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**Effect of Flow Distribution
on Scale Formation
in Plate and Frame Heat Exchangers**

A Dissertation

Submitted in Partial Fulfilment of the Requirements

for the degree of

Doctor of Philosophy

by

Teresa Chea San Kho

Supervisor: Professor Hans M. Müller-Steinhagen

Co-Supervisor: Dr. David T. Hughes

Department of Chemical & Process Engineering

University of Surrey

Guildford

England

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ABSTRACT

The uncertainty of fouling behaviour in plate heat exchangers is one of the main reasons (if not the main reason), why this heat exchanger type is not more widely installed in the chemical and process industry, despite the obvious potential for cost and energy savings. Recent investigations by Bansal (1994), have shown that similar heat exchanger plates with different corrugation patterns near the fluid inlet and outlet regions, fouled at very different rates under identical process conditions. As different corrugation patterns give rise to different flow patterns, the differences observed by Bansal (1994) are attributed to the effects of flow distribution on fouling rates in the plate channels. Consequently, in the present investigation, the effects of flow distribution on the precipitation fouling of calcium sulphate were studied experimentally using Alfa Laval P01 plates (diagonal flow), Alfa Laval M3 plates (diagonal and side flow), and 2-mm-thick stainless-steel flat plates (diagonal flow).

Effects of solution flow velocity, temperature, concentration, port position, plate surface roughness and distributor design on fouling were investigated. For corrugated diagonal flow plates (P01 and M3), the fouling rate was found to increase with decreasing solution flow velocity, and increasing solution temperature and concentration. Not only is the initial solution concentration an important factor in the fouling process, but also the amount of solution concentration being topped up so as to maintain the initial concentration during a run, and the time of doing so, have been shown to have a dramatic effect on the fouling process. When compared with their side flow counterparts, diagonal flow plates exhibit slightly lower fouling resistance and pressure drop values.

For the custom-made flat plates, increased fouling was observed with increasing solution concentration and plate surface roughness. In order to obtain a more even distribution of flow within the plate channel, low-flow-velocity and stagnant zones were avoided by using different distributor designs to channel fluid from the high-velocity regions to the low-velocity ones. In doing so however, all the 5 distributor designs examined created additional regions of low flow velocity in their wake.

Visual observation of flow distribution and fouling deposition in corrugated and flat plate channels were performed using a corrugated and a flat transparent polycarbonate plate respectively. From these observations, it became obvious that the two conditions necessary for precipitation fouling of calcium sulphate to occur, are high temperature and low flow velocity.

A few fouling mitigation methods were investigated. It was found that the addition of NaNO_3 to calcium sulphate solution reduced CaSO_4 fouling significantly. Velocity surges and the injection of air at a high flow rate for a short time period into the flowing solution, proved capable of removing an existing fouling layer. However, continuous air injection into the solution line, and the use of ion-sputtered plates and a magnetic device all failed to achieve any significant decrease in fouling.

Different ways of incorporating excess heat transfer area in plate heat exchangers for fouling provision were evaluated. In the presence of adhesion-controlled fouling, the traditional method of adding extra parallel plates proved to be inferior when compared with the incorporation of larger plates with different aspect ratios and additional plates in a second pass. On the other hand, if no flow control is possible, a two pass arrangement and plates with half the standard plate width should be avoided for reaction fouling.

Flow simulations were performed using the commercial Computational Fluid Dynamics (CFD) package, CFX. A flow passing through a 2-D channel with a single corrugation, and with multiple corrugations, has been modelled and interesting results have been obtained. Special emphasis was placed on the local flow distribution in a plate channel. This information was essential to improve the design of heat exchanger plates to guarantee longer operational cycles. Simulations were performed for different distributor designs and different plate shapes. Computational results compared well with those obtained experimentally.

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LIST OF SYMBOLS

<i>Symbol</i>	<i>Definition</i>	<i>[S. I. Unit]</i>
a_p	plate corrugation amplitude	m
A	heat transfer area	m ²
b_p	plate channel gap	m
B	body force	N
c	constant	
C	concentration	kg/m ³
c_p	heat capacity	J/kg·K
C_μ	empirical constant	
d_h	hydraulic diameter	m
E	empirical constant	
E_{act}	activation energy	J/mol
f	friction factor	-
G_b	generation of turbulence due to body force	kg/m·s ³
G_k	rate of production of turbulent kinetic energy	kg/m·s ³
h	static enthalpy	J/kg
H	total enthalpy	J/kg
H_s	step height	m
J_ϕ	flux of ϕ	
k	turbulent kinetic energy	m ² /s ²
k_{rxn}	reaction rate constant	
l_p	plate length	m
$\dot{m}$	mass flux	kg/m ² s
$\dot{M}$	mass flow rate	kg/s
M_m	molecular mass	g/mol
n	number of moles	-
n_{rxn}	order of growth rate	-
Nu	Nusselt number	-
p	pressure	Pa

<i>Symbol</i>	Definition	[S. I. Unit]
Pr	Prandtl number	-
$\dot{Q}$	heat transfer rate	W
R	resistance	m ² K/W
Re	Reynolds number	-
R_f	fouling resistance	m ² K/W
R_g	universal gas constant	J/mol·K
s	thickness	m
S	source or sink term of ϕ	
Sc	Schmidt number	-
Sh	Sherwood number	-
t	time	s
T	temperature	K
$\Delta T_{\log}$	log mean temperature difference	-
u	nominal channel flow velocity	m/s
	or flow velocity in x direction	m/s
U	overall heat transfer coefficient	W/m ² K
U_{ie}	internal energy	J/kg
U_0	initial overall heat transfer coefficient	W/m ² K
v	flow velocity in y direction	m/s
V	volume	m ³
$\dot{V}$	volumetric flow rate	m ³ /s
w	flow velocity in z direction	m/s
w_p	plate width	m
x	magnitude of vector component in i direction	m
$\dot{x}$	flow quality	-
y	magnitude of vector component in j direction	m
	or distance to wall	m
y^+	dimensionless wall sublayer thickness	-
z	magnitude of vector component in k direction	m

<i>Greek Symbol</i>	Definition	[S. I. Unit]
α	film heat transfer coefficient	W/m ² K
β	mass transfer coefficient	m/s
β_p	plate corrugation angle	°
ε	dissipation rate of turbulent kinetic energy k	m ² /s ³
ε_{gl}	flow hold-up	-
ϕ	dependent variable	
Γ	diffusion coefficient	kg/m·s
κ	von Karman's constant = 0.42	-
λ	thermal conductivity	W/m·K
λ_p	plate corrugation wavelength	m
μ	dynamic or molecular viscosity	kg/m·s
μ_e	enlargement factor	-
μ_{eff}	effective viscosity	kg/m·s
μ_t	turbulent or eddy viscosity	kg/m·s
ν_t	kinematic turbulent viscosity	m ² /s
ρ	density	kg/m ³
σ	stress	N/m ²
σ_k	Prandtl number governing turbulent diffusion of k	-
σ_ε	Prandtl number governing turbulent diffusion of ε	-
τ	shear stress	N/m ²
ψ_f	deposit strength	

<i>Subscript</i>	Definition
<i>ex</i>	excess
<i>d</i>	deposition
<i>f</i>	foul or fouling
<i>h</i>	hydraulic
<i>p</i>	plate
<i>r</i>	removal
<i>rxn</i>	reaction

1. INTRODUCTION

1.1 *Project Definition and Goals*

Plate and frame heat exchangers, more commonly known as plate heat exchangers (PHEs) have higher overall heat transfer efficiency and compactness when compared with other members of the heat exchanger family, such as conventional shell and tube heat exchangers. In spite of this and their potential for cost and energy savings, the installation of PHEs is still not as common in the chemical and process industry as that of the latter. One of the main reasons, if not the main reason behind this, lies in the uncertainty of the performance and reliability of PHEs when fouling is present.

A survey for the New Zealand Ministry of Commerce in 1990, covering 2000 heat exchangers from approximately 200 companies showed that almost 90% of the heat exchangers involved suffered from some form of fouling (Steinhagen *et al.*, 1990). Since about 14% of the heat exchangers surveyed were of plate-and-frame type, understanding the mechanisms behind the formation of fouling deposits, and improving the present PHE design to reduce fouling are of great importance.

Recent investigations by Bansal (1994) have shown the importance of plate corrugation patterns called distributors, near the fluid inlet and outlet regions. Flow into plate channels is distributed by the distributors in its entrance region, and guided to a certain extent by those in the flow outlet region. Under clean conditions, an improved inlet and outlet zone corrugation design of the APV TRI side flow plates resulted in higher heat transfer and lower pressure drop when compared to those of Alfa Laval P01 diagonal flow plates. Following the commencement of fouling, flow distribution deteriorated resulting in considerably more fouling on the former plates than the latter ones. Under such conditions, differences of a factor of 2 in heat transfer reduction and a factor of 3 in pressure drop increase, have been observed for these two plate types. It was also confirmed that the low-flow-velocity zones of heat exchanger plates have a much higher tendency to foul than other plate areas. Therefore, it can be said that the

differences of the two plate types mentioned above, are attributed to the effects of flow distribution on fouling rates in the plate channels.

It is therefore important to investigate flow distribution effects on fouling rates both experimentally and numerically. Flow characteristics in PHEs are extremely complicated and hence, very interesting. The situation is even more fascinating in the presence of fouling which is a dynamic process. The formation of fouling deposits may cause partial blockage of flow channels, which in turn may alter the flow patterns completely. The increase in pressure drop brought about by the presence of fouling may become the limiting factor in PHE operation, rather than the associated degradation in heat transfer.

As prevention is considerably more economical than cure, the best way to minimise fouling-incurred costs, is to prevent *their* occurrence altogether. To do this, it is necessary to redesign the plates so that the presence of low-flow-velocity zones is avoided, and a more even flow distribution in the plate channels can prevail. In view of the importance of the plate distributors on flow distribution, it is therefore essential to perform fouling experiments with different distributor designs. Furthermore, computational fluid dynamics (CFD) simulations need to be validated via comparisons with experimental results, before the local flow distribution in plate channels with different distributor designs and/or geometries can be simulated with confidence. It is anticipated that modifications of the present distributor design, and an asymmetric plate geometry may be required.

1.2 Overview of Dissertation

A brief overview is given on PHEs, and on how flow characteristics in plate channels affect heat transmission and pressure drop. Also highlighted in this dissertation are the various fouling mechanisms present, with emphasis on how crystallisation fouling of calcium sulphate (CaSO_4) affects the performance of PHEs. Next, the experimental apparatus of this investigation is presented, and a discussion is given on the results obtained. A number of methods for incorporating excess heat transfer area in PHEs and

their effect on fouling were examined. Following this, CFD modelling of flow distribution in plate channels with different geometries using a commercial CFD package, are described. Modelling of different plate distributor designs is also considered. Finally, the conclusions of this dissertation are presented together with some useful suggestions for future work.

2. THEORY & LITERATURE REVIEW

In this chapter, theories on heat transfer and fouling are presented together with a review of the literature on PHEs under clean and fouled conditions.

2.1 Plate Heat Exchangers

The first commercially successful PHEs were developed by Dr. Richard Seligman, founder of the Aluminium Plant and Vessel Company Ltd., now known as APV, in 1923 in England (Bond, 1981) (Carlson, 1992) (Dummett, 1964) (Seligman, 1964). Although they were originally developed for the dairy industry to pasteurise milk, PHEs now have their numerous advantages enjoyed by many other industries. Nevertheless, the food industry continues as the main user of PHEs.

In subsequent sections, the construction and operation of PHEs are presented together with a brief summary of their advantages and disadvantages. The effects of plate design including corrugation angle, wavelength, and amplitude, plate channel gap, port location, and distributor section, on flow, heat transfer and pressure drop characteristics of PHEs are then reviewed. Parameters pertinent to two phase flow are also mentioned.

2.1.1 Construction and Operation

A typical PHE consists of a pack of corrugated metal plates, assembled between a frame plate and a pressure plate, and compressed by tightening bolts. The plates are positioned by an upper carrying bar and a lower guiding bar, both of which are fixed to the support column. Figure 2.1 shows an 'exploded' view of a PHE.

The corrugated metal plates form the heat transfer surface of the exchanger. Each plate has four portholes for passage of the two heat-exchanging fluids, and is sealed at the outer edges and around its ports by gaskets. Depending on its plate and gasket pattern,

a particular plate can be either side flow or diagonal flow. For the former, the inlet and outlet ports of the fluid flowing over the plate are on the same side as shown in Figure 2.1, whereas for a diagonal flow plate, the inlet and outlet ports are on diagonally opposite plate corners. Moreover, a PHE can have either a co-current or a counter-current flow arrangement.

Figure 2.1: An 'exploded' view of a plate heat exchanger.

Figure 2.2: Schematic diagram of a chevron plate showing the corrugation angle, β_p , wavelength, λ_p , and amplitude, a_p .

The plate corrugations serve to increase the effective plate surface area and to promote turbulence, thus enhancing the heat transfer efficiency of the exchanger. Although various plate corrugation patterns are available commercially, the chevron pattern is now the standard type used in industry. Illustrated schematically in Figure 2.2 is a chevron plate and its corrugation angle β_b , corrugation wavelength or pitch λ_p , and corrugation amplitude, a_p . When chevron plates are assembled alternately with the 'arrowhead' pattern pointing upwards and downwards, the number of support points along the plates is greatly increased. This in turn improves plate rigidity and strength. Although the cross-sectional area between two chevron plates in the flow direction is constant as shown in Figure 2.3, the continuous changes in shape and direction of the flow passage, force the fluid to take a 'corkscrew' or criss-crossing flow path around the contact points between plates. This induces a higher degree of turbulence, resulting in higher heat transfer coefficients of the exchanger. Chevron plates were used to obtain a large portion of the experimental data in this dissertation.

Figure 2.3: (a) Chevron troughs through contact points
(b) Chevron troughs between contact points

However, chevron plates do have one drawback due to their many contact points, which causes them to be more prone to blockage by solid particles than other plate types (Carlson, 1992) (Kumar, 1988). Nevertheless, the flexibility of PHEs has been greatly increased by the introduction of plates with differing chevron angles, allowing the exchanger to be tailored to meet the exact heat transfer and pressure drop specifications required by industries. The design of a heat exchanger is determined by its fouling and hygienic aspects, flow rates, heat duty, operating pressure and pressure drop. The relative importance of these parameters differs greatly for different industries. As a result, a large number of plate designs with specific advantages for different applications, has been developed.

2.1.2 Advantages and Disadvantages

PHEs are gaining popularity over other heat exchanger types in many applications where appropriate. This is due to their many advantages, of which some of the most important are given as follows (Bond, 1981) (Carlson, 1992):

- (i) high heat transfer
- (ii) compactness
- (iii) economy
- (iv) lower fouling tendency
- (v) accessibility
- (vi) flexibility

PHEs owe their high efficiency in heat transfer to their plate design which features the presence of corrugations. Intense mixing of the fluid is induced and as a result, turbulence, normally expected at higher flow velocities, is induced at lower flow velocities and hence, at lower Reynolds numbers, Re . A higher level of turbulence leads to higher heat transfer coefficients, and consequently to a lower heat transfer area requirement. For the same heat duty, their required heat transfer area is considerably lower than that required by shell and tube heat exchangers. This means that, generally less material is needed for the construction of PHEs when compared with other types of heat exchangers and hence, more economic savings for industries that use PHEs when appropriate.

In addition, PHEs are reported to have a lower tendency to foul (Cooper, *et al.*, 1980). This is mainly due to the higher pressure drop across the plate channels, which causes higher shear forces capable of reducing deposition rate, and enhancing removal rate of the deposits. Owing to its accessibility, all parts of a PHE can be easily disassembled, inspected, cleaned and replaced if necessary, and reassembled with minimal downtime. PHEs are highly flexible such that their thermal duty can be altered easily with the addition of extra plates to or removal of existing plates from the exchangers. Process changes can also be readily implemented by the selection of suitable plate designs. Moreover, PHEs have a unique capability of embracing more than one heating or

cooling duty within the same frame, an impossibility for shell and tube units (Dummett 1964).

However, in all great inventions of man, there exists some limitations whether great or small. Unfortunately, the PHE is no exception. Most of the disadvantages of PHEs are caused by the following:

- (a) thin plate walls
- (b) gaskets
- (c) narrow channel gap

The thin walls of the plates offer many advantages, but are vulnerable if the heat exchangers are used for erosive or very corrosive duties. The greatest disadvantage of PHEs comes from their gaskets which cause them to be unsuitable for highly corrosive fluids, and fluids with temperature and pressure exceeding certain limits. This problem however, has been overcome in the latest gasket-free PHEs with laser-welded plate packs, for example, Alfa Laval's AlfaRex heat exchangers.

Because of their small channel gap, blockage may occur more easily under fouling conditions in PHEs than in shell and tube heat exchangers. Once blocked, the PHE will suffer from a tremendous pressure drop. This is especially so, if it is blocked near the inlet or outlet ports of the plates. Furthermore, changing the thermal duties of PHEs by adding more plates could cause flow maldistribution between the plate channels, or such already existing flow maldistribution to deteriorate. This adversity is only important for PHEs with more than 20 plate channels (Heggs and Scheidat, 1992).

Due to ongoing research, for many applications, the numerous advantages of PHEs usually outweigh the few disadvantages they possess. As a result, PHEs have become very important in industries due to the financial savings they bring, especially in terms of longer operational cycles made possible by their subdued fouling processes.

2.1.3 Heat Transfer

Figure 2.4: Schematic temperature profile across a clean heat transfer wall.

When heat is transferred from one fluid to another in a heat exchanger, it travels through a heat transfer wall whose purpose is to prevent the two fluids from mixing. A schematic diagram of the temperature profile across a clean heat transfer wall, when fouling is absent is shown in Figure 2.4.

The governing equation for heat transfer is:

$$\dot{Q} = U \cdot A \cdot \Delta T_{log} \quad (2.1)$$

$\dot{Q}$ is the heat transfer rate or heat duty, and U the overall heat transfer coefficient. The heat transfer area, A is the sum of all the plate areas through which heat is transferred. The effective driving temperature difference is equivalent to the log mean temperature difference, ΔT_{log} . In Equation (2.1), there is the assumption that U is nearly constant over the whole heat transfer area.

The rate of heat loss by the hot fluid and the rate of heat gain by the cold fluid can be expressed as follows:

$$\dot{Q} = \dot{M}_{hot} \cdot c_{p,hot} \cdot (T_{hot,in} - T_{hot,out}) \quad (2.2a)$$

$$\dot{Q} = \dot{M}_{cold} \cdot c_{p,cold} \cdot (T_{cold,out} - T_{cold,in}) \quad (2.2b)$$

$\dot{M}$ is the mass flow rate, c_p the heat capacity, and T the temperature. Equations (2.2a) and (2.2b) are based on the assumptions that a steady state process prevails and no heat is lost from the heat exchanger to the atmosphere. These assumptions however, do not introduce a significant degree of inaccuracy to the calculation of heat transfer, because in the absence of fouling, heat transfer is a steady state process, and even after

the onset of fouling, the process only becomes quasi-steady. In addition, PHEs usually have negligible heat losses, and they generally require no insulation.

To describe heat transfer in PHEs, a review of the pertinent parameters, namely the overall heat transfer coefficient U , log mean temperature difference ΔT_{log} , hydraulic diameter d_h , and film heat transfer coefficient α , is given below.

2.1.3.1 Overall Heat Transfer Coefficient

The process of heat transfer between two moving fluids separated by a metal wall involves the following individual heat transfer processes:

- (a) convective heat transfer from the hot fluid to the wall
- (b) conductive heat transfer through the wall
- (c) convective heat transfer from the wall to the cold fluid

These processes can be expressed in terms of individual thermal resistances, and then combined to give the overall thermal resistance, $R_{overall}$. In the case of a clean heat exchanger, the following equation is valid:

$$R_{overall} = R_{convective,hot} + R_{conduction,wall} + R_{convective,cold} = \frac{1}{U} \quad (2.3)$$

Since $R_{overall}$ is equivalent to $1/U$ (inverse of the overall heat transfer coefficient), U can be written as:

$$U = \frac{1}{\left(\frac{1}{\alpha_{hot}} + \frac{s_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_{cold}} \right)} \quad (2.4)$$

α is the film heat transfer coefficient, λ_{wall} the thermal conductivity of the heat transfer wall, and s_{wall} the wall thickness.

2.1.3.2 Log Mean Temperature Difference

Since the temperature difference between the hot and cold fluids varies from one end of the heat exchanger to the other, a logarithmic mean temperature difference, ΔT_{log} has to be used. For a true counter-current flow arrangement of a hot fluid and a cold fluid, ΔT_{log} can be defined as:

$$\Delta T_{\log} = \frac{(T_{hot,in} - T_{cold,out}) - (T_{hot,out} - T_{cold,in})}{\ln \frac{(T_{hot,in} - T_{cold,out})}{(T_{hot,out} - T_{cold,in})}} \quad (2.5)$$

In a PHE, the outlet temperature of the cold fluid may be higher than that of the hot fluid. This is called a temperature cross. For this to happen, either a large area for heat transfer or high turbulence capable of increasing the heat transfer coefficients, is required.

2.1.3.3 Hydraulic Diameter

The hydraulic diameter, d_h is defined as the ratio of four times the void volume between the plates, to the wetted surface area between the plates. For flow between two parallel flat plates with length l_p , width w_p and plate channel gap b_p , d_h takes the following form:

$$d_h = \frac{4 \cdot l_p \cdot w_p \cdot b_p}{2 \cdot (w_p + b_p) \cdot l_p} \quad (2.6)$$

Since in most cases, the plate width is much greater than the plate spacing, $w_p \gg b_p$, Equation (2.6) can be reduced to:

$$d_h = 2b_p \quad (2.7)$$

It should be pointed out that the plate channel gap, b_p is defined as the ratio of the hold-up volume in a plate channel to the projected surface area of one of the plates. For chevron plates, it is equivalent to the corrugation depth or twice the corrugation amplitude, $2a_p$.

Depending on the corrugation parameters, the plate heat transfer area increases by amounts varying from 15 to 25% (Cooper and Usher, 1992). To express this increase, an enlargement factor, μ_e defined as the ratio of the developed area to the projected area is used. μ_e varies between 1.1 and 1.25, with 1.17 being a typical average (Edwards *et al.*, 1974) (Kumar, 1988). For corrugated plates, d_h is expressed as:

$$d_h = \frac{2 \cdot b_p}{\mu_e} \quad (2.8)$$

Both Equations (2.7) and (2.8) are commonly used in PHE literature. As μ_e is difficult to measure, and the effect of d_h on the heat transfer coefficient is small, Equation (2.8) and $\mu_e = 1.17$ are used in many cases.

2.1.3.4 Film Heat Transfer Coefficient

The film or individual heat transfer coefficients, α depend on:

- (a) *fluid flow velocity*: higher velocities promote greater turbulence which in turn enhances energy transport to and from heat transfer surfaces.
- (b) *fluid physical properties*: density ρ , heat capacity c_p , thermal conductivity λ and viscosity μ
- (c) *heat transfer mechanisms*: natural convection or forced convection

The theoretical film heat transfer coefficients for PHEs can be calculated from correlations similar to those used for tubular heat exchangers. A typical equation is shown below, where μ_{wall} is the dynamic viscosity of the fluid at the wall.

$$Nu = c_1 \cdot (Re)^{c_2} \cdot (Pr)^{c_3} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{c_4} \quad (2.9)$$

where:

$$\text{Nusselt number, } Nu = \frac{\alpha \cdot d_h}{\lambda} \quad (2.10)$$

$$\text{Reynolds number, } Re = \frac{\rho \cdot u \cdot d_h}{\mu} \quad (2.11)$$

$$\text{Prandtl number, } Pr = \left(\frac{c_p \cdot \mu}{\lambda} \right) \quad (2.12)$$

For convective heat transfer, the Nusselt number Nu depends on the Reynolds number Re , the Prandtl number Pr , and the ratio of the dynamic viscosity of the bulk fluid to the dynamic viscosity of the fluid at the wall μ/μ_{wall} . The latter accounts for the effects of wall superheat, and requires an estimation of the wall temperature, T_{wall} . With reference to Figure 2.4, T_{wall} can be estimated via Equations (2.13), (2.14a) and (2.14b).

$$\dot{Q} = \alpha_{hot} \cdot A \cdot (T_{hot,average} - T_{hot,wall}) = \alpha_{cold} \cdot A \cdot (T_{cold,wall} - T_{cold,average}) \quad (2.13)$$

$$\therefore T_{cold,wall} = T_{cold,ave} + \left(\frac{\dot{Q}}{\alpha_{cold} \cdot A} \right) \quad (2.14a)$$

$$\therefore T_{hot,wall} = T_{hot,ave} - \left(\frac{\dot{Q}}{\alpha_{hot} \cdot A} \right) \quad (2.14b)$$

The values of the constant c_1 and the exponents c_2 , c_3 and c_4 are dependent on flow characteristics and plate geometries, and need to be determined experimentally. These values may vary considerably as the flow and turbulence characteristics differ appreciably for different plate designs.

It should be pointed out at this stage that, as can be seen from Equation (2.9), α is primarily determined by the fluid physical properties, which are often not well defined and change considerably in transit through the exchanger. In addition, while this relationship is based on the nominal velocity estimated using the bulk flow rate, true local velocities in the plate channel may be higher by a maximum factor of about four due to the effect of corrugations (Marriott, 1971).

Various relationships similar to Equation (2.9) are available in literature for laminar and turbulent flows. The transition from laminar to turbulent flow in PHEs occurs at a critical Reynolds number of between 10 and 400. A summary of typical values for the constant and exponents in Equation (2.9) for turbulent flow in PHEs is given by Marriott (1971):

$$0.15 \leq c_1 \leq 0.40$$

$$0.65 \leq c_2 \leq 0.85$$

$$0.30 \leq c_3 \leq 0.45 \text{ (usually 0.333)}$$

$$0.05 \leq c_4 \leq 0.20$$

McKillop and Dunkley (1960) investigated the effect of channel geometry on the Reynolds number exponent for Re between 4000 and 30000:

$$Nu = c_1 \cdot Re^{c_2} \quad (2.15)$$

Their experiments showed that the constant c_1 increases with increasing irregularity of the channel, while the exponent c_2 decreases. The following results were reported:

flat plates	$c_2 = 0.8$
regular uniform channels	$c_2 = 0.8$
semi-uniform flow	$c_2 = 0.7$
channels converge and diverge in all directions	$c_2 = 0.67$

Some researchers who have used Alfa Laval P01 plates, the same plates as those used in this dissertation, have published their findings. Branch *et al.* (1991) found that heat transfer to Kraft black liquor with dissolved solids concentration ranging from 0% to 68%, could be correlated as:

$$Nu = 0.36 \cdot (Re)^{0.64} \cdot (Pr)^{0.333} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.17} \quad (2.17)$$

Using an Alfa Laval P01 PHE to study crystallisation fouling of $CaSO_4$, Bansal (1994) has reported:

$$Nu = 0.34 \cdot (Re)^{0.66} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.17} \quad (2.18)$$

The Nu correlation published by Branch *et al.* (1991) and Bansal (1994) agree very well with each other.

Clark (1974) gave the following equation for flat straight passages where $l_p \geq 60 d_h$:

$$Nu = 0.036 \cdot (Re)^{0.8} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} \cdot \left(\frac{d_h}{l_p} \right)^{0.054} \quad (2.16)$$

For laminar flow, the exponent c_2 in Equation (2.9) would have a value of about 1/3 instead of about 0.7 for turbulent flow (Edwards *et al.*, 1974).

2.1.4 Pressure Drop

Pressure drop is one of the main parameters to be considered in the design of any heat exchanger type. The total pressure drop for a fluid flowing through a PHE can be taken as the sum of:

- (i) *plate pressure drop*: pressure loss in plate channels
- (ii) *port pressure drop*: pressure loss in entry and exit ports

To minimise port pressure drop, it is desirable to provide portholes which are as large as possible, but consistent with the flow rates required. However, the size of the portholes is controlled by the plate aspect ratio, which is defined as the ratio of the plate length l_p to the plate width w_p . This is because for a given total plate area, the aspect ratio governs the width of the plate available to accommodate the ports (Dummett, 1964).

Due to its complex geometry, it is very difficult to calculate the exact pressure drop across a PHE. As a result, plate pressure drop is predicted using an expression which is similar for that of pipe flow:

$$\frac{\Delta p}{\Delta l} = f \cdot \frac{2 \cdot \rho \cdot u^2}{d_h} \quad (2.19)$$

Δp is the pressure drop across the exchanger, and Δl is taken to be the vertical distance between the inlet and outlet port centres. f is the Fanning friction factor which is a function of Re . For turbulent flow, f has the following expression:

$$f = \frac{c_5}{Re^{c_6}} \quad (2.20)$$

For laminar flow, $c_6 = 1$ and f should take the following form:

$$f = \frac{c_5}{Re} \quad (2.21)$$

Clearly, from Equations (2.19) to (2.21), f decreases with increasing Re . The values of c_5 and c_6 depend on the plate type and flow regime. For turbulent flow in smooth pipes, $c_5 = 0.079$ and $c_6 = 1/4$, whereas for laminar flow in pipes, $c_5 = 16$ and $c_6 = 1$ (Douglas *et al.*, 1979).

Price and Fattah (1978) reported that for an APV Junior Paraflow heat exchanger, the pressure drop over the heat transfer section contributed between 69% to 83% of the total pressure drop over the whole plate, including the port entrance and exit sections. The corresponding pressure drop contributions from the entrance and exit ports, ranged between 6% to 16% per port. This holds for both up-flow and down-flow channels for a Re range of 3 to 3000.

2.1.5 Plate Design

Flow distribution, heat transfer and pressure drop characteristics in a PHE are all dependent on the plate design. Thonon *et al.* (1995) reported that heat transfer enhancement of corrugated channels over flat channels can be up to 6 times. However, the pressure drop can increase by two orders of magnitude from smooth to corrugated channels. Hence, heat transfer enhancement brought about by the presence of corrugations is generally paid for by the corresponding pressure drop increase.

As shown in Figure 2.2, the three important corrugation parameters for chevron plates are:

- (i) corrugation inclination angle to the flow direction, β_p
- (ii) corrugation pitch, or also known as corrugation wavelength, λ_p
- (iii) corrugation amplitude, a_p

In addition to the above corrugation parameters, port location and the distributor section are also critical plate design parameters. Sections 2.1.5.1 to 2.1.5.6 serve to analyse the effects these design parameters have on the performance of corrugated channels, with sinusoidal wave profiles similar to those in chevron plate channels. Although the various parameters are looked at in isolation, one should bear in mind that altering one aspect of the plate geometry may influence changes in other parts. A selection of publications pertinent to this review, are summarised in Table 2.1. In some of these investigations, different mass transfer measuring techniques were performed to obtain heat transfer results via an analogy between the two transfer processes. With the exception of the results from Saniei and Dini (1993a, 1993b), Price and Fattah (1978), and plates with $\beta_p = 90^\circ$, all the plate channels included in Table 2.1, have surfaces in contact with each other. Plate channels with $\beta_p = 90^\circ$ resemble the one shown in Figure 2.3(b).

