. The two main process constraints are achieving the target temperatures of the process streams and maintaining the existing heat exchanger areas (excluding the candidate heat exchangers to be replaced).

Objective Function: \( \text{Maximize Retrofit Profit} = \text{Profit from energy saving} - \text{Total cost of retrofit} \)

Constraints: \( A_{\text{ex}} = A_E \quad \forall \text{exchangers (excluded the candidate heat exchanges)} \)

\( TT_s = TT_E \quad \forall \text{streams} \)

Variables: heat load for all exchangers on a utility path

heat transfer coefficient for candidate heat exchanger

3. Case study

The base case is a simplified crude oil preheat train. The new HEN retrofit model is used to optimize the performance of the base case through applying plate heat exchangers in a fixed network structure. Figure 6 illustrates the existing HEN for the
There are in total 12 exchangers, 5 hot streams and 1 cold stream. The objective of this study is to maximize the retrofit profit by applying plate heat exchangers into the existing network without adding area and structure change. Table 3 summarises the cost parameters used in this study. The streams properties are assumed to be constant. To quantify the retrofit profit, the operating time is fixed at one year.

**Table 3. Cost data for case study**

<table>
<thead>
<tr>
<th>Utility cost data</th>
<th>Retrofit cost data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Utility Cost: 400 ($/kW y)</td>
<td>Cost of installing of plate heat exchangers: $2.5 \times 2070 \times A^{0.85}$</td>
</tr>
<tr>
<td>Cold Utility Cost: 5.5 ($/kW y)</td>
<td>Implementing By-pass: 5000 ($)</td>
</tr>
<tr>
<td></td>
<td>Cost of increasing heat exchanger area: $6000 + 200 \times A$ ($)</td>
</tr>
</tbody>
</table>

The first step is to identify the best heat exchanger for replacement. From Figure 6, only process heat exchanger 7 is not on the utility path. Thus, sensitivity analysis is performed on heat exchanger 1, 2, 3, 4, 5 and 6. From Figure 7, heat exchanger 5 is the most sensitive exchanger as it brings the greatest increase in the inlet temperature of the key utility exchanger. Thus, the heat exchanger 5 is selected to be replaced by a plate heat exchanger. To maximize retrofit profit, the heat duties of heat exchanger 5 and corresponding cold utility exchanger 10 are set as variables and rebalance the heat duty of all the heat exchangers.

The cold utility exchanger is selected as this represents a constraint on the maximum increase in the duty of heat exchanger 5 without violating the target temperature.
requirement. After inputting the process stream data of exchanger 5 into the optimization model of a single plate heat exchanger in GAMS, the maximum heat transfer coefficient can be derived with a value of 3.75 kW/m² °C, which is a significant increase compared with shell and tube heat exchangers i.e. 0.32 kW/m² °C. Then, the global solver is used to optimize the case in which optimization procedure is obtained to rebalance the HEN, implemented on LINDO Systems What’s Best!

The details of the heat exchanger area, heat transfer coefficient and heat load of each heat exchangers before and after the retrofit design of the HEN are listed in Table 4.

<table>
<thead>
<tr>
<th>Exchanger</th>
<th>A (m²)</th>
<th>Areplacement (m²)</th>
<th>U (kW/m² °C)</th>
<th>Ureplacement (kW/m² °C)</th>
<th>Q (kW)</th>
<th>Qreplacement (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>396.72</td>
<td>396.72</td>
<td>0.45</td>
<td>0.45</td>
<td>6141.33</td>
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<td>2</td>
<td>545.45</td>
<td>545.45</td>
<td>0.39</td>
<td>0.39</td>
<td>6134.83</td>
<td>6604.64</td>
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<td>3</td>
<td>633.85</td>
<td>633.85</td>
<td>0.14</td>
<td>0.14</td>
<td>5556.61</td>
<td>5640.23</td>
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<td>0.09</td>
<td>0.09</td>
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<td>2680.76</td>
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<td>5</td>
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<td>6</td>
<td>843.73</td>
<td>843.73</td>
<td>0.06</td>
<td>0.06</td>
<td>2291.88</td>
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<td>1141.81</td>
<td>1149.84</td>
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<td>-</td>
<td>14455.41</td>
<td>13723.76</td>
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</tbody>
</table>

The heat duty of exchanger 5 is increased by 816.94 kW and the utility consumption in exchanger 10 decreases to 0. Although the heat load of heat exchanger 5 increases, the area of exchanger 5 decreases from 183.85 m² to 23.83 m² by applying the plate heat exchanger. 3 by-passes are implemented to reduce the duty of heat exchangers. The results of retrofit cost and retrofit profit are detailed in Table 5 with the initial utility
cost of $5,801,396. This procedure is repeated until the net saving starts to decrease, since the retrofit objective is to maximize retrofit profit. According to the sensitivity analysis, the next heat exchanger to be replaced is heat exchanger 1. The results after replacement of heat exchanger 5 and heat exchanger 1 are listed in Table 5. The net saving after replacing heat exchanger 1 is smaller than only replacing heat exchanger 5, which violates the objective of maximizing the retrofit profit, thus the replacement is stopped after replacing heat exchanger 5. The main reason that limits the net saving is the amount of utility saving, which is restricted to the fixed area of heat exchanger 2 and heat exchanger 3.

Table 5. Retrofit results

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Replace E5</th>
<th>Replace E5&amp;E1 (allowing extra area on E2 &amp; E3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Replacement</td>
<td>$76,630</td>
<td>$272,893</td>
</tr>
<tr>
<td>Increasing Area</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>Implementing By-pass</td>
<td>$15,000</td>
<td>$15,000</td>
</tr>
<tr>
<td>Total Cost</td>
<td>$91,631</td>
<td>$287,893</td>
</tr>
<tr>
<td>Utility Savings</td>
<td>$296,684</td>
<td>$296,604</td>
</tr>
<tr>
<td>Net Saving</td>
<td>$205,054</td>
<td>$8,702</td>
</tr>
</tbody>
</table>

Thus, another trial allowing extra area on heat exchanger 2 and 3 is carried out. It can be clearly seen from Table 5 that the utility saving increases from $296,604 to $494,325. However, the high capital cost of the plate heat exchanger and high cost of increasing area leads to an increase in the payback time from 0.97yr to 1.09yr.

The comparisons between the application of plate heat exchanger and heat transfer enhancement [1] are also highlighted (see Table 6). The retrofit profit of replacement is $11,218 smaller than that obtained from enhancement because of the high installation cost of the plate heat exchangers. Nevertheless, in this case the utility saving in replacement is $59,431 larger than enhancement.

