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CREATIV D3.2.5 - HEAT EXCHANGERS FOR ABSORPTION SYSTEMS

Institute for Energy Technology

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Abstract

Compact heat exchangers have reduced volume and weight compared to conventional heat exchangers, and they often offer reduced cost and improved performance. Today, compact heat exchangers are becoming increasingly important elements in many industrial processes, usually in order to increase the energy efficiency. Numerous types of compact heat exchangers exist, and they are classified according to the kinds of compact elements that they employ.

Several promising heat exchanger geometries relevant for absorption systems are described in this report. Especially three of the discussed designs seem very suitable for the application:

- Plate heat exchanger with falling film type heat transfer.
- Plate heat exchanger with bubble type heat transfer.
- Hollow fibre membrane absorber heat exchanger.

As discussed in the report, different fin surfaces can also be used in order to enhance the performance of a heat exchanger.

Keywords:

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1 Introduction

The R&D Project funded by the Research Council of Norway *called KMB CREATIV – Competence project for Reduced Energy use through Advanced Technology InnoVations* has five different areas of work. This report is part of Sub-project 3; Utilisation and storage of thermal energy, and work package 2; *Concepts for surplus heat exploitation*, delivery D3.2.5.

There are several possible measures that can lead to more sustainable energy use. Energy efficiency requires careful selection and sizing of the components of the heat conversion technologies like e.g. heat pumps and chillers. Two aspects regarding heat exchangers can help to reduce the initial and life cycle costs of energy conversion systems. This involves compact heat exchangers and enhancement of the heat and mass transfer capabilities. Compact heat exchangers offer the following benefits:

- Improved efficiency
- Smaller volume and weight
- Lower initial costs
- Tighter control of the conditions (i.e. lower inertia in the system)
- Improved safety (i.e. reduced fluid inventory)

The usual heat exchanger design problem involves specifying a unit that will have a given heat transfer performance (i.e. effectiveness[1]) subject to a number of constraints. The latter could include low capital costs, low operating costs, limitations on size/shape/weight, or ease of maintenance. In a general design process, one typically has the freedom to choose the exchanger configuration (counterflow, cross-flow, multiple-pass, etc), the type of heat exchanger surface (coaxial tube, plate-fin, tube bank, etc), and characteristic dimensions of the surface (tube diameter, spacing in the tube bank, etc).

The objective of this report is to present a compilation of possible options for compact heat exchangers, used as evaporates, condensers, absorbers, desorbers, and in other roles for heat pumping equipment. The absorber is specifically addressed in the report. Chapter 2 gives a brief introduction to heat exchangers, with special focus on types of exchangers, energy balances, and elements relevant for design. Chapter 3 is concerned with compact heat exchangers, including a description of different types of compact heat exchangers. Advantages and disadvantages of different types are also discussed in this chapter, as well as operation limitations. Chapter 4 identifies promising heat

[1] Effectiveness is defined as the actual heat transfer divided by the maximum heat transfer obtainable in an infinitely long heat exchanger.

exchanger geometries relevant for absorption heat pumps and chillers. Some concluding remarks are given in chapter 5.

2 Heat exchangers

2.1 Introduction

A heat exchanger is a device built for efficient heat transfer from one medium to another. The energy transfer occurs between fluids at different temperature by different heat transfer modes (i.e. conduction, radiation, and convection). The media may be separated by a solid wall (unmixed) or they may be in direct contact. Heat exchangers are used in numerous applications, including e.g. space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, and natural gas processing.

2.2 Types of heat exchangers

A great variety of heat exchangers are commonly used. According to the design requirements, the following factors can vary:

- The geometry of the flow configuration
- The type of heat exchanger surface
- The materials of construction

Heat exchangers are often classified according to their flow arrangement, and some examples can be seen in Figure 1.

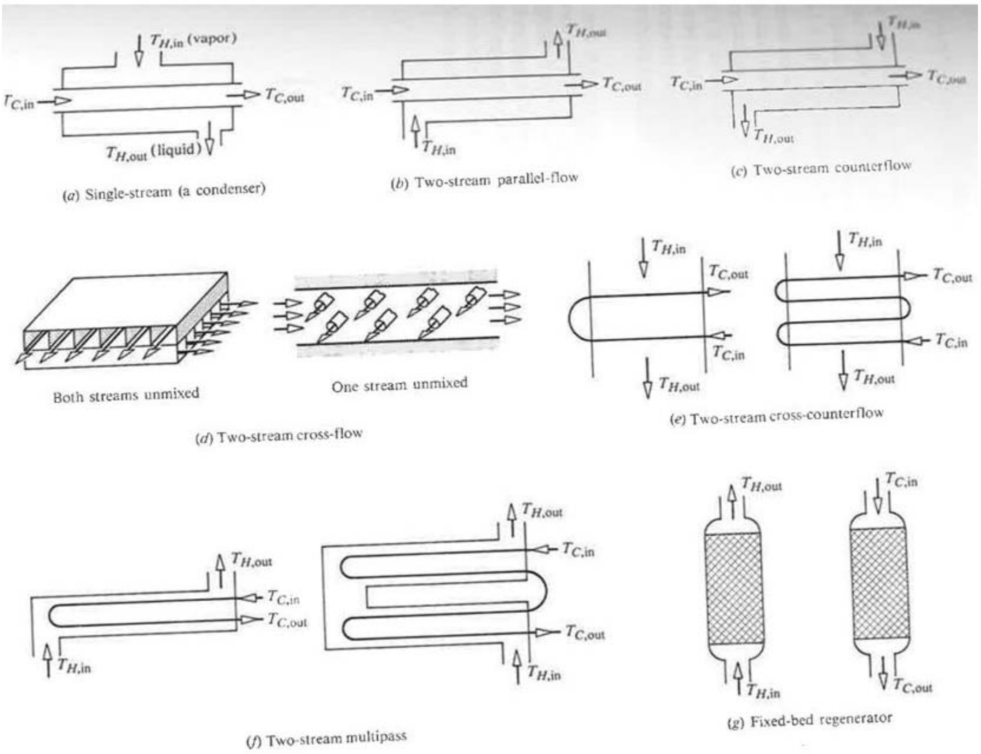

Figure 1 - Common heat exchanger geometries [1]

2.3 Heat transfer surfaces

2.3.1 Introduction

Heat transfer surfaces can take many forms. The most common design is a plain tube. Such a tube is usually circular and straight but can be bent or coiled. If the heat transfer resistance on one side of the tube is much larger than that on the other, the surface may be finned to increase the effective heat transfer area.

A wide variety of secondary surfaces are available. An example is given in Figure 2, which shows a plate-fin exchanger assembly. The purpose of adding fins is that they both serve as a secondary heat transfer surface and mechanical support for the internal pressure between layers.

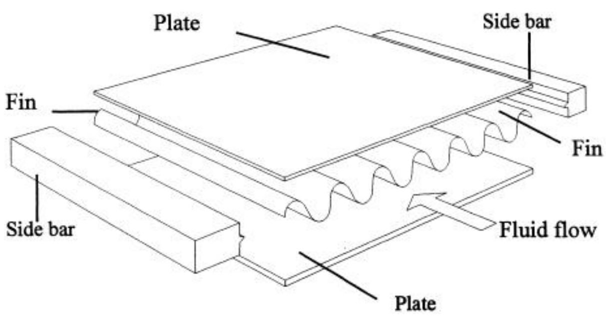

Figure 2 - Plate-fin exchanger assembly [2]

2.3.2 Ducts

The continuous duct is a very common class of surface for compact heat exchangers. The duct shape could be e.g. circular, rectangular, triangular, sine, regular polygonal, semicircular or trapezoidal. Formulas describing hydraulic diameter, friction factor and Nusselt number for different geometries can be found in various literature (e.g. [3]). Fully developed flow is a limiting case for all the design solutions. In the entrance region of ducts the flow will start from a condition depending on the leading edge. This will locally give infinite Nusselt number and wall shear stress. The flow will then start to develop under influence from the duct walls until it is fully developed. As a consequence, in real heat transfer ducts both friction and heat transfer parameters are higher than in a fully developed case.

2.3.3 Plate-fin surfaces

Plate-fin surfaces are by far the most common surface type of all compact types, and it is being used in numerous applications. Six basic fin surfaces are shown in Figure 3, including the plain rectangular fin (a), the plain triangular fin (b), the wavy fin (c), the offset strip fin (d), the perforated fin (e), and the louvered fin (f).

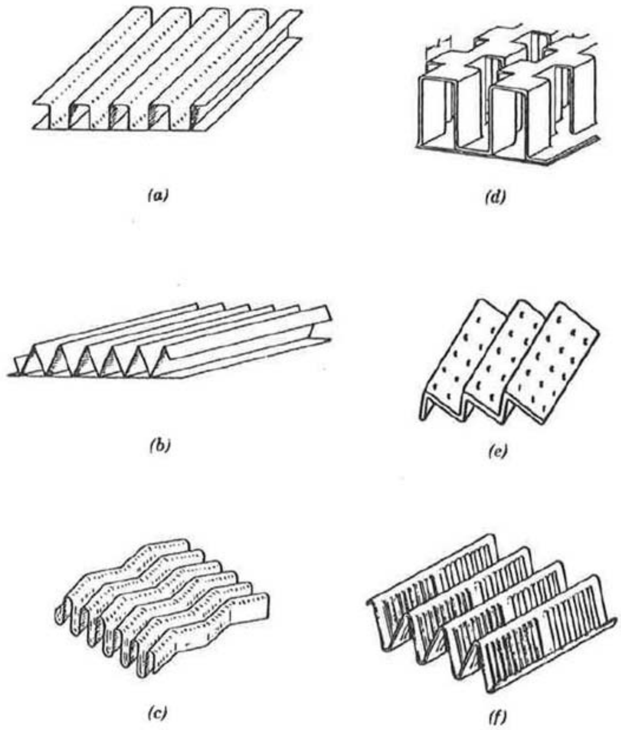

Figure 3 - Different fin geometries [3]

The plain fin (e.g. rectangular or triangular) can be characterised as having long uninterrupted flow passages. It is often designated by a number that indicates the number of fins per unit length.

The strip fins are also referred to as offset fins since they usually are offset at frequent intervals. Other designations include serrated fin or interrupted fin. The offset strip fin has a high performance, and has been subject to extensive experimental studies. It is often designated by a fraction that indicates the length of the fin in the flow direction, as well as a number that describes the number of fins per unit length.