Source	Plate Type	β_p [°]	λ_p [mm]	a_p [mm]	b_p [mm]	λ_p/a_p	b_p/a_p	Medium	Heat Transfer	Pressure Drop	Flow Pattern
Present work	α Laval P01 & M3	60°	10.3	1.2	2.4	8.58	2.0	liquid	✓ overall	✓ overall	✓
Bord (1981)		25°, 30°, 40°, 45°, 60°				8.0		liquid	✓ overall	✓ overall	
Edwards <i>et al.</i> (1974)	APV Junior Paraflo	55°	7.7	1.01	2.03	7.59	2.0	liquid	✓ overall	✓ overall	
Focke <i>et al.</i> (1985)		0°, 30°, 45°, 60°, 72°, 80°, 90°	10.0	2.5	5.0	4.0	2.0	liquid	✓ overall	✓ overall	✓
Focke & Knibbe (1986)		0°, 45°, 80°, 90°	10.0	2.5	5.0	4.0	2.0	liquid			✓
Gaiser & Kottke (1989)		19°, 27°, 45°, 58°, 69°, 77°				3.56		gas	✓ local		✓
		69°				10.0		gas	✓ local & overall	✓ overall	✓
Gaiser & Kottke (1990)		69°		constant		3.56, 5.5, 7.45, 10.0, 14.25		gas	✓ local & overall	✓ overall	✓
		18°, 27°, 45°, 58°, 69°, 77°				3.56		gas	✓ local & overall	✓ overall	✓
Haseler <i>et al.</i> (1992)	APV SR36	40°	4.27	18.4	3.67	2.33	2.0	liquid	✓ local & overall		✓
Heavner <i>et al.</i> (1993)		23°, 23°/45°, 45°, 23°/90°, 45°/90°						liquid	✓ overall	✓ overall	

Table 2.1: Summary of publications on plate design.

Source	Plate Type	β_p [°]	λ_p [mm]	a_p [mm]	b_p [mm]	λ_p/a_p	b_p/a_p	Medium	Heat Transfer	Pressure Drop	Flow Pattern
Okada <i>et al.</i> (1968)		90°	23.0	3.75	3.47, 4.35, 7.28	6.13	0.93, 1.16, 1.94		✓ overall	✓ overall	
Okada <i>et al.</i> (1972)		30°, 45°, 60°, 75°	10.0	2.0	4.0	5.0	2.0	liquid	✓ overall	✓ overall	
		60°	8.0, 10.0, 15.0	2.0	4.0	4.0, 5.0, 7.5	2.0	liquid	✓ overall	✓ overall	
		90°	17.0	4.25	5.0, 7.3, 10.0	4.0	1.18, 1.72, 2.35	liquid	✓ overall	✓ overall	
Price & Fattah (1978)	replica of APV Junior Paraflow	55°	7.7	1.14	2.28, 2.86, 4.04	6.75	2.0, 2.5, 3.54	liquid		✓ local & overall	✓
Quarini <i>et al.</i> (1995)	APV Junior Paraflow	55°	7.7	1.01	2.03	7.59	2.0	liquid	✓ local & overall		
Rosenblad & Kullendorff (1975)		5°, 15°, 30°, 45°, 55°, 60°, 65°, 70°, 75°, 80°, 85°	10.0	1.6	3.2	6.25	2.0	gas	✓ overall		
Saniei & Dini (1993a)	in-phase & out-of-phase	90°	66.7	6.35	101.6, 70.6, 57.2	10.5	9.01, 11.11, 16.0	gas	✓ local & overall	✓ overall	
Saniei & Dini (1993b)	corrugation on bottom wall only	90°	66.7	6.35	101.6	10.5	16.0	gas	✓ local & overall		
Stasiek <i>et al.</i> (1996)		15°, 30°, 37.5°, 39.5°	25.0, 29.5, 32.0	4.0, 4.82, 5.68	8.0, 9.65, 11.35	4.44, 6.12, 8.0	2.0	gas	✓ local & overall	✓ overall	✓
Thonon <i>et al.</i> (1995)	sinusoidal & flat	15°, 30°, 45°, 60°, 90°							✓ local & overall	✓ overall	✓

Table 2.1: Summary of publications on plate design (continued).

2.1.5.1 Corrugation Angle

The corrugation angle, β_b is defined as the angle between the plate corrugations and the overall flow direction. Plates with β_b varying between 0° to 90° are available commercially. With reference to Figure 2.3(a), out-of-phase plates with $\beta_p = 0^\circ$ would have flow paths that are composed of sinusoidal ducts. On the other hand, if plates with $\beta_p = 90^\circ$ are in phase, they would form a wavy channel as shown in Figure 2.3(b). This angle, β_b is believed to be a major thermo-hydraulic parameter in determining the performance of PHEs (Focke *et al.*, 1985) (Martin, 1996).

From the literature surveyed, there is a general consensus that at a given Reynolds number Re , both the heat transfer coefficient α , and the friction factor f , would increase with increasing β_p (Bond, 1981) (Focke *et al.*, 1985) (Gaiser and Kottke, 1989) (Gaiser and Kottke, 1990) (Heavner *et al.*, 1993) (Okada *et al.*, 1972) (Rosenblad and Kullendorff, 1975) (Stasiek *et al.*, 1996) (Thonon *et al.*, 1995). Those who investigated the effects of β_p over a wider range, observed both the heat transfer and pressure drop to go through an overall maximum between $70^\circ < \beta_p < 90^\circ$ (Focke *et al.*, 1985) (Rosenblad and Kullendorff, 1975) (Thonon *et al.*, 1995). Their data also confirm that excessive increase in pressure drop only occurs for plates with $\beta_p > 60^\circ$.

Fluid flow phenomena in plate channels which are responsible for the above mentioned variations in heat transfer coefficient and friction factor, have been the subject of numerous investigations (Price and Fattah, 1978) (Focke *et al.*, 1985) (Focke and Knibbe, 1986) (Gaiser and Kottke, 1989, 1990). The flow patterns shown in Figures 2.5(a), (b) and (c) are extracted from Focke and Knibbe (1986). Full and broken lines indicate flow along furrows (between the plate corrugations) above and below the plane of this paper respectively.

For $\beta_b = 45^\circ$, fluid flows predominately along the furrows of the plate. Its flow is reflected at the side walls and then follows the furrows of the opposite plate, as depicted in Figure 2.5(a). As a result, there are two criss-crossing streams in a flow channel, one going from left to right on one plate, and the other going from right to left

on the opposite plate. Some interactions occur between these streams as they cross one another, inducing secondary swirling motions and enhancing the heat transfer process even more. Similar flow patterns are expected for $30^\circ < \beta_b < 60^\circ$.

Figure 2.5(a): Flow characteristics of chevron plates with 45° corrugation angle.

Figure 2.5(b): Flow characteristics of chevron plates with 80° corrugation angles.

Figure 2.5(c): Flow characteristics of chevron plates with 90° corrugation angles.

Instead of flowing along the furrows for the complete plate width, the fluid is reflected whenever it approaches a contact point for $\beta_b = 80^\circ$. It then flows on the opposite plate in the other direction until it reaches the next contact point. This results in a zigzag flow pattern, also known as the corkscrew pattern. Since both heat transfer and pressure drop are at their highest at this β_b , there must be intense mixing of the flow at the furrow junctions.

When $\beta_b = 90^\circ$ such that the corrugations are perpendicular to the main flow as illustrated in Figure 2.5(c), flow separation was observed to occur at relatively low flow rates. The separated flow region increases with increasing flow rate until the point of reattachment almost coincides with the top of the downstream crest. This slow recirculating regions cause heat transfer and pressure drop to decrease.

Since Focke and Knibbe (1986) performed the above visualisation experiments in custom-made test sections, effect of the spines or 'vees' in chevron plates was not looked into. This missing information was filled by Haseler *et al.* (1992), who used commercial APV SR36 plates of $\beta_b = 40^\circ$ to show that fluid flowing on one side of the plate 'vee' acts independently to flow on the other side. This observation is most likely to be caused by flow reflection at the 'vees'.

2.1.5.2 Corrugation Wavelength

The corrugation wavelength, λ_p , describes the plate corrugation separation, which in turn determines the degree of turbulent mixing within the channel. As λ_p is decreased, a corresponding increase in heat transfer was observed by Gaiser and Kottke (1990), and Stasiek *et al.* (1996). This may be explained by the fact that corrugated plates with shorter wavelengths have greater ratios of developed to projected area, such that the criss-crossing streams meet more often per unit projected area. Therefore, a higher degree of overall mixing, and a larger pressure drop can be expected.

However, Stasiek *et al.* (1996) also reported that as λ_p/a_p increases, the corresponding friction factor increases asymptotically. As their findings were based on the combined effect of λ_p and a_p , such that λ_p/a_p was increased by increasing λ_p and decreasing a_p simultaneously, the friction factor trend observed is most likely to be the result of complex interactions between λ_p and a_p . Nevertheless, in the range of λ_p/a_p investigated, not only heat transfer enhancement was achieved but also pressure drop reduction. This optimum heat transfer and pressure drop condition suggests the existence of an ideal plate design for each individual heat exchange case.

Contradictory results have been reported by Okada *et al.* (1972), claiming that for the same a_p , the heat transfer coefficient is independent of any changes in λ_p , while the associated pressure drop increase with decreasing λ_p .

In an attempt to explain these experimental differences, three schematic diagrams of plate channels with decreasing λ_p are illustrated in Figures 2.6(a), (b) and (c). These plates have $\beta_p = 90^\circ$ and a constant a_p , and are similar to washboard heat exchanger plates. Although most heat exchanger plates have β_p other than 90° , this angle was chosen for easier flow envisagement. The gentle wavy flow path depicted in Figure 2.6(a), does not induce any rigorous fluid mixing and consequently, heat transfer and pressure drop are low. As λ_p is decreased as shown in Figure 2.6(b), the periodicity of the corrugations become frequent enough to cause flow separation and intensify fluid mixing. This enhances the heat transfer process, but inevitably causes the pressure drop to increase as well. Nevertheless, this corrugation geometry most probably gives the optimum balance between heat transfer enhancement and pressure drop increase. However, when the corrugations are very close to one another as shown in Figure 2.6(c), the fluid has to twist and turn to flow through the channel. As a result, the heat transfer rate is further augmented together with the pressure drop. The change in flow pattern as λ_p increases, is also dependent on the actual a_p and channel gap b_p .

Figures 2.6(a), (b), (c): Schematic diagrams showing decreasing corrugation wavelength for plate channels of 90° corrugation angle.

The results reported by Okada *et al.* (1972) concerning the effects of λ_p on heat transfer and pressure drop, depend on whether the range investigated has caused the fluid flow to switch from one of the aforementioned modes to another. If the flow has not switched modes, of course no heat transfer enhancement or deterioration would be observed. Although the results of Gassier and Kottke (1990), Okada *et al.* (1972) and Stasiek *et al.* (1996) show that λ_p influences both heat transfer and pressure drop in PHEs, this influence is not as great as that of β_p discussed previously.

It is worth mentioning that, λ_p is related to the enhancement factor, μ_e which in turn affects d_h according to Equation (2.8). Therefore, changes in λ_p are usually accompanied by variations in d_h , which would cause f , Nu , Re , and the Sherwood number Sh to change as well. Likewise, changes in a_p and b_p also provoke changes in d_h . Taking all these into account, extra caution must be taken whenever interpreting results involving changes in d_h .

2.1.5.3 Corrugation Amplitude

In common with β_p , and λ_p , the corrugation amplitude a_p is also an important parameter in enhancing turbulence. Turbulence can be generated via eddy formation by increasing a_p . However, increasing a_p beyond a certain limit can result in excessive pressure loss. Changes in a_p are usually accompanied by a change in the plate channel gap, b_p . This is especially true with chevron plates, as a_p is equivalent to half of b_p . However, a_p is usually investigated in conjunction with other plate corrugation parameters, and no experimental data have been found in the literature on the sole effect of a_p on heat transfer and pressure drop.

2.1.5.4 Channel Gap

The plate channel gap, b_p increases as d_h increases, and vice versa according to Equation (2.8). In turn, d_h is related to Re through Equation (2.11). For a given nominal flow velocity in the channel, u and constant fluid properties, Re increases with increasing b_p . u is related to b_p via the following equations:

$$\dot{M} = \rho \cdot \dot{V} = \rho \cdot u \cdot (b_p \cdot w_p) \quad (2.22)$$

$$u = \left(\frac{\dot{M}}{\rho \cdot b_p \cdot w_p} \right) = \left[\frac{\dot{V}}{b_p \cdot w_p} \right] \quad (2.23)$$

Substituting Equations (2.23) and (2.8) into Equation (2.11) for Re would give:

$$Re = \frac{2 \cdot \dot{M}}{w_p \cdot \mu_e \cdot \mu} = \frac{2 \cdot \rho \cdot \dot{V}}{w_p \cdot \mu_e \cdot \mu} \quad (2.24)$$

Therefore, if the mass flow rate $\dot{M}$ or the volumetric flow rate $\dot{V}$ is kept constant instead of u , Re would be independent of b_p . Otherwise, for Re to remain constant, an increase in b_p must correspond to a decrease in u . However, it is usually not mentioned in the literature, whether u , $\dot{M}$ or $\dot{V}$ was kept constant as b_p was varied.

Okada *et al.* (1968) noticed that for the same Re , both the heat transfer coefficient and the pressure drop of plate channels with $\beta_p = 90^\circ$ decreased with increasing b_p . Their heat transfer findings are supported by Saniei and Dini (1993a) who found Nu to decrease with increasing b_p . However, contradictory results published by Okada *et al.* (1972), claim that as b_p increases Nu increases as well. According to Equations (2.8) and (2.10), it is also possible that the increase in Nu values reported by Okada *et al.* (1972), is largely due to the increase in b_p , rather than the increase in heat transfer coefficient.

The pressure drop trend reported by Okada *et al.* (1968) is supported by Okada *et al.* (1972) and Price and Fattah (1978). The latter found friction factor, f of an APV Junior Paraflow heat exchanger to decrease, as b_p is increased from a standard channel with peak to peak contact, to wider channels where the corrugation peaks on opposite plates do not abut.

Shown in Figures 2.7(a), (b) and (c) are schematic diagrams of flow channels with increasing b_p , where $\beta_p = 90^\circ$ and a_p is constant. Due to the narrowness of the channel depicted in Figure 2.7(a), energy from the heated wall does not have to travel far to reach the coldest part of the fluid. As a result, fluid flowing through this narrow channel gets heated up much quicker than that flowing through wider channels, even though there is a higher degree of fluid recirculation and mixing in the latter channels.

Figures 2.7 (a), (b), (c): Schematic diagrams showing increasing plate channel gap for channels of 90° corrugation angle.

As b_p is increased, the fluid begins to experience the waviness of the channel as shown in Figure 2.7(b). When this happens, mixing and hence turbulence is induced, but heat from the wall has to be transported over greater distances to heat the fluid. Increasing b_p further such that the ratio b_p/a_p becomes significantly large, the fluid once again sees the channel as a long straight one. This is illustrated in Figure 2.7(c) where the corrugations are unable to induce enough fluid mixing across the height of the channel. When this happens, heat transfer is expected to drop. With reference to Equations (2.8) and (2.19), pressure drop should decrease as b_p increases.

2.1.5.5 Port Location

Flow patterns in diagonal flow plates differ from those in side flow plates. These patterns are illustrated in Figures 2.8(a) and (b) for a PHE system with water as the heating medium, and a solution as the process fluid. The solid arrows and the broken arrows represent the solution and water flow respectively. The length of an arrow is an indication of the magnitude of the local flow velocity.

Figures 2.8(a) and (b): Flow patterns in (a) diagonal and (b) side flow plate heat exchangers.

For diagonal flow plates, zones of identical velocity overlap each other in the adjacent channels, as shown in Figure 2.8(a). The high velocity zones in the solution channel exchange heat with the high velocity zones in the water channel. In the same way, the low solution velocity zones exchange heat with the low water velocity zones. This ensures high heat transfer because the higher local film coefficients in the high velocity zones on one side of the plate, are not off-set by the lower local film coefficients in the low velocity zones on the other side. However, for the side flow plates, high velocity zones exchange heat with low velocity zones in adjacent channels.

Nevertheless, side flow plates are beginning to replace diagonal flow plates because the same plate can act as the right-side plate (inlet and outlet on the right side) as well as the left-side plate (inlet and outlet on the left side) via a 180° rotation. Conversely, two sets of diagonal flow plates are required to assemble a PHE.

Price and Fattah (1978) performed dye injection to visualise the local flow patterns in side flow APV Junior Paraflow plates. A sinusoidal wavy pattern similar to the zigzag corkscrew pattern is observed for Re between 3 and 46. It was found that there is a significant difference between up-flow and down-flow channels. This however, could be attributed to the sensitivity of this flow visualisation method on the location of the dye injection.

Haseler *et al.* (1992) performed local heat transfer measurements with side flow APV SR36 plates. Two distinct sets of temperature were recorded for a heated stream, with the lower set corresponding to the side of the plate containing the inlet and outlet ports. This temperature difference may have originated from some form of flow maldistribution across the plate width. With reference to Figure 2.8(b), the lower temperature plate side should be the left side for the solution. Since this side of the plate represents the shortest path between the two ports, the flow here is expected to be faster than that on the right side. For a given heat flux, the heat transfer coefficient should increase with increasing flow velocity. According to Equation (2.1), this increase in heat transfer coefficient should be balanced with a corresponding decrease in temperature change. As a result of its higher flow velocity, the left side of the plate should have a lower temperature profile. In addition, flow visualisation via bubble injection revealed that flows in the two plate halves behave independently of each other.

However, using a side flow APV Junior Paraflow heat exchanger for local bulk temperature measurements, Quarini *et al.* (1995) reported no variation between the left and right sides of the plates. This may be due to the narrow width of the APV Junior Paraflow plates compared to that of the APV SR36 plates used by Haseler *et al.* (1992), which are 0.053 and 0.350 m respectively. Moreover, it is inconceivable that the presence of the thermocouples do not distort the flow in the small PHE channels used. As a result, what might have been observed is a distorted flow pattern which is different from that in 'normal' plate channels.

2.1.5.6 Distributor Section

Figure 2.9: Heat exchanger plate showing distribution triangle.

One of the most important factors affecting the performance of PHEs is the design of the corrugations or distributors near the inlet and outlet ports. As previously mentioned, these distributors determine to a certain extent, how well fluid distributes over the whole plate. The distribution regions are also important because they are mechanically the weakest region of the plate (Cooper and Usher, 1992). In addition, a significant percentage of the fluid pressure drop across the plate takes place in this region (Price and Fattah, 1978).

Haseler et al. (1992) analysed the flow and heat transfer characteristics in the distributor region of a PHE, refer Figure 2.9. From the inlet port, the flow width of the distribution

triangle increases, leading to significant changes in the mass flux and a mean flux higher than that in other plate regions. Although μ_e for the distributor section is less than that of the actual heat transfer section, the less effective heat transfer characteristics in the distributor section may be compensated for by the higher prevailing mass flux.

2.1.6 Two Phase Flow

For two phase flow of gas and liquid, the definition of two new terms, namely flow quality $\dot{x}$ and hold-up ϵ_{gl} becomes necessary. The former is defined as the ratio of the gas mass flow rate to the total mass flow rate:

$$\dot{x} = \frac{\dot{m}_g}{(\dot{m}_g + \dot{m}_l)} \quad (2.25)$$

The hold up, ϵ_{gl} is defined as the ratio of the gas volume to the total volume. However, in this dissertation, ϵ_{gl} is defined as follows:

$$\epsilon_{gl} = \frac{\dot{V}_g}{(\dot{V}_g + \dot{V}_l)} \quad (2.26)$$

$\dot{V}$ is the volumetric flow rate.

As liquid density usually does not vary much with slight fluctuations in temperature and pressure, its value can easily be looked up from the literature. This is generally not the case for the gas density. Nevertheless, the gas density can be calculated, provided that the prevailing temperature and pressure are known. This can be done by applying the ideal gas law, and using the known gas density value at standard temperature and pressure. Rearranging and substituting $\rho = (n \cdot M_m) / V$ into the ideal gas law, where n is the number of moles and M_m the molecular mass, the following equation can be obtained:

$$\left(\frac{p}{\rho \cdot T} \right)_1 = \left(\frac{p}{\rho \cdot T} \right)_2 \quad (2.27)$$

2.2 Fouling

Originally a descriptive expression used in the oil industry, the term 'fouling' has established itself in the literature to mean any unwanted deposition of materials, initially suspended and/or dissolved in fluids, onto heat transfer surfaces (Taborek *et al.*, 1972a). This undesirable process remains one of the major unresolved problems in heat exchanger design and operation.

Fouling is despised by many as it is responsible for additional capital cost, additional maintenance costs, higher energy requirement of heat exchangers, and production losses in industry (Epstein, 1983b) (Müller-Steinhagen, 1993a). Fouling has cost the industrial sector in some countries millions or even billions of US dollars each year (Steinhagen *et al.*, 1990, 1993). On the other hand, fouling is much loved by those who recognise it as a stimulating, thought-provoking problem waiting to be solved. As heat

exchanger fouling is an expensive inconvenience, even partially solving this costly problem will be a huge saving to industry. Total elimination of fouling is highly desirable but also impossible, so prevention of fouling may be a better alternative, or at present, the only alternative. To be able to prevent fouling, it is essential to understand how it occurs. Hence, there remains an urgent need to understand fouling mechanisms better.

This section describes the theory and literature of fouling and its effect on heat transfer and pressure drop. The five fouling mechanisms, five stages of fouling, and various fouling curves are also mentioned. In addition, the parameters influencing fouling, and various mitigation methods were looked into. It should be mentioned that attention has only been given to the formation of solid foulants at liquid-solid heat transfer interfaces.

2.2.1 Definition of Fouling Resistance

Figure 2.10: Schematic temperature profile across a fouled heat transfer surface.

Shown in Figure 2.10 is a schematic temperature profile across a fouled heat transfer surface. This is a simplified case where a fouling layer of thickness, s_f , with thermal conductivity, λ_f , is attached to one side of the wall only. This layer reduces the overall heat transfer coefficient, U as

follows (Epstein, 1983b) (Müller-Steinhagen, 1993a) (Müller-Steinhagen, 1993b):

$$\frac{1}{U} = \frac{1}{\alpha_{hot}} + \frac{s_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_{cold}} + R_f \quad (2.28)$$

R_f is called the fouling resistance and can be defined as:

$$R_f = \frac{1}{U} - \frac{1}{U_0} \quad (2.29)$$

U_0 is the clean overall heat transfer coefficient at time = 0, when there is no fouling. The ratio of U to U_0 is called the cleanliness factor, CF. This ratio gives an indication of how much U has decreased due to fouling.

$$\text{Cleanliness factor, CF} = \frac{U}{U_0} \quad (2.30)$$

An examination of Equation (2.28) will show that, for many typical heat transfer processes, the importance of further improvements in the art of predicting heat transfer coefficients will diminish unless accompanied by a corresponding increased understanding of fouling.

Service	Fouling Resistance R_f [$\text{m}^2\text{K/W}$]
Water:	
demineralised or distilled	1.7×10^{-6}
soft	3.4×10^{-6}
hard	8.6×10^{-6}
cooling tower (treated)	6.9×10^{-6}
sea (coastal) or estuary	8.6×10^{-6}
sea (ocean)	5.2×10^{-6}
river, canal, tube well, etc.	8.6×10^{-6}
engine jacket	1.03×10^{-5}
Steam	1.7×10^{-6}
Lubricating oils	$3.4 \times 10^{-6} - 8.6 \times 10^{-6}$
Vegetable oils	$3.4 \times 10^{-6} - 1.03 \times 10^{-5}$
Organic solvents	$1.7 \times 10^{-6} - 5.2 \times 10^{-6}$
General process fluids	$1.7 \times 10^{-6} - 1.03 \times 10^{-5}$

Table 2.2: Fouling resistances for plate heat exchangers
(Marriott, 1971).

Compared with the number of TEMA fouling resistances (TEMA, 1978) suggested for shell and tube heat exchangers, only a few design fouling resistances for PHEs have been reported. However, problems with fouling should not be overcome through excessive overdesign, because oversizing of PHEs may lead to lower flow velocities which in turn could increase fouling. Recommended fouling resistances for several process fluids in PHEs,

assuming operation at a pressure drop of about 30 kPa per Number of Transfer Units (NTU), are given in Table 2.2.

2.2.2 Effect of Fouling on Heat Transfer and Pressure Drop

In general, the presence of fouling can degrade the performance of operating heat exchangers in the following two ways:

- (i) The fouling deposit serves as an additional layer of resistance to heat transfer.
- (ii) The deposit causes pressure drop across the exchanger to increase.

2.2.2.1 Heat Transfer

Deposits	Thermal Conductivity λ [W/mK]
Biofilm	0.7
Ingredients of milk	0.5-0.7
Calcium carbonate	2.9
Calcium sulphate	2.3
Calcium phosphate	2.6
Magnesium phosphate	2.3
Magnetite	2.9
Hematite(boilers)	0.6
Calcite(boilers)	0.9
Gypsum(boilers)	1.3

Table 2.3: Thermal conductivities of some typical fouling deposits (Knudsen, 1984).

Due to their low thermal conductivity, fouling deposits have higher thermal resistances, called fouling resistances R_f , than those of the heat exchanger walls. For comparison purposes, the thermal conductivity of AISI 316 stainless steel, a common material PHEs are made of is 19.35 W/mK, while the thermal conductivities of some typical fouling deposits are given in Table 2.3.

In heat transfer operations, it is usual for either the heat flux or temperature of the heating medium to be kept constant. Temperature profiles across clean and fouled plates are shown in Figures 2.11 and 2.12 for constant heat flux and constant wall temperature respectively.

For constant-heat-flux heat exchangers, the interfacial temperature between the fouling deposit and the bulk fluid, T_{foul} remains constant with deposition. This is provided that

the film heat transfer coefficients remain unchanged, thus neglecting any surface roughness effects. However, the interfacial temperature between the wall and the deposit, $T_{wall, cold}$ will increase as fouling proceeds. As a result, part of the deposit will undergo additional crystallization ageing processes, which would affect the deposit strength, and thereby its removal rate (Suitor *et al.*, 1977).

Figure 2.11: Temperature profiles across plates with constant heat flux.

Figure 2.12: Temperature profiles across plates with constant wall temperature.

In applications where the heat transfer wall temperature, $T_{wall,hot}$ is maintained at a constant value, T_{foul} will decrease as fouling proceeds. Hence, with constant inlet parameters (flow velocity and temperature), the decrease in heat transfer coefficient due to fouling can be determined experimentally from exit temperature changes of the two heat-exchanging streams (Krause, 1993). A process called auto-retardation occurs in some cases, for instance precipitation fouling of inverse-solubility salts, whereby the decrease in T_{foul} brings about a corresponding decrease in the deposition rate.

2.2.2.2 Pressure Drop

The formation of fouling deposits on PHE walls leads to flow area reduction and increase in wall roughness, both of which are responsible for an increase in pressure drop. As a result, additional energy is usually required for pumping fluid through these narrow 'shrinking' flow channels.

Provided that fluid properties remain constant, such pressure drop increase can be caused by one or a combination of the following:

- (a) decrease in plate channel gap, b_p
- (b) increase in fluid velocity, u
- (c) increase in enlargement factor, μ_e due to increase in surface roughness

This is evident from Equation (2.31), which is obtained by substituting Equation (2.8) for d_h , into Equation (2.19) for frictional pressure drop:

$$\frac{\Delta p}{\Delta l} = f \cdot \frac{(\rho \cdot u^2)}{\left(\frac{b_p}{\mu_e}\right)} \quad (2.31)$$

An analysis on how a decrease in b_p affects the corresponding pressure drop can be made, by assuming that the fouling deposit is distributed evenly with a thickness of s_f over the plate channel walls. The resulting heat transfer rate across s_f can be written as:

$$\dot{Q} = -\lambda_f \cdot A \cdot \frac{\Delta T}{s_f} = \frac{1}{R_f} \cdot A \cdot \Delta T \quad (2.32)$$

Rearranging Equation (2.32) gives the following:

$$s_f = \lambda_f \cdot R_f \quad (2.33)$$

Since fouling usually occurs on both sides of a channel, the gap between the plates will be reduced by $2 \times s_f$. Therefore, if the fouling resistance, R_f and the thermal conductivity of the deposit, λ_f are known, the reduced fouled plate channel gap, b_{pf} can be calculated according to the following equation (Müller-Steinhagen, 1997):

$$b_{pf} = b_p - (2 \cdot s_f) = b_p - 2 \cdot \lambda_f \cdot R_f \quad (2.34)$$

Therefore, the pressure drop of a fouled PHE can be estimated by using b_{pf} instead of b_p in Equation (2.30):

$$\frac{\Delta p}{\Delta l} = f \cdot \frac{(\rho \cdot u^2)}{\left(\frac{b_{pf}}{\mu_e}\right)} \quad (2.35)$$

Changes in the fluid velocity as fouling proceeds also affect the pressure drop across the plate channel. The local velocity in a fouled channel is higher than that in a clean one, if the volumetric ($\dot{V}$) or mass ($\dot{M}$) flow rate is kept constant. These two flow rates are related to each other via Equation (2.22), which when substituted into Equation (2.35), gives the pressure drop in a fouled channel where the volumetric flow rate is constant:

$$\frac{\Delta p}{\Delta l} = f \cdot \frac{\left(\rho \cdot \left[\frac{\dot{V}}{b_{pf} \cdot w_p}\right]^2\right)}{\left(\frac{b_{pf}}{\mu_e}\right)} = \frac{f \cdot \rho \cdot \dot{V}^2 \cdot \mu_e}{b_{pf}^3 \cdot w_p^2} \quad (2.36)$$

The validity of Equations (2.35) and (2.36) is tested using a set of Bansal's (1994) data obtained without on-line filtration in Figure 2.13. Keeping u as a constant, the first equation deals with the effect changes in b_p have on pressure drop. The second equation keeps $\dot{V}$ constant, and is associated with changes in both b_p and u . The experimental data shown were acquired for solution and water inlet temperatures of about 51 and 87°C respectively. Flow velocity of the former and latter were 0.14 and 0.35 m/s respectively, while the CaSO_4 solution concentration was 3.0 g/l. The λ_f value

was taken from Table 2.3.

Figure 2.13: Effect of plate channel gap and flow velocity on pressure drop.

As can be seen in Figure 2.13, Equations (2.35) and (2.36) greatly underpredict fouling-related increase in pressure drop. This is because in reality, the deposit is not uniformly distributed over the whole heat transfer surface, but grows in localised areas of low flow velocity and high temperature (Bansal, 1994) (Cooper *et al.*, 1980). In addition, the deposits not only reduce the plate channel gap (from b_p to b_{pf}), but also reduce the effective width of the plates as well so w_p should be w_{pf} in Equation (2.23). Furthermore, this procedure for predicting pressure drop involves knowledge of the deposit thermal conductivity, λ_f which is difficult to estimate as it is time dependent.