Thus, heat transfer enhancement is a cost-effective retrofit method. However, when one of the film coefficients is controlling, applying plate heat exchanger to replace shell-and-tube heat exchanger can be regarded as an alternative option for retrofit, which can bring more energy saving at the same time.
Table 6. Replacement vs Heat transfer enhancement

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Replacement</th>
<th>Enhancement</th>
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</thead>
<tbody>
<tr>
<td>Retrofit Cost</td>
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</tr>
<tr>
<td>Replacement</td>
<td>$76,630</td>
<td>$3,981</td>
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<tr>
<td>Increasing Area</td>
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<td>$0</td>
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<tr>
<td>Implementing By-pass</td>
<td>$15,000</td>
<td>$15,000</td>
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<tr>
<td>Total Cost</td>
<td>$91,631</td>
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<tr>
<td>Retrofit Profit</td>
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</tr>
<tr>
<td>Utility Savings</td>
<td>$296,684</td>
<td>$237,253</td>
</tr>
<tr>
<td>Net Saving</td>
<td>$205,054</td>
<td>$216,272</td>
</tr>
</tbody>
</table>

4. Conclusions

A new retrofit method for heat exchanger networks using the integration of plate heat exchangers and conventional heat exchangers with different minimum approach temperature has been proposed in this work. The case study highlights the potential benefit of the new retrofit method with fixed structure. Compared to the traditional heat transfer enhancement technologies, replacing the shell and tube heat exchangers with plate heat exchangers can significantly increase the heat recovery and decrease energy consumption with a smaller heat transfer area. However, the installation cost of the plate heat exchanger is relatively high. Thus, plate heat exchangers can be used as an alternative retrofit option when one of the heat transfer films is not controlling. Applications of plate heat exchangers allow small $\Delta T_{\text{min}}$ in the network. A case study has been examined to introduce plate heat exchangers into the network with $\Delta T_{\text{min}}$ at 5 °C. Augmenting the existing shell-and-tube heat exchangers is more cost-effective than the replacement due to the reduced heat load. With fixed structure, the amount of energy saving is limited. The savings are constrained, as all of the existing matches, except the new plate heat exchangers, are being maintained at their original sizes.

Nomenclature

*Abbreviation*

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>HEN</td>
<td>heat exchanger network</td>
</tr>
<tr>
<td>PHE</td>
<td>plate heat exchanger</td>
</tr>
<tr>
<td>MINLP</td>
<td>mixed integer non-linear programming</td>
</tr>
<tr>
<td>NLP</td>
<td>non-linear programming</td>
</tr>
<tr>
<td>MILP</td>
<td>mixed integer linear programming</td>
</tr>
<tr>
<td>GAMS</td>
<td>General Algebraic Modelling System</td>
</tr>
</tbody>
</table>
Symbols

$\Delta T_{min}$  minimum approach time
$F_T$  correlation factor
$A_{ij}$  matrix of incident streams on exchangers
$S$  number of process and utility streams
$E$  number of heat exchanger in the existing HEN
$P$  column factor was added incidence matrix
$C_1$  cold utility 1
$C_2$  cold utility 2
$H_1$  hot stream 1
$H_2$  hot stream 2
$H_3$  hot stream 3
$H_U$  hot utility
$C_U$  cold utility
$Q$  heat load
$U$  overall heat transfer coefficient
$A$  surface area of the heat exchanger, m$^2$
$\Delta T_{LM}$  logarithmic mean temperature difference
$k$  flow arrangement
$R$  the ratio of flow heat capacities of streams
$TT$  target temperature
5. Reference


Chapter 5

Publication 3: New Methodology for Heat Exchanger Network Retrofit with Structural Modifications

New Methodology for Heat Exchanger Network

Retrofit with Structural Modifications

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Abstract

Improving the energy efficiency in heat exchanger networks (HEN) remains a significant industrial problem as obtaining cost effective retrofits is a challenge. Different retrofit methods have been proposed mostly centred around the use of structural modifications such as resequencing, adding new heat exchangers and stream splitting. The issues of identifying the best modification, the location within the network to apply the identified modification, and the cost associated with retrofit remains to be tackled. Using automated procedures to identify structural modifications provides no insight for selecting the best modification suited for a given network. In this work, a new step by step approach based on the network pinch is proposed to identify the best structure change.

Also, this work presents a new approach that seeks to utilize high efficiency heat exchangers such as plate heat exchangers (PHEs) for retrofit. Compared with conventional heat exchangers, PHEs allow a lower minimum approach temperature in heat exchanger networks, which can significantly enhance energy efficiency and decrease capital cost. However, the difference in minimum approach temperature poses an optimization problem for HENs. As such, a distinctive feature of the new proposed method is the ability to consider different minimum approach temperatures for the different types of exchangers used in the network within an optimization framework.

The objective of retrofit HENs in this paper is to maximize the heat recovery at certain minimum temperature difference with the minimum modifications step and retrofit cost. Three cases are studied by following the proposed step-by-step retrofit design method. To quantify the potential benefit of applying PHEs in heat exchanger network retrofit with structure modifications, the retrofit results are compared with the results using
shell-and-tube heat exchangers (STHX) following the same guidelines. This methodology and guidelines can be applied to a wider range of retrofit problems.

**Keywords:** Heat Exchanger Network Retrofit, Structure Modification, Plate Heat Exchanger, Network Pinch

**Highlights:**
- Applying plate heat exchanger into retrofit of heat exchanger network with structure modifications
- Optimization framework with different minimum approach temperature of different types of heat exchangers is proposed
- Step-by-step guidelines based on the pinch method
- Maximum heat recovery is achieved by minimum number of modifications and cost
- Potential benefit is quantified by comparing with adding shell-and-tube heat exchangers into heat exchanger network retrofit

**1. Introduction**

With increasing concerns regarding energy saving and greenhouse gas emissions, the process industries require better methods for the cost-effective retrofit of heat exchanger networks. Retrofit strategies currently mainly concentrate on adding heat transfer area and heat transfer enhancement technologies [1-3]. The heat recovery is limited by applying these methods. Plate heat exchangers (PHEs) are one of the most efficient types of heat transfer equipment [4]. The minimum approach temperature in PHEs is as low as 5 °C, which can dramatically improve heat recovery and energy efficiency. This work will focus on the retrofit of heat exchanger networks (HENs) with network structural modifications, which allow adding new heat exchangers, splitting streams and resequencing. The objective of retrofit design is usually to minimize the utility consumption under a given criterion. Pinch Analysis, optimization and hybrid methods are regarded as the three general conventional methodologies for HEN retrofit.

The concept of Pinch Analysis was first applied to retrofit of heat exchanger networks by Tjoe and Linhoff [5]. To achieve possible maximum heat recovery, the proposed approach identified the cross-pinch heat exchangers in the existing HENs and redesigned the network to remove cross-pinch heat transfer. A systematic method that eliminates cross-pinch heat transfer and re-locates the heat load based on pinch analysis was further proposed by Li and Chang [6]. The limitation of this pinch analysis
approach was that it relies on an experienced user when it comes to large scale networks, since the methodology fails to point out the specific number of steps for modifications and the exact place for application of modifications.

Optimization can be used to convert the retrofit problem into a mathematical programming model and solve the problem for different objectives. The network topology can be modified by introducing a superstructure [7] or matrix representation [8-10]. Based on the complexity of the retrofit problem, the optimization model can be formulated as linear programming (LP), non-linear programming (NLP), and mixed integer linear programming (MILP) or mixed integer non-linear programming (MINLP). Among these, HEN retrofit are generally MINLP problems, and this type of model was first developed by Yee at el. [11]. How to find a global optimum solution and how to simply the model to save the computation time are the two challenging problems with a MINLP model [12, 13]. Although the retrofit process is fully automated by formulating a mathematical model without relying on expert users’ experience, it is difficult to avoid local optimum solutions and prolonged computation workload with results that are sensitive to the assumptions.

The network pinch approach, proposed by Asante and Zhu [14] to retrofit HENs, combines pinch analysis and mathematical optimization methods while keeping their advantages. This approach, which allows a user interaction while ensuring a good retrofit design, involves two stages: diagnosis and optimization. The diagnosis stage analyses potential topology changes to achieve minimum energy consumption subject to a fixed minimum approach temperature through a MILP model. In the second optimization stage, the trade-off between capital cost and energy saving is optimized by developing a NLP model to select the desired modifications. Smith et al. [15] further modified the network pinch approach and combined topology changes and energy-capital trade-off optimization to a single step. However, this work still fails to identify the reason why that the selected retrofit option is better than the others. This may lead to inappropriate decisions. A multi-step methodology was proposed by Akpomiemie et al. [3] to bridge this gap by using the network pinch approach for HEN retrofit. The key significance of this approach was to stress guidelines on how to select the best retrofit topology change among all the options in order to obtain the maximum energy saving.