Wavy fins are characterised by a continuous curvature. The purpose of the waves in the surface is to interrupt the boundary layer by changing the flow direction. Other designations include corrugated or herringbone fin. According to Webb [4], this fin is competitive in performance with the offset strip fin. Two alternative designs are common for wavy fins. Either a folded fin strip between flat separating plates or a corrugated fin interlaced by flat or round tubes. Typically, the length of each corrugation is less than the length of the offset strip fin. Due to the absence of blunt leading edges, no wakes will be created. However, small dead zones will occur in the bottom of each corrugation, yielding a higher mean velocity and associated skin friction.

Perforated fins have a surface with holes in order to interrupt the boundary layer. These fins are designated by the number of fins per unit length. Perforated fins are often used in distributed sections of heat exchangers, as well as for boiling applications. Lateral migration of flow is obtained in distribution with less pressure drop than when using offset strip fins. They have also a slight performance improvement per unit surface area, but they lose some surface area, and they have generally a higher friction factor.

The louvered fin design has fins that are cut and bent into the flow stream at frequent intervals. The louvered fin is designated in the same way as the strip fin. It is formed by a rolling process. This makes it relative cheap to produce, and due to this the design is widely used for mass production markets (e.g. automotive heaters, radiators). The fin can be bonded to separating plates or to tubes. The former design is a form of conventional plate-fin surface. The performance of the louvered fin is comparable, or even better, to that of the offset strip fin. This is due to a longer flow length for mixing in a louvered surface. Here, the mixing is more complete between one fin and its downstream neighbour, giving a higher effective temperature difference.

2.3.4 Other surfaces

Pressed plate surfaces is another surface type used in compact heat exchangers. A well known example is the plate and frame exchanger. Here, the plates consist of a number of different variants of a chevron-corrugate form. One example is shown in Figure 4. It is common with variations in both the form of the chevron pattern and in the detailed shape of the corrugation itself.

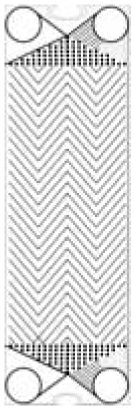

Figure 4 – Chevron configuration

An important parameter describing the corrugation form is the chevron angle. This is the angle between the overall flow direction and the line of the corrugation channels. 90 degrees represent flow directly across the corrugation, and 0 degrees represent flow along the corrugation. However, these limits are rarely used. Corrugations improve the heat transfer by generating turbulence. They also strengthen the plate, control the flow gap, and increase the effective area of the plates (e.g. 15 – 25 %). A wide variety of corrugation types are used in plate design. Both thermal and mechanical considerations must be taken into account when choosing the proper corrugation.

Other examples of surfaces include plate and shell surfaces, welded plates, and printed circuit heat exchanger surfaces.

2.4 Energy balances – Supporting concepts

2.4.1 Introduction

In order to analyse the performance of a heat exchanger, some supporting concept are needed. This chapter will briefly describe the following concepts:

- Overall heat transfer coefficient
- Logarithmic mean temperature difference
- Effectiveness & Number of transfer units

2.4.2 Overall heat transfer coefficient

Most heat exchangers involve transfer of heat from one fluid to another across a plate or tube wall. By using an electrical resistance analogy (common practice for heat transfer analysis), the local temperature profile and thermal circuit for heat flow through an exchanger tube is shown in Figure 5.

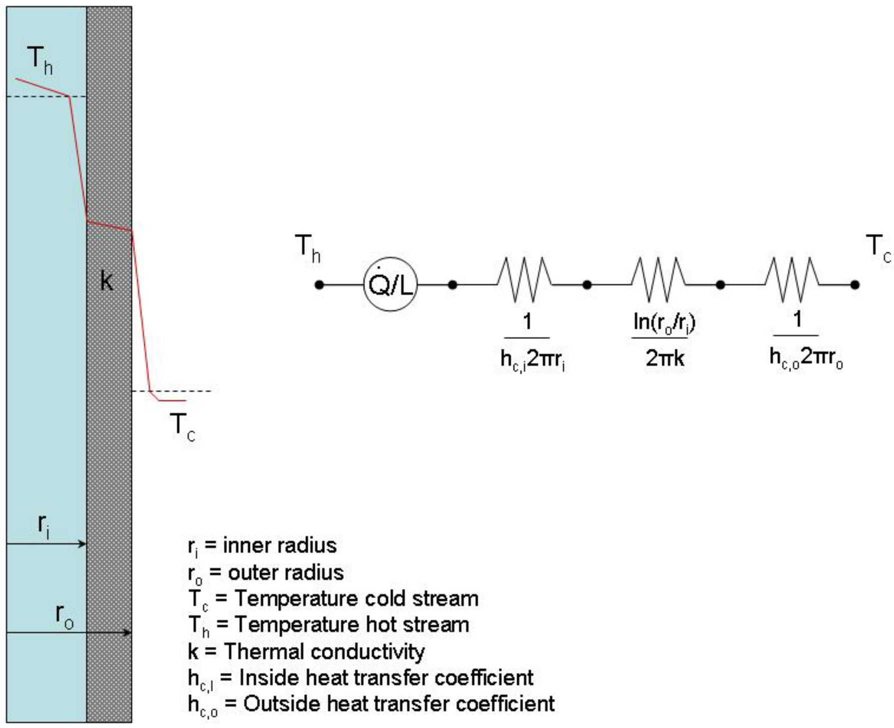

Figure 5 - Thermal circuit

For a clean tube, the product of overall heat transfer coefficient (U) times perimeter (*P*) can be expressed in the following way:

$$\frac{1}{\mathrm{U}P} = \frac{1}{\mathrm{h_{c,i}}\, 2\pi\, \mathrm{r_i}} + \frac{\ln(\mathrm{r_o}/\mathrm{r_i})}{2\pi\, \mathrm{k}} + \frac{1}{\mathrm{h_{c,o}}\, 2\pi\, \mathrm{r_o}} \tag{2.1}$$

The symbols in equation (2.1) are described in Figure 5. For an element of the exchanger that is Δx long, the heat transfer rate ($\Delta\dot{Q}$) can be described by the following equation:

$$\Delta \dot{\mathrm{Q}} = \mathrm{U}P\, \Delta \mathrm{x}\, (\mathrm{T_h} - \mathrm{T_c}) \tag{2.2}$$

The U*P* product can be based on either the inner or the outer diameter.

A heat exchanger that has been in service for some time may have deposits on the heat transfer surfaces; so-called fouled heat transfer surfaces. The overall heat transfer coefficient (U_f) for a fouled exchanger can be expressed by the following equation:

$$\frac{1}{\mathrm{U_f}P} = \frac{1}{\mathrm{U}P} + \frac{\mathrm{R_{f,h}}}{P_\mathrm{h}} + \frac{\mathrm{R_{f,c}}}{P_\mathrm{c}} \tag{2.3}$$

Here, $R_{f,h}$ and $R_{f,c}$ are the hot and cold stream fouling resistance, respectively. Recommended values for the fouling resistance can be found in various sources (e.g. [1]).

As discussed in chapter 2.3.3, fins can be added to a tube wall in order to increase the heat transfer surface. The effects of the fins can be included in the overall heat transfer coefficient. This means that the overall heat transfer coefficient for a clean tube can be described by the following equation:

$$\frac{1}{UP} = \frac{1}{h_{c,i} 2\pi r_i} + \frac{\ln(r_o/r_i)}{2\pi k} + \frac{1}{h_{c,o}(A_f/L)\eta_f + h_{c,o}(A_p/L)} \tag{2.4}$$

Here, (A_f/L) is the surface area of fins per unit length of tube, η_f is the fin efficiency, and (A_p/L) is the un-finned surface area per unit length of tube.

2.4.3 Logarithmic mean temperature difference

The most common type of heat exchanger is the two-stream steady-flow exchanger. The stream flow pattern in the exchanger can either be in parallel, counter-flow, or cross-flow. Two methods can be used to analyse these exchangers, the logarithmic mean temperature method and the effectiveness – number of transfer units method. The logarithmic mean temperature method is briefly described here.

The temperature difference for heat transfer from the hot to the cold fluid, $T_h - T_c$, varies along a heat exchanger (for both parallel flow and counterflow). The total heat transfer in an exchanger can be written as:

$$\dot{Q} = UP\, L\, \Delta T_{lm} \tag{2.5}$$

Here, UP is the product of the overall heat transfer coefficient (see section 2.4.2) and the transfer perimeter for the exchanger, L is the exchanger length, and ΔT_{lm} is an appropriate mean temperature difference between the hot and cold streams. ΔT_{lm} must be determined by analysis, but for parallel-flow and counterflow exchangers, the analysis is straightforward[2]. The result is shown in the equation below:

$$\Delta T_{lm} = \frac{(T_h - T_c)_L - (T_h - T_c)_0}{\ln\left[(T_h - T_c)_L / (T_h - T_c)_0\right]} \tag{2.6}$$

ΔT_{lm} is called the log mean temperature difference. For two-stream configuration other than the ideal case, a correction factor *F* is applied to the log mean temperature difference:

$$\dot{Q} = UAF\, \Delta T_{lm} \tag{2.7}$$

[2] Provided the following simplifying assumptions: 1) The heat exchanger is insulated from its surroundings (only heat exchange between fluids). 2) Axial conduction is negligible. 3) Potential and kinetic energy changes are negligible. 4) The fluid specific heats and the overall heat transfer coefficient are constant.

F factor charts can be found in various references (e.g. [5, 6]), and are commonly used in engineering practice.

2.4.4 Effectiveness - Number of transfer units (ε – NTU)

The performance of two-stream exchangers can be expressed in terms of effectiveness (ε) and number of transfer units (NTU). The ε-NTU formulation is rapidly becoming more popular, particularly because of its suitability for computer-aided design. The drawback with the ΔT_{lm} approach for heat exchanger analysis is that it requires both the inlet and outlet temperatures to be known. If the inlet or outlet temperature is to be calculated for a given exchanger and flowrates, the ΔT_{lm} approach requires an iterative solution procedure or specially constructed charts. The iterative procedure can be avoided if the ε-NTU approach is used instead.

It is the stream with the smallest flow thermal capacity that limits the amount of heat that can be transferred. The effectiveness is defined by the following equation:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{(\dot{m}C_p)_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,in})} = \frac{(\dot{m}C_p)_c(T_{c,out} - T_{c,in})}{C_{min}(T_{h,in} - T_{c,in})} \tag{2.8}$$

Here, Q is the actual heat transferred and Q_{max} is maximum possible heat that can be transferred (in an infinitely long heat exchanger).

Here the minimum flow thermal capacity, C_{min}, is defined by the following equation:

$$C_{min} = \min\left[(\dot{m}C_p)_h, (\dot{m}C_p)_c\right] \tag{2.9}$$

where m and C_p is the mass flow and the specific heat capacity of the hot and cold fluid.