In addition, it should be pointed out that the friction factor may also increase due to increase in surface roughness brought about by deposition. Corresponding to the cleanliness factor, CF for heat transfer, a similar factor called the pressure drop increase factor, PDIF is defined as the ratio of the instantaneous pressure drop, Δp , to the initial pressure drop at time = 0, Δp_0 , when the heat exchanger is clean. This ratio is particularly useful when comparing data with different initial pressure drops, for example, data with different flow velocities.

$$\text{Pressure drop increase factor, PDIF} = \frac{\Delta p}{\Delta p_0} \quad (2.37)$$

2.2.3 Fouling Mechanisms

In general, fouling mechanisms can be classified into the following six categories (Bott, 1990) (Epstein, 1981a) (Epstein, 1981b) (Epstein, 1983a) (Epstein, 1983b) (Garrett-Price *et al.*, 1985) (Müller-Steinhagen, 1993a):

- (i) precipitation fouling
- (ii) solidification fouling
- (iii) particulate fouling
- (iv) chemical reaction fouling
- (v) corrosion fouling
- (vi) biological fouling

(i) *Precipitation Fouling*

Precipitation fouling, or simply scaling, is the crystallisation of one or more dissolved inorganic salts from a solution onto heat transfer surfaces. This occurs when the dissolved substances in the fluid become supersaturated under the local process conditions. The solubility limit of fluids may be exceeded due to one or more of the following conditions:

- (a) evaporating solvents beyond the solubility limits of the dissolved salt or salts
- (b) supercooling normal solubility solutions below their solubility temperatures
- (c) heating solutions of inverse solubility salts, e.g. CaCO_3 , CaSO_4 , $\text{Ca}_3(\text{PO}_4)_2$, CaSiO_3 , $\text{Ca}(\text{OH})_2$, $\text{Mg}(\text{OH})_2$, MgSiO_3 , Li_2SO_4 , Li_2CO_3 , beyond their solubility temperatures.
- (d) mixing of different streams, and hence creating supersaturated conditions
- (e) changing the solution pH, for instance that of calcium carbonate solutions, and thus altering the solubility of carbon dioxide in water

(ii) *Solidification Fouling*

Solidification fouling occurs when the prevailing temperature falls below the melting

point of the fluid itself, or of one or more of its constituents, causing solids to form on heat transfer surfaces. Solidification fouling is also known as freezing fouling or may even be classified as a type of crystallisation fouling. All in all, the appropriate classification is very dependent on the specific fouling system.

(iii) Particulate Fouling

Particulate fouling is the deposition of finely suspended particles such as sand, dust, rust, onto heat transfer surfaces. In some cases, larger particles may settle due to gravity, resulting in sedimentation fouling. These larger particles may be formed as a result of agglomeration of smaller particles in suspension.

(iv) Chemical Reaction Fouling

Chemical reaction fouling or simply reaction fouling is the formation of a deposit onto the heat transfer surface via a chemical reaction. The heated surface does not participate in the reaction but may act as a catalyst.

(v) Corrosion Fouling

Corrosion fouling occurs when the heat exchanger material reacts with the fluid to form a thin oxide layer on its heat transfer surfaces. Unlike other fouling deposits, these oxide layers usually do not offer much resistance against heat transfer. This is because they are generally very thin and have a comparatively high thermal conductivity. However, these oxide layers have a tendency to change the structure of the heat transfer surfaces, which may promote other fouling mechanisms.

(vi) Biological Fouling

Biological fouling is the accumulation and growth of macro-organisms (algae, molluscs) and micro-organisms (bacteria) on heat transfer surfaces. Microbiological fouling poses an enormous problem to industries, as the microbes responsible for fouling can easily find their way into most systems, since they are present even in the air we breathe. They can cause problems in cooling towers since the temperature there ranges between 15°C to 50°C, which is almost ideal for microbial growth. Not only the live microorganisms but also the dead ones, can adhere to heat transfer surfaces and

form a layer of slime with low thermal conductivity comparable to that of water. Due to the filtering effect of biological deposits, there is a higher tendency for suspended particles in the fluid to amass onto these heat transfer surfaces.

With very few exceptions, fouling occurs preferentially on the side of the heated fluid. Fouling is not merely a function of the heat exchanging fluids, it is also dependent on the heat exchanger geometry and material, and the prevailing conditions. In practice, it is usual for two or more fouling mechanisms to occur sequentially or simultaneously as almost any pair of the above mentioned fouling mechanisms are synergistic, i.e. mutually reinforcing (Epstein, 1981a) (Epstein, 1981b) (Epstein, 1983a) (Epstein 1983b). This is particularly true of corrosion fouling in conjunction with any of the other mechanisms. However, in some unique cases, fouling mechanisms could be mutually weakening, for instance scaling accompanied by particle deposition tends to weaken an otherwise tenacious scale (Taborek *et al.*, 1972a). This is supported by Bansal (1994) who reported successful mitigation of CaSO_4 fouling with the addition of 1 μm alumina particles.

Based on a survey carried out in New Zealand (Steinhagen *et al.*, 1990), it was reported that the most common fouling mechanisms are corrosion fouling (45% out of the 2000 heat exchangers surveyed), followed by precipitation fouling (34%), particulate fouling (26%), and biological fouling (19%).

In this dissertation, precipitation fouling of calcium sulphate dihydrate, $\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$ (gypsum) is being studied. The inverse solubility property of CaSO_4 , together with the fact that calcium and sulphate are readily found in most water sources, make precipitation fouling of CaSO_4 an everyday problem for some industries.

2.2.4 Stages in Fouling

The different fouling mechanisms described above, usually proceed in five successive stages. They are described as follows (Epstein, 1983a) (Epstein, 1983b) (Garrett-Price *et al.*, 1985) (Müller-Steinhagen, 1993a):

- (i) initiation
- (ii) mass transport
- (iii) deposition
- (iv) removal and autoretardation
- (v) ageing

2.2.4.1 Initiation

Following the start-up of a clean heat exchanger, a certain period of time is required to elapse before any appreciable fouling is recorded. This period is known as the initiation period. Its duration varies depending on how conducive the surface structure of the heat exchanger and the prevailing conditions are, for the formation of deposit nucleation sites.

It has been frequently observed that immediately following the initiation period when fouling deposits start to grow, instead of decreasing, the heat transfer coefficient of the exchanger increases, resulting in a negative fouling resistance. This effect is caused by the changing flow characteristics near the heat transfer wall, whereby the deposits penetrate the laminar sublayer causing turbulence and hence the corresponding film heat transfer coefficient to increase. This process continues until the additional thermal resistance of the deposits outweighs the advantage of the increased turbulence. The period from the start of the process until the time at which the fouling resistance becomes zero again, is called the roughness delay time. Although this period is often observed, it is not present in every fouling situation.

The roughness delay time is the least reproducible and least predictable aspect of fouling for all categories. Its further study offers the prospect of developing optimum preventive measures against the initiation of fouling.

2.2.4.2 Mass transport

During the mass transport stage, the conditions required for deposition have been established, and the fouling material is being transported to the heat transfer surface. This transport process is usually driven by diffusion due to the concentration gradient

of the foulants or those of their constituents, between the fluid immediately adjacent to the heat transfer surface and the bulk of the fluid.

$$\dot{m}_d = \beta \cdot (C_{bulk} - C_{wall}) \quad (2.38)$$

The expression for the local deposition flux, $\dot{m}_d$ is given in Equation (2.38), where C_{bulk} and C_{wall} are concentrations pertinent to the fluid bulk and the heated wall respectively. The mass transfer coefficient, β can be determined for a given set of flow conditions from the corresponding correlations in the literature, provided that the diffusivity of the key component can be determined. Heat transfer relationships can be converted to mass transfer ones, by substituting Sherwood number Sh for Nusselt number Nu , and Schmidt number Sc for Prandtl number Pr .

2.2.4.3 Deposition

During deposition or attachment, the species responsible for fouling (which have already been transported to the heat transfer surface), either attach themselves to the heat transfer surface as in the case of particulate fouling, or react with each other to form a product or products which then deposit as a fouling layer.

In crystallisation processes, supersaturation is the driving force for deposition. In a solution of crystallising cations and anions of stoichiometric equality, the rate of deposition per unit surface area, $\dot{m}_d$ is commonly represented by the following expression:

$$\dot{m}_d = k_{rxn} \cdot (C_{wall} - C_{sat})^{n_{rxn}} \quad (2.39)$$

k_{rxn} is the reaction rate constant, C_{sat} the saturation concentration of the crystallising species at the wall temperature, and n_{rxn} the order of the growth rate. From a pool of experimental data from the literature, Konak (1974) noticed that most sparingly soluble salts grow under reaction-controlled conditions, and obey a second-order rate law at high supersaturations. This is supported by the findings of Amjad (1988) who studied the rate of $\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$ (gypsum) scale formation on heat exchanger surfaces from metastable supersaturated solutions.

The rate of deposition on the heat transfer surface is dependent upon the rate of the

diffusion-driven mass transfer to the surface, and the chemical reaction rate at the surface. To relate these two rates, Equation (2.38) is substituted into Equation (2.39) to give the following:

$$\dot{m}_d = \frac{(C_{bulk} - C_{sat})}{(1/\beta) + [1/\{k_{rxn} \cdot (C_{wall} - C_{sat})^{n-1}\}]} \quad (2.40)$$

If all fouling species that have diffused to the wall proximity, attach themselves there (e.g. crystal growth at sufficiently low fluid velocities), the rate of diffusion to the wall becomes the limiting factor for deposit formation. When this happens, deposition is said to be diffusion controlled, and the first term in the denominator of Equation (2.40) predominates over the second, giving Equation (2.41) which is identical to Equation (2.38):

$$\dot{m}_d = \beta \cdot (C_{bulk} - C_{sat}) \quad (2.41)$$

On the other hand, if the rate of chemical reaction at the surface is much slower than the diffusion rate (e.g. crystal growth at sufficiently high fluid velocities), then the deposition is reaction controlled. When this occurs, the second term predominates over the first in the denominator of Equation (2.40), and assuming that $C_{bulk} \approx C_{wall}$, the following can be deduced:

$$\dot{m}_d = k_{rxn} \cdot (C_{bulk} - C_{sat})^{n_{rxn}} \quad (2.42)$$

For all fouling categories, an Arrhenius type equation relates the reaction rate constant, k_{rxn} to the wall temperature T_{wall} :

$$k_{rxn} = c_7 \cdot e^{-E_{act}/(R_g \cdot T_{wall})} \quad (2.43)$$

c_7 is a constant, and E_{act} the activation energy.

2.2.4.4 Removal and Autoretardation

Removal of the fouling deposit can occur as soon as the deposit is formed. The removal rate is dependent on the deposit strength and the fluid shear forces. The removal process can occur via hydrodynamic forces causing erosion of the deposit, via thermal shock causing spalling of the deposit, and/or via chemical dissolution of the deposit. Moreover, it can be enhanced by randomly distributed turbulent bursts similar

to miniature tornadoes, which eject fluid out of the region adjacent to the wall into the main fluid stream (Cleaver and Yates, 1973, 1975, 1976).

Furthermore, the deposit may be affected by autoretardation mechanisms, such as the smoothing of a rough surface, giving rise to a thicker laminar sublayer of the fluid and reduced transport of the fouling species. A retardation of the deposition rate can also occur due to an associated temperature drop at the deposit-fluid interface under conditions of constant wall temperature. In addition, autoretardation can also be caused by deposit blockage which increases the scouring velocity, and hence suppresses deposition under constant mass flow rate conditions.

2.2.4.5 Ageing

Once the foulant is deposited, it undergoes an ageing process via chemical or physical changes, which may either strengthen or weaken the deposit. Under constant heat flux conditions, the temperature at the deposit-wall interface tends to increase, which may cause the strength of the deposit to change (refer Figure 2.11).

2.2.5 Fouling Curves

The net fouling deposition rate, $(dm_f)/(dt)$ is a combination of the deposition rate and the removal rate:

$$\frac{dm_f}{dt} = \dot{m}_d - \dot{m}_r \quad (2.44)$$

This model, originally proposed by Kern and Seaton (1959), has been widely accepted, with the recognition that the corresponding equations for the two mass fluxes, $\dot{m}_d$ and $\dot{m}_r$, are too simplistic and hence require further sophistication. As a result, various models are available in the literature (Epstein, 1983b) (Watkinson and Epstein, 1970) to describe these two fluxes specifically for each fouling mechanism. Depending on the operating conditions, different types of deposition and removal processes occur simultaneously, resulting in various forms of fouling curves. These curves are used to

illustrate the change of fouling resistance with time (Bott, 1995) (Epstein, 1981b) (Müller-Steinhagen, 1993a). Generally, there are four types of fouling curve and they are illustrated in Figure 2.14.

Figure 2.14: Schematic diagram showing the different types of fouling curves.

(i) *Linear Fouling Curve*

Linear fouling curves are obtained for deposits which not only have a strong coherence but also a strong adherence to the heat transfer surface. The removal rates of deposits which exhibit this kind of fouling curve are either negligible, or constant but slower than their corresponding deposition rates.

(ii) *Falling-Rate Fouling Curve*

Falling-rate fouling curves are typical for deposits of lower mechanical strength and for autoretardative deposits. With time, the former may suffer from an increase in the removal rate, while the latter may experience a deposition rate decrease, both resulting in a net deposition rate decrease.

(iii) *Asymptotic Fouling Curves*

The most commonly reported type of fouling curve is the asymptotic one. This trend is common for deposits where the removal rate increases with time and eventually becomes equal to the deposition rate. The removal rate increases with the fouling layer thickness, suggesting that the deposit strength is decreasing or other mechanisms deteriorating the stability of the layer are present. Hence, after a finite period of time,

the net deposition rate of the fouling process would become zero.

(iv) Sawtooth Curve

A sawtooth curve is obtained if relatively large chunks of the fouling deposit are detached from the heat transfer surface, whenever the deposit reaches a critical thickness or a critical residence time has elapsed.

2.2.6 Parameters Influencing Fouling in Plate Heat Exchangers

From an industrial view point, conditions affecting the impact of fouling on heat transfer or fluid flow can be classified as follows (Müller-Steinhagen, 1997):

- (i) operating parameters
- (ii) fluid properties
- (iii) plate parameters

2.2.6.1 Operating Parameters

Operating parameters such as flow velocity and temperature can be manipulated to mitigate fouling. This manipulation is only possible in the design stage because the operating parameters of a heat exchanger are usually predetermined by its required through-put.

2.2.6.1.1 Flow Velocity

An increase in flow velocity will cause wall shear stresses and hence the deposit removal rate to increase, resulting in lower fouling resistances. For most fouling mechanisms, the fouling rate decreases with increasing velocity. On the other hand, as velocity increases, the laminar sublayer close to the wall becomes thinner, and thereby generally reduces the resistance to diffusion from the bulk towards the wall. Although this promotes foulant transport to the heat transfer surface, the associated increase in shear forces may lead to an overall reduction of fouling for all but the most tenacious deposits. Hence, the effect of velocity on deposit removal is best characterised by wall shear stresses and the mechanical strength of the deposit.

It has been shown experimentally that $d(R_f)/dt$ in PHEs depends on the flow velocity in a way similar to that in shell and tube heat exchangers (Müller-Steinhagen, 1997). Hence, the following can be written for PHEs:

$$\frac{dR_f}{dt} \propto u^{-1.5} \quad (2.45)$$

This was confirmed by results of Bansal (1994), and those of this dissertation which are discussed in greater detail in Section 4.3.

2.2.6.1.2 Temperature

In heat exchanger fouling, the heat transfer surface temperature affects the solubility of dissolved substances, and the chemical reaction rate. For instance, precipitation fouling of inverse solubility salts will occur, if the temperatures near the wall rises above the solubility limit. Moreover, with reference to Equation (2.43), the corresponding reaction rate constant of such a process, increases exponentially with the wall temperature according to an Arrhenius type equation. In most cases, the effect of surface temperature on the reaction rate constant is more dominant than its effect on the solubility of dissolved salts. This leads to augmented deposition rates not only for inverse solubility salts, but also for normal solubility salts as long as the solution is supersaturated (Müller-Steinhagen, 1997).

Increasing the bulk fluid temperature will cause a corresponding increase in the wall temperature. For a given flow velocity and solute concentration, this will bring about an enhancement in the fouling rates as discussed above.

2.2.6.2 Fluid Properties

Both the nature of the fluid, and that of the species dissolved or entrained into the fluid, have a profound impact on fouling. Water quality is a key factor in fouling of cooling water systems. Salts with inverse solubility may lead to scaling. Suspended solids or ex situ corrosion products may settle onto heat transfer surfaces resulting in particulate fouling. Dissolved gases may induce corrosion of metal surfaces. The presence of micro-organisms and nutrients may lead to biological fouling. Regardless of what is fouling, the corresponding fouling rate usually increases with increasing foulant

concentration in the fluid bulk.

2.2.6.3 Plate Parameters

Plate parameters such as surface material, surface condition, and the actual plate design all have an influence on the resulting fouling behaviour.

2.2.6.3.1 Surface Material

In general, maximum adhesion occurs in systems experiencing a maximum decrease in surface energy (Walton, 1965, 1966). An example of this is, when a low energy fluid covers a high energy surface. Although this principle is developed for liquids on solids, Rankin and Adamson (1973) pointed out that it should be valid for solids upon solids as well. Interestingly, they discovered scale adhesion to be a strong function of surface energy, with adhesion increasing with substrate surface energy. They also confirmed experimentally that the heat transfer surface material has its greatest effect during the initial stage of fouling. Once the surface is completely covered by deposits, the original surface material will cease to be important.

Most recently, Müller-Steinhagen and Zhao (1997) demonstrated that scale formation can be greatly reduced if the metal surface energy is lowered by surface treatment such as Ion Beam Implantation or Magnetron Sputtering Ion Plating. The implantation of SiF_3^+ ions to a depth of between 0.2 to 0.5 μm , reduced calcium sulphate deposition by more than 50%.

2.2.6.3.2 Surface Condition

A rough surface possesses many projections and grooves. The former provide additional sites for nucleation, adsorption and chemical surface activity, while the latter provide regions for deposition that are sheltered from the mainstream velocity. Surface roughness also enhances the turbulence level near the wall, which facilitates transport of new material for deposition.

The initiation fouling period is not only a function of surface material, but that of

surface condition as well. A supporting observation was given by Watkinson and Epstein (1970), where an incompletely cleaned exchanger with already existing nucleation sites on the tube walls, fouled much faster than a properly cleaned one. In considering the possibility of high velocity flush-out as a method of fouling level reduction, Taborek *et al.* (1972b) discovered that during about an hour of flushing, fouling indeed decreased considerably. However, once the original velocity was restored, an extremely rapid increase in fouling was observed. They postulated that the crystalline structure which was not washed out, provided ideal nucleation sites for additional crystalline growth to occur. As depicted in Figure 2.15, the new asymptotic fouling level was higher than the probable original level without flush-out.

Figure 2.15: Fouling-time curve showing the effect of high velocity wash-out (Taborek *et al.*, 1972b).

2.2.6.3.3 Plate Design

A typical deposition pattern of CaSO_4 crystallisation fouling on a commercial Alfa Laval P01 plate, is shown in Figure 2.16 (Bansal, 1994). CaSO_4 solution was heated as it flowed from the bottom left plate corner to that of the top right.

Clearly, deposition is not uniform but concentrated in the corner opposite to the outlet port, and at the plate contact points. Depending on the prevailing conditions, a similar

pattern was sometimes found for the lower plate corner adjacent to the inlet port. However, the amount of deposit near the inlet of the plate was always less due to the lower wall and bulk temperatures. From Figure 2.16, it can be inferred that, in order to design for minimum fouling in PHEs, the flow across the plates should be as homogeneous as possible. The occurrence of fouling can be prevented by eliminating slow velocity zones in the plate channels.

Figure 2.16:
Typical CaSO_4
deposit pattern on
Alfa Laval P01 plate
(Bansal, 1994).

Due to the low local velocity and high local temperature around plate contact points, they play a key role in the initiation of fouling. The commencement of fouling at contact points in PHEs has been confirmed by Bansal (1994) and Delplace *et al.* (1994) for CaSO_4 and whey protein fouling respectively.

Substantial information is available in the literature regarding the influence of plate corrugation parameters on the performance of PHEs under clean conditions, but information under fouling conditions is scarce. As previously mentioned in Chapter 1, the investigations of Bansal (1994) confirmed the importance of the plate corrugation patterns near the fluid inlet and outlet regions. As plate corrugation angle affects the heat transfer, pressure drop and flow characteristics of a PHE, this angle must have an impact on the corresponding fouling rate. This is confirmed by Thonon and Grillot (1996) who investigated particulate fouling of CaCO_3 particles in two PHEs, one with 30° corrugation angle plates and the other with 60° plates. For flow velocities of 0.5 to 1.35 m/s, it was found that the asymptotic fouling resistances of the higher corrugation angle plates, were approximately four times

lower than those with the lower corrugation angle. Therefore, a high corrugation angle can lead to low fouling resistance in PHEs. As flow distribution together with heat

transfer characteristics determine the resultant fouling behaviour, the influence of different corrugation patterns on fouling is significant.

2.2.7 Fouling Mitigation

The most important steps in preventing the occurrence of fouling in heat exchangers are those taken in the design stage (Müller-Steinhagen, 1993a). They include:

- (i) the selection of an appropriate heat exchanger type
- (ii) efforts to ensure favourable operating conditions, e.g. flow velocity and heat transfer surface temperature
- (iii) optimum heat exchanger design

It is hardly surprising that the methods to mitigate fouling are focused on the parameters that influence fouling, which are discussed in the preceding section. As absolute prevention of fouling is yet to be achieved, numerous methods have been developed in recent years to control and mitigate fouling. They can be broadly classified into the more traditional chemical and mechanical mitigation methods, and new methods which do not fit into either of these two categories. However, all mitigation methods aim to reduce fouling via one of the following ways:

- (a) removing potential fouling precursors before their entry into the heat exchanger
- (b) increasing the solubilities of all potential foulants
- (c) weakening and removing fouling deposits

2.2.7.1 Removal Before Entry

The removal of one or more of the crucial species responsible for fouling prior to their entry into the heat exchanger, should reduce the overall fouling rate. For example, the installation of a filter or ion exchanger to remove certain fouling precursors.

2.2.7.2 Increasing Solubility

The addition of chemicals to keep fouling species in solution is one way of reducing fouling. However, this method may cause undesired product contamination or adverse

effects on the environment.

It is worth mentioning at this point, some fairly controversial devices currently available on the market, which claim to reduce fouling by magnetic treatment. Parkinson and Price (1984), Parsons *et al.* (1997) and Donaldson and Grimes (1988) reported that the installation of magnetic mitigation devices considerably reduced the deposition of CaCO_3 scales, whereas Hasson and Bramson (1985) and Sönel and Mullin (1988) found no effect under similar water conditions. Highly successful installations have been reported for an Australian UHT dairy plant (Magnets, 1991). However, laboratory measurements by Yoon and Lund (1994) did not show any beneficial effects on milk and calcium phosphate fouling in a PHE. In light of this evidence, it can be inferred that the influence of magnetic fields on deposition processes remains inconclusive.

2.2.7.3 *Weaken and Remove*

A number of mitigation methods have been developed to weaken and hence remove existing fouling deposits via mechanical means. In general, they are based on one of the following mechanisms (Müller-Steinhagen, 1993a, 1993b):

- (a) brief overheating of heat transfer surfaces, causing differential thermal expansion and hence deposit break-off
- (b) mechanical vibration of heat transfer surfaces
- (c) acoustic vibration of heat transfer surfaces
- (d) increased shear stress at solid-fluid phase boundary

An effective method to reduce fouling is to increase the wall shear stresses. Temporarily increasing flow velocity through the introduction of air at a high flow rate for a short time period resulted in the removal of a pre-existing fouling layer (Kho *et al.*, 1997). This is an effective and inexpensive mitigation technique provided that the deposits are not too tenacious.

It is known that single-salt solutions tend to form hard, tenacious deposits that are resistant to removal, as opposed to solutions containing salt mixtures which tend to

precipitate in an irregular pattern on heat transfer surfaces. The latter deposits are softer and less adherent. In addition, their irregular surfaces promote erosion brought about by the flow of the fluid (Garrett-Price *et al.*, 1985). Continuing with this line of reasoning, it should be possible to reduce fouling by promoting salt mixture precipitation instead of that of single salt.

Plate surfaces can be modified by reducing the surface roughness, and/or surface energy. This can be achieved by electropolishing and ion implanting the heat transfer surfaces respectively. The latter method for reducing fouling has been successful in developing low fouling heat transfer surfaces which still maintain or even improve the thermal and mechanical properties of the base materials. Another similar method is magnetron sputtering ion plating, which is a combination of magnetron sputtering and ion plating.

It should be borne in mind that each fouling process is unique and requires individual mitigation consideration. A very important aspect of the fouling mitigation methods proposed, is that they should be cost effective. In addition, perhaps its impact on the environment should be given conscientious consideration.

3. EXPERIMENTAL APPARATUS & PROCEDURE

In this chapter, the experimental apparatus and its recent modifications are described. Also mentioned are the data acquisition system used, and an error analysis of the variables. In addition, the preparation and concentration determination of CaSO_4 solution, and the experimental procedure are outlined.

3.1 *Experimental Apparatus*

The experimental set-up was originally built and used by Bansal (1994) for his Ph.D. dissertation on "Crystallisation Fouling in Plate Heat Exchangers", at the Department of Chemical and Materials Engineering, University of Auckland, New Zealand. For the present study, Bansal's set-up was disassembled and shipped to the United Kingdom, where it was rebuilt at the Department of Chemical and Process Engineering, University of Surrey, England. Numerous modifications were made to increase reliability and accuracy of the rig.

3.1.1 Flow Loop of Experimental Set-up

A photograph and a detailed flow loop of the set-up can be found in Figure 3.1 and Figure 3.2 respectively. Although there are two heat exchangers in the set-up, namely a PHE and a double pipe heat exchanger (DPHE), the former is the prime object of interest, with water as the heating medium and a process fluid of CaSO_4 solution. Both the water and the solution were circulated in closed flow loops. All components of the rig were made of stainless steel, except for some heat resistant polymer tubings and gaskets. Calibrated K-type thermocouples were used to measure all temperatures.

Figure 3.1: Photograph of the experimental set-up.

Figure 3.2: Detailed flow loop of the experimental set-up.

3.1.1.1 Calcium Sulphate Solution

The solution was stored in a 60-litre stainless steel tank, where its temperature was controlled automatically by a cooling-and-heating system. When the temperature exceeded the set-point, a control valve would increase the cooling water flow rate into the tank cooling coil. On the other hand, when the temperature dropped below the set-point, the heating pads attached to the outer tank surface would be switched on.

A centrifugal pump was used to circulate the solution through the rig. The solution flow rate was measured using a differential pressure transducer in conjunction with an orifice plate, and was automatically regulated by a pneumatic control valve which failed open. To simulate once-through industrial processes where solutions are not recycled, the solution was filtered, and hence broken crystals and impurities are removed prior to their entrance into the PHE. The filter was of 1- μ m pore size, 0.50 m long and made of polypropylene and polyethylene.

The pressure at the solution inlet of the PHE was measured by a mechanical pressure gauge, while the pressure drop across the exchanger was measured using an electronic pressure transducer. The solution leaving the PHE was fed into the annulus of the DPHE, after which samples could be taken before its re-entrance into the tank.

3.1.1.2 Hot Water

Water was stored in a 25-litre stainless steel tank where it was heated by three 3 kW immersion Domestic Aqualoy Heaters. The electrical circuitry and pipework of the tank are shown in Figure 3.3. The heaters were regulated by a controller via the water temperature and level in the tank. In order to maintain a constant water level in the tank, water was supplied to the tank via a solenoid valve, which was connected to a float switch in the tank. This set-up enabled the tank temperature to be controlled to within an accuracy of $\pm 1^{\circ}\text{C}$. Furthermore, having an overflow drainage system from the tank lid, not only kept the tank at atmospheric pressure, but also served as a precaution against tank overflow and overpressure.

Figure 3.3: Schematic of the hot water tank showing the electrical circuitry and pipework.

A centrifugal pump was used to circulate the water through the rig. The water flow rate was monitored by a magnetic flow meter, and regulated by a pneumatic flow control valve which failed closed. A rotameter was used to regulate hot water to the DPHE.

3.1.1.3 Compressed Air

The rig was also equipped with a compressed air supply for the two phase flow experiments. A rotameter and connections for a portable pressure transducer were installed. Compressed air was available from the main compressor unit, and this was throttled down to 80 psi on entering the rig air lines.

3.1.2 Plate Heat Exchanger

In this dissertation, Alfa Laval P01 and M3 chevron plates were used. Both plate types

Figure 3.4: Plate photographs showing
(a) P01 diagonal, (b) M3 side, (c) flat diagonal.

are identical in their design and dimensions, with the only difference being P01 is 0.6 mm thick while M3 has a thickness of 0.5 mm. Although Alfa Laval names their diagonal and side flow plates with 'P' and 'M' prefixes respectively, as symmetric plates, P01 and M3 can be either for diagonal or side flow depending on their fitted gaskets. Photographs of P01 diagonal, M3 side and flat diagonal flow plates can be seen in Figure 3.4. All plate types are made of AISI 316 stainless steel, and their dimensions and other relevant information are given in Table 3.1.

Plate type:	α Laval P01	α Laval M3	Flat
Corrugation angle, β_p	60°	60°	-
Corrugation amplitude, a_p	0.0012 m	0.0012 m	-
Corrugation wavelength, λ_p	0.0103 m	0.0103 m	-
Port diameter	0.0300 m	0.0300 m	0.0320 m
Plate length	0.4300 m	0.4300 m	0.4300 m
Plate width	0.1260 m	0.1260 m	0.1260 m
Effective plate length	0.3960 m	0.3960 m	0.3960 m
Effective plate width	0.1000 m	0.1000 m	0.1000 m
Nominal plate spacing, b_p	0.0024 m	0.0024 m	0.0024 m
Plate thickness	0.0006 m	0.0005 m	0.0020 m
Heat transfer area per plate	0.0320 m ²	0.0320 m ²	0.0270 m ²
Enlargement factor, μ_e	1.185	1.185	1.000
Average roughness on corrugation crest	0.530 μ m	0.590 μ m	-
Average roughness on corrugation trough	0.360 μ m	0.145 μ m	-

Table 3.1: Dimensions and relevant information of Alfa Laval P01, M3 and flat plates.

(Note: Effective sizes are sizes of heat transfer region.)

The PHE set-up was the simplest possible, with only three flow channels. This was done by using only 4 plates, of which two were end plates. The solution flowed in the middle channel between two channels of hot water. The flow was counter-current. A schematic diagram of the diagonal flow set-up can be seen in Figure 3.5. The only difference between this figure and one for a side flow set-up would be that, solution would exit from the top left ports, while water would enter via the top right ports.

Figure 3.5: Schematic diagram of the diagonal flow experimental set-up.

Figure 3.6:

Bolt tightening sequence for plate heat exchanger.