The objective of the present work is to propose a step-by-step network pinch approach to HEN retrofit, taking into account of using PHEs. The key significance of this work is to integrate the PHEs into the optimization process especially when it comes with
dealing with different minimum approach temperatures for different heat transfer equipment. To quantify the potential benefit to apply PHEs, the economic cost for retrofit is calculated by employing different retrofit techniques. This work also gives guidelines on how to identify the most suitable retrofit changes and the locations to apply these changes under different scenarios for existing HENs in terms of energy saving. In addition to adding new PHEs, resequencing and stream splitting are considered as the other two types of modification options for HENs retrofit. The dramatic feature of this approach is that it introduces the plate heat exchanger into the retrofit design of HENs and deal with two different minimum approach temperature in an existing HENs.

The early identification of favourable topology changes saves retrofit design time and calculation time and at the same time obtains a more robust result. The best retrofit design should maximize heat recovery while minimize the economic cost, including equipment pipework and civil engineering. Three case studies are used to illustrate the effectiveness of the proposed method. To quantify the potential economic and energy benefit by application of PHEs, the payback by using conventional shell-and-tube heat exchangers to achieve the same amount of energy saving is considered as the indicator to compare with those of the proposed method. Pinching match is referred as the approach temperature of a heat exchanger match is at the minimum allowable level after all the potential in existing loops and utility path in the network are exhausted. And network pinch is the point where the pinching match occurs.

2. Background of network pinch

For existing heat exchanger networks, pinching the network identifies the possibilities to reduce energy consumption by utilizing the degrees of freedom for a fixed network structure. The degrees of freedom are the utility paths and loops in the existing HEN. The connection between hot and cold utilities through process exchangers is termed a utility path and the closed path that starts from one exchanger and returns to the same exchanger is termed a loop. Redistributing the heat load in the loop can increase the driving force in the process exchangers, which reduces the utility consumption through the path.

3. Background on cross-pinche heat transfer
Equation 1 shows the correlations between the maximum heat recovery (as retrofit target $Q_{\text{max}}$) and the existing energy consumption ($Q_{\text{exist}}$). The difference between these two is mainly result from the cross-pinch heat transfer ($QP$).

$$Q_{\text{max}} = Q_{\text{exist}} + QP$$

(1)

The five different scenarios that could possibly cause the cross-pinch heat transfer are shown below and in Figure 1.

1. The hot end of a match is used below the pinch, while its cold end is used below the pinch (see Figure 8a) represented as $Q_{\text{PHP}}$.
2. The hot end of a match is used above the pinch, while its cold end is used above the pinch (see Figure 8b) represented as $Q_{\text{PCP}}$.
3. The hot end of a match is used below the pinch, while its cold end is used above the pinch (see Figure 8c) represented as $Q_{\text{HCP}}$.
4. Utility cooling above the pinch (see Figure 8d) represented as $Q_{\text{UCP}}$.
5. Utility heating below the pinch (see Figure 8e) represented as $Q_{\text{UHP}}$.

The total cross-pinch heat transfer is the sum of all five possibilities as shown in Equation 2. From Equation 2, a factor “PF” has been added to all variables. If any of the variables exists, a value of +1 is added, otherwise add zero.

$$QP = Q_{\text{PHP}} \times PF + Q_{\text{PCP}} \times PF + Q_{\text{HCP}} \times PF + Q_{\text{UCP}} \times PF + Q_{\text{UHP}} \times PF$$

(2)

Figure 1. Five different scenarios with cross-pinch heat transfer
For each process-to-process and utility exchanger transferring heat across the pinch, the cross-pincheck heat transfer under each scenario given in Figure 1 can be determined using Equation 3 to Equation 7.

\[ Q_{PHP} = CP_H(T_{PH,in} - T_{H,P}) \]  
\[ Q_{PCP} = CP_C(T_{PC,out} - T_{C,P}) \]  
\[ Q_{HCP} = CP_H(T_{PH,in} - T_{H,P}) - CP_C(T_{PC,out} - T_{C,P}) \]  
\[ Q_{UCP} = CP_H(T_{UH,in} - T_{H,P}) \]  
\[ Q_{UHP} = CP_C(T_{UC,out} - T_{C,P}) \]  

4. Retrofit Methodology

The pinch retrofit method has similar principles as the Network Pinch Approach, which is detailed by Smith et al. [15]. The objective of pinch retrofit method is to identify the best series of structural modifications for energy recovery with an advantage of providing insights such as identifying the features of the existing HEN that restricts energy recovery and the best location to apply the selected modifications to overcome these restrictions.

The best retrofit strategy is one which meets the retrofit target with the minimum number of structural modifications and cost. To achieve this, the modification and technology used should be one that ensures maximum energy recovery.

The methodology proposed in this work covers the identification of the best single modification for a given HEN based on energy recovery, the identification of the best series of modifications i.e. multiple modifications, the use of high efficiency exchangers in place of the conventional shell and tube heat exchanger for increased energy recovery, and the approach used are compared with the different retrofit methods.

4.1 Single modification

Outlined below is the step-by-step approach proposed for identifying the best single modification for a given HEN.

**Step 1:** pinch the network and identify features of the network restricting energy recovery
Pinching the existing HEN relative to the selected $\Delta T_{\text{min}}$ identifies the HEN features responsible for restricting energy recovery. Through pinching the existing network, pinched exchangers and cross-pincho heat exchangers, which are caused by inappropriate use of utilities and process-to-process heat exchangers, are identified. By reviewing the network, the locations of these exchangers are identified through checking the inlet and outlet temperatures of streams. By pinching the network, the existing structure is maintained while reducing the utility consumption.

**Step 2:** review the identified network features and identify the best type of modification based on energy recovery

Once the network is pinched, the best structural modification can be further determined based on the location of the cross pinch and pinched exchangers. To overcome the network pinch and eliminate cross pinch heat transfer, modification options considered are resequencing, stream splitting and adding new heat exchanger(s). Depending on the feature of the existing HEN, the best modification can be identified as explained below.

**Scenario 1:** no pinched exchangers in the network

- In this situation, adding a new heat exchanger to create utility path is beneficial. This is because for this type of network, the main restriction to energy recovery is the presence of cross-pincho exchangers.
- Therefore, by creating a utility path, the cross-pincho heat transfer can be effectively decreased or eliminated, and energy consumption can be significantly reduced.

**Scenario 2:** process heat exchangers located upstream of pinched exchangers

- In this situation, resequencing and adding a new heat exchanger are both possible. This is because energy recovery is restricted by the downstream pinched exchanger(s). Therefore, heat load can be moved from the process heat exchangers upstream to relieve the constraint on the pinched exchangers allowing for energy recovery.
- If moving the entire heat from the upstream heat exchanger does not violate the temperature driving force, resequencing is recommended. When there are more than one upstream process heat exchangers, the one with highest temperature driving force should be selected to relocate. It should be noted that in the event of
more than one exchanger having the same temperature driving force, the one with higher heat duty is selected.

- Adding a new heat exchanger is the most beneficial choice only when moving all load from the upstream heat exchanger across pinch violates the minimum temperature approach constraints. With this option, a new heat exchanger can be added downstream, allowing for some of the heat load to be moved thereby, reducing energy consumption.