The number of transfer units, NTU, is defined in the following way:

$$NTU = \frac{UPL}{C_{min}} \tag{2.10}$$

Depending on the exchanger configuration, different formulas exist in order to determine ε or NTU (see e.g. [1]). These expressions consist only of three parameters; R_c, NTU, and ε, where R_c is the capacitance ratio (C_{min}/C_{max}). These parameters are combined in different manners depending on the exchanger configuration. As an example, for a counterflow exchanger, the following two expressions apply:

$$\varepsilon = \frac{1 - \exp\left[-NTU(1 - R_c)\right]}{1 - R_c\exp\left[-NTU(1 - R_c)\right]} \tag{2.11}$$

$$\mathrm{NTU} = \frac{1}{1 - R_c} \ln\left(\frac{1 - \varepsilon R_c}{1 - \varepsilon}\right) \tag{2.12}$$

NTU and R_c are known for a given heat exchanger and flowrates. This means that the effectiveness can be calculated, and the unknown outlet temperature can be determined. On the other hand, in order to design a heat exchanger to have a specific outlet temperature, i.e. effectiveness, NTU must be treated as an unknown. It should be noted that the effectiveness and ΔT_{lm} formulation for two-stream exchangers are mathematically equivalent, and either is sufficient for the solution of a problem.

2.5 Elements of heat exchanger design

Heat transfer considerations alone do not determine the dimensions of a heat exchanger. Usually, a heat exchanger design problem requires the designer to specify a unit that will have a given heat transfer performance, i.e. effectiveness, subject to a number of constraints (see chapter 1). A possible design strategy (from [1]) involves the following steps:

1. Specify the required heat transfer effectiveness
2. Specify the allowable pressure drop for one or both streams
3. Choose a flow configuration
4. Choose a type of heat transfer surface
5. Choose the dimensions of the surface
6. Calculate the resulting dimensions of the unit
7. Evaluate the design with respect to constraints such as capital cost, size, weight, and maintenance

The dimension of the heat exchanger (obtained from step 6) may not be unique. In addition, for some configurations it will not be possible to choose the surface dimensions since these are fixed by the heat transfer and pressure drop requirements, e.g. for a coaxial-tube exchanger. Normally, the complete exchanger design problem involves optimisation.

3 Compact heat exchangers

3.1 Introduction

The feature of compact heat exchangers which distinguish them from other types is their high area density. Area density (in m^2/m^3) is defined as the ratio of heat transfer surface to heat exchanger volume. Compact heat exchangers have reduced volume and weight compared to conventional heat exchangers, and as a consequence, offers reduced cost and improved performance. The definition of a compact heat exchanger is related to the area density [7]:

- Greater than 700 m^2/m^3 for units operating in gas streams
- Greater than 300 m^2/m^3 for units operating in liquid or two-phase streams

Figure 6 shows a diagram relating area density to hydraulic diameters for various types of heat exchangers. Today, compact heat exchangers are becoming increasingly important elements in many industrial processes, usually in order to increase the energy efficiency. Some types of compact heat exchangers have also been used regularly for many decades. This includes e.g. units in car air conditioning systems. The heat transfer mode (see chapter 4) must also be developed in order to have a high overall heat transfer coefficient. This will reduce the size of the heat exchanger further.

Compact heat exchangers can be classified [8] according to the kinds of compact elements that they employ. Commonly, six classes of compact elements exist:

1. Circular and flattened circular tubes: This is the simplest design, and the design could be flow inside straight tubes, flow inside straight flattened tubes, or flow inside straight flattened dimpled tubes.
2. Tubular surfaces: Include arrays of tubes of small diameter used in service where the robustness and need for cleaning is less than in a conventional shell-and-tube exchanger.
3. Surfaces with flow normal to banks of smooth tubes: Different designs exist, e.g. round or flattened tubes expanded into fins that can accept several tube rows.
4. Plate fin surfaces: Numerous fin designs exist (see 2.3.3).
5. Finned-tube surfaces: Examples include circular tubes with fins or finned flat tubes.
6. Matrix surfaces: Typically used in rotating, regenerative heat exchangers.

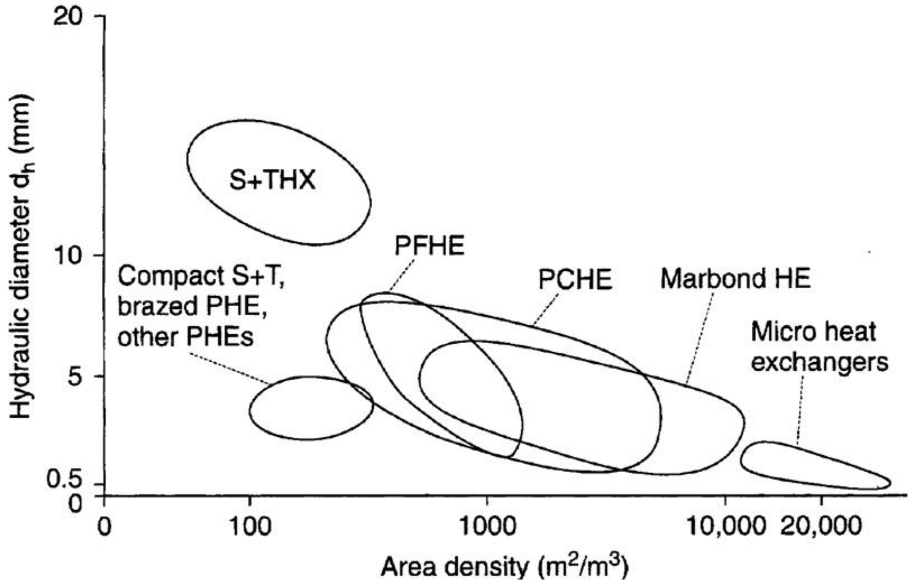

Figure 6 - Area density for different types of heat exchangers[3] [7]

Compact heat exchangers have several specific characteristics. They usually have extended surfaces, e.g. fins. Another characteristic property is the high heat transfer surface area per unit volume on at least one of the fluid sides. Compact heat exchangers have usually small hydraulic diameters. As a consequence, the fluids must be clean and relatively non-fouling because of small passages and difficulty in cleaning. Plain fins (see Figure 3) are suitable for use when moderate fouling occurs. In addition, pressure drop considerations are as important as the heat transfer rate, since the pressure drop affects the fluid pumping power. Due to thin fins and/or joining of fins to plates or tubes, the operating pressures and temperatures are somewhat limited compared to un-finned heat exchangers. Although enhanced heat transfer efficiency is accomplished by using very thin materials, problems are encountered because of the reduced mechanical strength of thin material fins.

Materials for the manufacture of fins for compact heat exchangers are limited by the operating temperature. Fins constructed of aluminium, copper or brass can be used for low to moderate temperature applications (typically up to approximately 200 ºC), and achieving high fin efficiency. For high temperature applications, stainless steel of heat resistant alloys can be used. This could give a possible reduction in the fin efficiency. The manufacture is normally done by brazing or welding. Mechanical bonding of fins is an alternative for low to medium temperature applications.

3.2 Compact heat exchanger types

Numerous types of compact heat exchangers exist today. This include among others:

- Plate and frame heat exchangers

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[3] S+THE = shell and tube heat exchanger. PHE = plate heat exchanger. PFHE = plate and fin heat exchanger. PCHE = printed circuit heat exchanger.

- Brazed plate heat exchangers
- Welded plate heat exchangers
- Spiral heat exchangers
- Plate-fin heat exchangers
- Printed circuit heat exchangers
- The Marbond heat exchanger
- Compact shell and tube heat exchangers
- Compact types retaining a “shell” (e.g. the plate and shell heat exchanger)

A comparison of important features of the most relevant heat exchangers can be found in Table 1, including compactness (in m^2/m^3), stream types (e.g. liquid-liquid, two-phase), materials (e.g. stainless steel, titanium, nickel alloys), temperature range, maximum pressure, cleaning methods (e.g. mechanical, chemical), corrosion resistance, multi-stream capability, and multi-pass capability. Often, the operational conditions constrain the choice of type and its surface to a small selection.

Table 1 – Important features of compact heat exchangers [3]

Type of Heat Exchanger	Area Density (m^2/m^3) (0)	Stream Types (1)	Materials (3)	Temperature Range (°C)	Maximum Pressure (bar) (2)	Fluid Limitations	Cleaning Methods	Corrosion Resistance (18)	Multi-stream Capability	Multi-pass Capability
Plate and frame (Gasketed)	→200	liquid-liquid gas-liquid two-phase	s/s, Ti, Incoloy Hastelloy, graphite, polymer	-25 to +175 Special –35 to +200	Normal 25 Special 40	Limited by gasket type. Not normal for gases	Mechanical (14) Chemical	Good (7)	Yes (9)	Yes
Partially welded plate	→200	gas-liquid liquid-liquid two-phase	s/s, Ti, Incoloy Hastelloy	-35 to +200	25	Few - Some types need clean fluids	Mechanical (4, 14) Chemical (6)	Good (7)	No	Yes
Fully welded plate (AlfaRex)	→200	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ti, Ni alloys	-50 to +650	40	Few - Some types need clean fluid	Chemical	Excellent	No	Yes
Brazed plate	→200	liquid-liquid two-phase	s/s	Cu braze –195 to +220 Ni braze →400	Cu braze 30 Ni braze 16	Must be compatible with braze	Chemical (5)	Good (8)	No	No (10)
Brazed plate-fin	800 – 1,500	gas-gas gas-liquid liquid-liquid two-phase	Al, s/s, Ti Ni alloy	Al –270 to +200 s/s cryogenic to +650	120	Low fouling Many limitations with Al	Chemical	Good	Yes	Yes
Diffusion-bonded plate-fin	700 - 800	gas-gas gas-liquid liquid-liquid two-phase	Ti, s/s, Ni	→400	200	Low fouling	Chemical	Excellent	Yes	Yes
Printed circuit	200 - 5,000	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ni, Ni alloys Ti	-200 to +900	500	Low fouling	Chemical	Excellent	Yes	Yes
Shell and tube (19)	→100	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ti, (shell also in c/s), many different materials	-100 to +600	Shell 300 Tubes 1400	Few	Mechanical (16, 14) Chemical (17)	Good	No	Yes

Notes for Table A3.2.3 → = up to, s/s = stainless steel, Ti = titanium, Ni = nickel, Al = aluminium, Cu = copper, c/s= carbon steel

(0) Area includes the secondary surface (such as fins)
(1) Two-phase includes boiling and condensing duties
(2) The maximum pressure capability is unlikely to occur at the higher operating temperatures, and assumes no pressure/stress-related corrosion
(3) Other special alloys are frequently available
(4) On gasket side
(5) Ensure compatibility with copper braze
(6) On welded side
(7) Function of gasket as well as plate material
(8) Function of braze as well as plate material
(9) Not common
(10) Not in a single unit
(11) On tube side
(12) Only when flanged access provided, otherwise chemical cleaning
(13) Five fluids maximum
(14) Can be dismantled
(15) Shell may be composed of polymeric material
(16) On shell side
(17) On plate or tube side
(18) Primarily a function of construction materials rather than the exchanger type
(19) Not a compact exchanger technology (given for comparison)

The capital cost of a heat exchanger is an important factor when deciding which exchanger to use. The capital cost is often expressed as a cost per unit surface area of heat transfer. This depends on the heat transfer coefficient for that surface, which again depends on the compactness and the degree of enhancement of the surface. Table 2 summarises some area costs for different compact heat exchangers.