As mentioned in Section 2.1, a fixed PHE geometry is physically maintained by the plate contact points. Thus, a standard PHE channel possesses a unique $f-Re$ relationship. Non-abutting corrugations, incorrect assembly or distortion of plates will no doubt alter the PHE performance. The former effect on pressure drop has been investigated by Price and Fattah (1978) and is mentioned in Section 2.1.5.4. Since the plate thickness and amplitude are known, the required distance between the frame and pressure plates can be determined. The 8 bolts of the PHE were always tightened using a torque wrench with 40 - 50 Nm in the systematic sequence depicted in Figure 3.6.

Figure 3.7: Picture of experimental set-up with polycarbonate plate.

To improve knowledge on the dynamic growth behaviour of fouling deposits under different conditions, a corrugated polycarbonate plate with a metal support-frame was constructed for visual observation. Corrugations with dimensions similar to those of the P01 plates, were milled into one side of the polycarbonate plate with a programmable mill. It should be noted that the latter plate has corrugations which are rectangular rather than sinusoidal as in the Alfa Laval plates. The polycarbonate plate could be experimented with either P01 or M3 plates, for diagonal or side flow, and could withstand temperatures up to 120°C. A picture of this set-up is shown in Figure 3.7. As can be seen, only one side of the solution channel could be heated.

For enhancing the understanding of flow distribution in plate channels, some experiments were performed with flat plates. Apart from the absence of corrugations, the main difference between the Alfa Laval plates and the flat plates is that, the latter has a thickness of 2 mm for improved structural stability and rigidity. The dimensions of the flat plate are given in Table 3.1. Prior to most experiments, the flat plates were sanded with 240 emery paper in the main flow direction. This is to ensure that the surface condition was always identical at the start of each experiment. Using a Hommel Tester T4000 (Hommelwerke GmbH, Germany), plate roughness with different surface pretreatments, and in different directions were determined, see Table 3.2.

Flat Plate type:	Original	240 sanded	600 sanded
Average roughness in flow direction	0.130 mm	0.080 mm	0.030 mm
Average roughness in transverse direction	0.120 mm	0.110 mm	0.040 mm

Table 3.2: Roughness of flat plates with different pretreatments.

To investigate the role of the distributors adjacent to the plate inlet and outlet ports, on the resulting flow distribution between the plates, a series of different distributor designs were incorporated onto the flat plates. These distributors varied in length and shape, and were constructed from either the gasket material supplied by Alfa Laval or sheets of 3-mm-thick ethylene propylene rubber. Gasket materials were all glued onto the plates using a toughened single component epoxy adhesive (Permabond ESP110). For the glue to set, it has to be cured for 4 hours at 120°C. A flat polycarbonate plate was also manufactured for visualisation of the flow and fouling patterns in the flat plate channels.

3.1.3 Data Acquisition

A computer controlled data acquisition system, consisting of a Hewlett Packard PCIB interface, a HP 61013A digital multimeter (DMM) and two HP 61011A relay multiplexers (RELAY.MUX.01 and RELAY.MUX.02) was used to record all experimental data. The eight-input multiplexer had a temperature reference for thermocouple applications. Flow rates and temperatures of all streams entering and leaving the heat exchangers, pressure drop across the PHE and the solution concentration in the tank were measured and recorded throughout each experiment.

3.1.4 Error Analysis

The maximum error in the temperature measurements (type K thermocouples) was $\pm 0.75^\circ\text{C}$. Water and solution flow rate measurements all have a maximum error of $\pm 5\%$. A maximum error of $\pm 3\%$ was obtained in the pressure drop measurements across the PHE. The acquisition of a set of data required about a minute, resulting in a fouling resistance data fluctuation of approximately $\pm 2.5 \times 10^{-6} \text{ m}^2\text{K/W}$.

3.1.5 Modifications of Experimental Set-up

To further enhance the smooth operation of the rig and increase data reliability, some

parts of the original set-up (Bansal, 1994) were modified, improved and/or replaced. Some of the changes are listed below:

- (i) *PHE Solution Inlet Temperature:* Regulation of the PHE solution inlet temperature was achieved via the automatic control of the solution tank temperature, which was made possible by the installation a temperature controller, a cooling water control valve and heating pads.
- (ii) *PHE Hot Water Inlet Temperature:* Instead of having another PHE for the production of hot water with steam, as was done previously in New Zealand, an electrical heating system for the water tank was installed. This modification helped to stabilise the PHE hot water inlet temperature.
- (iii) *Data Acquisition Program:* The original computer was replaced, and the GW Basic data acquisition program was revised.
- (iv) *Air Supply System:* To perform two phase flow experiments, an air supply system for injecting air into the PHE solution inlet line was installed
- (v) *PHE Pressure Drop:* The differential pressure transducer which measured pressure drop across the PHE was replaced.
- (vi) *PHE Solution Inlet Flow Rate:* During a typical PHE fouling experiment, the effective plate channel gap decreased following the commencement of fouling, resulting in greater resistance against flow through the channel. As a result of this, the solution flow rate decreased because the pump was only operating to overcome a constant head. This problem used to be rectified by manually adjusting the valve from the solution tank to the PHE. As most fouling experiments in this dissertation have a duration of 7000 minutes (about 5 days), maintaining a constant solution flow rate manually was a tedious task. Consequently, a pneumatic flow control valve together with a flow controller were installed to regulate the PHE solution inlet flow rate automatically. The control valve was installed downstream of the

PHE differential pressure transducer and upstream of the PHE itself. This control valve was specified to fail open, because if it were to fail closed and heat was still being supplied to the stagnant solution in the plate channel, a severe build-up of fouling deposits would be inevitable.

(vii)*PHE Hot Water Inlet Flow Rate:* To maintain a stable PHE hot water inlet flow rate, the original rotameter was replaced by an electromagnetic flow meter together with a pneumatic flow control valve and a controller. In addition, distilled water was used instead of tap water in the hot water flow loop. This was to prevent the formation of hard water scales on the heating elements of the hot water tank.

(viii)*On-Line Monitoring of CaSO_4 Concentration:* The CaSO_4 concentration in the solution tank was monitored on-line with a carbon conductivity probe. This is discussed in greater detail in Section 3.2.

3.2 Calcium Sulphate

Calcium sulphate can exist in three different crystal forms depending on its water content. These three forms are gypsum ($\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$), calcium sulphate hemihydrate ($\text{CaSO}_4 \cdot 1/2\text{H}_2\text{O}$) and calcium sulphate anhydrite (CaSO_4). Their solubility curves can be seen in Figure 3.8. Gypsum starts to exhibit an inverse-solubility behaviour at temperatures above 30 to 40°C with a maximum solubility of 2.2 g/l (Gmelin, 1962) (Krause, 1993) (Linke, 1958) (Partridge and White, 1929).

To simplify the analysis of the experimental data, it was important to ensure that the fouling deposits comprised of a single crystal structure. Hence, experiments were carried out under conditions where the solution was unsaturated with respect to $\text{CaSO}_4 \cdot 1/2\text{H}_2\text{O}$, but supersaturated with respect to $\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$ and calcium sulphate anhydrite. However, the crystallisation of calcium sulphate anhydrite can be assumed to be negligible as it is very slow (Partridge and White, 1929). Consequently, only

$\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$ could crystallise out of the solution. Since the solution used was calcium sulphate and the deposit obtained was $\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$, for convenience both of these are simply referred to as ' CaSO_4 ' from now onwards.

Figure 3.8: Solubility curves for gypsum ($\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$), calcium sulphate hemihydrate ($\text{CaSO}_4 \cdot 1/2\text{H}_2\text{O}$) and calcium sulphate anhydrite (CaSO_4) (Gmelin, 1962).

3.2.1 Solution Preparation

As CaSO_4 is only sparingly soluble in water, it was difficult to prepare a solution of high concentration directly. If any undissolved CaSO_4 salts were left in the solution during an experiment, they could promote particulate fouling by providing extra nucleation sites for crystallisation to take place. To overcome this, the solution was prepared in an indirect way with more soluble salts, namely calcium nitrate tetrahydrate, $\text{Ca}(\text{NO}_3)_2 \cdot 4\text{H}_2\text{O}$, and sodium sulphate, Na_2SO_4 . The overall reaction for this can be written as:

To prepare the solution, the required amounts of sodium sulphate and calcium nitrate were first dissolved separately, each in 1 litre of distilled water. Then one after another,

they were added to the remaining water in the solution tank. Two extra grams of sodium sulphate were always added to be sure that there were enough sulphate ions to react with the calcium ions to form CaSO_4 , according to Equation (3.2).

Since the CaSO_4 concentration was determined by measuring the concentration of calcium, the latter should be the limiting factor in the solution rather than the sulphate concentration. The investigated CaSO_4 concentration range was between 2.8 and 3.2 g/l, which translates to 2800 and 3200 ppm, and to 0.0206 and 0.0235 mol/l, respectively.

It should be pointed out that the presence of Na^+ and NO_3^- ions in the solution could affect the crystallization behaviour of CaSO_4 to a certain extent. This is because these ions could alter the activity of the reacting ions. The addition of certain salts to an aqueous solution can either decrease or increase the resulting solubility causing a salt-out or salt-in effect respectively.

3.2.2 Concentration Determination

The concentration of CaSO_4 solution was determined by measuring the calcium (II) concentration using Ethylene-Diamine-Tetra-Acetate (EDTA) titration (Fritz and Schenk, 1987). The stoichiometric reaction is as follows:

The titrant (H_2Y^{2-}) is EDTA solution, and has a concentration of 0.01 mol/l. Eriochrome Black T and Calmagite were used as indicators. When all the free calcium ions have reacted with the EDTA molecules, a few extra drops of EDTA would initiate the end-point reaction. This would result in a colour change of the indicator from pink to blue:

Unfortunately, the end-point of this titration is not very sharp. As a result, a small amount of magnesium sulphate (MgSO_4) solution was added to the CaSO_4 solution. The stoichiometric reaction for magnesium (II) is the same as that for calcium (II), as shown in Equation (3.3). Since EDTA molecules do not distinguish between calcium (II) and magnesium (II), the added amount of magnesium (II) needs to be subtracted from the total EDTA consumption. Finally, the pH of the solution has to be adjusted by a buffer solution to a value of 10. Preparation methods for the various solutions necessary for titration are outlined in Appendix A.

To perform the titration, 25 ml of distilled water, 3 ml of the buffer solution, and 3 ml of the 0.01 M MgSO_4 solution were added to 25 ml of CaSO_4 solution. A generous amount of the indicator was added to the mixture giving a bright pink colour. This was essential to ensure a distinct end-point colour change. This solution was then titrated with 0.01 M EDTA solution until the expected colour change occurred. In calculating the CaSO_4 solution concentration, the amount of EDTA required for the 3 ml of MgSO_4 added was subtracted from the total amount of EDTA used.

3.2.3 On-line Concentration Measurement

Generally, in once-through industrial processes where solutions are not recycled, solution inlet concentrations for heat exchangers remain fairly constant. However, in experimental rigs such as that installed for this dissertation, where closed flow loops were used, the initial solution concentrations would start to decrease following the onset of fouling.

To simulate industrial processes, the CaSO_4 concentration was maintained to within 10% of its initial value for as long as possible, during the course of an experiment. This was done by manually adding the required amounts of $\text{Ca}(\text{NO}_3)_2 \cdot \text{H}_2\text{O}$ and Na_2SO_4 into the solution tank. The initial concentrations of CaSO_4 investigated were 2.8, 3.0 and 3.2 g/l. Therefore, a device which could detect with confidence a difference of about 0.3 g/l in the solution was essential. As it was aimed to maintain a constant

concentration of CaSO_4 during a run, rather than the actual concentration values, the parameter monitored was the change in concentration with time.

The possibility of using a calcium ion selective electrode (ISE) for the aforementioned purpose was investigated. Figure 3.9 shows two calibration curves of the electrode performed 3 days apart, for a CaSO_4 concentration range of 0.3 to 3 g/l. As calcium was the main component of interest, the solutions used for the calibrations were made only from $\text{Ca}(\text{NO}_3)_2 \cdot 4\text{H}_2\text{O}$. From this figure, two discouraging points are obvious. First, there was a huge drift in the electrode reading over a period of only 3 days, when some of the fouling experiments were ran for as long as 5 days. Second, for the concentration to drop 10% from 3.0 to 2.7 g/l, a voltage change of only 1 mV was exhibited. The drift in the electrode reading over time is further confirmed in Figure 3.10, where an average drift of 1.5 mV every 10 hours is evident. This together with the small voltage range to be controlled, makes it necessary to calibrate the electrode several times each day, making ISE a highly unfeasible choice.

Figure 3.9: Calibration curves of calcium ion selective electrode.

After trials with ISE, CaSO_4 concentration in the solution tank was finally measured on-line with confidence using a conductivity probe. The probe consisted of a carbon conductivity cell slot where solution and bubbles in the tank could flow through freely, thus avoiding unnecessary fluctuations. The probe was connected to a conductivity

meter which was in turn linked into the data acquisition unit. Since the make-up of CaSO_4 solution required two salts namely, $\text{Ca}(\text{NO}_3)_2 \cdot \text{H}_2\text{O}$ and Na_2SO_4 , in order to maintain a constant concentration of CaSO_4 automatically, a controller with a floating set-point would be needed. The incorporation of such an on-line dosing device in the experimental set-up would allow for complete automation of the experimental runs.

Figure 3.10: Variation of calcium ion selective electrode reading with time at a CaSO_4 concentration of 3.0g/l.

3.3 Experimental Procedure

The experimental procedure can be divided into 3 categories, namely start-up, the experimental run, and shut-down. Tables outlining start-up and shut-down procedures in detail are given in Appendix A.

3.3.1 Start-Up Procedure

Before measurements could be taken, the following procedure was required to start-up the experimental apparatus:

- (i) The data acquisition program, FOUL was loaded and run.
- (ii) 60 and 25 litres of distilled water were poured into the solution and hot water tanks respectively.

- (iii) The required amounts of $\text{CaNO}_3 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 were weighed and added into the solution tank, during which the solution was recycled back into the tank at a high flow rate. This was to ensure that the salts were dissolved before the solution was allowed to circulate through the rig.
- (iv) The hot water tank heaters were switched on.
- (v) A sample of the solution was taken and its concentration determined using the procedure described in Section 3.2.2. Steps (iii) and (v) were repeated until the desired solution concentration was reached.
- (vi) When the hot water reached its desired temperature, it was pumped to the PHE to heat the solution up to its required temperature. The conductivity probe was inserted into the solution tank and the conductivity meter was switched on.
- (vii) Once the PHE solution and water inlet temperatures and flow rates stabilised, the data acquisition program was started and the experiment begun.

3.3.2 The Experimental Run

During an experimental run, solution samples were taken at least once a day and their concentrations determined. Judging from the conductivity meter readings, the solution was topped up accordingly to maintain a concentration of within 10% of its initial value. To top up the solution, a litre of distilled water was used to dissolve the required amounts of $\text{CaNO}_3 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 separately, and the two solutions were mixed gently before being quickly added into the solution tank.

3.3.3 Shut-Down Procedure

To end an experimental run, the following shut-down procedure including some clean-up steps were followed:

- (i) A sample was taken.
- (ii) The data acquisition program was stopped.
- (iii) After the solution pump was switched off, the solution was drained from the tank. The conductivity probe was removed from the tank to be stored with its tip immersed in distilled water.

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- (iv) The heaters in the hot water tank followed by the hot water pump were switched off. The hot water was then drained from its tank.
 - (v) The heat exchanger plates were removed from the PHE for visual inspection of the fouling layer. Clean plates were used to reassemble the PHE.
 - (vi) The solution flow loop was flushed with tap water to remove any CaSO_4 deposits or solution remaining in the system. This cleaning process was allowed to run overnight.
 - (vii) The hot water flow loop was also flushed with tap water.
 - (viii) Photographs of the fouled plates were taken. Deposit samples were then taken from the plates for future analysis with SEM and/or XRD.
 - (ix) The PHE was disassembled to check if everything was clean. It was then reassembled with clean plates ready for the next experimental run.

The CaSO_4 solution flow loop used to be cleaned after each experiment with 3 to 4 % acetic acid solution, which was later replaced by tap and distilled water. By inspecting the PHE, the DPHE and the filter in the cartridge which have been flushed overnight with water, it was observed that this cleaning procedure was sufficient.

4. EXPERIMENTAL RESULTS & DISCUSSION

The experimental results of this dissertation are divided into two main categories, namely, those obtained from commercial Alfa Laval P01 and M3 plates (Sections 4.1 to 4.8), and those from custom-made flat plates (Sections 4.9 to 4.14).

Heat transfer and pressure drop analyses were performed under clean conditions for all the aforementioned plate types. Using Alfa Laval P01 and M3 plates, the effects of solution flow velocity, concentration, and temperature on fouling were investigated. In addition, fouling behaviour of diagonal and side flow plates was also studied. Visualisation of flow and fouling patterns were carried out, together with the experimentation of several fouling mitigation methods

On the other hand, for the flat plates, the effects of surface roughness, and solution concentration on fouling were looked into. Furthermore, flow and fouling pattern visualisation were performed in conjunction with a flat polycarbonate plate. The effects of five different distributor designs on fouling were also examined.

4.1 Alfa Laval P01 and M3 Plate Heat Exchangers

Heat transfer and pressure drop analyses were performed under clean conditions using Alfa Laval P01 and M3 plates, for side and diagonal flow.

4.1.1 Heat Transfer

To compare clean experimental heat transfer measurements with correlations from the literature, a series of runs with different flow velocities were performed for a hot water/water system. Conditions of the hot side were kept constant, with a flow velocity and inlet temperature of 14.6 cm/s and 87°C respectively. On the cold side, the inlet temperature was maintained at 51.5°C, while the flow velocity was varied from 20 to 90 cm/s. The measured overall heat transfer coefficients, U_m and the curve-fitted

overall heat transfer coefficients, U_c were determined using Equations (2.1), (2.2) and (2.5), and Equations (2.4) and (2.9) to (2.14) respectively. Physical properties of water and stainless steel AISI 316 were taken from VDI-Wärmeatlas (1994). As the ratio μ/μ_{wall} in Equation (2.9) has a maximum of only 1.1 for all the heat transfer runs performed, its effect was ignored by setting c_4 to zero. Furthermore, c_3 was kept constant at 0.33 in curve-fitting the values of c_1 and c_2 in Equation (2.9).

Figure 4.1 shows the comparison between U_m and U_c at different Reynolds numbers, Re , for P01 and M3 plates, both diagonal and side flow. The constants c_1 and c_2 obtained, are 0.38 and 0.65 respectively, which agree well with those of Bansal (1994) and Branch *et al.* (1991) mentioned in Section 2.1.3. The Nu correlation can thus be written as follows:

$$Nu = 0.38 \cdot (Re)^{0.65} \cdot (Pr)^{0.33} \quad (4.1)$$

Figure 4.1: Comparison of measured and curve-fitted overall heat transfer coefficient at different Reynolds number for P01 and M3 plates.

To avoid cluttering, U_m for M3 *side* flow plates is included without error bars. The M3 side flow plates display similar heat transfer characteristics to the M3 diagonal flow plates. This is contrary to the theory mentioned in Section 2.1.5.5 and depicted in

Figure 2.8, which states that higher heat transfer is expected for diagonal flow plates than side flow ones.

Although both data sets for diagonal flow in Figure 4.1 obey Equation (4.1), the data pertaining to P01 exhibit lower overall heat transfer coefficients. To determine whether this discrepancy is due to the difference in thickness between P01 and M3 plates, U was calculated using Equation (2.4) for α_{hot} , α_{cold} , λ_{wall} , and s_{wall} values similar to those of U_c in Figure 4.1. The resultant variation of U with α_{cold} is plotted in Figure 4.2 for the two plate types. As can be seen, the difference between P01 and M3 shown in Figure 4.2 matches that in Figure 4.1. Since both plate types are small, the maximum difference in their U value is only about 3%.

Figure 4.2: Calculated overall heat transfer coefficient for P01 and M3 diagonal flow plates.

Figure 4.3 shows the variation of $Nu \cdot Pr^{-0.33}$ with Re for P01 and M3 plates. The hot and cold side of each plate type and flow direction set-up lies on a straight line, with a slope of 0.65.

Figure 4.3: Variation of $Nu \cdot Pr^{0.33}$ with Reynolds number for P01 and M3 plates.

4.1.2 Pressure Drop

Apart from being conducted at room temperature (about 20 - 25°C), all pressure drop measurements were obtained under similar operating conditions as those for heat transfer. Using Equations (2.19) and (2.20), the pressure drop for a clean PHE can be correlated. Δl was taken to be 0.356 m, which is the vertical distance between the inlet and outlet port centres. The measured experimental friction factors, f at different Re are displayed in Figure 4.4.

The curve-fitted values for diagonal and side flow plates are $c_5 = 1.40$, $c_6 = 0.05$, and $c_5 = 1.45$, $c_6 = 0.05$ respectively. Bansal (1994) did not perform such a pressure drop analysis.

$$\text{For M3 diagonal flow PHE: } f_{dia} = \frac{1.40}{Re^{0.05}} \quad (4.2)$$

$$\text{For M3 side flow PHE: } f_{side} = \frac{1.45}{Re^{0.05}} \quad (4.3)$$

Figure 4.4: Variation of friction factor with Reynolds number for M3 diagonal and side flow plates.

Figure 4.5: Variation of pressure drop with nominal flow velocity for M3 diagonal, side, and indented side flow plates.

As can be seen from Figure 4.4 and Figure 4.5, both the friction factor and pressure drop are higher for side flow plates than diagonal ones. To account for this difference,

attention was first given to the way f was calculated via Equation (2.19), which can be rearranged to give:

$$\Delta p = f \cdot \frac{2 \cdot \rho \cdot u^2}{d_h} \cdot \Delta l = f \cdot c_s \cdot u^2 \quad (4.4)$$

$$\text{where } c_s = \left(\frac{2 \cdot \rho \cdot \Delta l}{d_h} \right)$$

Since the data were acquired at room temperature, changes in ρ were minimal. In addition, both Δl and d_h have fixed values, and hence the assumption can be made for c_s to be constant. It is worth noting the ambiguity in the length of the actual flow path for diagonal and side flow channels. As a constant Δl value was used for both types of channel, a difference in the actual flow length could tip the balance for the corresponding f values. Another questionable parameter used is the flow velocity, u which was taken as the average velocity across the heat transfer section of the exchanger. With reference to Figure 2.8, as diagonal and side flow plates have different flow paths within their respective channels, their corresponding local velocities must be different as well. It is possible that the slight discrepancy in pressure drop observed, has been caused by different degrees of local flow maldistribution in the channels. From the literature review on plate design in Section 2.15, when the heat transfer is enhanced by changes in one aspect of the plate geometry, a boost in the pressure drop is usually unavoidable. This trend is also exhibited by the M3 side flow plates, which have similar overall heat transfer coefficients and higher friction factors, when compared with their diagonal flow counterparts.

As a result of overtightening, some of the M3 plates have their contact points indented. Figure 4.5 shows that these plates exhibit a pressure drop of about 10% higher than those of their non-indented counterparts. Although it is very important to make sure that the plates are in contact with each other so as to have a fixed geometry, it is equally important not to overtighten the plates and hence distort the fixed geometry.

4.1.3 Summary of Experimental Parameters

A summary of the constants c_1 to c_6 for M3 diagonal and side flow plates, and flat diagonal flow plates, is given in Table 4.1. The flat plates are discussed in greater detail in Section 4.9. Table 4.2 gives a summary of the experimental parameters, while a list of the experimental conditions for all fouling runs performed can be found in Table 4.3.

	M3 (diagonal)	M3 (side)	Flat (diagonal) set A
c_1	0.38*	0.38*	0.0245*
c_2	0.65*	0.65*	0.80*
c_3	0.33	0.33	0.33
c_4	0	0	0.14
c_5	1.40*	1.45*	0.64*
c_6	0.05*	0.05*	0.20*

Table 4.1: Summary of plate constants c_1 to c_6 for heat transfer and pressure drop (* tested constant).

Plate type	M3 (diagonal)	Flat (diagonal)
Flow velocity [cm/s]	20, 35, 65	35
Solution Concentration [g/l]	2.8, 3.0, 3.2	3.0, 3.2
Solution Inlet Temperature [°C]	42, 51.5	41.5
Hot Water Inlet Temperature [°C]	87	87

Table 4.2: Summary of experimental parameters.

All the experiments were performed with in-line filtration, where M3 diagonal flow plates were mainly used. Most experiments were continued for either 3000 or 7000 minutes, with the latter being more common. However, some experiments were stopped before the usual period because the pressure drop was too high for a constant flow rate to be maintained. These experiments show that increase in pressure drop due to deposition can become the limiting factor in PHE operation, rather than the corresponding degradation in heat transfer.

Run ID.	Plate Type	Time [min]	H ₂ O Vel [cm/s]	Soln Vel [cm/s]	Initial Soln Conc [g/l]	H ₂ O In Temp [°C]	Soln In Temp [°C]	Surface
250796	P01(dia)	7200	14.5	35	3.3	86.1 - 89.7	51.0 - 52.6	SS
020896	P01(dia)	7070	14.5	35 (flush 52)	3.3	85.7 - 88.0	51.3 - 52.2	CrN (IS)
120896	P01(dia)	7000	14.5	35	3.3	87.0 - 87.9	50.8 - 52.5	SS
200896	P01(dia)	3000	14.5	20	3.3 (top)	87.1 - 87.6	51.2 - 52.1	SS
110996	P01(dia)	2900	14.5	20	3.3	87.0 - 87.5	51.3 - 51.9	SS
061196	P01(dia)	7400	14.6	35	2.81	86.6 - 87.6	51.5 - 51.9	SS
131196	M3(dia)	7200	14.6	35	2.80	86.6 - 87.6	51.4 - 51.9	DLC (IS)
271196	M3(dia)	8080	14.6	35	2.81	86.9 - 87.5	51.4 - 51.9	DLC + HMDSO
041296	M3(dia)	8200	14.6	35	2.80	86.9 - 87.5	51.4 - 51.8	DLC + C ₂ F ₂ (IS)
111296	M3(dia)	8280	14.6	35 (flush 70)	2.79	86.8 - 87.6	51.3 - 51.8	SS
040197	M3(dia)	7200	14.6	35 (flush 86)	2.80	86.8 - 87.3	51.5 - 51.8	DLC + C ₂ F ₂ (IS)
120197	M3(dia)	7200	14.6	35 (flush 50)	3.01 (top)	87.0 - 87.5	51.5 - 51.9	SS
190197	M3(dia)	1720	14.6	35 (flush 44)	3.01 (top)	87.0 - 87.4	51.5 - 51.9	DLC + C ₂ F ₂ (IS)
220197	M3(dia)	1740	14.6	35 (flush 52)	3.00 (top)	87.0 - 87.4	51.5 - 51.8	DLC + HMDSO
260297	M3(dia)	7110	14.6	35 (flush 80)	3.00	87.1 - 87.7	51.4 - 52.0	SS
060397	M3(dia)	5800	14.6	20	3.00	87.1 - 88.0	51.2 - 52.2	SS
130397	M3(dia)	5080	14.6	35	3.20 (top)	87.2 - 87.8	50.5 - 53.5	SS
250397	flat(dia)	4400	14.6	35	3.20 (top)	87.2 - 87.7	41.5 - 41.8	SS (original)
020497	M3(dia)	7140	14.6	35 (flush 80)	2.80	87.0 - 88.0	51.2 - 52.1	SS
090497	flat(dia)	6000	14.6	35	3.21 (top)	87.1 - 87.9	41.5 - 43.7	SS (original)

Table 4.3: Summary of experimental conditions.

Run ID.	Plate Type	Time [min]	H ₂ O Vel [cm/s]	Soln Vel [cm/s]	Initial Soln Conc [g/l]	H ₂ O In Temp [°C]	Soln In Temp [°C]	Surface
150497	M3(dia)	930	14.6	35	3.20	87.3 - 88.0	51.5 - 51.9	SS
170497	flat(dia)	3000	14.6	35	3.00 (top)	87.1 - 88.5	41.4 - 43.0	SS (240 sand)
230497	flat(dia)	7000	14.6	35	3.02 (top)	87.6 - 88.2	41.9 - 42.4	SS (240 sand)
100597	flat(dia)	7000	14.6	35	3.21 (top)	87.2 - 88.0	41.6 - 42.2	SS (240 sand)
160597	flat(dia)	3000	14.6	35	3.20 (top)	87.4 - 88.0	41.7 - 42.2	SS (240 sand)
290597	M3(dia)	7000	14.6	35	3.00	87.6 - 88.5	51.7 - 52.4	SS
040697	flat(dia) & poly(2mm)	7000	29.2	35	3.21 (top)	87.4 - 88.3	37.6 - 42.3	SS (240 sand)
100697	M3(dia)	10000	14.6	65	3.00	87.4 - 88.4	51.7 - 52.3	SS
190697	M3(dia)	3000	14.6	20	3.00 (top) particulate	87.3 - 87.9	51.6 - 52.0	SS
240697	M3(dia)	3000	14.6	20	3.00 (top)	87.2 - 88.8	51.6 - 52.2	SS
270697	M3(dia)	2700	14.6	35	3.21 (top)	86.9 - 87.7	51.6 - 52.0	SS
300697	M3(dia)	3000	14.6	35	3.21 (top)	86.5 - 89.0	51.7 - 52.1	SS
030797	M3(dia) & magnet	7000	14.6	35	2.80	86.5 - 88.6	51.7 - 52.2	SS
090797	M3(dia) & poly	3020	14.6	35	3.20 (top)	87.2 - 87.9	51.8 - 52.4	SS
120797	M3(side) & poly	3020	14.6	35	3.22 (top)	87.2 - 87.8	51.8 - 52.4	SS
150797	M3(side)	3000	14.6	35	3.22 (top)	87.7 - 88.7	51.8 - 52.4	SS
180797	flat(dia) & poly	7000	14.6	35	3.20 (top)	87.1 - 88.0	41.9 - 42.4	SS (240 sand)
260797	flat(dia) Design A	7020	14.6	35	3.20 (top)	87.1 - 88.2	41.6 - 42.5	SS (240 sand)
040897	M3(dia)	3000	14.6	20	3.00 (top) + NaNO ₃	87.5 - 87.9	51.8 - 52.2	SS
070897	M3(dia)	1000	14.6	20 + air (1.5 l/min)	3.01 (top)	87.5 - 88.3	51.8 - 52.2	SS

Table 4.3: Summary of experimental conditions (continued).