**Scenario 3**: No process heat exchangers located upstream of pinched exchangers

- In this scenario, adding a new heat exchanger and stream splitting are considered as the best modifications.
- Only when there are more than two pinched heat exchangers adjacent to each other, is introducing stream splitting beneficial. This is because the lower temperature of cold stream and the higher temperature of hot stream ease the driving force constraint of pinched heat exchangers.
- Otherwise, adding a new heat exchanger is recommended.

### 4.2 Multiple modifications

After the best single modification is applied, further modifications steps need to be identified as shown in Figure 2a. However, Figure 2b shows that by repeating the best single modification, the moat benefit retrofit solution might not be attained. As such, the best retrofit strategy is one which can identify the best series of modifications that ensures the retrofit target is met with minimum modification steps and minimum capital cost.

![Figure 2. Paths for multiple modifications](image)

In this work, the step-by-step algorithm shown in Figure 3 is proposed. The retrofit methodology is repeated, and the network data is updated until the retrofit energy target is met. The best modification at each stage is obtained based on the updated network
data and the key features of the new network. Note that the stopping criteria can be different based on the user requirements. The stopping criterion used in this study is achieving the maximum energy recovery under a certain minimum approach temperature for a given HENs.

**Figure 3.** Step-by-step algorithm for multi-modification

### 4.3 Integration of plate heat exchanger into HEN retrofit

Plate heat exchangers, as one of most-efficient heat transfer equipment, allow for lower minimum approach temperature (say 5 °C) compared with conventional heat
Chapter 5

Publication 3

Exchangers. Lower approach temperatures make it possible to achieve greater heat recovery and energy saving. The main challenge of integrating of plate heat exchanger is how to tackle the problem of allowing different minimum approach temperatures in the same network. This paper presents a methodology to deal with the different ΔT required if different exchanger types are used in the same network. The strategy is to maintain the existing ΔT\text{min} in the network as global minimum approach temperature for all the shell-and-tube heat exchangers, and set the local minimum approach temperature as 5°C for all the new plate heat exchangers as shown in Equation (8) to (10).

\[ TT_i = TT_{out,i} \quad i \in \text{process streams} \]  
\[ ΔT_{glocal,j} = \text{Existing} \ ΔT_{min} \quad j \in S\&TH \]  
\[ ΔT_{local,k} \geq 5°C \quad k \in \text{PHE} \]  

When integrating plate heat exchangers into existing HENs, the capital cost of PHE needs to be taken into the consideration. The cost of the PHE is dependent on the area. As such, to obtain the minimum economic installation cost of PHE, the area of PHE needs to be minimized. The heat transfer area of plate heat exchangers is determined as follows:

\[ A = \frac{Q}{ΔT_{LM'}} = \frac{Q}{\left(\frac{\left(ΔT_{H1} - ΔT_{C2}\right) - \left(ΔT_{H2} - ΔT_{C1}\right)}{\ln\left(\frac{ΔT_{H1} - ΔT_{C2}}{ΔT_{H2} - ΔT_{C2}}\right)}\right)} \]  

Where Q is the total heat load, U is the heat transfer coefficient, A is the heat transfer area of PHE, T_{H1} and T_{H2} are the inlet and outlet temperatures of the hot stream, T_{C1} and T_{C2} are the inlet and outlet temperatures of cold streams. Only the heat transfer coefficient (U) is not determined and is possible to be further optimized. The heat transfer coefficient of chevron plate PHEs is dependent on the plate type, flow arrangement, and chevron angle. Thus, the optimization model can be set up as below.

Minimize: \[ U = f(X_h, X_c, \beta, n, P_{type}) \]

Subject to \[ Q \geq Q^0 \]  
\[ ΔP \leq ΔP_{max} \]

Where \( X_h \) is the number of passes for the hot stream, \( X_c \) is the number of passes for the cold stream, \( \beta \) is the chevron angle, \( n \) is the number of plates in each block, \( P_{type} \) is the type of plate, \( Q^0 \) is process heat transfer requirement, and \( ΔP \) is the pressure drop,
ΔP_{\text{max}}$ is the maximum pressure drop allowance. An mixed-integer non-linear programming (MINLP) model is set up in GAMs, which is detailed by Xu et al.[16].

Under some circumstances, even when using plate heat exchangers, the approach temperature of that heat exchanger could be much greater than the minimum allowed owing to the restrictions created by the network structure.

With the use of PHE, the objective of the optimization model is to achieve the maximum heat recovery at the existing minimum approach temperature with minimum modification steps and capital costs. The stopping criterion is to meet the minimum energy consumption at specified minimum approach temperature. This is because although adding plate heat exchanger potentially can save more energy due to its relative high heat transfer coefficient, to further quantify the economic benefit of applying PHE, the maximum heat recovery at the existing minimum approach temperature is used. Because adding a new heat exchanger into a heat exchanger network might lead to an excess over the maximum heat recovery, a constraint for heat recovery is required. Equation 13 shows the boundary of hot and cold utility in order to obtain the maximum heat recovery, where $Q_H$ is the hot utility consumption, $Q_{H_{\text{min}}}$ is the minimum hot utility consumption at the existing minimum approach temperature, $Q_C$ is the cold utility consumption, $Q_{C_{\text{min}}}$ is the minimum cold utility consumption at the existing minimum approach temperature.

$$Q_H \geq Q_{H_{\text{min}}} \text{ or } Q_C \geq Q_{C_{\text{min}}}$$ (13)

4.4 Compare with different retrofit technologies

To quantify the potential benefit of integrating PHEs into retrofit design of HENs, the retrofit results with application of shell-and-tube heat exchangers to achieve the same amount energy saving are calculated. The utility cost and retrofit cost data are shown in Table 1. The total retrofit cost and utility cost, payback is calculated as shown in Equation (14) to (16).

$$TRC = HX_{\text{new}} + BP + AA$$ (14)

$$UC = HU + CU$$ (15)

$$\varepsilon = \frac{TRC}{UC}$$ (16)

Where TRC represents the total retrofit cost, BP is the cost of implementing by-pass, AA is the cost of adding area,
UC is the total cost of utility saving, HU is the total cost of hot utility saving, CU is the total cost of cold utility saving, ε is the payback of the retrofit strategy.

Table 1. Cost data for case studies

<table>
<thead>
<tr>
<th>Utility cost data</th>
<th>Retrofit cost data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Utility Cost ($/kW y): 50</td>
<td>Cost of Installation of PHE ($) : $2.5*(16,000+210*A^{0.95})</td>
</tr>
<tr>
<td>Cold Utility Cost ($/kW y): 1.2</td>
<td>Cost of Installation of S&amp;T HE ($) : $C_e = 2.5*(28,000+54*A^{1.2})</td>
</tr>
<tr>
<td></td>
<td>Implementing By-pass ($) : 15,000</td>
</tr>
<tr>
<td></td>
<td>Cost of resequencing ($) : 75,000</td>
</tr>
<tr>
<td></td>
<td>Cost of adding heat exchanger area ($) : 6000 + 200 * A</td>
</tr>
</tbody>
</table>

5. Case study

5.1 Case study 1

The first case study was introduced by Tjoe and Linnhoff [5]. Since there are some inconsistencies in the data, it was modified by Li and Chang [6]. The detailed modified data are listed in Table 2 and the HEN structure shown in Figure 4. In general, there are three hot streams and two cold streams. The structure consists of four process heat exchangers, one hot utility exchanger and two cold utility exchangers. The minimum approach temperature of the existing heat exchanger network is 19°C. The required minimum hot and cold utility assuming this $\Delta T_{\text{min}}$ is 12,410 kW and 10,323 kW, while the hot utility and cold utility in the existing HEN is 17,597 kW and 15,510 kW respectively. The objective of retrofit is to reduce the existing utility consumption to the least with the existing $\Delta T_{\text{min}}$ of the HEN and at the same time minimize the modification cost. Two utility paths provide the two degrees of freedom of the network. All analysis has been carried out using SPRINT v.2.9, which is developed by the University of Manchester.