Table 2 – Cost per unit area of compact heat exchangers [3]

Heat Exchanger	Q/ΔT (W/K)	Cold Stream	Hot Stream	U (W/m^2K)	C (GBP/W/K)
S+THX	1,000	Water	Water	938	1.12
PHE (Gasket)	1,000	Water	Water	3,457	0.036
S+THX	30,000	Water	Water	938	0.14
PHE[1] (Gasket)	30,000	Water	Water	3,457	0.033
PFHE[3]	100,000	Medium pressure gas	Condensing hydrocarbon	402	0.227
PCHE[4]	100,000	Medium pressure gas	Condensing hydrocarbon	1090	0.55
PHE (Welded)	100,000	Medium pressure gas	Condensing hydrocarbon	1518	0.186
S+THX	30,000	Water	Water	938	0.14
PHE (Welded)	30,000	Water	Water	9,100	0.147
PHE[2] (Gasket)	30,000	Water	Water	3,500	0.02
PCHE[5]	30,000	Water	Water	3,230	0.4

Notes:
1 & 2 Different sources of data.
3 Aluminium. Stainless steel costs are 3 times greater. Titanium costs are greater by a factor of 5.
4 Stainless steel.
5 Treated water on one side, 30% triethylene glycol on other side.

3.3 Plate heat exchangers

3.3.1 Introduction

The plate heat exchanger (PHE) is used in numerous applications, including:

- Liquid-food heat treatment (e.g. pasteurisation of beer, sterilisation of milk, and cooling of milk)
- Closed-circuit cooling applications (e.g. on ships, on oil platforms, and in power stations)
- Steam heater
- Evaporator
- Part of air-conditioning plants
- Wet gas cooler
- Refrigeration plants

Wherever a close temperature approach is required, weight or space is very important, or corrosion-resistant materials are needed, the PHE is a prime candidate for heat exchanger selection.

3.3.2 Advantages of a plate heat exchanger

Some significant advantages with the PHE are described below. Primarily, plate heat exchangers achieve high NTU values. The PHE attains high heat transfer coefficients, and achieves almost a full couter-current flow arrangement. This enables small end temperature differences. These features allow high recuperative efficiencies, and for a single pass, a NTU value of 6 can often be achieved [9].

Often, the ability to modify an existing heat exchanger is desirable. This could be due to requirements for increased capacity, altered applications, or adjustments to handle unexpected fouling of the heat exchanger surface. Such requirements can not be met by most types of heat exchanger equipment. In a gasketed or partially welded PHE, the arrangement and the surface can easily be altered to conform to such needs.

A gasketed or partially welded PHE is relatively easy to dismantle, giving access to all working surfaces for inspection and cleaning. This is a huge advantage when fouling is not asymptotic in nature and in-place chemical cleaning systems are not available.

PHE uses essentially rectangular channels with narrow gaps. This leads to a compact construction and low liquid holdup. This feature is enhanced by high heat transfer coefficients, something that reduces the surface area requirements compared with e.g. a shell-and-tube exchanger. Low liquid holdup also leads to low weight and a short start-up time.

Simple pressing combined with no welding means a low cost per unit area, especially if more expensive corrosion-resistance materials are used. These features are also shown in Table 2.

If special connector plates are used for intermediate headers, several heat exchange duties can be housed in a single frame. This feature gives reduced cost, weight, and space requirements.

Because the high heat transfer coefficients in a PHE are generated by turbulence, the surface shear stresses are very high. This allows a high fouling removal rate. Compared with a shell-and-tube exchanger, the fouling resistance are very low for most types of fouling.

3.3.3 Limitations of a plate heat exchanger

As discussed in section 3.3.2, the advantages of plate heat exchangers are numerous. However, the capability range is limited compared with a shell-and-tube exchanger. Primarily, a PHE faces limitations regarding pressure. According to Kumar [9], some plate heat exchangers can operate at pressures around 25 bar. However, a more typical limitation is 16 bar. A shell-and-tube exchanger can withstand higher pressures than this. In addition, PHEs are often associated with high pressure drops.

PHEs also experience some limitations regarding gaskets. Although a wide variety of elastomers exists, the maximum temperature limit for PHE gaskets is around 200 ºC [9], something that also can be seen in Table 1. In addition, corrosion or safety aspects for some operating fluids may rule out their use.

Low density vapours are also a challenge for plate heat exchangers. PHEs can act as condensers and vaporisers provided that a low vapour density is not coupled with a high flowrate.

3.3.4 Types of plate heat exchangers

3.3.4.1 Introduction

There are several types of plate heat exchangers available today. The most common PHE is the plate and frame, or gasketed, type.

3.3.4.2 Plate and frame heat exchanger

The plate and frame heat exchanger is currently only second to the shell-and-tube heat exchanger in terms of market share. The most common variant of the plate and frame heat exchanger consists of a number of pressed, corrugated metal plates compressed together into a frame. Gaskets are used to seal the spaces between adjacent plates, and partly to distribute the media between the flow channels. Stainless steel is usually used as plate material.

The heat transfer surface consists of a number of thin corrugated plates pressed out of a high grade metal (see Figure 7). The purpose of the pressed pattern is to induce turbulence, minimise stagnant areas, and fouling. The plates of plate and frame heat exchangers are mass-produced using expensive dies and presses. As a consequence, all plate and frame heat exchangers are made with a limited range of plate designs.

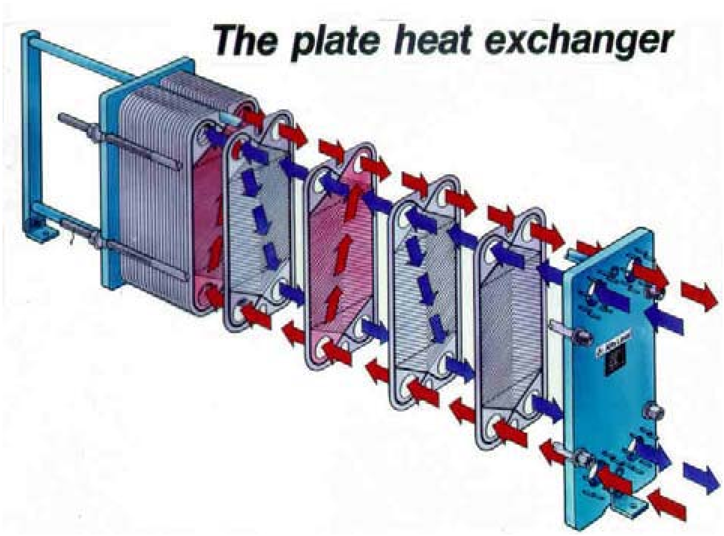

Figure 7 – Exploded view of a plate and frame heat exchanger [10]

Even though all plate and frame heat exchangers are made from standard parts, each one is custom designed. This is done by altering the corrugation angle, flow path or flow gap, which again alters the performance of the heat exchanger. The plate pack is clamped together in a frame, and gaskets are fitted to seal the plate channels and interfaces. The frame consists of a fixed frame plate at one end and a moveable pressure plate at the other. This makes access for cleaning or exchanging the heat transfer surfaces possible. In addition, this type of heat exchanger has the ability to add or remove surface area if necessary.

The plates can be produced from all pressable materials. Metals are most commonly used, but non-metallic materials can be used if corrosion is a problem. The latter materials are used to some extent in absorption cycle machines. Gasket properties have a critical bearing on the capabilities of a plate and frame exchanger, in terms of its tolerance to temperature and pressure. Special care should be taken in locating the gaskets during reassembly, since an imperfect sealing is the main disadvantage of the plate and frame heat exchanger.

The operating limits of gasketed plate and frame heat exchangers vary to some extent from manufacturer to manufacturer. A typical operating temperature range for metal plates is from -35 ºC to 200 ºC. Design pressure up to 25 bar is common, but some heat exchangers have experienced test pressures up to 40 bar. The heat transfer areas per plate vary from 0.02 m^2 to 4.45 m^2. Flow rates up to 3500 m^3/h is common in standard units, rising to 5000 m^3/h if a double port entry is being used.

3.3.4.3 Partially welded plate heat exchangers

From the outside, partially welded plate heat exchangers resemble a fully-gasketed plate and frame unit. The main difference is that the plate pack has alternating welded channels and gasketed channels. The advantage of welding the plate pair is that other materials are eliminated (except for a small gasket around the ports), and corrosion is slightly reduced.

The plate construction materials are more or less the same as for the gasketed plate and frame exchanger, and the type of material is chosen for its resistance to corrosion. The operating limits are as for the gasketed plate and frame type, but with the added protection from leaks. This means that the welded section should tolerate higher pressures. In addition, partially welded plate heat exchangers can tolerate more aggressive media than the gasketed plate and frame heat exchanger. Partially welded plate heat exchangers are used for the evaporation and condensation of refrigerants (e.g. ammonia), implying that industrial heat pump applications could benefit from the use of these heat exchangers.

3.3.4.4 Brazed plate heat exchangers

Brazed heat exchangers consist of a pack of pressed plates that have been brazed together. This eliminates the use of gaskets, and the frame can also be omitted. Brazed plate heat exchangers are usually used in niche applications (e.g. refrigeration). These heat exchangers have heat transfer capabilities up to 600 kW, depending on the manufacturer.

The corrugated plates inside the heat exchanger induce a highly turbulent flow, which again reduces surface deposits in the heat exchanger. Stainless steel is usually used as the plate material, and typically a very high content copper braze is used. Copper brazed units are available for temperatures up to 225 ºC and a maximum pressure of 30 bar. One disadvantage with a copper brazed unit it its incompatibility with some working media. As an alternative, nickel brazed units can be used. These units are available for temperatures up to 400 ºC, and a maximum operating pressure of 16 bar[4].

3.3.4.5 The AlfaRex welded plate heat exchanger

The AlfaRex was the first full-size, gasket-free heat exchanger [10]. It consists of a herringbone plate design which creates channels with high fluid turbulence that increases the thermal efficiency and minimises the risk of fouling. During construction, the herringbone pattern is laser-welded together to form a plate pack in which both media are in full counter-current flow.