Run ID.	Plate Type	Time [min]	H ₂ O Vel [cm/s]	Soln Vel [cm/s]	Initial Soln Conc [g/l]	H ₂ O In Temp [°C]	Soln In Temp [°C]	Surface
090897	M3(dia)	3000	14.6	20 + air (30 l/min)	3.02 (top)	87.4 - 88.2	50.4 - 52.0	SS
120897	M3(dia)	2100	14.6	20 + air (10 l/min)	3.01 (top)	87.7 - 88.4	50.1 - 52.3	SS
150897	M3(dia)	3220	14.6	20 + air flush (50l/min)	3.02 (top)	87.6 - 88.1	51.9 - 52.3	SS
180897	M3(dia)	1520	14.6	20 + air (10 l/min)	3.02 (top)	87.8 - 88.3	51.7 - 52.2	SS
200897	M3(dia)	1500	14.6	20	3.01 (top)	87.4 - 88.2	51.7 - 52.5	SS
220897	flat(dia) Design B	7000	14.6	35	3.21 (top)	87.4 - 88.0	41.7 - 42.6	SS (240 sand)
280897	flat(dia) Design C	7020	14.6	35	3.19 (top)	87.4 - 87.9	41.7 - 42.5	SS (240 sand)
030997	flat(dia) Design D	7000	14.6	35	3.22 (top)	87.2 - 87.8	41.8 - 42.4	SS (240 sand)
100997	M3(dia)	7000	14.6	35	3.02 (top)	77.4 - 78.1	42.6 - 43.3	SS
160997	flat(dia) Design E	7100	14.6	35	3.21 (top)	87.3 - 88.0	41.8 - 42.6	SS (240 sand)
031097	M3(dia)	7000	14.6	35	2.81 (top)	87.2 - 88.0	51.6 - 52.5	ta-C (IS)

Note: dia = diagonal

poly = polycarbonate

IS = ion-sputtered

flush = run has been flushed (with a higher flow rate) prior to its termination

top = solution concentration has been topped up during run

particulate = particulate and crystallisation fouling have occurred

magnet = a magnetic device has been installed to mitigate fouling

2mm = channel gap was 2 mm

original = original flat plate surface was used

240 sand = 240 emery paper has been used to sand flat plates.

Table 4.3: Summary of experimental conditions (continued).

4.2 Fouling Data and Deposit Pattern in Alfa Laval PHEs

A typical experiment with Alfa Laval plates, using the current set-up is presented. The corresponding heat transfer and pressure drop trends are also highlighted. Comparisons were made with respect to the degree of data scatter, between a set of current experimental results and that of a preliminary run. In addition, ways of modifying the present rig so that it becomes fully automatic are also discussed.

4.2.1 Current Experiments

For a typical fouling run, Figures 4.6(a) and (b) show that the fouling resistance, R_f and the pressure drop, Δp increase with time, respectively. Therefore, the cleanliness factor (CF) decreases with time, and the pressure drop increase factor (PDIF) increases with time, as illustrated in Figure 4.6(c). Using a moving average of 3 data points, reduces the scatter in R_f but does not improve that of the pressure drop. This is because the latter has a low degree of scatter, which made moving average unnecessary. Towards the end of the run, R_f has almost reached an asymptotic value.

Figure 4.6(a): Variation of fouling resistance with time for a typical fouling run.

Figure 4.6(b): Variation of pressure drop with time for a typical fouling run.

Figure 4.6(c): Variation of cleanliness factor and pressure drop increase factor with time for a typical fouling run.

As mentioned previously in Section 2.2.4.1, when fouling deposits start to grow, they penetrate the laminar sublayer causing turbulence to increase, and a roughness delay period to occur. Indicated by negative R_f values, the latter is however not always registered by the data acquisition system. This is because the data acquisition program calculates R_f by using the first overall heat transfer coefficient calculated as the clean overall heat transfer coefficient, according to Equation 2.29. If this overall heat transfer

coefficient is unexpectedly high or low due to process fluctuations, the shape of the fouling curve, and the absolute R_f values would be affected to a certain extent.

Unless otherwise stated, only data points directly obtained from the data acquisition program are shown in this chapter. In most cases, M3 diagonal flow plates were used, with a solution inlet temperature of 51.5°C. The hot water inlet temperature and velocity were usually maintained at about 87°C and 14.6 cm/s (or 70 ml/s) respectively. The run identification numbers for the experiments are shown in the legend of their respective figures. This is so that they can be easily identified with the help of Table 4.3. These numbers mark the experiment start dates, for instance run number [120197] was started on the 12th of January, 1997.

In order to simulate once-through processes in industries, not only was an on-line filter installed as mentioned in Chapter 3, but in some experiments the solution concentration was also maintained by manual top-ups to within 10% of its initial value. For the above mentioned typical run, the word 'top' is indicated in its legend in Figure 4.6(a) for topping up of concentration, while the top-up times and concentrations are indicated by '**' in Figure 4.6(b). The label 'top' is given to experiments with solution concentration conforming to this 10% criterion, regardless of how long this criterion was met, and regardless of whether topping up was required or not to meet this criterion. The topping up times are indicated by sharper increase in pressure drop as shown in Figure 4.6(b). However, there is no such corresponding increase in R_f . This is because the pressure drop increase was most likely to be caused by deposit formation near the outlet region of the plate which does not contribute much to heat transfer.

A picture of the fouled plates is shown in Figure 4.7, where the flow enters and leaves through one of the bottom and top ports respectively. It should be pointed out that, at the end of this run, the plate channels were flushed at a high flow rate of 50 cm/s for about an hour. This was to investigate the removal of a pre-existing fouling layer, and is discussed in detail in Section 4.9. As a result, some of the deposits on the fouled plates may have been removed. Nevertheless, comparing Figure 4.7 with Figure 2.16, it can be concluded that fouling deposits are found in the top portion of the plate where the

temperature is high, particularly in the top corner of the plate where the flow velocity is low as well.

Figure 4.7: Picture of a typical pair of fouled plates [120197].

Figure 4.8: Formation of fouling deposits around plate contact points [061196].

This deposit pattern is confirmed by Thonon and Grillot (1996). They experimented with particulate fouling of CaCO_3 particles in PHEs, and found no proportionality between the deposited mass and R_f . This implies that while deposition occurred in areas where the local heat transfer coefficients are low, high heat transfer coefficient regions have little or no deposit.

Furthermore, the deposit patterns in the present investigation confirms the findings of Bansal (1994) and Delplace *et al.* (1994), which claim that fouling commences at PHE contact points, as mentioned in Section 2.2.6.3.3. This is shown in Figure 4.8. In addition, the fouling deposits obtained in the present study, were analysed using Scanning Electron Microscopy (SEM), Energy Dispersive X-Ray Analysis (EDX) and X-Ray Diffraction Analysis (XRD), and were found to be pure gypsum crystals ($\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$). This agrees with the CaSO_4 fouling deposit analyses conducted by Bansal (1994).

4.2.2 Preliminary Experiments

A series of preliminary experiments was carried out after the solution flow rate control system was installed, but before those of the hot water flow rate and solution tank cooling water were incorporated as part of the rig. These modifications are discussed in Section 3.1.5.

Figure 4.9 shows the variation of R_f with time for a preliminary experiment and that for a current experiment. These two experiments were performed under similar operating conditions, except that the former had a concentration of 3.3 g/l, while the latter had one at 3.2 g/l. In addition, the preliminary experiment is a non-top one, while the current experiment was topped up.

Figure 4.9: Comparison between preliminary and current experiments for solution inlet velocity of 35 cm/s.

There is good agreement between the preliminary and current experiments, which demonstrates the repeatability of the experiments. It can be seen in Figure 4.9 and in the rest of this chapter that R_f of the current experiments fluctuated by about $\pm 2.5 \times 10^{-6} \text{ m}^2\text{K/W}$, while the values of the preliminary experiments fluctuated by about $\pm 5 \times 10^{-6} \text{ m}^2\text{K/W}$. The higher degree of scatter for the latter was brought about by greater

fluctuations in the hot water flow velocity and the solution tank temperature. The current experimental set-up not only lowers the degree of scatter, but is also semi-automatic, and hence, more reliable and less tedious to run.

4.2.3 Future Experiments

The current set-up is only semi-automatic because the solution concentration still needs topping up manually. Figure 4.10 shows the variation of solution conductivity with time for run [300697], which is a typical 'top' experiment. Each time the concentration is topped up, the conductivity of the solution rises above the previous value for the same CaSO_4 concentration. This is because the solution used is made up of both $\text{Ca}(\text{NO}_3)_2 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 , so when Ca^{2+} and SO_4^{2-} precipitate out of the solution, Na^+ and NO_3^- remain and accumulate in the solution with each top-up.

Figure 4.10: Variation of solution conductivity with time for a typical 'top' experiment.

For the future, with slight modifications to the current set-up, the conductivity meter can be used to maintain the concentration of the tank automatically. This can be done such that when the concentration drops by 10% below its set-point, which would be for instance from about 7.70 to 7.50 mS with reference to Figure 4.10, a controller will

drive two peristaltic pumps to dose fixed amounts of concentrated $\text{Ca}(\text{NO}_3)_2$ and Na_2SO_4 solutions into the solution tank separately, thus increasing the solution conductivity from 7.50 to 8.10 mS. Both 7.70 mS at the start of the experiment, and 8.10 mS at 1000 minutes, correspond to a concentration of 3.2 g of CaSO_4 per litre of solution (3.2 g/l). Every time the conductivity drop by approximately 0.2 mS, fixed amounts of Ca^{2+} and SO_4^{2-} enough to make up for their 10% loss in the solution circulating the rig, will be dosed. This stepwise dosing method is proposed instead of a continuous one, because of the changing set-point of the solution conductivity. Hence, the controller required for this automatic dosing set-up needs to have a floating set-point.

4.3 Effect of Solution Flow Velocity on Fouling in Alfa Laval PHEs

In this section, the influence of solution flow velocity on CaSO_4 fouling in Alfa Laval PHE was investigated. Using diagonal flow plates, three different solution inlet flow velocities were considered, namely 25 cm/s, 35 cm/s and 65 cm/s. An initial CaSO_4 concentration of 3.0 g/l was used.

Figure 4.11(a): Variation of fouling resistance with time for solution flow velocities of 25 cm/s, 35 cm/s and, 65 cm/s.

Figure 4.11(b): Variation of pressure drop with time for solution flow velocities of 25, 35 and 65 cm/s.

Figure 4.11(c): Variation of pressure drop increase factor with time for solution flow velocities of 25, 35 and 65 cm/s.

A comparison of the dynamic variations of R_f for the three different flow velocities is shown in Figure 4.11(a). As the solution inlet velocity increases, a general decrease in

the overall R_f and in the gradient of the fouling curves can be observed. The curves for 35 and 65 cm/s seem to have reached their respective asymptotic values, while the 20-cm/s curve shows no sign of an asymptotic value at the end of its 3000-minute run. However, due to the data scatter, no apparent roughness delay period was observed for the fouling processes.

As mentioned in Section 2.1.4 and demonstrated in Equation 2.19, the clean pressure drop increases with increasing solution velocity. The pressure drop for 20 and 35 cm/s started to increase at about 1000 minutes, while that for 65 cm/s remained fairly constant throughout the entire experiment, as illustrated in Figure 4.11(b). Figure 4.11(c) shows that, the higher the flow velocity, the smaller the amount of pressure drop increase.

It should be noted that the pressure drop increase is affected by the solution concentration top-up times. Almost immediately following a top up, the pressure drop would start to increase more rapidly. Its increase slowly tapers off as the concentration drops. Although an effort was made for top-up times of similar experiments to be as identical as possible, there were occasions when topping up was not required until a later time or when it was not possible to observe the experiment fully.

Displayed in Figures 4.12(a), (b) and, (c) are photographs of fouled M3 diagonal flow plates for 20 cm/s, 35 cm/s and, 65 cm/s respectively. These photographs confirm that fouling deposits are highly concentrated in the upper corner adjacent to the outlet port of the plate, where the flow velocity was low and the temperature was high. As expected, there was not much deposit on the fouled plates for 65 cm/s even though it was run for the longest time of 10000 minutes, when compared with the run times for 20 cm/s (3000 minutes) and 35 cm/s (7000 minutes). There appears to be more deposit on the 35-cm/s fouled plates than those for 20 cm/s because the latter was obtained in less than half the run-time of the former.

Figures 4.12(a), (b) and, (c):
Photographs of fouled M3 plates for
solution flow velocities of 25, 35 and 65
cm/s respectively.

This set of results showed that an increase in solution velocity induces a reduction in fouling. As mentioned in Section 2.2.6.1.1, higher flow velocities bring about stronger shear stresses and enhance the turbulence level, making it difficult for deposits to grow. They could also remove any deposits which have successfully attached themselves to the plate channel walls. Plotting R_f values of the three velocities shown in Figure 4.11(a) against their corresponding flow velocities, u , at 2000 minutes, 3000 minutes and the asymptotes (only for 35 and 65 cm/s), revealed that R_f is proportional to u to the power of between -1.4 and -1.6. This agrees well with Equation 2.45 ($R_f \propto u^{-1.5}$), which seems to be valid for all heat exchanger types and fouling mechanisms.

The general trends of this set of results in terms of R_f and pressure drop, match those of Bansal (1994) obtained under similar process conditions. However, compared with Bansal's results, the R_f and pressure drop values of the current investigation are higher by up to about 50% and 230% respectively. These discrepancies could be partly due to the different quality of distilled water used and the dissimilar grade of $\text{Ca}(\text{NO}_3)_2 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 crystals used to make up the CaSO_4 solution.

4.4 Effect of Solution Concentration on Fouling in Alfa Laval PHEs

Amongst other operating parameters, the effect of solution concentration on fouling was investigated by Bansal (1994). Fouling was found to increase with increasing solution concentration. Interestingly, this concentration effect was discovered to be important only in the initial stages of fouling. Concentration topping up after deposits were formed, affected neither R_f nor the pressure drop in the exchanger.

To study the influence of solution concentration on fouling in PHEs, experiments were performed by varying the CaSO_4 concentration from 2.8 to 3.2 g/l. It can be seen from Figure 3.8 that between 30 to 60°C, the prevailing saturated concentration does not vary much but remains at a value of about 2.2 g/l. As all experiments using Alfa Laval plates were performed with either a 42 or 51.5°C solution inlet temperature, the corresponding supersaturation is between 27% and 45%. The solution concentration effects were examined from three perspectives, namely initial concentration, concentration topping-up, and time and amount of top-up. These are discussed in Sections 4.4.1 to 4.4.3 respectively.

4.4.1 Initial Solution Concentration

Figures 4.13(a) and (b) show the effect of solution concentration on fouling. For these 'top' experiments, R_f increases with increasing solution concentration. The fouling curves of 2.8 and 3.2 g/l each exhibits a clear roughness delay period. Moreover, topping-up times are indicated by a sharper increase in pressure drop. This set of results suggests that an increase of 9% in supersaturation from 2.8 to 3.0 g/l, causes the corresponding R_f and pressure drop to increase tremendously. However, when the supersaturation was given a further 9% increase from 3.0 to 3.2 g/l, hardly any difference in R_f was observed. Nevertheless, the pressure drop for the 3.2-g/l run is significantly higher for the first 1700 minutes than those for 3.0 and 2.8 g/l. Its pressure drop eventually levelled off due to a drop of more than 10% of the CaSO_4 concentration below its initial value.

Figure 4.13(a): Effect of solution concentration on fouling resistance for 35 cm/s solution flow velocity.

Figure 4.13(b): Effect of solution concentration on pressure drop for 35 cm/s solution flow velocity.

The effects of increasing the initial solution concentration can also be realised from an inspection of the fouled plates, refer Figures 4.14(a), (b) and (c). It is worth mentioning that the 3.2-g/l experiment has only half the run-time of the other two experiments. In

addition, both the 2.8- and 3.0-g/l experiments were flushed at higher flow rates before being terminated. Nevertheless, pictures of the fouled plates can be used to elucidate the observed R_f and pressure drop trends. It is likely that at a concentration of 2.8 g/l, fouling deposits grow in regions which do not contribute significantly to heat transfer, refer Figure 4.14(a). By creasing the concentration from 2.8 to 3.0 g/l, the deposits are able to invade plate regions which are important to heat transfer, refer Figure 4.14(b). However, with a further increase in concentration from 3.0 g/l to 3.2 g/l, deposition could not be established on a larger region of the main heat transfer plate area, and hence no further increase in R_f is displayed. Nevertheless, fouling increases with increasing concentration, which can be readily observed from the corresponding pressure drop increase.

Since the deposition rate did not increase with increasing solution flow velocity (refer Figure 4.11), but increased with increasing solution concentration (refer Figure 4.13), it is likely that the deposition rate is reaction-controlled and not diffusion-controlled. Although the R_f and pressure drop trends agree with those reported by Bansal (1994), the corresponding absolute values obtained in the present study are higher.

4.4.2 Topping up of Solution Concentration

Figures 4.15(a) and (b) show the effects of solution concentration topping-up for an initial concentration of 3.0 g/l and a solution flow velocity of 35 cm/s, on the variation of R_f and pressure drop with time.

Figure 4. 15(a) Effect of solution concentration top-up on fouling resistance for 3.0 g/l initial concentration and 35 cm/s solution flow velocity.

Figure 4.15(b): Effect of solution concentration top-up on pressure drop for 3.0 g/l initial concentration and 35 cm/s solution flow velocity.

It is apparent that the fouling process is strongly influenced not only by the initial concentration, but also by changes in concentration once the run has started. This observation does not agree with the findings of Bansal (1994) who found solution concentration only to be important in the initial stages of fouling. This difference may be due to the way the salts were added. In the present study, the two salts were dissolved separately with a total of 1 litre of distilled water, and were usually mixed gently before being quickly added to the solution tank. Bansal (1997) used the same procedure except that he did not mix the two salt solutions prior to their entry into the tank. Therefore, it is possible that mixing the two solutions may have caused minute crystals to form, which act as nuclei for subsequent fouling. However, no signs of particulate fouling have been observed in the present study, provided that no more than 11% of the initial concentration was topped up via the above mentioned method, where the solutions were mixed before they were added into the tank. Moreover, similar fouling trends were observed in the present study for experiments performed using this topping-up method and that practised by Bansal.

4.4.3 Time and Amount of Top-Up

Figures 4.16(a) and (b) show the effects of solution concentration top-up times and amounts, for an initial concentration of 3.2 g/l and a solution flow velocity of 35 cm/s. The corresponding top-up concentrations and amounts are shown in Figure 4.16(c). An 'over-top' experiment is one whose concentration was topped up back to its initial value, after it has dropped more than 10% below this value. Run [130397] was 'over-topped' at about 1300 minutes and again at 2300 minutes as shown in Figure 4.16(c). The danger in doing this is that tiny CaSO_4 crystals may form when the two salt solutions are mixed, as mentioned in the preceding section. Even with the filter on-line, crystals smaller than the filter pore size may still reach the PHE, causing particulate fouling to occur. The presence of this fouling mechanism can only be confirmed by inspecting the structure of the deposits on the plates after the run is terminated. If only crystallisation fouling has occurred, clear facets of the crystalline deposit can be seen. Conversely, when both crystallisation and particulate fouling are present, the fouling

deposit is made up of a combination of clear crystal facets and rounded structures of adhering particles.

Figure 4.16(a): Effect of solution concentration top-up times and amounts on fouling resistance (3.2 g/l initial concentration; 35 cm/s solution flow velocity).

Figure 4.16(b): Effect of solution concentration top-up times and amounts on pressure drop (3.2 g/l initial concentration; 35 cm/s solution flow velocity).

On the other hand, an 'under-top' experiment is one whose concentration was not topped up anymore once it has dropped more than 10% below its initial value. It is apparent in Figure 4.16(c) that if no top-up attempts were made, the solution concentration eventually dropped to an asymptotic value of about 20% below its initial value.

Figure 4.16(c): Variation of solution concentration with time for 3.2 g/l initial concentration and 35 cm/s solution flow velocity.

4.5 Effect of Solution Inlet Temperature on Fouling in Alfa Laval PHEs

All experiments performed using Alfa Laval plates except run [100997], were carried out with the solution inlet temperature being 51.5°C. On the other hand, the flat plate experiments (Sections 4.9 to 4.14) were performed with a 41.5°C solution inlet temperature. This is because the lower heat transfer efficiency of the flat plates made it impossible to have a higher solution inlet temperature, using the present rig. To investigate the effect of this 10°C temperature difference, an experiment (run [100997]) was performed using M3 diagonal plates with 42°C solution inlet temperature. The results obtained are presented in Figures 4.17(a) and (b).

Figure 4.17(a): Effect of solution inlet temperature on the variation of fouling resistance with time.

Figure 4.17(b): Effect of solution inlet temperature on the variation of pressure drop with time.

As the solution inlet temperature is decreased, R_f and its associated pressure drop also decrease. Figures 4.18(a) and (b) confirm that more deposition is obtained when the

solution inlet temperature is higher. With reference to Figure 3.8, the solubility of CaSO_4 remains fairly constant over the temperature range of 30 to 60°C. Beyond this range, the solubility drops with increasing and decreasing temperature. Nevertheless, increasing the solution inlet temperature would result in an increase in the wall temperature. Hence, the augmented deposition rate is most probably caused by the corresponding change in the reaction rate constant, k_{rxn} which increases exponentially with the wall temperature according an Arrhenius type equation (refer Equation (2.43) and Section 2.2.6.1.2). These results agree well with trends reported by Bansal (1994).

Figures 4.18(a) and (b): Photographs showing fouled plates of 42 and 51.5°C solution inlet temperature respectively.

4.6 Effect of Port Position on Fouling in Alfa Laval PHEs

To study the effects of port position, experiments were performed using M3 diagonal and side flow plates. The solution inlet temperature and flow velocity are 51.5°C and 35 cm/s respectively. The initial solution concentration is 3.2 g/l. From Figures 4.19(a) and (b), side flow plates seem to display slightly higher R_f and pressure drop values than their diagonal flow counterparts. In addition, the former exhibits a 200-minute roughness delay time which is about 3 times shorter than that of the latter. Both plate types reached their respective asymptotic values for R_f and pressure drop.

The side flow plates have a higher clean pressure drop than the diagonal flow plates. This is consistent with the pressure drop analysis in Section 4.1. Although the side flow

plates maintained a higher overall pressure drop throughout *their* 3000-minute run, the overall change in pressure drop is about the same for both plate types. Furthermore, a clean heat transfer coefficient, U_o of about $5400 \text{ W/m}^2\text{K}$ was registered for side flow, which is slightly higher than the $5300 \text{ W/m}^2\text{K}$ for diagonal flow. This result conforms approximately with the analysis given for heat transfer in Section 4.1, where similar U_o was obtained for both plate types.

Figure 4.19(a): Effect of port position on the variation of fouling resistance with time.

Figure 4.19(b): Effect of port position on the variation of pressure drop with time.

Different port positions give rise to dissimilar flow distribution in the plate channels, which in turn result in the formation of unlike fouling patterns, as displayed in Figures 4.20(a) and (b). Both plate types have flows entering and leaving via one of their bottom and top ports respectively. For the side flow plates, the flow is most likely to be faster in the region directly between the two ports. Consequently, this main flow path is generally free from deposits. The flow is slower away from this main path, while the temperature is higher in the upper part of the plate. Therefore, more deposits are found in the top corner of the plate opposite the outlet port.

Figures 4.20(a) and (b): Photographs showing fouled diagonal and side flow plates respectively.

The only side flow experiments performed by Bansal (1994) were with APV TR1 plates, instead of Alfa Laval P01 or M3 plates. Hence, no direct comparisons were given on the effect of port position on fouling.

4.7 Flow and Fouling Patterns in Alfa Laval PHEs

To enhance the understanding of deposit growth, the following three approaches were taken:

- (i) visual observation of flow patterns
- (ii) visual observation of fouling pattern development
- (iii) performance of fouling experiments with various run-times

Approaches (i) and (ii) were performed using M3 diagonal and side flow plates in conjunction with a corrugated polycarbonate plate. The former approach was performed with a suspension of Acrylonitrile Butadiene Styrene (ABS) particles in water at room temperature. ABS particles can form a suspension with water because of their similar densities. The particles used have a size distribution of 300 to 700 μm .

As mentioned in Section 3.1.2, only one side of the solution channel could be heated with this set-up for approach (ii) (refer Figure 3.7). Moreover, the polycarbonate plate has rectangular corrugations instead of sinusoidal ones. On the other hand, only M3 diagonal plates were used in approach (iii). Nevertheless, all three approaches were carried out under similar operating conditions, whereby the solution flow velocity and concentration were 35 cm/s and 3.2 g/l respectively.

4.7.1 Diagonal Flow

Using approach (i) for the diagonal flow set-up, the velocity was observed to be fastest between the inlet and outlet ports of the plate, and slower away from this main flow area. Referring to Figure 2.8(a), the upper left and lower right corners of the plate are the two regions of low flow velocity. The flow patterns were recorded using a Sony DXC930P 3CCD True Colour camera, with Leica Monozoom 7 lens, on video tapes for future reference.

Figure 4.21(a): Variation of fouling resistance with time for M3 diagonal flow and polycarbonate plates.

Figure 4.21(b): Variation of pressure drop with time for M3 diagonal flow and polycarbonate plates.

Figure 4.21(c): Variation of pressure drop increase factor with time for M3 diagonal flow and polycarbonate plates.

In association with approach (ii), the variations of R_f , pressure drop and pressure drop increase factor (PDIF) with time are shown in Figures 4.21(a), (b) and (c) respectively. The M3-and-polycarbonate set-up exhibits no apparent roughness delay time, while the M3 set-up displays a clear period of negative R_f . The R_f values for the former set-up are more scattered than those of the latter. In addition, the M3-and-polycarbonate set-up exhibits higher clean pressure drop and overall pressure drop increase than those of the M3 set-up. The latter set-up seems to have reached an asymptotic pressure drop value, whereas that of the former is still increasing when the run was terminated. These discrepancies could be attributed to the many differences between the two set-ups, for example:

- (a) solution channel for M3-and-polycarbonate set-up was heated from one side only
- (b) polycarbonate plate has rectangular corrugations
- (c) M3 and polycarbonate plate have different surface structures

As photographs taken throughout the M3-and-polycarbonate experiment showing the chronological development of the deposit were not very distinct, a series of schematic diagrams were produced to illustrate this development. These diagrams are shown in

Figures 4.22 (a), (b) and (c) to illustrate the fouling pattern observed at 500, 1500 and 3000 minutes respectively. The solution enters and leaves via the bottom right and top left ports respectively.

Figures 4.22(a), (b) and (c):
Schematic diagrams showing the
chronological development of CaSO_4
deposit in diagonal flow M3-and-
polycarbonate set-up.

For the M3-and-polycarbonate set-up, the fouling layer was observed to grow in the following chronological sequence:

- (a) Deposits first appeared in the upper right region of the plate adjacent to the gasket as shown in Figure 4.22(a). This is the low-velocity, high-temperature area of the plate. It is also the plate region with the best conditions for fouling to occur.
- (b) From this plate corner, the deposits spread across to regions directly below the outlet port, as shown in Figure 4.22(b). Flow visualisation with ABS particles revealed that low flow velocity predominates in the upper right and lower left plate corners. Low shear stresses and hence, low removal rates are also expected to occur in these regions. Of these two corners, a higher temperature prevails in the upper one, which corresponds to a higher reaction rate and perhaps a higher degree of solution supersaturation (refer Figure 3.8), resulting in a higher deposition rate. Although flow velocity is high near the outlet port, the corresponding temperature is also high.
- (c) The deposits proceeded to establish themselves in the entire upper half of the plate, as shown in Figure 4.22(c).

High temperature and low flow velocity are the two criteria necessary for deposition to occur. When only one of the two is present, deposition may still occur but its rate would be very much dependent on the local conditions, such as surface roughness, surface material and degree of supersaturation.

It could be argued that due to the above listed differences between the normal set-up and that with the polycarbonate plate, the latter is inappropriate for studying fouling behaviour. As a result, approach (iii) was employed, in which a series of fouling experiments performed with the normal set-up, were terminated prematurely to examine the fouling pattern at various run-times. Figures 4.23 (a) and (b) show two pairs of fouled M3 plates with different run-times, namely (a) 930 minutes and (b) 3000 minutes. As the growth pattern displayed agrees well with that revealed via the M3-and-polycarbonate set-up, it confirms the validity of the above results.

Figures 4.23(a) and (b): Photographs showing fouling pattern in M3 diagonal flow plates at 930 minutes [150497] and 3000 minutes [300697] respectively.

4.7.2 Side Flow

Visual analysis of the side flow set-up reveals that, the fastest velocity prevails in the region directly between the inlet and outlet ports. Away from this flow region, slower velocity predominates. With reference to Figure 2.8(b), the upper and lower right plate corners are regions of low flow velocity.

Figure 4.24(a): Variation of fouling resistance with time for M3 side flow and polycarbonate plates.

Figure 4.24(b): Variation of pressure drop with time for M3 side flow and polycarbonate plates.

Figure 4.24(c): Variation of pressure drop increase factor with time for M3 side flow and polycarbonate plates.

With respect to approach (ii), the corresponding variations of R_f , pressure drop and PDIF with time are shown in Figures 4.24(a), (b) and (c) respectively. Both R_f and pressure drop increased steadily with time. Compared with the diagonal flow set-ups presented in the preceding section, some differences in R_f and pressure drop trends are observed. For instance, the M3 set-up has higher overall R_f and pressure drop increase than those of the polycarbonate plate set-up.

Figures 4.25(a), (b) and (c) show the chronological development of CaSO_4 deposit in side flow M3-and-polycarbonate set-up at about 500, 1500 and 3000 minutes respectively. The fouling deposit was observed to form in the following chronological sequence:

- Deposit growth commenced at the upper left region of the plate as shown in Figure 4.25(a). This area has both of the criteria necessary for fouling to occur, namely high temperature and low flow velocity.
- Deposition continued from this plate region downwards, and across to the area directly below the outlet port.
- Deposition occurred on the entire upper half of the plate.

Figures 4.25(a), (b) and (c):
Schematic diagrams
showing the chronological
development of CaSO_4
deposit in side flow M3-
and-polycarbonate set-up.

4.7.3 Effect of Port Position in Polycarbonate Plate Experiments

A comparison of the M3-and-polycarbonate results for diagonal and side flow with those of the M3 set-up discussed in Section 4.6, reveals several differences. For the latter set-up, side flow R_f values are higher than those of diagonal flow. An examination of Figure 4.26(a) reveals that the opposite is true for the polycarbonate runs. In addition, the side flow M3 set-up has higher pressure drop than its diagonal flow counterpart. Conversely, the opposite is shown to be valid for the polycarbonate runs in Figure 4.26(b).

These discrepancies might have been brought about by a combination of the differences between the two set-ups, which are listed in Section 4.7.1. Despite the opposite trends, the polycarbonate set-up reveals flow and fouling patterns in M3 PHEs, and hence provides new insight into the effect of flow distribution on fouling in PHEs.

Figure 4.26(a): Effect of port position on the variation of fouling resistance with time for polycarbonate plate set-ups.

Figure 4.26(b): Effect of port position on the variation of pressure drop with time for polycarbonate plate set-ups.

4.8 Fouling Mitigation in Alfa Laval PHEs

Several methods for mitigating fouling, such as salt addition, velocity surge, air injection, ion sputtering and magnetic treatment were looked into. The results obtained are presented and discussed in the following sections.