Figure 4. The Original Heat Exchanger Network for case study
Table 2. Stream data of case study 1

<table>
<thead>
<tr>
<th>Stream Name</th>
<th>T_s (°C)</th>
<th>T_T (°C)</th>
<th>Q (kW)</th>
<th>CP (kW/°C)</th>
<th>H.T.C.(kW/m²°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>159</td>
<td>77</td>
<td>18,737.0</td>
<td>228.5</td>
<td>0.40</td>
</tr>
<tr>
<td>H2</td>
<td>267</td>
<td>80</td>
<td>3,841.8</td>
<td>20.4</td>
<td>0.30</td>
</tr>
<tr>
<td>H3</td>
<td>343</td>
<td>90</td>
<td>13,611.4</td>
<td>53.8</td>
<td>0.25</td>
</tr>
<tr>
<td>C1</td>
<td>26</td>
<td>127</td>
<td>9,423.3</td>
<td>93.3</td>
<td>0.15</td>
</tr>
<tr>
<td>C2</td>
<td>118</td>
<td>265</td>
<td>28,826.7</td>
<td>196.1</td>
<td>0.50</td>
</tr>
</tbody>
</table>

Best Single Modification Result

Heat exchangers 1 and 4 are identified as both cross process pinch exchangers and pinched exchangers, as shown in Figure 5. Based on structure analysis, there are two pinched exchangers and they are not adjacent. The best single modification with the largest amount of energy saving and lowest cost is adding a new plate heat exchanger to create a utility path. The pinch temperature for the hot and cold streams are 159°C and 140°C.

![Figure 5. Pinched heat exchanger for case study 1](image-url)

Applying the retrofit methodology, the pinched heat exchangers are located on the stream with hot utility. However, no heat exchanger is located on the stream with cold utility. No heat exchangers are located upstream from the pinched exchangers. Therefore, resequencing and adding a new heat exchanger to create a loop are not recommended. The energy saving is limited by applying stream splitting since there are only two pinched exchangers in the HEN. The best location to add the new heat
exchanger to form the utility path should be the streams with highest utility consumption.

First, Figure 6 shows that the new plate heat exchanger N is added on the upstream of Exchanger 3 on the hot stream as Exchanger 3 transfers heat across the pinch, and upstream of the pinched exchangers on the cold stream to create a path.

![Figure 6. Add new plate heat exchanger for case study 1](image)

To quantify the potential benefit of adding a PHE, a shell-and-tube heat exchanger is added by following the same algorithm to compare with the plate heat exchanger. The result of adding shell-and-tube heat exchanger as a retrofit option is shown in Figure 7.

![Figure 7. Add new shell-and-tube heat exchanger for case study 1](image)

Compared with the results shown in Figure 6, location of adding the new shell-and-tube heat exchanger and the heat load of the exchanger are the same as that for the new PHE as shown in Figure 7. This is mainly due to the restriction of the existing network structure. Economic cost is another factor that needs to be considered. By employing the equations shown in Table 1, the installation cost of adding a PHE is $66,296, which is $133,072 less than the installation cost of adding shell-and-tube heat exchanger. Thus, adding plate heat exchanger is preferred.
An alternative option for structure modification is to apply stream splitting, the result of which is shown in Figure 8. Compared with adding a new plate heat exchanger, the energy saving for stream splitting is 5.9% lower. Accordingly, adding a new plate heat exchanger to create a path is the most beneficial retrofit option for the single modification.

![Figure 8. Apply stream splitting for case study 1](image)

**Best Retrofit Solution (Multiple Modifications)**

By pinching the network after the first modification, heat exchangers 1, 3 and 4 are identified as pinched exchangers, and heat exchangers 2 and 4 are identified as cross pinch exchangers. The retrofit methodology is applied until the retrofit target is met. In terms of the application of the methodology, although there are three pinched heat exchangers in total, they are not adjacent to each other in the network. In this situation, stream splitting is not a reasonable option. Since the pinched heat exchangers are on the streams with the utility that has the highest duty, adding a new plate heat exchanger to create a path is not the most energy effective option as the amount of energy that can be recovered will be restricted by the pinched exchangers on the streams. For resequencing, heat exchanger 2 is located downstream from the pinched heat exchanger 1. Therefore, exchanger 2 cannot be moved to relieve the constraint on exchanger 1.

Figure 9 shows the best retrofit solution when a PHE is used for the new heat exchanger required for retrofit. As shown in Figure 9, heat exchanger N2 is added next to heat exchanger 2 in order to create a loop. The input process stream data of the heat exchanger and the maximum heat transfer coefficient from the single optimized MINLP model for plate heat exchanger N2 is then derived [16]. The detailed optimization model for single PHE is proposed by Xu et al.[16]. After rebalancing the network using a
nonlinear optimization algorithm, the minimum utility consumption is achieved through adding two plate heat exchangers, which allow the $\Delta T_{\text{min}}$ of the PHEs to be as low as 5°C and the $\Delta T_{\text{min}}$ of other heat exchangers in the existing network is maintained (19°C).

Figure 9. Best retrofit solution by applying PHE for Case Study 1

To quantify the economic benefit of applying PHEs in retrofit, Figure 10 shows the best retrofit solution by following the same proposed methodology with conventional shell and tube heat exchanger technology instead of PHE. To achieve same amount of energy saving, conventional technologies need three steps of structure modifications (Add 2 shell and tube heat exchangers and add 1 stream split).

Figure 10. Best retrofit solution by applying S&T HE for Case Study 1

It can be clearly seen from Figures 9 and 10 that the location of adding the two different types of new heat exchanger are the same. This is because same algorithm is applied to select the best modification step and technologies. However, only two structural modifications are required with the use of PHEs compared to three when shell and tube exchangers are used. Table 3 compares the detailed economic costs of two different retrofit methodologies.

As shown in Table 3, some additional area and by-passes are added to exchangers because of constraints in the network structure. The installation cost of plate heat
exchanger is 31% of the installation cost of shell and tube heat exchanger. This is because the higher heat transfer coefficient of plate heat exchanger leads to less area requirement. The payback time of the new retrofit method (with PHE) is 1.88yr. However, to achieve the same energy saving the payback time of the traditional retrofit method (with shell and tube exchanger) is 5.39yr.

Table 3. The comparison of two different methodologies

<table>
<thead>
<tr>
<th>Modification</th>
<th>New retrofit method</th>
<th>Traditional retrofit method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Installation cost of adding new HX ($)</td>
<td>310,130</td>
<td>996,715</td>
</tr>
<tr>
<td>Adding area ($)</td>
<td>124,200 (E1,4,5,6,7)</td>
<td>416,920 (E1,3,4,5,7)</td>
</tr>
<tr>
<td>Cost of stream splitting ($)</td>
<td>75,000</td>
<td>45,000</td>
</tr>
<tr>
<td>Total retrofit cost ($)</td>
<td>509,330</td>
<td>1458,635</td>
</tr>
<tr>
<td>Total utility saving ($)</td>
<td>270,573</td>
<td>270,573</td>
</tr>
<tr>
<td>Payback (yr)</td>
<td>1.88</td>
<td>5.39</td>
</tr>
</tbody>
</table>

5.2 Case Study 2

A simplified crude oil pre-heat train was studied by Akpomiemie and Smith [17] and the detailed heat exchanger network structure is shown in Figure 11. Table 4 lists the stream properties for case study 2. In total there are 5 hot streams and 1 cold stream and 7 process heat exchangers and 5 utility heat exchangers. Six utility passes and three loops give nine degrees of freedom. The minimum approach temperature is 10°C and the corresponding minimum hot and cold utilities consumptions are 10,958 kW and 0 kW respectively. However, in the existing heat exchanger network, the hot and cold
utilities consumptions are 14455 kW and 657 kW. The objective is to apply the proposed retrofit approach to achieve the maximum heat recovery at the existing $\Delta T_{\text{min}}$.