The heat exchanger has a design operating temperature range from -50 ºC to 350 ºC at pressures up to 40 bar. The exchanger is capable of handling flowrates up to 800 m^3/h. The heat exchanger is suitable for e.g. evaporation and condensation of ammonia in heat pump and adsorption systems.

3.3.4.6 AlfaNova plate heat exchanger

The AlfaNova is the world's first plate heat exchanger made of 100 % stainless steel. The heat exchanger is based on a new, innovative technology called AlfaFusion. The technology is patented by Alfa Laval, and it is used in the brazing process during manufacturing. It consists of a new stainless steel brazing filler and a new method of brazing the plates. The brazed design holds the plate pack together internally with contact points. External plate frames at design pressures of up to 16 bar are not needed. The exchanger is distributing the load across many separate contact points. Due to this, it provides excellent resistance to pressure fatigue. AlfaNova HP 400 is a high pressure version that has a design pressure of 28 bar at 175 ºC.

Compared to e.g. shell-and-tube heat exchangers, the AlfaNova is extremely compact in relation to its capacity. Specially designed corrugated plates offer optimised heat transfer with low hold-up volumes. The channels in the plates are designed to ensure optimal distribution of the media.

The AlfaNova heat exchanger can withstand temperatures of up to 550 ºC. The 100 % stainless steel construction ensures high resistance to corrosion. Natural refrigerants, such as ammonia and other cooling media that are corrosive to copper, can be used in the heat exchanger. The AlfaNova can be cleaned chemically.

[4] The design pressure depends on the temperature. A higher design pressure can therefore be obtained at lower temperatures.

3.4 Plate-fin heat exchangers

3.4.1 Introduction

Plate-fin heat exchangers (PFHEs) consist of a matrix of flat plates and corrugated fins in a sandwich construction. Brazed aluminium PHFEs exhibit the following certain interesting features and characteristics:

- A very large heat transfer area per unit volume of heat exchanger, typically from 850 to 1500 m^2/m^3. The surface area is composed of primary and secondary (finned) surfaces (see Figure 8).

- A single heat exchanger can incorporate several different process streams and the plate-fin construction allows these to enter and exit the exchanger at intermediate points, and not just at the ends.

- Very close temperature approaches between streams can be obtained, typically down to 1 to 3 ºC.

- The combination of high thermal efficiency, use of aluminium and multi-stream capability gives a compact, low-weight structure.

PFHEs show great versatility. In addition, they are manufactured in a variety of other materials. The combination makes them suitable for both heat pumping systems and absorption cycle units.

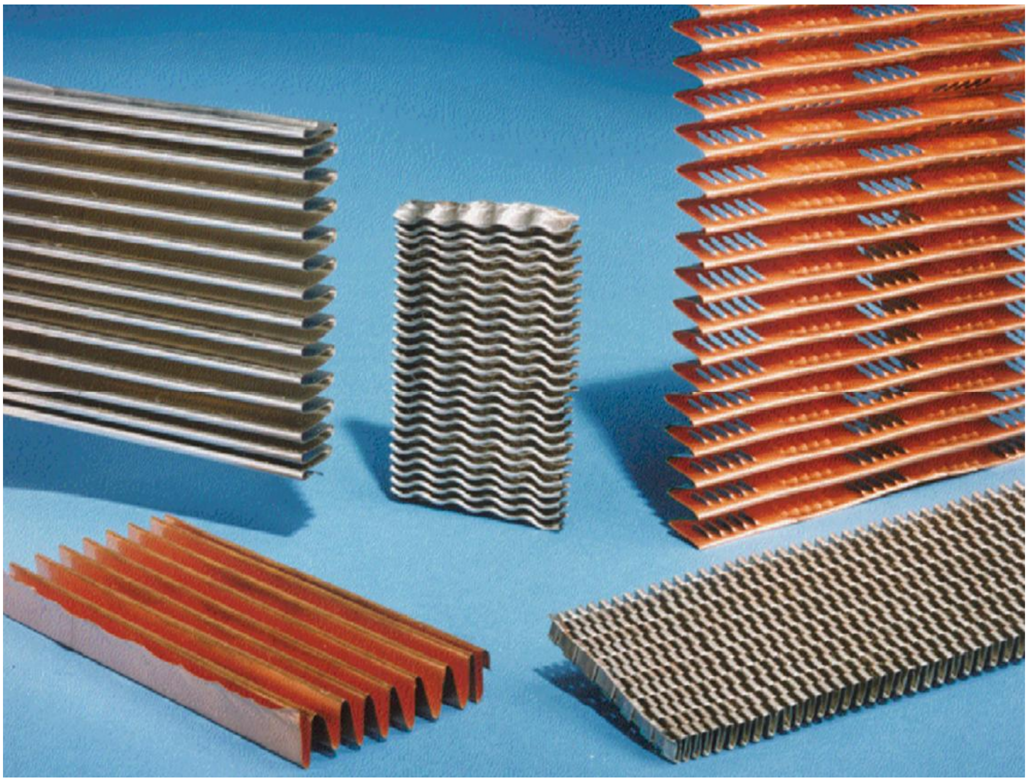

Figure 8 – Fin structures used between the plates in PFHEs [10]

3.4.2 Operating limits

The maximum operating temperature of a plate-fin heat exchanger depends on its construction materials. Aluminium based PFHEs can be used from -270 ºC to 200 ºC (depending on the pipe and header alloys). Stainless steel PFHEs can operate up to 650 ºC, whereas titanium units can tolerate temperatures approaching 550 ºC.

In certain configurations, aluminium brazed units can operate up to 120 bar. Stainless steel PFHEs are currently limited to 50 bar, but future developments will extend their capability to 90 bar [10]. Even higher pressures can be tolerated by using a diffusion-bonded structure. Some features of different fin types are given in Table 3.

Table 3 – Brazed plate-fin types [10]

Fin Type	Application	Features	
		Relative Δp	Relative Heat Transfer
Plain	General	Lowest	Lowest
Perforated	Boiling streams	Low	Low
Herringbone	Gas streams with low allowable ΔP High pressure streams Gas streams for hydrocarbon and natural gas applications	High	High
Serrated	Gas streams in air separation applications General	Highest	Highest

3.5 Printed circuit heat exchangers

3.5.1 Introduction

Printed circuit heat exchangers (PCHEs) have several advantageous features. They are very compact (see e.g. Figure 6), corrosion resistant, and have wide operating limits (se section 3.5.2). The PCHE design offers a unique combination of innovative manufacturing technology and potential breadth of application. The heat exchanger can also be used in a variety of other unit operations, including reactors, mass transfer and mixtures and as a structural member if required.

PCHEs are constructed from flat alloy plates. The fluid flow passages are thereafter photochemically machined into them, and the process is similar to manufacturing of electronic printed circuit boards. Hence,the name of the heat exchangers. A herringbone flow pattern is shown as an example in Figure 9.

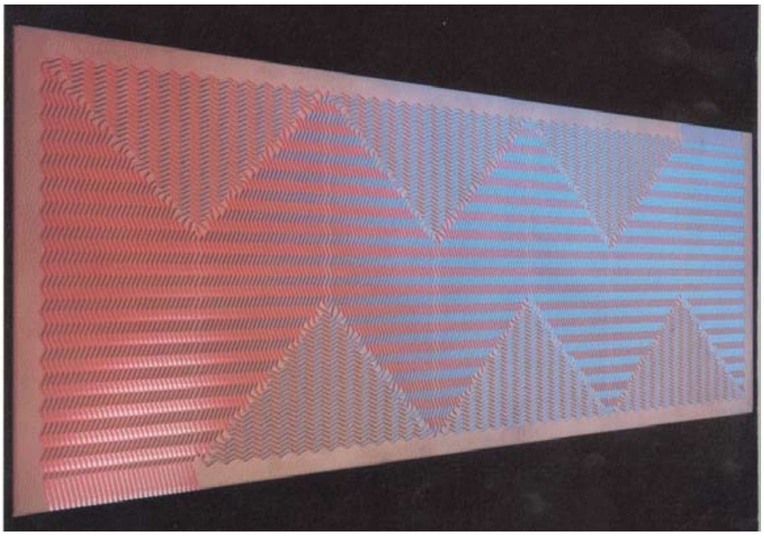

Figure 9 – Fluid flow paths on a typical PCHE plate

3.5.2 Operating limits

The mechanical design of the PCHE is flexible; each pattern can be adjusted according to given needs. This flexibility makes the exchanger able to withstand substantial pressures. 200 bar are often used, with values in the range of 300-500 bar is possible. The all-welded construction gives the possibility to withstand very high temperatures. The use of austenitic steel allows cryogenic operation. The temperature range of a PCHE can therefore be from -200 ºC to 900 ºC. The upper limit depends on the selected material and the pressure duty.

The heat transfer surface density can be as high as 2500 m^2/m^3. This is higher than the prime surface densities in gasketed plate exchangers.

3.5.3 Applications

PCHEs can be used in applications where pressure, temperature or corrosion prevents the use of conventional plate heat exchangers. PCHEs can handle gases, liquids, and two-phase flows. Typical applications for PCHEs include chillers/condensers, cascade condensers, and absorption cycles.

3.6 Advanced compact heat exchangers

3.6.1 The ”Chart-flo” heat exchanger

The Chart-flo is one of the latest truly innovative designs of compact heat exchangers. The heat exchanger, which is produced by Chart Energy & Chemicals, extends the option for those who are looking for high integrity, highly compact units able to operate over a range of pressures and temperatures not met with more conventional compact heat exchangers.

The construction of the Chart-flo heat exchanger allows the use of small passage ways, something that greatly increases the porosity of the heat exchanger core. This can result in a substantially higher area density than the PCHE. A doubling of the porosity when all other factors are kept constant will result in a halving of the volume for a given surface area.

Figure 10 shows an example of plates used in the Chart-flo heat exchanger. The heat exchanger can e.g. be used in order to reduce the size of absorption cycle refrigeration plants. The high porosity of the Chart-flo unit in addition to the use of low hydraulic diameters, gives it advantages regarding compactness compared to other compact heat exchangers. In extreme cases, the volume of a Chart-flo heat exchanger could be as low as 5 % of the equivalent shell-and-tube heat exchanger.