4.8.1 Effect of Added Salt on Fouling

As mentioned in Chapter 3, the CaSO_4 solution used in the present investigation was prepared from $\text{CaNO}_3 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 , instead of from $\text{CaSO}_4 \cdot x\text{H}_2\text{O}$ only. It is uncertain how different these two systems, namely $\text{CaNO}_3/\text{Na}_2\text{SO}_4/\text{H}_2\text{O}$ and $\text{CaSO}_4/\text{H}_2\text{O}$, would behave. However, it is well established that the solubility of a solution can be either decreased or increased by the addition of certain soluble salts, causing a salt-out or salt-in effect respectively. Adding a salt with a salt-in effect to CaSO_4 solution before introducing it into the PHE, would result in reduced fouling.

As the system used in this present study is $\text{CaSO}_4/\text{NaNO}_3/\text{H}_2\text{O}$, it is important to analyse the effect of NaNO_3 on CaSO_4 fouling. To do this, a computer program called SOLMIN88 (Alberta Research Council, 1988) was used to determine the change of CaSO_4 solubility in the presence of differing amounts of NaNO_3 .

Figure 4.27: Variation of CaSO_4 solubility with added NaNO_3 at 62°C (SOLMIN88).

As the solution entering and leaving the PHE has an average temperature of about 62°C , Figure 4.27 shows the calculated solubility of CaSO_4 at this temperature in the presence of added NaNO_3 . It is apparent that the solubility of CaSO_4 increases with increasing NaNO_3 .

Consequently, this effect was experimentally investigated with a solution inlet temperature and velocity of 51.5°C and 35 cm/s respectively.

As the CaSO_4 solution used was made from $\text{Ca}(\text{NO}_3)_2 \cdot 4\text{H}_2\text{O}$ and Na_2SO_4 , the addition of 30.0 g/l (or 0.35 mol/l) of NaNO_3 at the start of an experiment with 3.0 g/l (or 0.02 mol/l) initial CaSO_4 concentration, would give a total initial NaNO_3 concentration of 33.75 g/l (or 0.40 mol/l). Figures 4.28(a) and (b) show the effect of NaNO_3 addition on the variation of R_f and pressure drop with time respectively. Both R_f and pressure drop remained fairly constant throughout the course of the experiment, where the former fluctuated around $0.0\text{ m}^2\text{K/W}$. An inspection of the plates after the experiment, confirmed that only negligible deposition had occurred. Hence, it can be inferred that the addition of NaNO_3 has a significant influence on the crystallization fouling of CaSO_4 .

Figure 4.28(a): Effect of NaNO_3 (30.0 g/l) addition on the variation of fouling resistance with time.

Figure 4.28(b): Effect of NaNO_3 (30.0 g/l) addition on the variation of pressure drop with time.

Although the addition of NaNO_3 presented itself as a useful chemical fouling mitigation method for CaSO_4 , the amount of NaNO_3 used (30.0 g/l) was ten times more than the initial concentration of CaSO_4 (3.0 g/l). This is a very expensive way of mitigating fouling. In addition, disposal or treatment of the solution stream would also prove to be environmentally challenging. In order to verify the fouling mitigating feasibility of NaNO_3 further, more experiments are needed to confirm the CaSO_4 solubility trend predicted in Figure 4.27.

4.8.2 Effect of Velocity Surge on Deposit Removal

The effect of increasing the solution flow rate for a short time period at the end of a run was investigated. This velocity surge process which can also be called 'flushing', was to study whether the resulting increase in shear forces at the deposit interface could enhance the removal rate, such that existing fouling deposits in the PHE may be dislodged.

Figure 4.29(a): Effect of flushing on the variation of fouling resistance with time for a 3.0 g/l initial solution concentration.

Figure 4.29(b): Effect of flushing on the variation of pressure drop with time for a 3.0 g/l initial solution concentration.

Variation of R_f and pressure drop with time are shown in Figures 4.29(a) and (b) respectively, for an initial solution concentration of 3.0 g/l. Flushing was initiated immediately after 7000 minutes for an hour, after which the flow velocity was decreased back to its initial value for another hour, before the run was terminated. Both flush velocities were the fastest velocity achievable by the rig under the fouling conditions. The 'top' flush velocity (50 cm/s) was lower than that of the 'non-top' one

(80 cm/s), because the former run has a higher degree of blockage in its plate channel such that a faster flush velocity was impossible to attain.

When the solution velocity is increased, a corresponding increase in the heat transfer coefficient is induced, resulting in an apparent drop in R_f , as can be seen in Figure 4.29(a). According to Equation (2.19), this increase in flow velocity is accompanied by an increase in pressure drop, as shown in Figure 4.29(b). For both the 'top' and 'non-top' runs, the R_f values seem to be unaffected by the flushing process, whereas the corresponding pressure drops were found to decrease after the original velocities were resumed. This implies that flushing had removed some of the fouling deposits blocking the solution flow in the plate channel. Moreover, the removed deposits were probably located in regions where the contribution to heat transfer was insignificant. Photographs of the flushed plates can be seen in Figures 4.30(a) and (b).

Figure 4.30:

Photographs of flushed plates for

(a) 3.0g/l(top) [121097],

(b) 3.0g/l [260297],

(c) 2.8g/l(top) [020497].

The effects of flushing was also investigated for an initial solution concentration of 2.8 g/l. Figures 4.31(a) and (b) show the variation of R_f and pressure drop with time respectively, for such a run. It is apparent that flushing did not bring about a significant

change in the pressure drop. This could be explained by examining a photograph of the flushed plates, as shown in Figure 4.30(c). As can be seen, less deposits are present for removal in this run, than those in the 3.0-g/l runs. In addition, the deposits present are located in the top corner away from the main flow path of high velocity. Hence, they are sheltered to a certain extent from the effects brought about by the flush. It can therefore be inferred that, as a deposit removal method, velocity surge seems to be more effective if more deposits are present, initially.

Figure 4.31(a): Effect of flushing on the variation of fouling resistance with time for a 2.8 g/l initial solution concentration.

Figure 4.31(b): Effect of flushing on the variation of pressure drop with time for a 2.8 g/l initial solution concentration.

4.8.3 Effect of Continuous Air Injection on Deposit Removal

Experiments with continuous air injection into the PHE solution flow were performed, to determine whether this could be used as a method for mitigating fouling. It was anticipated that the presence of air bubbles in the solution may further enhance the turbulence level in the plate channel, such that the shear stresses and the removal rate may also increase. Referring to Figure 3.2, air was injected into the solution line upstream of its inlet to the PHE. This was done on a continuous basis at various air flow rates for the full duration of the experiment.

$\dot{V}_g$ [l/min]	p_g [bar]	ρ_g [kg/m ³]	$\dot{m}_g$ [kg/min]	$\dot{x}$ flow quality	ϵ_{gl} hold up
60	3.75	4.38	0.2628	0.0844	0.95
50	3.23	3.77	0.1885	0.0620	0.94
40	2.55	2.98	0.1192	0.0401	0.93
30	2.11	2.46	0.0738	0.0252	0.91
20	1.68	1.96	0.0392	0.0136	0.87
10	1.45	1.69	0.0169	0.0059	0.77
5	1.33	1.55	0.0078	0.0027	0.63
4	1.32	1.54	0.0062	0.0022	0.58
2	1.31	1.53	0.0031	0.0011	0.41
1	1.30	1.52	0.0015	0.0005	0.25

Table 4.4: Range of air flow rate, flow quality ($\dot{x}$) and hold-up (ϵ_{gl}).

Table 4.4 shows the range of air flow rates achievable by the rig and their corresponding flow quality $\dot{x}$, and hold-up ϵ_{gl} , which were calculated using Equations (2.25) and (2.26) respectively. Experiments with continuous air injection were performed with air flow rates of 1.5 l/min, 10 l/min and 30 l/min. The superficial solution liquid flow velocity was measured to be 20.0 cm/s before air was injected. Therefore, the actual solution velocity in the PHE should be higher.

The variation of R_f , pressure drop and pressure drop increase factor (PDIF) with time are shown in Figures 4.32(a), (b) and (c) respectively. At 30 l/min, the air flow rate was not very stable. This caused the R_f values to be too scattered to show any existing trend. As a result, these data were ignored. Despite this, the two-phase-flow R_f values appear to be slightly lower than those of the single phase flow. Furthermore, the corresponding pressure drop increases with increasing air flow rate.

Figure 4.32(a): Effect of continuous air injection on the variation fouling resistance with time.

Figure 4.32(b): Effect of continuous air injection on the variation pressure drop with time.

Figure 4.32(c): Effect of continuous air injection on the variation of pressure drop increase factor with time.

It was found that due to the presence of fouling, higher air flow rates could not be kept constant throughout the entire course of the experiment, e.g. 10 l/min [120897] and 30 l/min [090897]. As deposits build up in the plate channel, the pressure drop across the channel increases, resulting in a decrease in the air flow rate which was supplied at a constant pressure of 80 psi. Hence, for runs where a constant air flow rate was not maintained, the pressure drop data may be misleading. Figure 4.33 shows the actual air flow rates over the duration of the runs.

Figure 4.33: Variation of air flow rate with time for continuous air injection runs.

Surprisingly, an inspection of the fouled plates reveals that, instead of mitigating fouling, continuous air injection promoted more deposition than single phase flow. Bearing in mind the different run-times of the experiments, photographs of their respective fouled plates are shown in Figures 4.34(a) to (e). A closer look at the fouling deposits indicates that the two-phase-flow deposit patterns are rather different from those of the single phase, as highlighted in Figure 4.34(f). This may be due to flow phenomena in the distribution of air and water, and complex interactions between two phase flow and fouling. Nevertheless, it is conceivable that there may be an air flow rate range for two phase flow to be capable of mitigating fouling in PHEs.

Figure 4.33: Photographs of single-phase-flow and two-phase-flow fouled plates.

4.8.4 Effect of Air Flush on Deposit Removal

For experimentation with air flush, a high air flow rate was introduced into the PHE solution flow for an hour after fouling had occurred. Before the run was terminated, only the solution was allowed to circulate the rig, at its original flow rate for another hour. Similar to the continuous air injection runs, the presence of air bubbles in the solution is expected to increase shear forces at the deposit interface, and hence increase the removal rate. Air was injected into the solution line upstream of its entry into the heat exchanger as shown in Figure 3.2.

Figure 4.34(a): Effect of air flush on the variation of fouling resistance with time.

Figure 4.34(b): Effect of air flush on the variation of pressure drop with time.

The superficial solution flow velocity was 0.20 m/s, and the initial solution concentration was 3.0 g/l. Air at a flow rate of 50 l/min ($8.33 \times 10^{-4} \text{ m}^3/\text{s}$) was introduced into the solution line at 3000 minutes. The variation of R_f and pressure drop with time are shown in Figures 4.34(a) and (b) respectively. The two experiments shown ([240697] and [150897]) have identical operating conditions, and apart from the air flush, can be seen to be quite reproducible. The pressure drop decreased significantly after the introduction of the large air flow rate, and did not increase again before the run was terminated. It is worth mentioning that the CaSO_4 concentration was only 2.7 g/l before, during and after the air flush. Since the removed fouling deposits may not have redissolved, the concentration of CaSO_4 may not have been high enough for CaSO_4 deposition to occur again. Nevertheless, air flush was found to remove a pre-existing fouling layer successfully, and hence can be considered as an option for mitigating fouling.

4.8.5 Ion-Sputtered Plates

Four different combination of ions were sputtered onto M3 plates for the investigation of a novel fouling mitigation method. The four types of ion-sputtered plates tested are diamond-like-carbon (DLC), diamond-like-carbon and a silicon polymer (DLC & HMDSO), diamond-like-carbon and C_2F_2 (DLC & C_2F_2), and tetrahedral-amorphous carbon (ta-C). As mentioned in Sections 2.2.6 and 2.2.7, the ion-sputtered plates have lower surface energies, and hence are less adhesive for the deposits.

This series of runs were performed with a 35 cm/s solution flow velocity and a 2.8 g/l initial solution concentration. Each of the runs exhibited a clean overall heat transfer coefficient, U_o ranging between 5250 to 5300 $\text{W}/\text{m}^2\cdot\text{K}$, and a clean pressure drop of 0.200 ± 0.005 bar. Hence, it can be said that all the runs were performed under identical process conditions. The variations of R_f and pressure drop with time are displayed in Figures 4.35(a) and (b) respectively. Despite the data scatter, there seemed to be no apparent difference in fouling behaviour between the untreated stainless steel (SS) plates and the ion-sputtered plates, with the exception of ta-C sputtered plates. The latter has the lowest overall R_f and pressure drop values. This is further confirmed via

an examination of the various fouled plates, of which some are shown in Figures 4.36(a), (b) and (c). The fouled ta-c sputtered-plates have the least amount of deposit present, when compared with those of the untreated SS and DLC plates.

Figure 4.35(a): Effect of ion-sputtering on the variation of fouling resistance with time.

Figure 4.35(b): Effect of ion-sputtering on the variation of pressure drop with time.

Figures 4.36(a), (b) and (c):
Photographs of fouled plates for
(a) untreated stainless steel [020497],
(b) DLC [131196] and
(c) ta-C [031097].

Although the results pertaining to ta-C sputtered plates appear promising, repeatability of their performance needs to be established before conclusions can be drawn on their fouling mitigation potential. As for the other ion-sputtered plate types investigated, it was observed that CaSO_4 deposits tend to grow on ~~the surfaces~~ rather than on clean surfaces. Hence, once deposition is initiated, the surface plays a less important role in the deposit growth process. In addition, the numerous contact points in PHEs provide excellent initiation areas for fouling to occur, as mentioned in Section 2.2.6.3.3. As a result of the unique combination of the plate geometry, and the fouling behaviour of CaSO_4 , ion-sputtered plates have yet to prove themselves capable of mitigating fouling,

During ion-sputtering, the presence of the plate corrugations made it impossible for ion sources to reach all plate surfaces evenly. This may lead to the production of inhomogeneous or patchy layers, which could mask the advantage of the lower stickiness of the plates. It is therefore of vital importance to establish the roughness of ion-sputtered plates, and the homogeneity of the coating layer.

4.8.6 Effect of Velocity Surge on Ion-Sputtered Plates

Although ion-sputtering does not seem to be able to reduce fouling, it was hoped that velocity surge or 'flushing' would at least be able to remove existing deposits more efficiently than for the untreated stainless steel plates. Results and discussions for flushing untreated plates can be found in Section 4.8.2.

Figure 4.37(a): Effect of velocity surge on the variation of fouling resistance with time for a 2.8 g/l initial solution concentration.

Figure 4.37(b): Effect of velocity surge on the variation of pressure drop with time for a 2.8 g/l initial solution concentration.

Variation of R_f and pressure drop with time for DLC & C_2F_2 ion-sputtered plates are shown in Figures 4.37(a) and (b) respectively. The solution flow velocity was 35 cm/s, and a 2.8 g/l initial concentration was employed. Run [040197] was flushed at 7000 minutes for an hour with a solution flow velocity of 86 cm/s. The original flow velocity was reinstated for an hour before the run was finally terminated. There seems to be little difference in R_f as a result of the flush, which is compatible with similar experiments on untreated plates, see Section 4.8.2. In addition, the pressure drop was reduced by only a small fraction of its value.

Results obtained with the untreated plates suggest that velocity surge may be more efficient when a larger amount of deposit is present for removal. Bearing this in mind, the next set of runs with ion-sputtered plates were performed with a higher concentration i.e. 3.0 g/l, at 35 cm/s solution flow velocity. Figures 4.38(a) and (b) show the dynamic variations of R_f and pressure drop respectively.

Figure 4.38(a): Effect of velocity surge on the variation of fouling resistance with time for a 3.0 g/l initial solution concentration.

Figure 4.38(b): Effect of velocity surge on the variation of pressure drop with time for a 3.0 g/l initial solution concentration.

Referring to Figures 4.38(a) and (b), run [190197] was flushed twice at 1585 and 1665 minutes with a flow velocity of 44 cm/s, while run [220197] was given a 52-cm/s flush at 1640 minutes. Each flush process lasted one hour, before the original velocity was restored. Unexpectedly, both R_f and pressure drop continue to increase, following the resumption of the original velocity. Although this is contrary to the results obtained for the untreated plates, it is consistent with the findings of Taborek *et al.* (1972b) mentioned in Section 2.2.6.3.2. They found an extremely rapid increase in fouling once the original velocity was restored after an hour of flushing. They postulated that crystals which have endured the flush, provided perfect sites for additional nucleation to occur.

Newly sputtered plates were used to test the validity of ion-sputtering as a fouling mitigation method, see Section 4.8.5. The fouled plates were then soaked and rinsed with distilled water before being used again for the flush runs. As suggested by Taborek *et al.* (1972b), it is therefore possible that some nuclei still remained on the surfaces of the plates used for the flush runs. Although the ion-sputtered plates have not been cleaned by any abrasive means, deposition and removal processes may have

altered the surface condition of these plates. This would probably explain why the used ion-sputtered plates have patchy surfaces. This further emphasises the importance of plate surface condition on fouling. Photographs of the fouled plates are displayed in Figures 4.39(a), (b) and (c).

Figures 4.39(a), (b) and (c):
Photographs of fouled plates for
(a) untreated stainless steel [120197],
(b) DLC & C₂F₂ [190197] and
(c) DLC & HMDSO [220197].

It can be concluded that ion-sputtering has not proved itself capable of mitigating fouling in PHEs, and also flushing of deposits attached to such plates has not been successful in terms of fouling reduction. However, these results may have been influenced by the surface condition of the plates as they undergo ion-sputtering and cleaning.

4.8.7 Effect of Magnetic Treatment on Fouling

The set-up of the commercial magnetic device for reduction of fouling is shown in Figure 4.40. The induction coil was wrapped around the double pipe heat exchanger (DPHE) twice for a certain number of turns. Apparently, this coil could be placed

Figure 4.40: Photograph of magnetic device set-up.

anywhere along the solution circuit. The field power selection available ranged from 3.9 to 72.9 mA. In order to utilise the full effect of this device, the highest power was chosen. Depicted in Figure 4.41, is a schematic diagram describing how the magnetic device works.

Illustrated in Figures 4.42(a) and (b) are the dynamic variations of R_f and pressure drop respectively. The experiment with the magnetic device installed appears to foul slightly more than the untreated run. However, since only one run with the magnetic device was performed, it is

inconclusive if such a device could reduce the formation of CaSO_4 fouling in PHEs.

Figure 4.41: Schematic diagram showing how the magnetic device works.

Figure 4.42(a): Effect of magnetic treatment on the variation of fouling resistance with time.

Figure 4.42 (b): Effect of magnetic treatment on the variation of pressure drop with time.

4.9 Flat Plate Heat Exchangers

Distinguishing features of corrugated channels include unique geometry, intricate flow patterns, complex temperature distribution, and non-uniform deposit formation. As a result, analysis of the plate design-fouling relationship can be greatly simplified by experimentation with flat plates.

A typical flat plate used in this investigation is shown in Figure 3.4(c). Other relevant plate details are given in Section 3.1.2. Free of corrugations and contact points, flat plates differ from their corrugated counterparts in a number of ways:

- (i) no support points to maintain plate channel gap
- (ii) lack of mechanical stability; therefore thicker plates are required
- (iii) less mixing, lower turbulence level, and hence lower thermal efficiency
- (iv) lower pressure drop across channel
- (v) elimination of local low-velocity areas around contact points

With nothing to maintain the channel gap, flat plates have a tendency to bend when assembled as a flat PHE. Although the overall distance of the plate pack (distance between pressure plate and frame plate) was measured and uniformity was ensured for all runs, it is impossible to guarantee intra-channel and inter-channel gap uniformities.

Two identical sets of flat plates (set A and set B) were used for heat transfer and pressure drop analyses. Each set consists of 4 plates with two of them being end-plates. Set A3 refers to a set-up of 3 channels with set A plates. The same applies for set A2, set A1, and set B3.

4.9.1 Heat Transfer

A heat transfer analysis similar to those for the Alfa Laval P01 and M3 plates presented in Section 4.1.1, was performed for the flat plates. In the absence of corrugations, the flat plates have an enlargement factor, μ_e of unity. Moreover, only diagonal flow was investigated, with a 41.5°C cold side inlet temperature.

The curve-fitted overall heat transfer coefficients, U_c were determined using Equations (2.4), (2.9), (2.14) and (2.16). The latter can be rearranged into the form of Equation (2.9) by combining the constant 0.036 with $(d_h/l)^{0.055}$:

$$\begin{aligned}
 Nu &= 0.036 \cdot (Re)^{0.8} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} \cdot \left(\frac{d_h}{l} \right)^{0.055} && \text{from (2.16)} \\
 &= \left[0.036 \cdot \left(\frac{d_h}{l} \right)^{0.055} \right] \cdot (Re)^{0.8} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} \\
 &= c_1 \cdot (Re)^{0.8} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} && \text{similar to (2.9)}
 \end{aligned}$$

With lower turbulence, the ratio μ/μ_{wall} attained a maximum of 1.45. Consequently, in curve-fitting the values for c_1 and c_2 in Equation (2.9), c_3 and c_4 were kept constant at 0.33 and 0.14 respectively, according to Equation (2.16). Figure 4.43 shows the comparison between U_m and U_c for a Reynolds numbers, Re range of 1600 to 7700. For set A3, the curve-fitted values of c_1 and c_2 are 0.0245 and 0.80 respectively:

$$Nu_{flat(A3)} = 0.0245 \cdot (Re)^{0.80} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} \quad (4.5)$$

In order to curve-fit the lower U_m values pertaining to set B3, corrections were made to the flow velocities, u and hydraulic diameters, d_h . This is to account for its non-uniform inter-channel gaps, and is substantiated by the improvement in the corrected curve-fit (curve-fitted & corrected). Instead of 2.4 mm, the solution/cold channel gap was adjusted to 3.3 mm, and hence the water/hot channel gaps were adjusted to 1.95 mm each. As constant mass flow rates were maintained in adjusting the flow velocities, the corresponding Re numbers remain unchanged according to Equation (2.24). However, the film heat transfer coefficient changes with b_p according to Equation (2.10). For set B3, c_1 and c_2 are 0.0215 and 0.80 respectively:

$$Nu_{flat(B3)} = 0.0215 \cdot (Re)^{0.80} \cdot (Pr)^{0.33} \cdot \left(\frac{\mu}{\mu_{wall}} \right)^{0.14} \quad (4.6)$$

The curve-fits for both sets A and B, compare favourably with the Dittus-Boelter equation for forced convective turbulent flow in tubes (Collier, 1992):

$$Nu_{DB} = 0.023 \cdot (Re)^{0.8} \cdot (Pr)^{0.33} \quad (4.7)$$

All fouling experiments using flat plates (Sections 4.10 to 4.14) were performed with a solution/cold flow velocity of 35 cm/s, and a corresponding Re of 2800. As both Set A and set B plates were used in the fouling experiments without distributors, the corresponding clean overall heat transfer coefficient, U_o has a range of 850 to 1050 W/m^2K , refer Figure 4.43.

Figure 4.43: Comparison of measured and curve-fitted overall heat transfer coefficients as a function of Reynolds number for flat diagonal flow plates.

Figure 4.44 serves to verify the difference in heat transfer between the 2 mm thick plates used in this investigation, and those of 1 mm thick used by Bansal (1994). For a solution/cold side velocity of 35 cm/s, there is however only a 5% difference between the two plate types. Due to data scatter, the difference in heat transfer between these two plate types, can thus be assumed to be negligible.

Figure 4.44: Effect of plate thickness on heat transfer performance.

Figure 4.45: Variation of $Nu \cdot Pr^{-0.33}$ with Re for flat and M3 diagonal flow plates.

Figure 4.45 shows the variation of $Nu \cdot Pr^{-0.33}$ with Re for flat (sets A and B) and M3 diagonal flow plates. The latter is included to show the heat transfer enhancement brought about by the presence of corrugations and reduction in plate thickness. With reference to Figures 4.1 and 4.43, for the same nominal flow velocity of 35 cm/s ($Re \sim 2800$), the corrugated commercial plates have an overall clean heat transfer coefficient of about $5.25 \times 10^3 \text{ W/m}^2\text{K}$, which is approximately 5 times higher than that of the flat plates, namely about $1.0 \times 10^3 \text{ W/m}^2\text{K}$ only.

4.9.2 Pressure Drop

For the same reasons that flat plates possess a lower thermal efficiency than commercial corrugated plates, the former exhibit lower pressure drops than the latter. Figure 4.46 shows the variation in friction factor for the two plate types. Similar to the analysis of heat transfer, pressure drop analysis for the flat plates was also performed in accordance to those of the Alfa Laval plates in Section 4.1.2. The measured pressure drop values include port losses, and losses due to changes in flow direction. For set A3 plates, the best-fit values for c_5 and c_6 of Equation (2.20) are 0.64 and 0.20 respectively:

$$f_{flat} = \frac{0.64}{Re^{0.20}} \quad (4.8)$$

Figure 4.46: Comparison of friction factor for flat and M3 diagonal flow plates.

With a wider solution/cold channel gap, the uncorrected data for pressure drop versus flow velocity of set B3 are lower than those of set A3, as can be seen in Figure 4.47. Since experimentation with flat plates (without distributors) was performed with both set A and set B plates, the pressure drop registered was always about 0.025 ± 0.002 bar. This is approximately 8 times lower than those registered by the Alfa Laval plates (about 0.205 bar) under identical process conditions. In addition, a series of pressure

drop runs were performed using set A3 plates, with different number of channels set up in the PHE. It is apparent that pressure drop decreases with decreasing number of channels. Provided that a uniform channel gap was maintained, this set of results indicates the influence of port pressure loss on the overall pressure loss. However, this cannot be validated without local pressure drop measurements, which cannot be obtained with the present experimental rig.

Figure 4.47: Variation of pressure drop with nominal flow velocity for flat diagonal flow plates with different number of channels.

A summary of the constants c_1 to c_6 for flat (set A3) diagonal plates is given in Table 4.1. Table 4.2 provides a summary of the operating parameters, while the experimental conditions for all runs performed can be found in Table 4.3.

The pressure drop values reported by Bansal (1994) are all higher than those of the present investigation by up to a maximum of 530%, for example, at 19, 35 and 70.5 cm/s, he reported pressure drop values of 0.063, 0.063 and 0.115 bar respectively. As his pressure drop values did not decrease when the flow velocity was decreased from 35 to 19 cm/s, the lower limit of the pressure transducer used may have been exceeded. Therefore, pressure drop values reported by Bansal (1994) for flat plates should be referred to with caution.

4.10 Fouling in Flat Plate Heat Exchangers

In this section, a typical set of flat PHE runs is shown and discussed. Furthermore, comparisons were made with an Alfa Laval M3 diagonal flow PHE experiment performed under identical process conditions.

Figure 4.48(a): Variation of fouling resistance with time for flat and M3 diagonal flow plates.

Figure 4.48(b): Variation of pressure drop with time for flat diagonal flow plates.

Similar to the commercial plate experiments, only raw data points are displayed for the flat plates, unless otherwise mentioned. In addition, only diagonal flow was investigated with a 35-cm/s solution flow velocity, 14.6-cm/s hot water flow velocity, 41.5°C solution inlet temperature, and 87°C hot water inlet temperature. Figures 4.48(a) and (b) show the dynamic variation of R_f and pressure drop respectively. All the runs exhibited have an initial solution concentration of 3.0 g/l. The flat plates were sanded with 240 emery paper. One of the flat plate experiments shown was terminated prematurely at 3000 minutes, while the other one was allowed to continue for 7000 minutes. Although there are discrepancies in the R_f values for these two runs, their pressure drop data agree quite well.

For the 7000-minute experiment with flat plates, R_f decreases rapidly and attains a negative asymptote at about 600 minutes. Together the R_f and pressure drop trends imply that, fouling did not occur along the main flow path until after 4000 minutes. This is substantiated by Figures 4.49(a) and (b) which show photographs of the fouled plates at 3000 and 7000 minutes respectively. It is interesting to note that, throughout the entire course of the run, the increased turbulence brought about by the deposits penetrating into the laminar sublayer, outweighs their additional thermal resistance. Although the flat plates have lower thermal efficiency when compared to the M3 plates, R_f values of the former are lower than those of the latter.

On the other hand, pressure drop across the flat PHE is significantly lower than that of the M3 PHE. It is apparent from Figure 4.48(b) that the pressure drop for the former commenced at 0.025 bar and only increased to 0.031 bar. With reference to Figure 4.6(b), the corrugated PHE has an initial pressure drop of 0.200 bar, which increased to a value of 0.820 bar at 7000 minutes.

Figure 4.49(a) indicates that deposition was initiated at the top plate corner opposite the outlet port. This agrees with fouling patterns of the commercial plates, see Figure 4.7. As fouling proceeds, the deposit front propagates from the side walls towards the main flow region. An interesting fouling pattern is observed at the plate inlet region, where the deposits formed a wavy front. As this pattern is more pronounced when a

higher solution concentration of 3.2 g/l was used, the fouling pattern of run [100597] is shown in Figure 4.49(c).

Figures 4.49(a) and (b):
Photographs showing fouled flat plates with run-times of
(a) 3000 minutes and
(b) 7000 minutes.

Figure 4.49(c): Fouling pattern at plate inlet region [100597].

Although plate corrugations are effective in increasing turbulence, they also bring about a corresponding increase in pressure drop. Furthermore, they give rise to contact points which can initiate fouling. However, in terms of heat transfer performance, the corrugated plates are unmatched by their flat counterparts. As a result of fouling, the overall heat

transfer coefficient, U of the M3 PHE shown in Figure 4.48(a) dropped from 5350 to 4850 W/m²·K. Conversely, U increased from 940 to 1000 W/m²·K in the flat PHE due to fouling. Nevertheless, the lowest M3 U is still about 5 times higher than the highest U for flat plates.

4.11 Effect of Surface Condition on Fouling in Flat PHEs

As mentioned in Section 2.2.6.3.2, the surface condition of heat exchangers has a profound effect on the resulting fouling behaviour. Therefore, fouling experiments were performed using two sets of plates with different surface roughnesses, namely original smooth finish plates of average roughness in flow direction, $R_a = 0.130 \mu\text{m}$, and plates sanded with 240 emery paper, $R_a = 0.080 \mu\text{m}$. A comparison of different plate roughness can be seen in Table 3.2.

The effects of surface roughness on the variation of R_f and pressure drop with time are displayed in Figures 4.50(a) and (b) respectively. The initial solution concentration is 3.2 g/l. It is apparent that the original plates have significantly higher R_f and pressure drop values than those of the sanded plates. Hence, it can be inferred that rougher plates fouled substantially more than smoother ones. This is confirmed by photographs of the fouled plates in Figures 4.51(a) and (b). As can be seen, deposition can be found over most of the original plate heat transfer area, while the sanded plates only have deposition concentrating in the plate corners adjacent to the ports.

Figure 4.50(a): Effect of surface roughness on the variation fouling resistance with time.

Figure 4.50(b): Effect of surface roughness on the variation of pressure drop with time.

(a) $R_a = 0.130\ \mu\text{m}$

(b) $R_a = 0.080\ \mu\text{m}$

Figures 4.51(a) and (b):
Photographs showing
the effect of surface
roughness on fouling

Following the start of the respective experiments, the R_f values of both plate types decreased. Unlike the sanded plates which have negative R_f values throughout the entire run, R_f of the original plates eventually attained positive values at around 4500 minutes, thus defining what is known as a roughness delay time. As the main flow path is a crucial region for heat transfer, it is possible that when ample fouling occurs in this area of the original plates, the corresponding R_f values become positive.