**Table 4.** The process data of Case Study 2

<table>
<thead>
<tr>
<th>Stream Name</th>
<th>$T_s$ (°C)</th>
<th>$T_T$ (°C)</th>
<th>$Q$ (kW)</th>
<th>$CP$ (kW/°C)</th>
<th>H.T.C.(kW/m$^2$°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>310</td>
<td>95</td>
<td>18,490.0</td>
<td>86.0</td>
<td>0.5</td>
</tr>
<tr>
<td>H2</td>
<td>299</td>
<td>120</td>
<td>3,830.6</td>
<td>21.4</td>
<td>0.5</td>
</tr>
<tr>
<td>H3</td>
<td>273</td>
<td>250</td>
<td>4,248.1</td>
<td>184.7</td>
<td>0.5</td>
</tr>
<tr>
<td>H4</td>
<td>230</td>
<td>95</td>
<td>3,172.5</td>
<td>23.5</td>
<td>0.5</td>
</tr>
<tr>
<td>H5</td>
<td>206</td>
<td>178</td>
<td>3,623.2</td>
<td>129.4</td>
<td>0.5</td>
</tr>
<tr>
<td>C1</td>
<td>52</td>
<td>360</td>
<td>4,432.1</td>
<td>143.9</td>
<td>0.5</td>
</tr>
</tbody>
</table>

After pinching the HEN (see Figure 12), the heat duty of cold utilities C1, C3 are decreased to 0kW. Thus, the total number of utility paths decreases to four. There are two utility pinch temperatures. These are 62°C (hot) and 52°C (cold), and 40°C (hot) and 30°C (cold). The pinched exchangers identified are Exchangers 2, 4 and 6. There were no cross process pinch exchangers, but exchangers C2 and C4 were identified as cross utility pinch exchangers.

![Figure 12. The pinched HEN for Case Study 2](image)

**Best Single Modification Result**

Based on the proposed retrofit methodology, the most beneficial options for retrofit will be resequencing or adding a new plate heat exchanger to create a loop. This is as a result
of the presence of heat exchangers 3 and 7 upstream from the pinched exchangers. Next, the network is analysed to determine the feasibility of moving the entire heat load of the heat exchanger with the highest $\Delta T_{LM}$ further downstream. The heat exchanger with the highest $\Delta T_{LM}$ was identified to be Exchanger 7.

Moving Exchanger 7 downstream does not violate the network temperature constraint, as the inlet temperatures of the pinched exchangers on the cold stream C1 decreases, which in return allows for them to take up more heat load. Therefore, the best single modification for this case study was to resequencing Exchanger 7. Exchanger 7 is then moved to the outlet of the pinched exchanger furthest downstream and still on a viable utility path i.e. Exchanger 4. Figure 13 shows the network for the best single modification.

![Network Diagram]

**Figure 13.** The best modification for first step retrofit of HEN

**Best Retrofit Solution (Multiple Modifications)**

After pinching the network again, Exchangers 2, 4 and 6 are identified as pinched heat exchangers and utility exchangers C2 and C4 transfer heat across the pinch. From Figure 13, it can be noted that Exchanger 3 is located upstream from pinched exchangers. As such, both resequencing and adding a new plate heat exchanger to create a loop are the most beneficial options.

Next the feasibility of moving all the heat load of Exchanger 3 is examined. By doing this, the network constraint is violated when Exchanger 3 is moved to the outlet of the pinched exchanger furthest downstream. Therefore, the most beneficial option is to add a new plate heat exchanger to create a loop with exchanger 3.
After adding a new plate heat exchanger N, the maximum heat recovery at the existing $\Delta T_{\text{min}} (10^\circ \text{C})$ is achieved as shown in Figure 14. The maximum heat transfer coefficient of plate heat exchanger N is derived based on the input stream data from the single optimized MINLP model.

Figure 14. The best retrofit solution for Case Study 2

To quantify the potential benefit of applying plate heat exchangers, by following the retrofit methodology the best retrofit solution with conventional heat exchangers is shown in Figure 15. To achieve the same amount of energy saving, the alternative solution can also be achieved by two-step modifications. The detailed results of the two different approaches are listed in Table 5.

Figure 15. The best retrofit solution for Case Study 2 by adding S&THX

These two approaches provide the same number of modifications, but the heat load distribution is different. The heat load of the new plate heat exchanger is 1068kW, higher than the traditional shell and tube heat exchanger, which benefits the downstream
Exchanger 3 allowing it to release heat. The heat transfer of the new plate heat exchanger N is limited due to its presence in the utility path H-1-2-N-3-C1 with the heat load of heat exchanger C1 already reduced to 0. Even with more heat load on the new heat exchanger, the installation cost of adding a plate heat exchanger is $310,130. The total retrofit cost of new proposed methodology is $449,330, which is $1,084,305 lower than the traditional retrofit cost to achieve the same amount of energy saving. The payback of the traditional retrofit cost is 8.56yr, and the payback of the new proposed method is 2.51yr.

**Table 5. The comparison result of two different technologies**

<table>
<thead>
<tr>
<th>Modification</th>
<th>New retrofit method</th>
<th>Traditional retrofit method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capital cost of adding new HX ($)</td>
<td>310,130</td>
<td>996,715</td>
</tr>
<tr>
<td>Adding area ($)</td>
<td>124,200 (E1,4,5,6,7)</td>
<td>416,920 (E1,3,4,5,7)</td>
</tr>
<tr>
<td>Cost of stream splitting ($)</td>
<td>75,000</td>
<td>45,000</td>
</tr>
<tr>
<td>Cost of resequencing</td>
<td>75,000</td>
<td>75,000</td>
</tr>
<tr>
<td>Total retrofit cost ($)</td>
<td>449,330</td>
<td>1,533,635</td>
</tr>
<tr>
<td>Total utility saving ($)</td>
<td>178,996</td>
<td>178,996</td>
</tr>
<tr>
<td>Payback (yr)</td>
<td>2.51</td>
<td>8.56</td>
</tr>
</tbody>
</table>

### 5.3 Case study 3

A more complex case [17] is studied to further illustrate the methodologies of retrofitting the HENs. The structure of the heat exchanger network is shown in Figure 16. In this case, there are 5 hot streams and 5 cold streams. The physical properties of the process streams are listed in Table 6. As shown in Figure 16, there are 5 process heat exchangers and 3 hot utilities and 4 cold utilities. Three utility paths and two loops give the heat exchanger network five degrees of freedom. The minimum approach temperature of the existing heat exchanger network is 12°C, and the hot and cold utility consumptions are 28,168kW and 35,143kW respectively.
By pinching the network (see Figure 17), the minimum hot and cold utility consumptions obtained are 8,300kW and 15,275kW at the existing $\Delta T_{\text{min}}$. Exchangers 3 and 4 were identified as pinched exchangers after pinching the network. The pinch temperatures for hot streams and cold streams are 330°C and 320°C respectively. Exchangers 1, 2 and 3 were identified as cross process pinch exchangers, and Exchanger H2 was identified as utility pinch exchanger.
Figure 17. The pinched HEN for Case Study 3

Best Single Modification Result

To identify the best single modification, the retrofit methodology is applied. By reviewing the pinched network, it is not possible to add a new plate heat exchanger to create a loop or resequence since there are no process heat exchangers upstream from the pinched exchangers. The two pinched exchangers are not adjacent, which makes the implementation of stream splitting an infeasible solution.