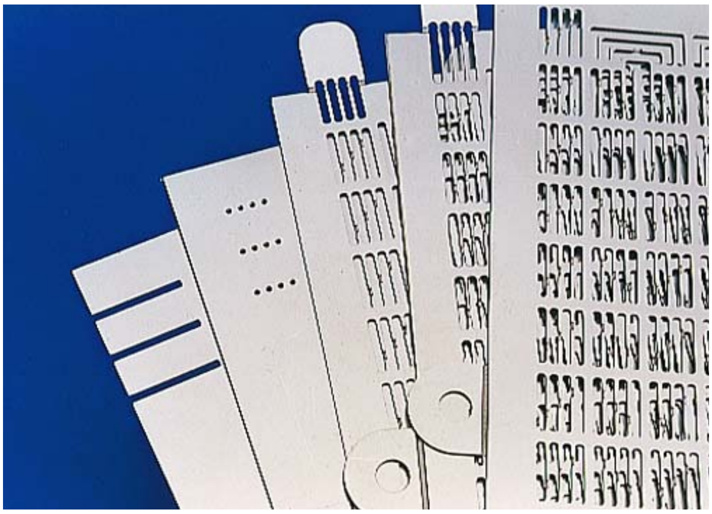

Figure 10 – Plates used in the Chart-flo exchanger [10]

3.6.2 Foam heat exchangers

Rigid foam can used for heat transfer enhancement, and can also be used as a basis of heat exchangers. One example of its use is in a gas-liquid heat exchanger. Here, the tubes penetrate the bulk of the foam, where the foam replaced conventional extended surfaces. The tubes are sintered into the foam, giving good contact.

In general, foam could be put inside tubes or between plates. The foam can be described as a three-dimensional extended surface. Advantages with foam heat exchangers include a choice of materials to 1000 ºC, low weight, compact, and the ability to be formed in complex shapes.

4 Heat exchangers for absorption cycles

4.1 Introduction

The mixture of water and ammonia has a number of attractive advantages which makes it suitable for use in e.g. absorption heat pumps and chillers. Such heat pumps have some properties that make them attractive, including:

- The capability of utilising low-temperature heat sources

- The use of a refrigerant that will not cause the ozone depletion problem

The performance of an absorption heat pump cycle is highly related to the heat and mass transfer of major components, especially the absorber and desorber. The absorber is often designated as the bottleneck in a vapour absorption cycle, that is, in terms of efficiency, cost, and size. The purpose of the absorber is to absorb the refrigerant's vapour into the weak solution, as well as to distribute the heat of absorption to a coolant.

The driving force for the absorption process is the difference between the weak solution concentration and saturation at a given temperature and pressure. During the process, the weak solution enriches and the temperature rises because of the exothermal absorption. This means that the driving force for the absorption decreases. As a consequence, heat must be distributed to the coolant in order to obtain high mass transfer.

The absorption takes place in a small area between the bulk vapour and the solution. A challenge related to absorber design is to obtain efficient heat transfer to the coolant. Design of a small and efficient absorber requires therefore efficient heat and mass transfer properties.

4.2 Two-phase gas/liquid mixture

Special care must be taken when designing the desorber and absorber heat exchangers (of an absorption cycle) due to the non-linear properties of the ammonia/water mixture during phase change processes. Compact heat exchangers can be used in order to reduce the physical dimensions of the components. One factor that strongly influences the performance of compact heat exchangers is the degree of flow rate uniformity in the various parallel channels where the heat transfer occurs.

In heat exchangers where evaporation and condensation takes place, the problem of ensuring uniform distribution of the two-phase flow arise. Uneven two-phase distribution can occur both inside each channel and inside the header. Flow mal-distribution in heat exchangers can reduce thermal and fluid-dynamic performances. Some cases of bad distribution have little effects on heat exchanger performances, whereas others result in significant loss of performance and/or mechanical failure of the device.

For heat transfer in a binary mixture (e.g. ammonia-water) in absorption systems, the heat transfer mode should be carefully selected in order to reduce the heat and mass transfer resistance which exist in both liquid and vapour regions.

4.3 Plate heat exchanger falling film type absorber

In ammonia-water absorption heat pump systems, falling film heat transfer is one alternative option that can be used in order to enhance heat and mass transfer performances. Thin falling film heat transfer modes provide relatively high heat transfer coefficients and are stable during operation. One challenge with this design is related to wettability, and as a consequence, good liquid distributors are needed at the inlet of the liquid flow.

Figure 11 shows an example of a possible configuration for a falling film type absorber. Here, the ammonia-water liquid solution flows down from the top inside of the plate heat exchanger while vapour solution flows up (i.e. in counterflow) to the liquid flow. In some designs, the liquid solution flows in parallel with the vapour solution. In the given figure, the hydronic fluid (e.g. water) flows upwards in counterflow to the liquid solution flow. This means that in the given example exists two counter current flows:

- One between the liquid and vapour solution flows
- One between the liquid solution and the coolant

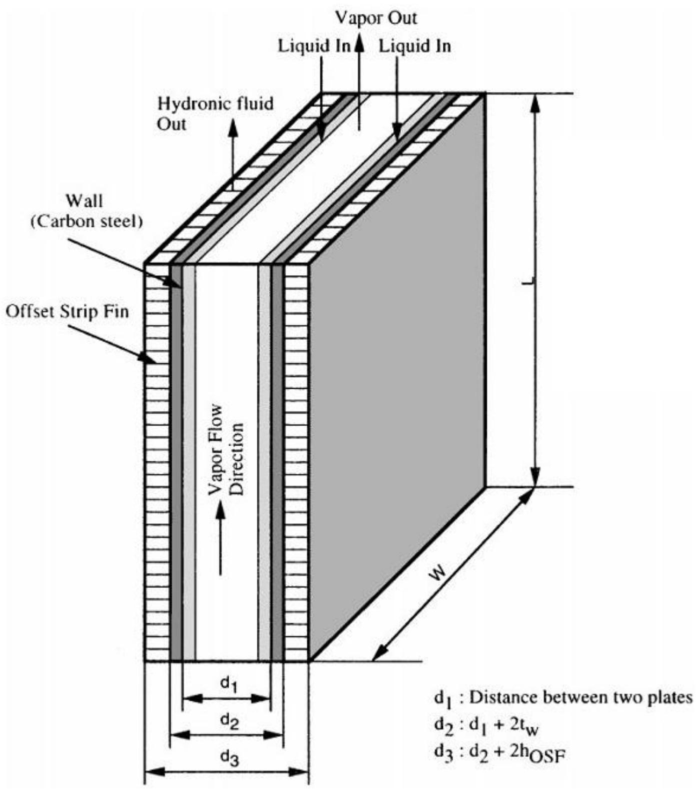

Figure 11 - Schematic diagram of a plate heat exchanger with falling film mode [11]

For a falling film type of heat exchanger, a liquid distributor is required to provide a good wettability at the top of the exchanger. In Figure 11, offset strip fins are inserted in order to enhance the heat transfer coefficient on the coolant side. According to various sources (e.g. [3]), this form of fin surface gives the highest performance. Figure 12 presents a schematic diagram of the offset strip fin.

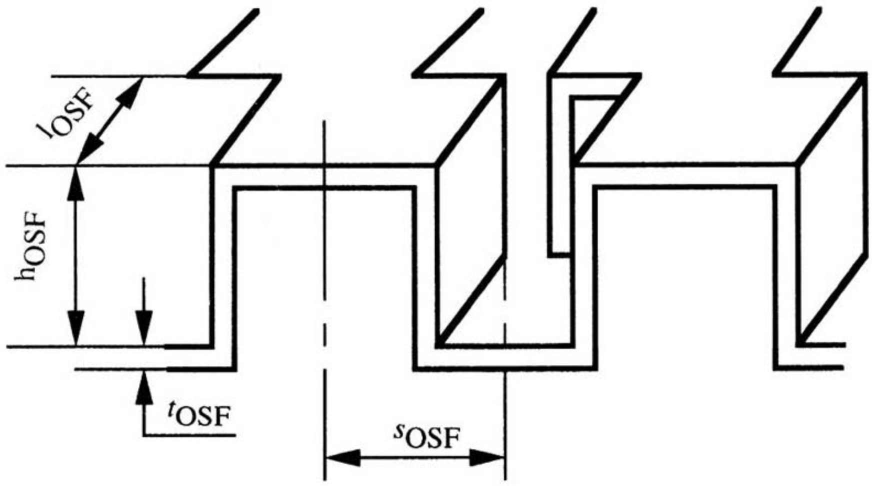

Figure 12 - Offset strip fin [11]

Generally, the liquid and vapour are completely separated in the falling film mode. As a consequence, the heat transfer resistance within the vapour flow is more significant in the falling film mode than in other modes. Another feature of the falling film mode is that both heat and mass transfer areas are constant through the entire length of the heat exchanger. This will often lead to a larger heat exchanger than e.g. a bubble mode heat exchanger (see section 4.4).

Normally, the mass transfer resistance is dominant within the effective mass transfer region in the liquid flow, while both heat and mass transfer resistances are considerable in the vapour flow. The heat transfer coefficients (for coolant, liquid, and vapour) have a significant effect on the heat exchanger size in the falling film mode. According to Kang et al. [11], the heat transfer coefficient in the coolant has the most significant effect on the heat exchanger size.

4.4 Plate heat exchanger bubble type absorber

In ammonia-water absorption heat pump systems bubble type heat exchanger is another alternative design that can be used in order to enhance heat and mass transfer performances. The advantage with this design is not only the high heat transfer coefficients, but also good wettability and mixing between the liquid and the vapour. Unlike the falling film alternative, bubble type heat transfer requires vapour distribution instead of liquid distribution. This is usually an easier task to accomplish. Special care must be taken to avoid stability problems and flooding.

Figure 13 shows an example of a possible configuration for a bubble type absorber. Here, the ammonia-water liquid solution flows down from the top inside of the plate heat exchanger while vapour solution flows up to the liquid flow. In other designs, the liquid solution flows in parallel with the vapour solution. In the given figure, the coolant flows up in counterflow to the liquid solution flow.