On the other hand, Bansal (1994) only used flat plates with their original surfaces in his experiments. Under identical process conditions, the corresponding R_f values were about 50% higher than those of the present investigation. Since Bansal did not report a surface roughness value for his plates, it is difficult to compare his data to those displayed in Figure 4.50(a). Nevertheless, in his experiments with Teflon coatings of different roughnesses on flat plates and on APV TR1 plates, Bansal (1994) also observed that rougher surfaces fouled significantly more than smoother ones. Therefore, it can be concluded that fouling increases with increasing plate roughness.

4.12 Effect of Solution Concentration on Fouling in Flat PHEs

The effects of solution concentration on the variation of R_f and pressure drop with time are displayed in Figures 4.52(a) and (b) respectively. All the plates used for this set of results were sanded with 240 emery paper in the main flow direction. Increasing the initial solution concentration from 3.0 to 3.2 g/l, produced hardly any difference in the corresponding R_f values and trends. However, it is apparent from Figure 4.52(b) that pressure drop increases with increasing solution concentration. Therefore, it can be inferred that similar to the Alfa Laval plates, the flat plates experience increased fouling with increasing solution concentration.

Figure 4.52(a): Solution concentration effect on fouling resistance variation with time.

Figure 4.52(b): Solution concentration effect on pressure drop variation with time.

(a) 3.0 g/l [230497] (b) 3.2 g/l [100597]

Figures 4.53(a) and (b): Photographs showing the effect of solution concentration on fouling.

From the photographs of the fouled plates shown in Figures 4.53 (a) and (b), more deposits appear to have formed, especially in the main heat transfer area, on the plates with 3.0 g/l solution concentration than on those with 3.2 g/l. Therefore, for the two sets of plates to exhibit similar R_f values, deposition in the main heat transfer area does not

seem to contribute towards the enhancement/reduction of heat transfer. From the pressure drop trends it can be inferred that, although the deposits for the 3.2 g/l solution concentration run occurred in regions unimportant for heat transfer, they were able to offer resistance against fluid flow, thus resulting in a higher pressure drop.

4.13 Flow and Fouling Patterns in Flat PHEs

In order to visualise flow and fouling patterns in flat PHEs, a flat polycarbonate plate was constructed, and assembled with the flat metal plates. These plates were set up in a similar manner to the Alfa Laval M3 plates and corrugated polycarbonate plate as mentioned in Sections 3.1.2 and 4.7. These experiments serve to improve understanding of the dynamic behaviour of fouling.

4.13.1 Flow Patterns

(a) 35 cm/s

(b) 100 cm/s

Figures 4.54(a) and (b):

Flow patterns of ABS particles in flat plate channels at different velocities.

Fluid flow characteristics in flat plate channels have been investigated via experimentation with ABS particles in water. This was done following the procedure and conditions outlined in Section 4.7 for Alfa Laval M3 diagonal and side flow plates. In addition to the low speed camera (Sony DXC930P 3CCD True Colour), a high speed Kodak Ektapro Motion Analyser HS4540 camera with Leica Monozoom 7 lens was also used.

Flow visualisation was performed at two different velocities, namely 35 and 100 cm/s, as illustrated in Figures 4.54(a) and (b) respectively. Similar to flow patterns in Alfa Laval M3 PHEs, the flow was observed to be fastest between the bottom inlet and top outlet ports, and slower away from this main

flow path, especially in the left lower and upper right plate corners. The two flow patterns exhibited are very similar, each with a large recirculation zone in the lower plate corner, and a much smaller one in the upper corner. At 35 cm/s flow velocity, the

former recirculation zone has a vertical length of about 10 cm. This length becomes slightly longer, measuring approximately 11 cm at a flow velocity of 100 cm/s.

With the high speed camera, an interesting flow pattern is observed at the port inlet region. As illustrated in Figure 4.54(c), apart from the large recirculating zone on the far left, there are two regions of higher flow velocity. One of these regions is next to the recirculation zone, and the other is at the far right next to the plate gasket. Between these regions, a lower flow velocity zone exits.

Figure 4.54(c):

Schematic flow pattern at port inlet region of a flat plate heat exchanger.

Flow entering through the inlet port, hits the end plate and is forced to flow upwards. Instead of flowing upwards immediately, some of the fluid first flows towards the space between the inlet port and the gasket surrounding it. After hitting the gasket and being forced by fresh inflowing fluid, they then make their way upwards along this space. As a constant flow rate is maintained, in making this detour, the flow velocity increases, hence, giving rise to this unique flow pattern. Fouling patterns near the inlet port region as shown in Figure 4.49(c), support this flow pattern. As deposition

decreases with increasing flow velocity, more deposits are found in the recirculating and the low-flow-velocity zones than in the two high-flow-velocity regions.

Due to the local corrugation design near Alfa Laval plate ports, the spaces between the ports and their surrounding gaskets are dead flow zones. As a result, the flow pattern illustrated in Figure 4.54(c) should be non-existent in Alfa Laval PHEs. This was verified using the high speed camera.

4.13.2 Fouling Patterns

Fouling experiments were performed with the polycarbonate plate, at an initial solution concentration of 3.2 g/l. The variation of R_f , pressure drop and pressure drop increase factor (PDIF) with time are shown in Figures 4.55(a), (b) and (c) respectively.

Figure 4.55(a): Variation of fouling resistance with time for flat diagonal and polycarbonate plates.

Figure 4.55(b): Variation of pressure drop with time for flat diagonal and polycarbonate plates.

Figure 4.55(c): Variation of pressure drop increase factor with time for flat diagonal and polycarbonate plates.

As the solution channel of the flat-and-polycarbonate set-up was heated from only one side, it was expected to foul less than the standard flat PHE set-up. However, on the contrary, the former displays higher R_f values than the latter. Although the flat metal plates and the polycarbonate plate have different surface structures, fouling was observed to grow on the former rather than the latter. Photographs of the fouled plates show that deposition for the polycarbonate set-up occurred over a larger plate area, see Figures 4.56(a), and (b). It is unclear why more deposition occurred for the polycarbonate set-up than for the standard set-up.

In addition, lower clean pressure drop and lower overall pressure drop increase are exhibited by the polycarbonate set-up. Because the standard set-up has 3 channels, while the polycarbonate set-up has only two, the latter may have less port pressure loss and hence a lower clean pressure drop, as illustrated in Figure 4.47 and discussed in Section 4.9. It is also possible that the two set-ups have different channel gaps. From Figures 4.56(a), and (b), it can be seen that for the polycarbonate set-up, fouling occurred away from the main flow path and is absent in areas near the ports. This may be the reason why it has a lower overall pressure drop increase than that of the standard set-up.

(a)

(b)

Figures 4.56(a), and (b):
Photographs of fouled plates for
different set-up
(a) standard
(b) flat-and-polycarbonate

(a) 1500 minutes

(b) 3600 minutes

(c) 7000 minutes

Figures 4.57(a), (b)
and (c):
Photographs showing
the development of
 CaSO_4 fouling deposits
in flat diagonal flow
plate channels.

Shown in Figures 4.57(a), (b) and, (c) are photographs taken during the polycarbonate experiment, showing the development of the CaSO_4 fouling deposit in flat diagonal flow plates. Figures 4.58(a), (b) and (c) show the corresponding schematic diagrams for Figures 4.57(a), (b) and (c) respectively.

Figures 4.58(a), (b) and (c):
Schematic diagrams showing the development of CaSO_4 fouling deposits in flat diagonal flow plate channels.

For the flat diagonal flow plates, the fouling layer was observed to grow in the following chronological sequence:

- (i) Fouling first occurred in the top right plate corner where high temperature and low flow velocity prevailed. Following this, deposits started to appear in the other low-flow velocity region, namely the bottom left plate corner.
- (ii) Deposition propagated towards the plate centre.
- (iii) Deposits continued to grow towards the main flow path, with a faster rate in the top half of the plate where the prevailing temperature is higher.

Similar to fouling pattern development on Alfa Laval plates, fouling on flat plate occurs in regions where the flow velocity is low and where the temperature is high.

4.13.3 Effect of Plate Channel Gap on Fouling Patterns

To examine the effect of plate channel gap, h_f on fouling patterns, an experiment was performed using a polycarbonate set-up of 0.0020-mm channel gap instead of the standard 0.0024-mm. The latter channel gap was maintained for all results presented in

Section 4.13.2. This reduced channel gap was chosen because it produces approximately the same clean pressure drop as that of the standard set-up.

Figure 4.59(a): Variation of fouling resistance with time for different plate channel gap.

Figure 4.59(b): Variation of pressure drop with time for different plate channel gap.

Figure 4.59(c): Variation of pressure drop increase factor with time for different plate channel gap.

Figure 4.60:

Fouled plate with 0.0020-mm channel gap.

The variation of R_f , pressure drop and PDIF with time are displayed in Figures 4.58(a), (b) and (c) respectively, for $b_f = 0.0020$ mm and $b_f = 0.0024$ mm polycarbonate, and standard flat-plate set-ups. All these experiments were conducted under identical process conditions, with a 3.2-g/l initial solution concentration and a 35-cm/s nominal flow velocity. Lower R_f and lower overall pressure drop increase were observed for the set-up with lower b_f than those of higher b_f . This is confirmed by comparing Figure 4.56(b) and Figure 4.60. With a constant flow rate, a smaller b_f will result in a faster flow velocity, higher turbulence and shear stresses, and consequently higher removal rates. Therefore, fouling decreases with decrease in b_f , provided that a constant mass or volumetric flow rate is maintained.

The chronological development of CaSO_4 fouling deposit in the polycarbonate set-up with $b_f = 0.0020$ mm, is very similar to that of the set-up with $b_f = 0.0024$ mm presented in Section 4.13.2.

4.14 Effect of Distributor Designs on Fouling in Flat PHEs

In order to investigate the effect of distributors adjacent to plate inlet and outlet ports on flow distribution, and hence the resulting fouling behaviour in the plate channels, a series of different distributor designs were incorporated onto the flat plates. This was done by means of gaskets as mentioned in Section 3.1.2. The fouling patterns obtained are compared with those simulated using CFD in Chapter 6. Five distributor designs were examined, designs A to E can be see in Figures 4.62, 4.64, 4.66, 4.68 and 4.70 respectively. All the distributor experiments were performed under identical process conditions, with a 3.2-g/l initial solution concentration. They are also compared with an experimental run using (bare) flat plates without any distributors.

4.14.1 Distributor Design A

Compared to the bare plates, design A exhibits lower R_f , significantly higher clean pressure drop and slightly larger overall pressure drop increase, as displayed in Figures 4.61(a), (b) and (c) respectively.

Figure 4.61(a): Variation of fouling resistance with time for distributor design A.

Figure 4.61 (b): Variation of pressure drop with time for distributor design A.

Figure 4.61(c): Variation of pressure drop increase factor with time for distributor design A.

Photographs of the fouled plates can be seen in Figures 4.62(a), (b) and (c), where fouling appears to occur in the wake of the far right inlet distributor. More deposition occurred on these plates than on the bare plates, refer Figure 4.53(b). However,

contrary to the fouling pattern on the bare plates, no deposits are formed in the plate corner opposite the inlet port for design A. This is because the distributors serve to direct flow to the two plate corners adjacent to the ports, where the prevailing flow velocity is low. In doing so however, they may have created additional zones of low flow velocity in their wake. Since there are no distributors or blockages in the main flow area of the plate, there is nothing to break up these additional low-velocity zones.

In addition, the distributors also increase the clean pressure drop of the channel. This applies for all plates with distributors attached. Although design A has a slightly larger overall pressure drop increase (0.17 bar) than that of the bare plates (0.11 bar), the corresponding PDIF values for design A are much lower. This is because the PDIF values of design A are masked by its large clean pressure drop. Therefore, PDIF can be misleading unless the corresponding values for pressure drop are referred to as well.

Figures 4.62(a), (b) and (c): Photographs of fouled plates with distributor design A.

As a constant solution flow rate was maintained, it was anticipated that a higher flow velocity would prevail in the fouled channel than in the clean channel. This is because the effective plate width of the fouled plates is less than that of clean plates. As the flow velocity increases, an enhancement in heat transfer can be expected, together with a higher removal rate. Consequently, a corresponding reduction in the deposition rate

should occur. In altering the effective width of the plate channel, fouling has also changed the fluid flow characteristics in the channel completely.

4.14.2 Distributor Design B

The variation of R_f , pressure drop and PDIF with time for design B are shown in Figures 4.63(a), (b) and (c) respectively. As with design A, distributor design B aims to eliminate areas of low flow velocity in the vicinity of the plate corners opposite the ports. The distributors of design B are fewer in number and longer in length, than those of design A, as can be seen in Figures 4.64(a), (b) and (c). Once again, more fouling occurred on these plates than on the bare plates, refer Figure 4.53(b). However, no deposition was formed in the lower plate corner opposite the inlet port of design B. From Figure 4.64(b), it can be confirmed that fouling occurs in the wake of the far right inlet distributor. Furthermore, a larger plume of deposits tend to form in the wake of longer distributors, thus making them inferior to shorter ones.

Figure 4.63(a): Variation of fouling resistance with time for distributor design B.

Figure 4.63(b): Variation of pressure drop with time for distributor design B.

Figure 4.63(c): Variation of pressure drop increase factor with time for distributor design B.

Figure 4.63(a) shows that the R_f values of design B are comparable to those of the bare plates. Deposition occurred on the main heat transfer regions of design B plates. However, the deposits formed were able to penetrate the laminar viscous sublayer,

resulting in the increased turbulence not being outweighed by their addition thermal resistance. On the other hand, the increase in overall pressure drop is significantly higher for design B than for the bare plates, as can be seen in Figure 4.63(b). This is due to severe blockage of the plate outlet region of design B by deposits, see Figure 4.64(a).

Figures 4.64(a), (b) and (c): Photographs of fouled plates with distributor design B.

4.14.3 Distributor Design C

With reference to Figures 4.65(a) and (b), design C exhibits higher R_f , and higher overall pressure drop increase than those of the bare plates. The long inlet distributors provide regions of low flow velocity for fouling to occur. Initiated from the far right inlet distributor, a huge plume of deposit was formed. Deposition is evident along the main flow path of the plate which is crucial for heat transfer. This is why higher R_f values are displayed by design C.

The fouling patterns of design C once again confirm the disadvantage of having long distributors especially in the inlet region. As inlet distributors distribute inflowing fluid across the plate channel, while outlet distributors only 'collect' flow into the outlet port, the former is more influential on the resulting flow distribution, and hence the

fouling pattern. Therefore, the small round outlet distributors did not contribute much towards the reduction of fouling. Furthermore, much of the outlet region is blocked by deposits, and therefore, an increase in pressure drop was recorded.

Figure 4.65(a): Variation of fouling resistance with time for distributor design C.

Figure 4.65(b): Variation of pressure drop with time for distributor design C.

Figure 4.65(c): Variation of pressure drop increase factor with time for distributor design C.

Figures 4.66(a), (b) and (c): Photographs of fouled plates with distributor design C.

4.14.4 Distributor Design D

The long inlet distributors in designs B and C are replaced by a string of small round distributors in design D. This resulted in better heat transfer and less fouling for design

D, which displayed lower R_f values than those of the bare plates, see Figure 4.67(a). However, it is apparent from Figures 4.68(a) and (b) that fouling still forms in the wake of the far right inlet distributor. Once deposition is initiated, it continues to propagate until a large portion of the main heat transfer area is covered. As most of the plate outlet region is blocked, a substantial increase in pressure drop is recorded for design D.

Figure 4.67(a): Variation of fouling resistance with time for distributor design D.

Figure 4.67(b): Variation of pressure drop with time for distributor design D.

Figure 4.67(c): Variation of pressure drop increase factor with time for distributor design D.

Figures 4.68(a), (b) and (c): Photographs of fouled plates with distributor design D.

4.14.5 Distributor Design E

The variation of R_f and pressure drop with time for design E are shown in Figures 4.69(a) and (b) respectively. Instead of having distributors in the inlet and outlet plate

regions, design E has its main heat transfer area divided into three flow channels of equal width. This was done by using two strips of gasket as shown in Figures 4.70(a), (b) and (c). Being different in material and surface structure to the metal plates, the gaskets appear to initiate fouling. Nevertheless, increased turbulence brought about by the deposits penetrating into the laminar sublayer still surpasses their associated increase in thermal resistance. Since there are no distributors in the inlet and outlet regions to force the fluid to flow in certain directions, the corresponding pressure drop is comparable to that of the bare plates.

Figure 4.69(a): Variation of fouling resistance with time for distributor design E.

Figure 4.69(b): Variation of pressure drop with time for distributor design E.

Figures 4.70(a), (b) and (c): Photographs of fouled plates with distributor design E.

4.14.6 Comparisons of All Distributor Designs

The variation of the overall heat transfer coefficient, R_f , pressure drop and PDIF with time for designs A to E are shown in Figures 4.71(a), (b), (c) and (d) respectively. From Figure 4.71(a), it can be seen that all the experiments with different distributor designs have similar clean overall heat transfer coefficients, U_o , except for design A which registers a U_o value of only 700 W/m²K. It is possible that a plate channel gap of approximately 2.4 mm was not maintained in design A.

Designs A to D have distributors in the plate inlet and outlet regions, while design E has 'distributors' in its main heat transfer area only. Compared with the bare plates without any distributors, more fouling occurred on all the plates with distributors. This is due to the prevailing low flow velocity in the wake produced by the distributors. Instead of being broken up, this wake is allowed to propagate into the main heat transfer area, hence promoting the occurrence of fouling.

Considering all the distributor designs involving the inlet and outlet plate regions (designs A to D), design B is best in terms of heat transfer, R_f and pressure drop, as can be seen from Figures 4.71(a) to (d). A comparison of all the designs, reveals that

although design E only has 'distributors' in its main heat transfer area, it has lower R_f and pressure drop values than design B. Conversely, design A not only exhibits the lowest R_f values, it also displays the highest pressure drop.

Figure 4.71(a): Variation of overall heat transfer coefficient with time for different distributor designs.

Figure 4.71(b): Variation of fouling resistance with time for different distributor designs

Figure 4.71(c): Variation of pressure drop with time for different distributor designs.

Figure 4.71(d): Variation of pressure drop increase factor with time for different distributor designs.

It is difficult to draw conclusions from this set of designs, on the relationship between distributor design and the resulting fouling pattern for commercial plates. Therefore, future experiments should include distributors in both the inlet/outlet region and main heat transfer area of the plates.

4.15 Repeatability of Data

Data scatter brought about by fluctuations in the operating conditions was common. Even with the installations of the solution and hot water flow control valves, and temperature controllers for both the solution and hot water tanks, there remained some unavoidable fluctuations in the flow velocities and temperatures. Care was always taken to ensure that the heat exchanger plate surfaces were not altered during cleaning and handling. A number of experiments were therefore performed to establish the repeatability of the data.

For the Alfa Laval plates, experimentation with 35 cm/s solution inlet velocity, 51.5°C solution inlet temperature, and 2.8 g/l initial solution concentration was performed with P01 and M3 diagonal flow plates. As can be seen in Figures 4.72(a) and (b), although the R_f values are comparable, there are differences in the corresponding pressure drops. Nevertheless, within the fluctuations of the data, these two runs performed about 5 months apart agree well with each other.

Figure 4.72(a): Variation of fouling resistance with time for commercial Alfa Laval P01 and M3 diagonal flow plates.

Figure 4.72(b): Variation of pressure drop with time for commercial Alfa Laval P01 and M3 diagonal flow plates.

To confirm the repeatability of fouling data for the flat plates, two experiments one lasting 3000 minutes and the other 7000 minutes were performed with 35 cm/s solution inlet velocity, 41.5°C solution inlet temperature, and 3.2 g/l initial solution concentration. Both sets of plates were sanded with 240 emery paper, and were conducted 6 days apart. The variation of fouling resistance and pressure drop with time can be seen in Figures 4.48(a) and (b) respectively. Contrary to the commercial plates, there are discrepancies in the R_f values for the flat plates, but their corresponding pressure drops agree well. Due to the absence of contact points for the flat plate, the two runs may have differing plate channel gaps, which could result in different R_f and pressure drop values. In any case, within the degree of data fluctuations, fouling data of the flat plates are reasonably repeatable.

5. FOULING & EXCESS HEAT TRANSFER AREA

To make provision for fouling, most heat exchangers are oversized in terms of excess heat transfer area. In PHEs, this excess area is usually provided by adding extra parallel plates. For constant flow rates however, this procedure reduces the flow velocity in individual plate channels. As a result, the prevailing wall shear stress is reduced, and an increase in deposit formation can thus be expected.

It is therefore, of great interest to compare this method of incorporating the required excess area with other possible ways. This chapter serves to investigate the overall effect on heat transfer and fouling when this excess area is provided by one of the following ways:

- (i) additional parallel plates
- (ii) larger plates of different aspect ratios
- (iii) additional plates in a second pass

A standard PHE of a particular configuration was first defined, and its number of plates and inlet conditions were specified. The outlet conditions of this exchanger were then determined, and the variation of its fouling resistance with time was computed using an approximate fouling model, which is described in detail in Section 5.2.

For a given design fouling resistance, the excess area required can be calculated. Adding this excess area in the ways mentioned above to the standard exchanger resulted in configurations with different sizes, shapes and/or flow arrangements. The outlet conditions of these 'oversized' exchangers were estimated for the same mass flow rates and inlet temperatures of the two heat-exchanging fluids as those for the standard exchanger. By applying the same fouling model, the dynamic fouling behaviours of these exchangers could be determined and compared with that of the standard one.

5.1 The Standard Plate Heat Exchanger

<i>Plate heat exchanger</i>	Alfa Laval M3	
	flow direction	diagonal
	total number of plates	40
	total number of channels	39
<i>Service fluid</i>	Cooling water	
	flow arrangement	parallel
	number of passes	1
	number of channels	20
	inlet velocity	0.35 m/s
	inlet temperature	20°C
<i>Process fluid</i>	Heat transfer oil (Dowtherm J)	
	flow arrangement	parallel
	number of passes	1
	number of channels	19
	inlet velocity	0.45 m/s
	inlet temperature	80°C

Table 5.1: Details of the standard plate heat exchanger.

As can be seen from Table 5.1, an Alfa Laval M3 diagonal flow PHE in a single pass (parallel flow) arrangement was chosen as the standard exchanger. The reason behind this choice lies in the fact that most of the experiments pertaining to this dissertation were performed with this particular configuration. Therefore, the available data on heat transfer and fouling can be used when necessary. For instance, the dimensions of Alfa Laval M3 plates can be found in Table 3.1, while their clean film heat transfer coefficients, α for diagonal flow can be estimated using the Nusselt number correlation in Equation (4.1). However, no distinction was made as to whether the exchanger has a U- or Z-arrangement. This is because only the overall heat transfer effects are of primary interest to this assessment, and its application would only be limited by 'overspecific' details.

For this excess area assessment, the service and process fluids chosen were cooling water and Dowtherm J, a non-fouling heat transfer oil, respectively. Hence, fouling occurred on the cooling water side only. Together with the associated Nu correlation in Equation (4.1), the known inlet conditions of the standard exchanger can be used to calculate their corresponding outlet conditions. This is essential for a full evaluation of the heat transfer performance, and later for estimating the excess area requirement. Taking just the overall heat transfer effect into account, the procedure for determining the outlet temperatures of the cold cooling water and the hot heat transfer oil, is outlined as follows:

- (a) guess the two outlet temperatures, $T_{cold,out}$ and $T_{hot,out}$.
- (b) calculate the average temperatures of the two fluids, $T_{cold,ave}$ and $T_{hot,ave}$, using an arithmetic mean of the appropriate inlet and outlet temperatures. It was based on these average temperatures that the corresponding fluids' physical properties were estimated from relevant data sources (VDI-Wärmeatlas, 1994).
- (c) calculate the heat transfer rate, $\dot{Q}$, according to Equation (2.2).
- (d) calculate the log mean temperature difference, ΔT_{log} , via Equation (2.5).
- (e) calculate the clean overall heat transfer coefficient, U , using Equation (2.1):

$$\dot{Q} = U \cdot A \cdot \Delta T_{log} \quad \text{from (2.1)}$$

- (f) calculate the clean film heat transfer coefficients, α_{cold} and α_{hot} , via Equation (4.1):

$$Nu = \frac{\alpha \cdot d_h}{\lambda} = 0.38 \cdot (Re)^{0.65} \cdot (Pr)^{0.33} \quad \text{from (4.1)}$$

- (g) calculate the two wall temperatures, $T_{cold,wall}$ and $T_{hot,wall}$, using Equations (2.14a) and (2.14b) respectively. These temperatures are necessary to account for the wall superheat effect in the evaluation of α_{cold} and α_{hot} , see Equation (2.9). However, with reference to Equation (4.1), this effect was negligible for the present set-up, and hence the wall temperatures were mainly required to estimate the fouling resistance via an approximate fouling model.
- (h) calculate the clean overall heat transfer coefficient, U , using Equation (2.4):

$$U = \frac{1}{\left(\frac{1}{\alpha_{hot}} + \frac{s_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_{cold}} \right)} \quad \text{from (2.4)}$$

- (i) calculate the outlet temperatures, $T_{hot,out}$ and $T_{cold,out}$ via Equations (2.2a) and (2.2b) respectively, where $\dot{Q}$ is determined according to Equation (2.1) using the U values calculated in (h).
- (j) substitute the calculated values of the outlet temperatures in (i) for their guessed counterparts in (a), and repeat this procedure until these two sets of values agree with each other to within $\pm 0.1^\circ\text{C}$.

Using this procedure, the outlet temperatures of the cooling water and the heat transfer oil were calculated to be 37 and 44°C respectively. In addition, the U values calculated in steps (f) and (h) agree well with each other.

5.2 An Approximate Fouling Model

Most fouling models (Kern and Seaton, 1959) (Taborek *et al.*, 1972a) have two separate terms for the deposition and removal rates, which combine to give a net deposition rate as shown in Equation (2.44).

$$\frac{dm_f}{dt} = \dot{m}_d - \dot{m}_r \quad \text{from (2.44)}$$

In order to develop an approximate fouling model, the following assumptions were made:

- (a) deposit properties such as density ρ_f , thermal conductivity λ_f , and strength ψ_f , remain constant with time
- (b) deposition rate is reaction-controlled, and of second order, $n_{rxn} = 2$
- (c) $C_{bulk} \approx C_{wall}$
- (d) mass flow rates of both fluids remain constant
- (e) flow velocities of both fluids remain constant
- (f) wall temperatures of both fluids remain constant
- (g) experimental heat transfer and fouling data in this dissertation can be used to determine certain constants and parameters necessary for the application of this model

Based on assumption (a), the net deposition rate can be related to the fouling resistance as follows:

$$\left(\frac{dR_f}{dt}\right) = \frac{1}{\rho_f \cdot \lambda_f} \cdot \left(\frac{dm_f}{dt}\right) \quad (5.1)$$

From the experimental results of Sections 4.3 and 4.4, and the conclusions drawn, it is reasonable to assume that the deposition rate is reaction-controlled and not diffusion-controlled. Hence, $C_{bulk} \approx C_{wall}$ is also a reasonable assumption. Since the reaction is of second order, $n_{rxn} = 2$, the deposition rate can be expressed as follows:

$$\dot{m}_d = k_{rxn} \cdot (C_{bulk} - C_{sat})^2 \quad \text{from (2.39)}$$

The reaction rate constant, k_{rxn} is related to the wall temperature, T_{wall} via an Arrhenius type equation:

$$k_{rxn} = c_7 \cdot \exp\left(-\frac{E_{act}}{R_g \cdot T_{wall}}\right) \quad \text{from (2.43)}$$

Neglecting the removal rate for the time being, and substituting Equations (2.39) and (2.43) into Equation (5.1), the rate of change of the fouling resistance, dR_f/dt can be written as follows:

$$\begin{aligned} \left(\frac{dR_f}{dt}\right) &= \frac{1}{\rho_f \cdot \lambda_f} \cdot \left(\frac{dm_f}{dt}\right) \\ &= \frac{1}{\rho_f \cdot \lambda_f} \cdot (\dot{m}_d) \\ &= \frac{c_7}{\rho_f \cdot \lambda_f} \cdot \exp\left(-\frac{E_{act}}{R_g \cdot T_{wall}}\right) \cdot (C_{bulk} - C_{sat})^2 \end{aligned} \quad (5.2)$$

It has been previously stated in Section 2.2.6.1.1 that the $d(R_f)/dt$ is inversely proportional to the flow velocity, u raised to the power of 1.5. This relationship has also been confirmed using the experimental results in Section 4.3 of this dissertation.

$$\frac{dR_f}{dt} \propto u^{-1.5} \quad \text{from (2.45)}$$

Therefore, instead of modelling the removal rate with a separate term as in Equation (2.44), it can be incorporated as part of the fouling model by substituting Equation (2.45) into Equation (5.2). Assuming that the flow velocity and deposit properties such as density ρ , thermal conductivity λ , and strength ψ , remain constant with time, the fouling resistance can be expressed as follows by integrating Equation (5.2):

$$R_f = \frac{c_9 \cdot \exp\left(-\frac{E_{act}}{R_g \cdot T_{wall}}\right) \cdot (C_{bulk} - C_{sat})^2 \cdot t}{u^{1.5}} \quad (5.3)$$

The constant, c_9 and the activation energy, E_{act} can be estimated using the fouling data from Section 4.5. For the interval after the initiation period and before asymptotic fouling rate was reached, E_{act} was estimated to be about 75 kJ/mol, while c_9 was approximated to be $20 \text{ m}^{7.5} \text{ s}^{0.5} \text{ K/kg}^3$. The values of C_{sat} at the prevailing temperature can be obtained from Gmelin (1962), for example, between 30°C to 40°C, $C_{sat} = 2.2 \text{ kg/m}^3$.

For processes with constant flow rates, the cross-sectional area of the plate channel will decrease as fouling proceeds, thus bringing about an increase in the flow velocity. This effect would cause the fouling resistance in Equation (5.3) to exhibit an asymptotic behaviour as in the models proposed by Kern and Seaton (1959). However, as constant mass flow rates and flow velocities of both fluids were assumed, an asymptotic behaviour is not reflected in this model. Moreover, the wall temperatures were also assumed to be constant. In reality, as deposition occurs, not only the flow velocity changes, but the prevailing wall temperature should change as well.

This modified fouling model is comparable to the transfer-adhesion model proposed by Watkinson and Epstein (1970) for particulate fouling:

$$\begin{aligned} \frac{dm_f}{dt} &= (\text{mass flux}) \cdot (\text{sticking probability}) \\ &= (\beta [C_{bulk} - C_{sat}]) \cdot \left(\frac{c_{10} \cdot \exp\left[-\frac{E_{act}}{R_g \cdot T_{wall}}\right]}{f \cdot u^2} \right) \end{aligned} \quad (5.4)$$

According to their model, the net deposition rate is proportional to the product of particle mass flux from the bulk to the heat transfer surface, and the sticking probability of particles attaching to the same surface. The latter is a ratio of the adhesive forces binding a particle to the surface, to the hydrodynamic forces acting on the particle when it reaches the surface. c_{10} is a proportionality constant, and f the friction factor.