Figure 18 shows the best single modification for retrofit of the existing HEN. The new PHE is located on the utility path to connect the utility exchangers with highest duty (C1 and H2). Also, the new exchanger has been added after the pinched exchanger 4 and it does not transfer heat across the pinch as indicated in the guidelines. It has also been added after Exchanger 5 on stream C3 because Exchanger 5 is not on a utility path or loop. This ensures that the network energy balance is maintained. The heat transfer coefficient for the single plate heat exchanger is derived from the single optimized MINLP model in GAMS after input the process stream data. The new plate heat exchanger N allows the minimum approach temperature to be as low as 5°C.
Figure 18. The first-step modification for retrofit HEN by adding new plate heat exchanger

To quantify the potential benefit, the retrofit guidelines are applied with the addition of a shell and tube heat exchanger as shown in Figure 19. The heat transfer is limited due to the restriction of the existing network structure. Under this circumstance, whether it is necessary to install plate heat exchanger needs to be tested (economic cost).

Figure 19. The first-step modification for retrofit HEN by adding new shell-and tube heat exchanger

By employing the equations in Table 1, the installation cost of adding PHE ($33,472) is three times lower than the installation cost of adding shell and tube heat exchanger ($92,142).

Best Retrofit Solution (Multiple Modifications)

The retrofit stopping criteria is to achieve the minimum utility consumption at the existing $\Delta T_{\text{min}}$ of 12°C. Pinching the network again, Exchangers 1, 2 and 3 are
identified as cross process pinch exchangers. Based on the proposed methodology, the best second modification is to add a new plate heat exchanger to create a loop with the cross process pinch exchanger having the highest cross-pincher heat transfer (Exchanger 1). This is because there are no upstream heat exchangers from the pinched exchangers and the pinched exchangers are not adjacent to each other, which make resequencing an infeasible solution. Also, since the pinched exchanger is located on the streams with the most viable utilities (H3 and C4), adding a new plate heat exchanger to create a path is not a feasible solution, as the temperature constraint of that heat exchanger will be violated.

Based on these reasons, adding a new plate heat exchanger to create a loop is the best second modification. As shown in Figure 20, Exchanger N2 is added to connect utility exchangers H3 and C4 since only the cold utility C4 is not connected to a pinched exchanger. The same amount of energy saving, is achieved by adding the second new shell and tube heat exchanger.

![Figure 20. The best second modification for retrofit HEN](image)

After the second modification, the maximum heat recovery is not achieved. Therefore, the retrofit methodology is repeated until this is obtained. The final retrofit solution is shown in Figure 21. To achieve the minimum utilities consumption at the existing $\Delta T_{\text{min}}$, 7 new plate heat exchangers are added in total.
Chapter 5

Figure 21. Best retrofit solution for Case study 3 with application of plate heat exchangers

An alternative option for HEN retrofit with conventional heat exchangers to achieve the same heat recovery is obtained by following the proposed methodology. As shown in Figure 22, the best retrofit result requires 9 modifications and 9 new traditional exchangers are added to create a loop or utility path. The comparisons of two different retrofit options are listed in Table 7.

Figure 22. Best retrofit solution for Case study 3 with application of conventional heat exchangers

The difference between retrofit solutions of adding new plate heat exchangers and adding new shell-and-tube heat exchangers results in differences that are highlighted as follows:

1. In total, 7 new plate heat exchangers are added to achieve the maximum heat recovery at the existing ΔT_{min}. Two more heat exchangers are needed when using shell and tube heat exchangers to obtain the same amount of energy saving. This is because plate heat exchangers allow lower minimum approach temperature. Based on the retrofit result, the ΔT_{min} of new heat exchangers N3, N4 and N6 are 5°C.
2. As it can be clearly seen from Figures 21 and 22, the new heat exchangers N1, N2, N3, N4, N5 are added in the same location but with different heat loads. Since after adding new plate heat exchanger N4, the heat load of cold utility C3 is already reduced to 0. There is no need to add additional new heat exchangers, as shown in Figure 21.

3. The total installation cost of adding new plate heat exchangers is 18% of the total installation cost of adding shell and tube heat exchangers. Adding on the cost of extra area and installation of by-passes, the total retrofit cost of the new retrofit method is 41% of the total retrofit cost of traditional retrofit method.

<table>
<thead>
<tr>
<th>Table 7. The comparison result of two different technologies</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Modification</strong></td>
</tr>
<tr>
<td>Capital cost of adding new HX ($)</td>
</tr>
<tr>
<td>Adding area ($)</td>
</tr>
<tr>
<td>Cost of stream by-pass ($)</td>
</tr>
<tr>
<td>Cost of resequencing</td>
</tr>
<tr>
<td>Total retrofit cost ($)</td>
</tr>
<tr>
<td>Total utility saving ($)</td>
</tr>
<tr>
<td>Payback (yr)</td>
</tr>
</tbody>
</table>

6. Conclusion

A novel step-by-step methodology of HEN retrofit is proposed through integration of plate heat exchanger into conventional heat exchanger networks with structure modifications. The proposed method provides insight into how to identify the best types of modification and location by applying pinch analysis based on the existing network structure. A multi-step retrofit methodology towards the maximum heat recovery at the existing minimum approach temperature is developed.

The best HEN retrofit should not only achieve the energy target in the fewest number of modification steps and lowest retrofit cost. A methodology of dealing with two different minimum approach temperatures within a HEN is presented. Three case studies are used to provide insight into the potential benefit of the proposed step-by-step retrofit methodology. The results show that to achieve the same amount of energy saving, the utilization of plate heat exchangers can significantly reduce total retrofit cost compared with using shell and tube heat exchangers. In some circumstances, the energy saving of applying plate heat exchangers at each stage is limited due to the structural restrictions.
However, the utilization of plate heat exchangers is always beneficial due to its low capital cost. The proposed methodology can be applied into various HEN objectives and stopping criteria.
Nomenclature

Abbreviation

PHE plate heat exchanger
HEN heat exchanger network
LP linear programming
NLP non-linear programming
MILP mixed integer linear programming
MINLP mixed integer nonlinear programming
STHE shell and tube heat exchanger

Symbols

\( Q_{\text{max}} \) maximum heat recovery
\( Q_{\text{exist}} \) existing energy consumption
\( Q_P \) cross-pincher heat transfer
\( \Delta T_{\text{min}} \) minimum temperature approach
\( CP \) heat capacity flowrate
\( Q_H \) hot utility consumption
\( Q_{H_{\text{min}}} \) minimum hot utility consumption
\( Q_C \) cold utility consumption
\( Q_{C_{\text{min}}} \) minimum cold utility consumption
\( TT_{\text{out},i} \) target temperature of process stream \( i \)
\( \Delta T_{\text{local},k} \) local minimum approach temperature for PHE
\( \Delta T_{\text{global},j} \) global minimum approach temperature for STHE
\( HU \) hot utility
\( CU \) cold utility
\( \Delta T_{LM} \) logarithmic mean temperature difference
\( TRC \) total retrofit cost
\( BP \) cost of implementing by-pass
\( AA \) cost of adding area
\( UC \) total cost of utility saving
\( Q \) total heat load
\( X_h \) number of passes for the hot stream
\( X_c \) number of passes for the cold stream
\( n \) the number of plates in each block
\( P_{\text{type}} \) type of plate
\( Q^0 \)  
process heat transfer requirement