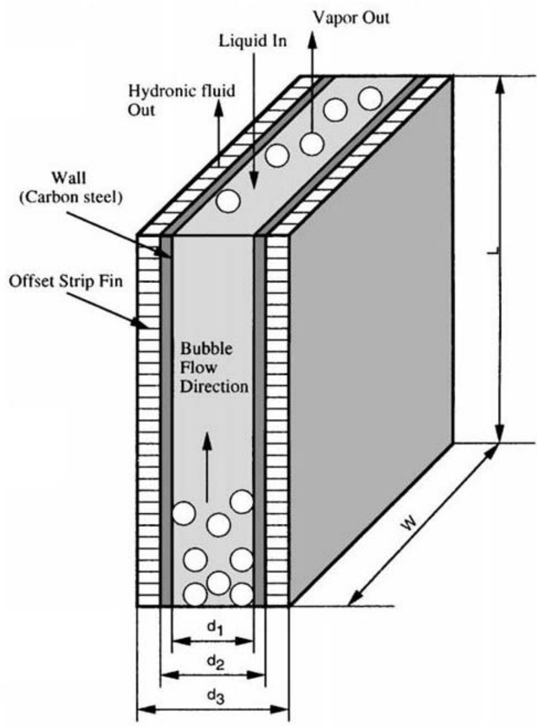

Figure 13 - Plate heat exchanger with bubble absorption mode [11]

In the bubble mode, a vapour distributor (e.g. several orifices) is needed in order to obtain a good mixing rate between liquid and vapour at the bottom of the exchanger. As a consequence, good mixing between the liquid and the vapour is achieved in this design. During the bubble absorption process, the heat transfer area remains constant while the mass transfer area varies with the length of the heat exchanger. The bubble diameter and the gas hold-up will affect the mass transfer area. According to Kang et al. [11], a heat exchanger utilising bubble mode absorption will be of smaller size than in e.g. the falling film mode. This is mostly due to the variation of the mass transfer area, which typically is larger than the heat transfer area. For the bubble absorption mode, the heat transfer resistance is dominant in the vapour region while the mass transfer resistance is dominant within the effective mass transfer region in the liquid flow. If one compares the bubble mode with the falling film mode, then the local absorption rate will be greater for the bubble mode. This is mainly due to the larger mass transfer area, a better mixing between the liquid and the vapour, and the higher heat transfer coefficients for the bubble mode. As also stated above, this will typically result in a smaller heat exchanger for a bubble mode absorber.

As shown by Kang et al. [11], in the bubble mode, the heat transfer coefficient of the coolant has the most significant effect on the absorber size. Unlike the falling mode, the mass transfer coefficients have more significant effect on the heat exchanger size in the bubble mode.

4.5 Hollow fibre membrane

A hybrid hollow fibre membrane absorber and heat exchanger has been theoretically analysed by Chen et al. [12]. According to Gableman et al. [13], hollow fibre membrane contactors made of non-selective porous membranes are very suitable for use in gas-liquid or liquid-liquid devices. Such membranes have some great advantages:

- Large, undisturbed, known and constant interfacial area available in small volume
- Independent gas and liquid velocities which avoid flooding, loading, and turn down ratio problems from two-phase flows.
- Scale-up advantages

According to experimental results by Zarkadas et al. [14], extremely large ratios between surface area and volume makes polymeric hollow fibre heat exchangers more efficient than metal heat exchangers for certain applications. A hollow fibre membrane heat exchanger device utilises porous fibres for the heat and mass transfer between the absorption solution phase and the vapour phase. Nonporous fibres are used to facilitate the heat transfer between the absorption solution phase and the cooling fluid phase. Figure 14 and Figure 15 show a schematic diagram of a hollow fibre membrane module as an absorber heat exchanger. As seen in Figure 14, a central baffle is used to deflect the liquid flow on the shell side in the module and not the flows inside the fibres. The absorption solution enters the centre tube of the exchanger, flows through the gap between the baffle and the module housing, and then exits from the centre tube in the opposite half of the module. This leads to a cross-flow arrangement between the solution and the fibres in both halves of the module.

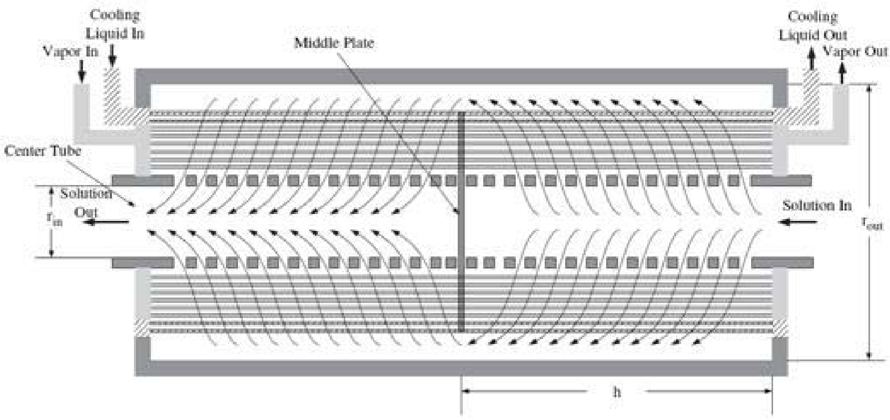

Figure 14 - Hollow fibre membrane absorber heat exchanger [12]

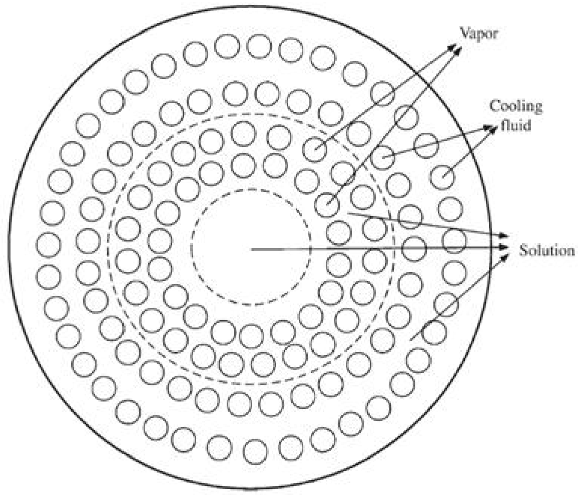

Figure 15 – Cross-section of a hollow fibre membrane absorber heat exchanger [12]

As shown in Figure 15, the vapour flows inside the porous fibres and the cooling fluid flows inside the nonporous fibres. In the given design, the cooling fluid and the absorption solution are counterflow (and some cross-flow), while the vapour and the absorption solution are countercurrent (and some cross-flow), but can also be in parallel.

Chen et al. [12] compared the hollow fibre membrane approach with the falling film mode (in a plate heat exchanger). By demanding the same absorption solution-cooling fluid heat transfer interfacial area for the two designs, the device volume of the hollow fibre membrane heat exchanger was 31 % of that of its falling film counterpart. The vapour-absorption solution mass transfer interfacial area of the hollow fibre membrane approach was 4.3 times that of the falling film approach. In addition, the total amount of ammonia absorbed by the hollow fibre membrane heat exchanger was almost 2 times of that of the plate heat exchanger falling film absorber.

Compared to the plate heat exchanger falling film absorber, the hollow fibre membrane approach has generally lower transfer coefficients. The improved absorption performance for the hollow fibre membrane absorber is related to the significant larger mass transfer interfacial area. Although the performance of using a hollow fibre membrane absorber exchanger has been thoroughly analysed by Chen et al. [12], further confirmation of the feasibility of the hollow fibre membrane approach by experimental study is needed.

4.6 Shell and tube vertical tubular absorber

A tubular absorber approach is another possible design for the absorber heat exchanger. Such an absorber consists of a tube bundle in a shell (a so-called shell and tube heat exchanger). A possible design is seen in Figure 16. Here, the ammonia vapour and the liquid solution are distributed at the bottom of the absorber and circulate co-currently upwards within the tubes. The vapour is distributed into the tubes through small diameter nozzles. The liquid solution enters the tube through the free area between tubes and nozzles. The absorption process progresses upwards inside the tubes as the vapour and the liquid come together. As a result, a strong liquid solution is removed at the top of the absorber.

As seen in the Figure 16, numerous baffles are introduced transverse to the tube bundle in order to drive the coolant, but also to stiffen up the absorber. The coolant flows downwards, external and transverse to the tube bundle, in a typical “shell and tube manner”. According to Fernández-Seara et al. [15], the hydrodynamics within the absorber is characterised by a changing two-phase flow. The mass flow of vapour and liquid change continuously until the absorption process is completed at the top of the absorber. The heat and mass transfer coefficients, the specific interfacial area, and the two-phase flow void fraction depend strongly on the hydrodynamics within the absorber. As a consequence, a detailed knowledge of the two-phase flow behaviour is required in order to analyse the performance of a tubular absorber.

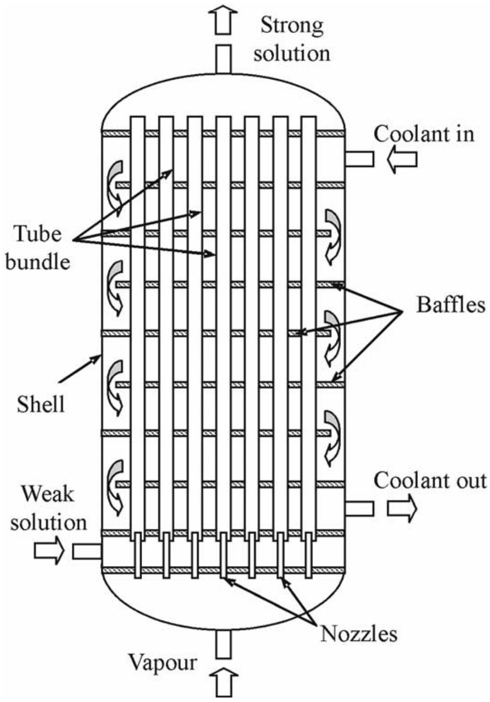

Figure 16 - Vertical tubular absorber [15]

Churn, slug, and bubbly flow patterns occur in vertical tubular absorbers with co-currently upward flow. The churn flow pattern is an entrance effect [16] due to the inlet nozzles. This causes an unstable flow region with a co-current upward flow of the liquid and vapour phases. The slug flow pattern can be described as the vapour phase rising as bullet shape bullets separated by slugs of liquid. The bubbly flow pattern is characterised by isolated spherical bubbles rising co-currently with the liquid. This flow pattern occurs at the top end of the absorber.

The vapour mass flow gives a good indication of the absorber performance. The absorption is considered complete when the vapour mass flow reaches a certain pre-defined value. The length from the bottom to the point where the absorption process is completed is defined as the absorption length.

In the vertical tubular absorber, the liquid and vapour temperatures are always lower than the interface temperature. The liquid temperature lies closer to the interface temperature than the vapour temperature. This indicates that the heat transfer resistance is dominant in the vapour phase. Through simulations, Fernández-Seara et al. [15] demonstrated that the increase in the bulk liquid concentration is larger in churn and slug flows than in bubbly flows. In addition, the total molar flux and the ammonia molar flux are always from the vapour to the liquid phase.

4.7 Helical coil geometry

A falling film helical coil absorber is another possible design for the absorber heat exchanger. An example of such a design is given in Figure 17. In this design, the coolant flows inside the coil from the bottom to the top of the heat exchanger. The solution enters at the top, and is being distributed by numerous nozzles attached at the top of the exchanger. In the design shown in the figure, the vapour can enter through the top or the bottom of the heat exchanger. The solution flow is always downwards. As a consequence, a counterflow is obtained if the vapour is supplied at the bottom of the heat exchanger. On the other hand, a parallel flow is formed if the vapour is supplied from the top of the heat exchanger. In the current design, an inner cylinder is installed in order to increase the vapour velocity.

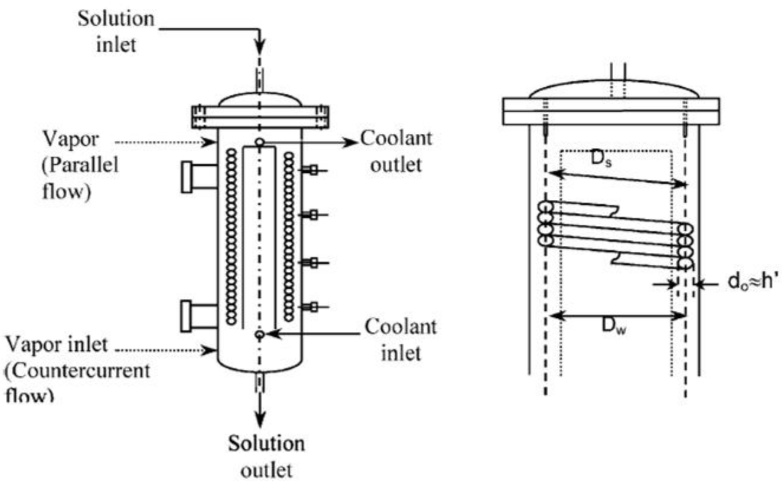

Figure 17 – Helical coil heat exchanger [17]

Results obtained by Jeong et al. [17] indicate that the total and sensible heat transfer rate for the parallel flow is slightly higher than that for the counterflow. The differences in heat transfer between the two flow approaches become clearer when the solution concentration of ammonia decreases. The effect of the vapour flow direction on the heat transfer therefore increases with decreasing solution concentration. The differences between the heat transfer coefficient for parallel and counterflow decrease with increasing concentration of ammonia solution. According to experiments performed by Jeong et al. [17], the mixing of the solution and the wetting of the coils improve as the solution flow rate increases. As a consequence, the Nusselt number increases with increasing solution flow rate. It was also found that the interfacial shear stress had a significant effect on the heat transfer for the counterflow arrangement.

4.8 Falling film heat transfer over horizontal tubes

Garimella [18] proposed a novel, miniaturised absorber design resulting in substantial size reduction over conventional geometries. The absorber consists of a stacked lattice-style arrangement where the coolant flows in tubes of very small diameter. These tubes are arranged in a horizontal plane to form a square array (see Figure 18). These arrays are then placed one above the other in such a way that the tubes in each array are perpendicular to those in the adjacent arrays. A distribution structure holds the tubes in the desired arrangement and distributes the coolant. The pressure drop considerations will determine the exact pass arrangement of the coolant. The liquid solution enters at the top and is distributed as droplets to the top array of the horizontal tubes. The liquid proceeds to the bottom of the absorber by alternating film and droplet flow. The vapour flow enters at the bottom of the absorber, and consequently, a counterflow arrangement with the liquid solution is achieved. The coolant also starts at the bottom and flows more or less in counterflow to the falling liquid solution.

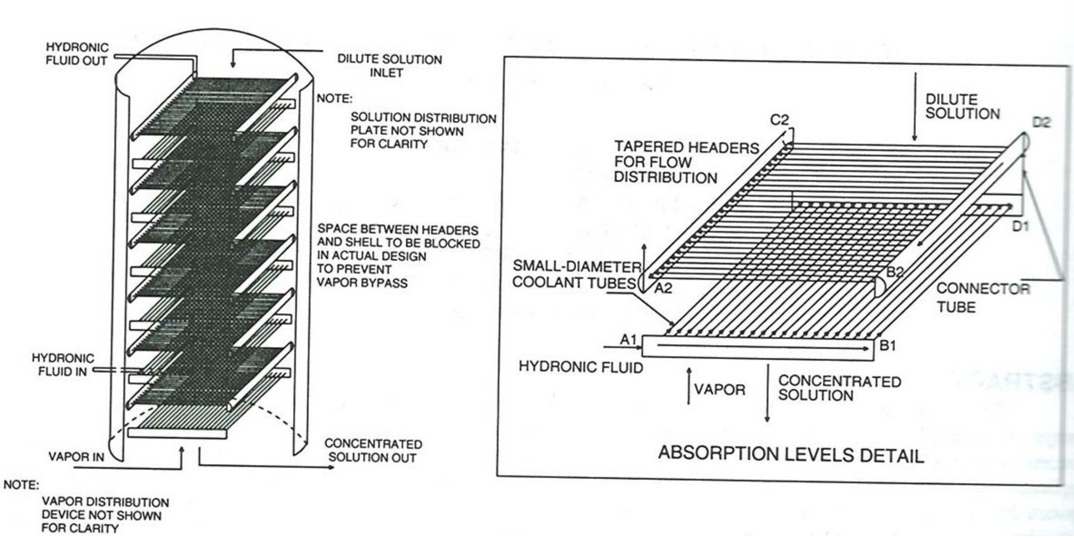

Figure 18 – Binary-fluid heat and mass exchanger (schematic)

Garimella [18] developed a model of the absorber in order to predict its overall performance. In addition, the model can be used to obtain information regarding local conditions and fluxes within the absorber for one specific design layout at typical operating conditions. According to results obtained by using the model, the heat transfer resistance in the coolant is very low. This is due to very small tube-side hydraulic diameters. In the main part of the absorber, a small amount of water is desorbed as ammonia is absorbed. Near the top of the absorber, a larger rate of water absorption occurs. This is due to the reduced ammonia concentration in the vapour and the larger temperature difference.

4.9 Membrane utilised in a compact absorber

A compact thermally driven absorption chiller with membrane technology has been investigated by Ali et al. [19]. Although the working mixture is lithium bromide and water, the results from the study are relevant for other mixtures. The main focus of the study was to investigate the factors that influence the water vapour transfer flux into a lithium bromide-water solution in confined narrow channels under vacuum conditions, as well as the property limits for utilisation in compact absorber design.

As discussed before in the report, the absorber is one of the major components of an absorption chiller, and has therefore direct effect on the size of the whole unit. By using polymeric, hydrophobic, microporous membranes in the absorber, a highly compact design can be achieved. In the current case, the membrane pores are filled with water vapour (refrigerant) while the hydrophobic nature of the membrane prevents penetration of an aqueous solution into the pores. Therefore, only water vapour is transported through the membrane into the aqueous solution. By using the membrane, the mass transfer area per unit volume is increased at the refrigerant side (due to narrowly confined aqueous solution flow channels). In addition, favourable mass transfer at the liquid vapour interface by forced convection in narrow confined channels can lead to a reduction of the absorber unit size and weight.

The aim of the study by Ali et al. [19] was to investigate experimentally and analytically the characteristics and properties of commercially available microporous, hydrophobic membranes. The purpose was to study the influence on the water vapour (refrigerant) mass transfer flux into thin films of aqueous lithium bromide-water solution, as well as the limits for utilisation in compact absorber design for absorption chillers. Ali et al. [19] carried out experimental measurements and developed a mathematical model for heat and mass transfer of water vapour transfer into aqueous solution through hydrophobic, microporous membrane sheets.

Regarding the purpose of the study, the following conclusions were made:

- Necessary with high permeability to water vapour and hydrophobic to the aqueous solution

- No capillary condensation of water vapour should occur inside the membrane pores to avoid pore blocking

- A high liquid inlet pressure was needed in order to avoid wettability of membrane pores by the aqueous solution

- A certain membrane layer thickness was needed as a compromise between required mechanical stability and low resistance to water vapour transfer

5 Concluding remarks

Compact heat exchangers have reduced volume and weight compared to conventional heat exchangers, and they often offer reduced cost and improved performance. Today, compact heat exchangers are becoming increasingly important elements in many industrial processes, usually in order to increase the energy efficiency. As described in chapter 3, numerous types of compact heat exchangers exist today, and they are classified according to the kinds of compact elements that they employ.

As discussed in chapter 4, several design solutions exist for the absorber heat exchanger. Especially three of the discussed designs seem very suitable for use in e.g. the absorber of an absorption cycle (absorption heat pump, absorption chiller, heat transformer, compression/absorption heat pump). This include the plate heat exchanger with falling film type heat transfer, the plate heat exchanger with bubbly type heat transfer, and a heat exchanger utilising a hybrid hollow fibre membrane. The different design solutions are among others chosen due to good results for similar applications, that is, related to stability of operation and effectiveness.

The plate heat exchanger with a falling film type heat transfer is a common design for the absorber heat exchanger. The advantages with this design are relatively high heat transfer coefficients and stable operation. With this design, good liquid distributors are necessary at the liquid flow inlet in order to obtain satisfactory wettability. The liquid and vapour are usually completely separated in the falling film mode. In addition, the heat and mass transfer areas are constant through the entire length of the absorber heat exchanger.

A plate heat exchanger with a bubble type heat transfer is an alternative design for the absorber heat exchanger. The advantages with this design are high heat transfer coefficients, good wettability and mixing between the liquid and the vapour. Bubble type heat transfer requires vapour distribution instead of liquid distribution to obtain a good mixing rate between liquid and vapour in the exchanger. An additional advantage with the bubble mode, when compared to the falling film mode, is that the dimensions of the heat exchanger will be smaller. The reason is mostly due to a varying mass transfer area in the bubble mode. A challenge with this design is stability and flooding problems.

A hollow fibre membrane absorber heat exchanger is another alternative design of the absorber heat exchanger. Such a design has some great advantages, including large and constant interfacial area, reduced problems related to two-phase flows, and scale-up advantages. Compared to a falling film mode plate heat exchanger with similar performance, the device volume of the hollow fibre membrane heat exchanger is considerable smaller. The hollow fibre membrane approach has generally lower transfer coefficients but significantly larger mass transfer interfacial area than the falling film approach. One uncertainty with the hollow fibre membrane approach is the lack of experimental results, and therefore its suitability for use in e.g. a hybrid heat pump needs to be validated.

Different fin surfaces can also be used in order to enhance the performance of a heat exchanger. The offset strip fin is one example of a suitable design with high performance. These fins have a high degree of surface compactness, and heat transfer enhancement is obtained due to complex flow patterns (e.g. boundary layers and wakes). However, an increase in pressure drop is experienced. Wavy fins can also be used in order to enhance the performance of a heat exchanger. These fins are comparable in performance to the offset strip fin. A third promising alternative for enhancing the performance of the absorber heat exchanger is by using louvered fins. The performance of the louvered fin is comparable, and sometimes even better, to that of the offset strip fin. The high performance of the louvered fin is due to a longer flow length for mixing.

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