\( \Delta P \)  
pressure drop

\( \Delta P_{\text{max}} \)  
maximum pressure drop allowance

\( A \)  
heat transfer area of PHE

\( U \)  
heat transfer coefficient

Greek symbol

\( \varepsilon \)  
payback time

\( \beta \)  
chevron angle

7. Reference


Chapter 6  Conclusions and Future work

6.1 Conclusions

The aim of this research was to develop a new optimization method for a single multi-pass heat exchanger design and integrate plate heat exchanger and shell-and-tube heat exchangers into a heat exchanger network retrofit. The existing methodologies of HEN retrofit and the identification of gaps from the literature review provide an insight to how to achieve this goal. Firstly, an automated systematic approach of the design multi-pass plate heat exchangers, which involve in gasket plate heat exchanger and welded plate heat exchangers, has been proposed. However, lack of reliable design methods to quantify the potential benefit of applying plate heat exchanger limits the application of plate heat exchangers. The small minimum approach temperature of plate heat exchangers increases the energy saving, but the installation cost of plate heat exchangers is relatively high. Therefore, a methodology of application of plate heat exchangers into the retrofit design of conventional heat exchanger network retrofit considering the trade-off between capital cost and energy conservation with fixed structure has been developed. However, the energy saving is limited with the fixed structure. Therefore, this thesis considers the structure modification in the retrofit of heat exchanger networks. A step-by-step retrofit methodology of integrating plate heat exchanger into heat exchanger networks to achieve the maximum heat recovery with the minimum retrofit cost, which allows topology changes, has been proposed. The main contributions of this research that achieve the three objectives of the thesis are detailed as follows.

6.1.1 Automated optimization methodology of plate heat exchangers design

A computer-aided optimization methodology of multi-pass plate heat exchanger design, including gasket and welded plate heat exchangers, has been proposed to obtain the optimum exchanger configuration. A MINLP problem has been formulated considering standardized plate sizes and the unique pressure drop and heat transfer characteristics
for numbers of plate patterns and flow arrangement. The integer variables involved in the basic plate geometries and the number of plates is determined to obtain the minimum heat transfer area of plate heat exchangers. Through employing an enumeration method, the flow arrangement selection is integrated into the thermal-hydraulic model. Compared with published literature, this new proposed methodology gives a better design solution with less computation effort.

6.1.2 Application of plate heat exchanger into retrofit with fixed structure

Heat transfer enhancement has been used to some extent in the retrofit design of HENs as it can significantly decrease the cost of the retrofit based on the literature survey. However, for enhancement to be effective, at least 50% of the overall heat transfer resistance must be either the tube-side film coefficient or the shell-side film coefficient. Owing to the limitations of heat transfer enhancement, other types of high-efficiency heat exchangers, such as plate heat exchangers, have been considered for HEN retrofit. Thus, a novel methodology of application of plate heat exchanger into the retrofit design of HENs has been proposed.

The objective is to maximize the retrofit profit, which is the profit of energy saving minus the cost of retrofit, while meeting the constraints. A significant challenge of integration of plate heat exchangers and conventional heat exchangers is dealing with different minimum approach temperatures and identifying the best heat exchangers to replace. To achieve these goals, a three-stage optimization approach has been developed. The first stage is to identify the candidate heat exchangers and the best sequence for replacing the plate heat exchangers. Only the exchangers on utility paths are considered for replacement based on the principle of energy balance. The incidence matrix approach is used for the identification of candidate heat exchanger on utility paths. Sensitivity analysis is used to identify the best heat exchangers to replace based on the passive response a heat exchanger has on the network. The second stage is to replace the identified heat exchanger with plate heat exchanger. To obtain the minimum capital cost of plate heat exchangers, the maximum heat transfer coefficient is derived from the model proposed in the first paper based on the process stream data. The third stage is to develop a NLP model to rebalance the network and maximize the retrofit profit after the replacement by varying the heat duty of exchangers on the utility path and the heat transfer coefficient of exchangers not on utility paths.
The proposed method combines heuristics and optimization, which encourages user interaction and ensures the optimal heat recovery is achieved. The results show that compared to the heat transfer technologies, application of plate heat exchangers makes it possible to increase the heat recovery significantly due to the small minimum approach temperature. But the installation cost of the plate heat exchangers is relatively high. Therefore, plate heat exchangers can be used as an alternative option for the retrofit design of PHEs. The savings are constrained mainly due to the maintained original sizes.

6.1.3 Application of plate heat exchanger into retrofit with topology change

A step-by-step novel methodology of utilizing the high efficiency plate heat exchangers into the retrofit design of PHEs, which allows for structure modification, has been proposed. The objective of this research is to achieve the maximum heat recovery at a certain minimum temperature difference with the minimum steps of modifications and retrofit cost. The distinctive feature of this approach is handling the different minimum approach temperatures for the different types of exchangers within an optimization framework. Pinch analysis is used to provide insight into how to identify the best types of modification and their location. For the best HEN retrofit solution, the minimum modification steps and lowest retrofit cost is required. Thus, an algorithm to identify the most beneficial modification methods depending on the different scenario has been developed, which significantly saves the computation time for the optimization process.

The results show that, compared with shell and tube heat exchangers, the utilization of plate heat exchangers is always beneficial and dramatically reduces the retrofit cost to achieve the same amount of energy savings. The proposed methodology can be applied into various HENs retrofit based on the users’ objectives.

6.2 Future work

Although the objectives in this thesis have been met, further research should address the following problems.

1. The optimization of a single plate heat exchanger design considered two stream heat transfer. However, it is possible for plate heat exchangers to handle heat transfer with multiple streams. The proposed methodology for thermal-hydraulic model of plate
Chapter 6

Conclusion and Future work

heat exchanger can be further extended to the multi-stream heat transfer by dividing the overall heat exchanger into several two-stream blocks.

2. The optimization approach of plate heat exchanger design for single phase has been proposed with constant physical properties. However, the heat transfer process can involve phase change. Future work can consider phase change in the optimization model of design a plate heat exchanger.

3. The thermal-hydraulic model of a single plate heat exchanger used in this work fails to consider flow maldistribution. However, in practical applications, maldistribution is not negligible, especially in the multi-pass flow arrangements, which may affect the heat transfer behavior. Since each block has different flow types, the maldistribution mostly happens in the channels between different blocks. It can be assumed that there are two different temperatures in the upper and lower layer of channel, and use the average temperature as the real temperature of the fluids in the channel. The model can be then optimized by following the same approach proposed in Chapter 3.

4. In this work, the fouling issue of plate heat exchangers has been neglected. Compared with traditional shell and tube heat exchangers, plate heat exchangers are less prone to fouling due to turbulence in the flow and high shear stress in plate heat exchangers. It might be beneficial to consider the fouling issue when applying plate heat exchangers in the retrofit design of HENs and quantifying the potential benefit.

5. The approach to retrofit in this work has mainly focused on the integration of a mixture of plate and shell-and-tube heat exchangers in heat exchanger networks. The pressure drop is a key factor that cannot be neglected. The pressure drop is recommended in future work when dealing with two different types of heat exchangers.
References


ALFALAVAVal Gasketed plate-and-frame heat exchangers.