Equations (5.3) and (5.4) are similar in the sense that, the deposition rate is in the numerator and is dependent on the wall temperature according to an Arrhenius equation, while the removal rate is in the denominator and is dependent on the flow velocity. Therefore, the method of excess area incorporation which gives the lowest wall temperature and the fastest flow velocity, will result in the least fouling. Since the wall temperature is related exponentially to the deposition rate and hence to the fouling resistance, it can be expected to have a stronger effect than the flow velocity on the corresponding fouling behaviour. The following relationship can thus be written for fouling resistance:

$$\frac{dR_f}{dt} = \text{function}(T_{\text{wall}}, u) \quad (5.5)$$

5.3 Excess Heat Transfer Area Evaluation

The excess heat transfer area required, A_{ex} is defined according to Equation (5.6) as the difference between the total area required under fouling conditions, A_f , and the area required when the exchanger is clean, A_{clean} . A_f can be predicted using Equation (5.7) via a known design fouling resistance, R_f .

$$A_{\text{ex}} = A_f - A_{\text{clean}} \quad (5.6)$$

$$\frac{A_f}{A_{\text{clean}}} = 1 + R_f \cdot U_{\text{clean}} \quad (5.7)$$

For the same heat duty, a larger clean overall heat transfer coefficient, U_{clean} , would lead to a smaller A_{clean} being calculated, see Equation (2.1). This in turn will result in a larger A_f/A_{clean} ratio, for the same R_f . Since U_{clean} for PHEs is generally larger than those for many other heat exchanger types, extra caution should be exercised when

estimating A_{ex} for PHEs. A larger-than-necessary R_f would result in a severely oversized heat exchanger.

The design fouling resistance recommended by TEMA for treated cooling water where the outlet temperature is less than 50°C, lies between 1.8×10^{-4} and 5.3×10^{-4} m²K/W (TEMA, 1978). On the other hand, Marriott (1971) recommended a design fouling resistance of 6.9×10^{-6} m²K/W for PHEs. As can be seen, there is a difference of almost 2 orders of magnitude between these two sources. In order to have a large enough excess area for the following calculations, a design fouling resistance whose magnitude lies between these two sources was chosen, namely 7×10^{-5} m²K/W.

For this design R_f , the corresponding A_{ex} was calculated to be 0.249 m². This is equivalent to 8 extra M3 plates, which is an increase of 20% in the heat transfer area. Most manufacturers of PHEs recommend that the excess area to be incorporated should not exceed 25% of the area calculated for the clean duty (Müller-Steinhagen, 1997a). If the upper limit of the TEMA design fouling resistance was used for this assessment, 20 extra M3 plates would be required, giving rise to a 50% increase in the heat transfer area.

5.4 Excess Heat Transfer Area Incorporation

In this section, the different ways of incorporating the excess heat transfer area mentioned above are described in detail. Their influence on flow velocities, and hence heat transfer and fouling are also discussed.

The same fluid inlet temperatures and mass flow rates imposed on the standard exchanger, were applied to the 'oversized' exchangers. The cooling water mass flow rate and that of the heat transfer oil were both calculated to be 1.68 kg/s. The outlet conditions of the 'oversized' exchangers were estimated using the same procedure as for the standard exchanger described in Section 5.1. A summary of their inlet and outlet conditions is given in Table 5.2, where Pass A#1 and Pass A#2 are two

passes pertaining to the same heat exchanger A, while Pass B#1 and Pass B#2 belong to another heat exchanger B with a different configuration.

		Std	Para- llel	std AR	std w_p	$\frac{1}{2}$ std w_p	Pass A#1	Pass A#2	Pass B#1	Pass B#2
Overall										
no. of plates		40	48	40	40	40	24	24	24	24
no. of channels		39	47	39	39	39	23	24	23	24
U	W/m ² K	2.92 $\times 10^3$	2.61 $\times 10^3$	2.77 $\times 10^3$	2.93 $\times 10^3$	4.34 $\times 10^3$	4.01 $\times 10^3$	3.76 $\times 10^3$	4.08 $\times 10^3$	3.66 $\times 10^3$
Water										
no. of channels		20	24	20	20	20	12	12	12	12
u_{in}	m/s	0.35	0.29	0.32	0.35	0.70	0.58	0.59	0.58	0.58
T_{in}	°C	20	20	20	20	20	20	35	28	20
T_{out}	°C	37	37	38	38	42	35	38	41	28
T_{ave}	°C	28	29	29	29	31	27	36	34	24
T_{wall}	°C	36	36	36	36	36	36	38	41	28
α	W/m ² K	1.30 $\times 10^4$	1.16 $\times 10^4$	1.23 $\times 10^4$	1.30 $\times 10^4$	2.04 $\times 10^4$	1.74 $\times 10^4$	1.89 $\times 10^4$	1.89 $\times 10^4$	1.74 $\times 10^4$
Oil										
no. of channels		19	23	19	19	19	11	12	11	12
u_{in}	m/s	0.45	0.37	0.41	0.45	0.90	0.78	0.69	0.78	0.70
T_{in}	°C	80	80	80	80	80	80	49	80	53
T_{out}	°C	44	43	42	41	34	49	41	53	35
T_{ave}	°C	62	61	61	60	57	64	45	66	44
T_{wall}	°C	39	39	39	39	41	40	39	45	30
α	W/m ² K	4.18 $\times 10^3$	3.69 $\times 10^3$	3.93 $\times 10^3$	4.18 $\times 10^3$	6.43 $\times 10^3$	6.02 $\times 10^3$	5.27 $\times 10^3$	6.02 $\times 10^3$	5.29 $\times 10^3$

Note: std $\equiv$ standard plates; parallel $\equiv$ parallel plates; std AR $\equiv$ plates with standard plate aspect ratio; std w_p $\equiv$ plates with standard plate width; $\frac{1}{2}$ std w_p $\equiv$ plates with half the standard plate width.

Table 5.2: Summary of flow conditions and configurations for standard and 'overdesigned' plate heat exchangers.

Results displayed in Table 5.2 do not include any process control to maintain constant outlet temperatures by regulating the cooling water flow rates. These results will be explained in more detail in the following paragraphs.

5.4.1 Additional Parallel Plates

Müller-Steinhagen and Middis (1989) have looked into the effects of adding more parallel plates as a way of providing the excess area. Figure 5.1 shows the calculated heat duty of an Alfa Laval P20 PHE as a function of the heat transfer area for constant flow rates. The inlet temperatures of the hot water and solvent X-2 are 70°C and 30°C respectively. The concentration of particles suspended in the solvent is 300 ppm.

Figure 5.1: Heat flow rate versus parallel heat transfer area for clean and fouling fluids (Müller-Steinhagen and Middis, 1989).

With reference to Figure 5.1, increasing the number of parallel plates, results in an increase in the heat transfer area, but at the same time causes the flow velocities of both fluids to decrease. Under clean conditions, the heat flow rate increases at a

decreasing rate, with increasing heat transfer area. On the other hand, the heat flow rate calculated under fouling conditions exhibits a maximum as the area is increased. Therefore, adding parallel plates to the heat exchanger is only beneficial to the left of this maximum.

The above results indicate that excess area should be provided by methods other than the addition of more parallel plates. In order to maintain the flow velocity in the plate channels for constant flow rates, incorporation of excess area can be done by using larger plates and/or by increasing the number of flow passes in the PHE.

5.4.2 Larger Plates of Different Aspect Ratios

With the same total number of plates as for the standard PHE, three types of larger plates are considered:

- (i) plates with the standard plate aspect ratio (std AR)
- (ii) plates with the standard plate width (std w_p)
- (iii) plates with half the standard plate width ($\frac{1}{2}$ std w_p)

<i>Plate Characteristics</i>		Std	std AR	std w_p	$\frac{1}{2}$ std w_p
actual heat transfer area	m ²	0.032	0.039	0.039	0.039
projected heat transfer area	m ²	0.027	0.033	0.033	0.033
cross-sectional flow area	m ²	2.40×10^4	2.63×10^4	2.40×10^4	1.20×10^4
heat transfer width	m	0.100	0.110	0.100	0.050
heat transfer length	m	0.270	0.296	0.325	0.651
aspect ratio	-	2.7	2.7	3.3	13.0

Table 5.3: Dimensions and other details of standard and larger plates.

The dimensions of the larger plates are compared with those of the standard plate in Table 5.3. All the plates have an enlargement factor similar to that of Alfa Laval M3 plates, i.e. $\mu_e = 1.185$, which was used to determine their respective projected areas.

Consequently, their equivalent plate lengths for heat transfer, and plate aspect ratios AR , can also be estimated.

It can be seen from Table 5.3 that among the larger plates, the cross-sectional flow area is greatest for plates with the standard plate aspect ratio, and smallest for plates with half the standard plate width. Consequently, the flow velocity would be slowest for the former and fastest for the latter, as is shown in Table 5.2. As a result, the exchanger with plates of half the standard plate width is least likely to foul. However, its associated pressure drop would be higher than those with lower plate aspect ratios.

5.4.3 Additional Plates in a Second Pass

Another way of providing the excess area required is by adding more plates, and by increasing the number of flow passes at the same time. Since the passes are always in series to one another and are usually comparable in size, increasing the number of passes would cause the number of parallel plates per pass to decrease. This would cause the flow velocities of both fluids to rise, which in turn would promote the rate of removal of deposits. As a result, an enhancement in the film heat transfer coefficients can be expected together with a reduction in the corresponding fouling resistance. However, these benefits do not come without a price, namely an associated increase in pressure drop. This is because the fluids not only flow faster but do so through a longer winding channel.

This method of incorporating the excess area was investigated by having two passes with different entrance and exit positions for the two heat-exchanging fluids:

- (i) configuration A where the two fluids enter and leave at the same pass
- (ii) configuration B where the two fluids enter and leave at different passes

Standard plates were used for both configurations, where pass number one (#1) is similar in size to pass number two (#2). This is to minimise pressure drop differences between the two passes. With reference to Table 5.2, after incorporating the required excess area, an even number of cooling water channels and an odd number of oil

channels were necessary. Therefore, although pass #1 and pass #2 have exactly the same number of water channels, pass #1 has one less oil channel than pass #2.

The flow arrangements and directions for configurations A and B are illustrated schematically in Figure 5.2 and Figure 5.3 respectively. In configuration A, both the cooling water and hot oil enter the exchanger via pass number one (A#1) and leave via pass number two (A#2). On the other hand, in configuration B, the hot oil enters and leaves the exchanger through pass number one (B#1) and two (B#2) respectively, while the cooling water enters the exchanger via pass B#2 and leaves via pass B#1.

Figure 5.2: Configuration A.

Figure 5.3: Configuration B.

A careful examination of Figure 5.2 and Figure 5.3 reveals that for each of the configurations, all adjacent channels have counter-current flow except the first hot oil channel in the second pass (A#2 or B#2) and the last water channel of the first pass (A#1 or B#1) which have flows co-current to each other. Nevertheless, this co-current effect was neglected for both configurations, due to the larger number of counter-current flow channels present.

The overall results for configuration A were taken as the average of those obtained for pass A#1 and pass A#2. The same applies for configuration B, pass B#1 and pass B#2.

5.4.4 Fouling Trends

Under the prevailing conditions for the experiments performed in the present study, the saturated CaSO_4 concentration, C_{sat} has a value of about 2.2 g/l, refer Figure 3.8 (Gmelin, 1962). Therefore, a CaSO_4 concentration of 3.2 g/l would give a 1000 ppm difference between the bulk concentration and the saturated concentration, $(C_{bulk} - C_{sat})$. Using the approximate fouling model described in Section 5.2, and $(C_{bulk} - C_{sat}) \equiv 1000$ ppm, the variation of water side fouling resistance, R_f with time was computed for all the above mentioned methods of incorporating the required excess heat transfer area. This variation is illustrated in Figures 5.4, 5.5 and 5.6, while Figures 5.7, 5.8 and 5.9 show the corresponding variation of the ratio of fouled heat transfer rate $\dot{Q}$ to clean heat transfer rate $\dot{Q}_0$ with time, for activation energies, E_{act} of 40, 75 and 150 kJ/mol respectively. At the same time, the flow velocity, u has been raised to the power of 2.0, 1.5 and 0.0, whereas the corresponding values of c_9 in Equation (5.3) are 1.4×10^{-5} , 20 and $4.7 \times 10^{14} \text{ m}^{7.5} \text{ s}^{0.5} \text{ K/kg}^3$ for comparable R_f values of the standard heat exchanger, respectively. Since fouling occurs on the cooling water side only, flow velocities and temperatures discussed in this section refer to those of the cold side unless otherwise mentioned.

E_{act} and the power of u were varied in order to demonstrate the shift in sensitivity between the wall temperature influence, and the flow velocity influence on R_f . For chemical reaction fouling and crystallisation fouling with hard adhesive deposits, the effect of flow velocity on fouling is not as profound as that of temperature. This is because in addition to their higher stickability, these fouling mechanisms also have higher activation energies. In order to examine which method of excess area incorporation is best for this type of reaction-controlled growths, E_{act} and the power of u were set at 150 kJ/mol and 0.0 respectively.

Conversely, for particulate fouling where the resultant deposits are not very tenacious, the velocity has a dominant effect on the fouling behaviour. With a lower activation energy, this type of fouling does not have a strong dependence on the temperature.

Therefore, E_{act} and the power of u were set at 40 kJ/mol and 2.0 respectively, in order to investigate the best method of incorporating the excess area for this type of adhesion-controlled deposition. In the following figures, the words 'standard' and 'configuration' have been abbreviated to 'std' and 'conf' respectively.

Figure 5.4: Variation of water side fouling resistance with time for $E_{act} = 40 \text{ kJ/mol}$.

Figure 5.5: Variation of water side fouling resistance with time for $E_{act} = 75 \text{ kJ/mol}$.

Figure 5.6: Variation of water side fouling resistance with time for $E_{act} = 150$ kJ/mol.

With reference to Figure 4.13(a), for 3.2 g/l CaSO_4 solution concentration, 35 cm/s solution inlet velocity, 51.5°C solution inlet temperature, 0.146 m/s hot water inlet velocity, and 87°C hot water inlet temperature, the corresponding R_f value is approximately 1×10^{-5} m²K/W at 3000 minutes (or about 2 days). As previously mentioned in Section 5.2, E_{act} was estimated to be about 75 kJ/mol using the fouling data from Section 4.5. For the same E_{act} , it can be seen from Figure 5.5 that the R_f value for the standard heat exchanger is significantly less than 1×10^{-5} m²K/W at the second day of operation. However, this discrepancy is reasonable considering the lower cold and hot side inlet temperatures employed in this excess heat transfer area analysis, refer Table 5.1. In addition, constant flow velocities and wall temperatures (interface between deposit and fluid) have also been assumed in this analysis.

It should be mentioned at this point that most of the trends shown in Figures 5.4 to 5.9, can be elucidated by examining the results displayed in Table 5.2. As the activation energy increases from 40 to 150 kJ/mol, and the power of u decreases from 2.0 to 0.0, the fouling process becomes more influenced by the wall temperature and less by the flow velocity. This can be clearly seen from the changes in the R_f and $(\dot{Q} / \dot{Q}_0)$ values

for pass B#1 in Figures 5.4 to 5.9. Although pass B#1 has a relatively high cooling water flow velocity, it has the highest average wall temperature for all the exchangers examined. Therefore, configuration B is unsuitable when the prevailing deposition mechanism is reaction-controlled, as it would suffer from a severe pressure drop when pass B#1 fouls significantly more than pass B#2.

Figure 5.7: Variation of the ratio $\dot{Q} / \dot{Q}_0$ with time for $E_{act} = 40$ kJ/mol.

Figure 5.8: Variation of the ratio $\dot{Q} / \dot{Q}_0$ with time for $E_{act} = 75$ kJ/mol.

Figure 5.9: Variation of the ratio $\dot{Q}/\dot{Q}_0$ with time for $E_{act} = 150$ kJ/mol.

Figure 5.10: Variation of heat transfer rate with time for $E_{act} = 40$ kJ/mol.

Among all the methods investigated for excess area provision including the standard exchanger case where no excess area has been provided for, the addition of parallel plates gives the highest fouling resistance for the two lower-activation-energy cases investigated. It is only at very high activation energies where flow velocity has little or

no effect on fouling that pass B#1 which has the average highest wall temperature among all the 'overdesigned' exchangers, exhibits the highest fouling resistance. This is reasonable as adding extra parallel plates resulted in the lowest flow velocity among all the exchangers investigated.

All the 'overdesigned' exchangers with larger plates, have higher outlet temperatures and slightly higher average wall temperatures than those of the standard exchanger. This is because when compared with the latter, they all have more heat transfer area, and hence could pick up more heat from the hot heat transfer oil. Therefore, in the presence of adhesion-controlled fouling, these 'overdesigned' exchangers would suffer from more fouling, unless they have a higher flow velocity than that of the standard exchanger, refer Figures 5.4 and 5.7. Conversely, for reaction fouling where velocity has little or no effect on the resulting fouling behaviour, there is little difference between the performance of these 'overdesigned' exchangers, except for the exchanger with half the standard plate width which has the highest outlet temperature, refer Figures 5.6 and 5.9.

The consequences of having a two pass arrangement are mostly encouraging as can be seen from Figures 5.4 to 5.9, and Table 5.2. The flow velocities are higher for both passes of both configurations when compared with that of the standard exchanger. However, all the corresponding average wall temperatures are higher than that of the standard exchanger, except for pass B#2. Since the flow velocities of all the individual passes are comparable to each other, whether a pass would foul more than another is dependent on the corresponding wall temperatures. In configuration A, the two fluids enter from the same pass, namely pass A#1, therefore the cooling water in pass A#2 has higher temperatures than that in pass A#1. As a result, Pass A#2 fouls more than pass A#1. The same can be said of configuration B, where pass B#1 fouls more than pass B#2. The two fluids enter the exchanger at different passes, such that having entered the exchanger via pass B#2, the cooling water has higher temperatures when it reaches pass B#1.

Among all the oversized exchangers and their individual passes, pass B#2 not only has the lowest average wall temperature, but it also has a relatively high flow velocity. As a result, it has the lowest fouling resistance and highest $(\dot{Q} / \dot{Q}_0)$ for all the three activation energies investigated. Despite its high average wall temperature, pass B#1 maintains a low fouling profile at low activation energies. It is only at very high activation energies that pass B#1 exhibits high fouling resistances and low $(\dot{Q} / \dot{Q}_0)$, as shown in Figures 5.6 and 5.9.

Fouling is the reason for incorporating excess area into PHEs, however, the way this excess area is provided in turn affects the fouling behaviour in the exchanger. Although 'oversized' exchangers with high flow velocities foul less in the presence of adhesion-controlled fouling, their corresponding pumping costs are higher as well. As can be seen from Table 5.2, the clean pumping cost should be highest for the plates with half the standard plate width, and lowest for the parallel plates.

When adhesion-controlled fouling is expected, the excess area is best incorporated by having a two pass arrangement, or by using plates with half the standard plate width. These methods result in higher heat transfer rates than the standard exchanger by as much as 50% under clean conditions, and 65% under fouled conditions for 50 days, see Figure 5.10. It is interesting to note that incorporating the excess heat transfer area by adding parallel plates results in a lower heat transfer rate than the standard exchanger under fouled conditions. This traditional method of excess area incorporation should therefore be avoided especially in the presence of adhesion-controlled fouling.

The occurrence of reaction fouling can be significantly reduced by lowering the prevailing wall temperature on the cooling water side. This can be readily achieved by the installation of a flow control system to increase the cooling water flow rate. However, this would bring about a corresponding increase in the pumping cost. Therefore, if no flow control is possible, a two pass arrangement and plates with half the standard plate width should be avoided. Since all the other 'oversized' heat exchangers have comparable average wall temperatures, there is not much difference in their heat transfer performance in the presence of reaction fouling.

6. FLOW SIMULATION

The unique geometry of heat exchanger plates, created by the presence of corrugations, is responsible for the formation of fluid flow patterns of extreme complexity in the plate channels. These patterns have been the focal point of numerous experimental and numerical investigations. The former have been reviewed and discussed in Section 2.1.5. Although there are many reports on heat transfer and pressure drop characteristics, the experimental data accumulated offer little information on fouling behaviour, especially in terms of local deposition and removal rates. Implications for fouling behaviour made from these data are based on experimental evidence given by Bansal (1994) and by results of this dissertation. It has been confirmed that for solutions of inverse solubility, fouling predominates in regions of high temperature and low flow velocity.

Figure 6.1: Flow characteristics in a 2-D channel with a single corrugation on the bottom surface (Bansal, 1994).

Bansal (1994) simulated simplified flow arrangements to increase the fundamental understanding of fouling behaviour in PHEs. All simulations were performed for incompressible, isothermal, laminar and steady-state flow. Using PHOENICS, Figure 6.1 was obtained for a flow passing through a two-dimensional (2-D) channel with a single corrugation on the bottom surface. There is separation at the rear side of the corrugation giving rise to recirculation. In this region, not only are the local flow velocities lower, but the local heat transfer coefficients are lower as well. This in turn would result in low shear stresses and high wall temperatures, causing the corresponding removal rate of fouling deposits to decrease, and the deposition rate to increase.

Flow patterns in a clean flat PHE channel and in a fouled channel as shown in Figures 6.2 and 6.3 respectively, were also simulated by Bansal (1994). The circular inlet and outlet ports of the actual plate have been approximated by flat ends. Low velocity zones were predicted in the plate corners adjacent to the inlet and outlet ports. These are the regions where fouling deposits tend to form, as shown in the numerous photographs of fouled flat plates in Chapter 4 (for example, see Figure 4.53(b)). Figure 6.3 shows that partial blockage of the channel due to deposit formation significantly changes the flow pattern. These changes make it very difficult to design a PHE when fouling is expected.

Figure 6.2: Simulated flow pattern in a clean flat plate heat exchanger (Bansal, 1994).

Figure 6.3: Simulated flow pattern in a fouled flat plate heat exchanger (Bansal, 1994).

Mehrabian (1996) and Poulter *et al.* (1994) have attempted macroscopic CFD calculations of PHEs, in which the bulk flow through a flat channel with geometry similar to that of an APV junior paraflow plate was modelled. No recirculation was

observed in the simulations obtained for turbulent flow using the standard k - ϵ model. This is because the corners of the associated plate geometry are more streamline than the Alfa Laval ones simulated by Bansal (1994).

Figure 6.4: 'Repeat' building block of corrugated plate channels.

In order to resolve the complex geometry of PHEs with existing computational resources, a small segment of the exchanger was modelled. The smallest volume that was worthwhile resolving is one which represents a 'repeat' building block. This building block consists of two simple corrugated plates, arranged such that their principle axes intersect at

twice the corrugation angle, β_p as shown in Figure 6.4. Flow through this segment has been attempted by Ciofalo *et al.* (1996), Hugonnot *et al.* (1989), Mehrabian (1996), Mehrabian *et al.* (1997), Quarini *et al.* (1995) and Quarini *et al.* (1996). Although these attempts shed some new light on the local flow patterns between the corrugations, they fail to show the overall cork-screw pattern inside a plate channel. Information on how the flow zigzags across the heat exchanger length can only be revealed if a three-dimensional (3-D) flow simulation of a corrugated plate channel is performed. Such a model would no doubt require much computational space, memory and Central Processing Unit (CPU) time.

What seems to be missing from most, if not all of the literature on flow phenomena in PHE, is CFD modelling of the local flow patterns in different heat exchanger plate designs, and their effect on the corresponding heat transfer, pressure drop and fouling behaviour. The effect of distributor designs near the inlet and outlet region of the plates have not been looked into computationally. These questions/concerns led to the commencement of the study detailed in this dissertation, where extensive CFD modelling was undertaken to examine flow in a flat PHE channel.

For the present study, the main objectives of the flow simulations were twofold. Firstly, the simulations were aimed to investigate the extent of agreement between CFD predictions and experimental observations. The latter comprises ABS particle trackings and deposit growth patterns, observed via a polycarbonate plate used in conjunction with the flat metal plates, as previously described in Section 4.13. Secondly, it was then intended to use these results to examine the issue of distributor designs on heat exchanger plates. Hence, the strategy for flow simulations is to validate the CFD model through comparison with the experimental results first, and then to use the validated CFD model to simulate the different plate distributor designs.

6.1 Mathematical Formulation

In this section, the governing equations for laminar and turbulent flows are presented in Cartesian tensor notation. Turbulence models such as the k - ϵ model and the Reynolds Stress Model are also discussed, together with wall boundary conditions for turbulent flow.

Of the commercially available codes based on the finite volume method, the most widely used are CFX, FLUENT, STAR-CD and PHOENICS. CFX has been used throughout the course of this work, and AEA Technology (1995), Fluent Inc. (1996), and Versteeg and Malalsekera (1995) are referred to in this section.

6.1.1 Laminar Equations

The basic set of equations that require solving if a description of laminar fluid flow is required, comprise of equations for:

- (i) conservation of mass
- (ii) conservation of momentum (in the three co-ordinate directions)
- (iii) conservation of energy.

The general form of the mass continuity equation as shown below is valid for compressible and incompressible flows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad (6.1)$$

ρ is the fluid density, t the time, and u the flow velocity

The conservation of momentum in the i th direction in an inertial (non-accelerating) reference frame is given by:

$$\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = \frac{\partial \sigma_{ij}}{\partial x_j} + B_i \quad (6.2)$$

B_i is a body force in the i th direction. It includes contributions from gravitational acceleration and external body forces. The stress tensor, σ_{ij} is given by:

$$\sigma_{ij} = -p\delta_{ij} + \mu \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \frac{2}{3} \mu \frac{\partial u_i}{\partial x_i} \delta_{ij} \quad (6.3)$$

with the final term being the effect of volume dilation (zero for an incompressible fluid). The pressure and molecular viscosity are denoted by p and μ respectively.

In addition, a general scalar advection-diffusion equation for a dependent variable, ϕ is given by:

$$\frac{\partial \rho \phi}{\partial t} + \frac{\partial}{\partial x_i} \left(\rho u_i \phi - \Gamma \frac{\partial \phi}{\partial x_i} \right) = S \quad (6.4)$$

Γ is the diffusion coefficient, and S a source or sink term representing creation or destruction of ϕ . CFX allows the solution of any user defined scalars that obey a transport equation of this form.

For fully compressible flow, the energy transport equation is solved for the total enthalpy, H according to:

$$\frac{\partial \rho H}{\partial t} + \frac{\partial}{\partial x_i} \left(\rho u_i H - \lambda \frac{\partial T}{\partial x_i} \right) = \frac{\partial p}{\partial t} \quad (6.5)$$

λ is the thermal conductivity, and T the temperature. H is expressed in terms of the static enthalpy, h according to:

$$H = h + \frac{1}{2} u^2 \quad (6.6)$$

$$h = U_{ie} + p/\rho \quad (6.7)$$

For weakly compressible and incompressible flow, the kinetic energy term ($\frac{1}{2}u^2$) is assumed to be negligible compared to the internal term, U_{ie} . The pressure work term, may also be safely ignored.

The above equations, also known as the Navier-Stokes equations, represent five transport equations in the seven unknown field variables u, v, w, p, ρ, h, T . The thermal equation of state provides a sixth equation relating density, ρ to temperature, T and pressure, p (in CFX, by default this is the Ideal Gas Law for compressible gases). The seventh equation required to close the entire system is a thermodynamic relation between state variables. In CFX, this equation defines the function of static enthalpy in terms of temperature and pressure, i.e. $h = h(T, P)$. The fluid is assumed to be thermally perfect, and hence the static enthalpy is a function of temperature only.

6.1.2 Turbulence Equations

6.1.2.1 Turbulence Characteristics

At higher Reynolds numbers, flows become turbulent and exhibit a chaotic and random state of motion with continuously changing velocity and pressure characteristics. Particles of fluid initially separated by a significant distance can be brought closer together by the eddying motions in turbulent flows, and therefore heat, mass and momentum can be very effectively exchanged. Due to the large amount of eddies in a typical turbulent flow, direct calculation of fluid motion would require extremely large computational resources. Therefore, using a time averaging procedure commonly known as Reynolds averaging, conservation equations used for turbulent flows are obtained from those for laminar flows.

Each flow property $\phi(t)$, may be considered to be the sum of a mean flow property Φ , and a random time varying component $\phi'(t)$ such that:

$$\phi = \Phi + \phi' \quad (6.8)$$

where time averaging gives:

$$\bar{\phi} = \frac{1}{\Delta t} \int_t^{t+\Delta t} \phi(t) dt = \Phi \quad (6.9a)$$

$$\bar{\phi'} = \frac{1}{\Delta t} \int_t^{t+\Delta t} \phi'(t) dt = 0 \quad (6.9b)$$

Δt should in theory be taken as the limit towards infinity, but in practice it is sufficient that it exceeds the time scales of the slowest variations of flow property $\phi(t)$. With the fluctuating components substituted into the Navier-Stokes equations, and utilising standard rules governing the averages of the fluctuating properties, one obtains:

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = \frac{\partial \sigma_{ij}}{\partial x_j} + \frac{\partial}{\partial x_j}(-\rho \overline{u'_i u'_j}) + B_i \quad (6.10)$$

A comparison of Equations (6.2) and (6.10) reveals that the time-averaging process has introduced new terms involving products of fluctuating velocities which constitute convective momentum transfer due to the velocity fluctuations. These extra stress terms or Reynolds stresses produced consist of three normal and three shear stresses, and are usually large compared with the viscous stresses in turbulent flow. The convective transport due to turbulent velocity fluctuations will thus act to enhance mixing over and above that caused by thermal fluctuations at the molecular level. The scalar transport equation exhibits similar additional terms, and the enthalpy equation will contain a contribution from the turbulent kinetic energy, k as well as the mean kinetic energy, where:

$$k = \frac{1}{2} \overline{u'_i u'_i} \quad (6.11)$$

Turbulence models provide closure for the flow equations by providing models for the computation of the Reynolds stresses and the scalar transport terms.

6.1.2.2 The k - ϵ Turbulence Model

This eddy-viscosity type model relies on the assumption that there exists an analogy between the action of viscous stresses and Reynolds stresses on the fluid flow. The Reynolds stresses are assumed to be proportional to the mean velocity gradients with

the turbulent or eddy viscosity, μ_t as the constant of proportionality. Utilising this Boussinesq hypothesis, one therefore obtains:

$$-\rho \overline{u'_i u'_j} = -\rho \frac{2}{3} k \delta_{ij} + \mu_t \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \frac{2}{3} \mu_t \frac{\partial u_i}{\partial x_i} \delta_{ij} \quad (6.12)$$

Similar to the molecular viscosity μ , the turbulent viscosity μ_t is equal to the product of density and kinematic turbulent viscosity ν_t , i.e. $\mu_t = \rho \cdot \nu_t$.

By substituting the expressions for σ_{ij} and $-\rho \overline{u'_i u'_j}$ given in Equations (6.3) and (6.12) respectively, into Equation (6.10), it follows that:

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p'}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu_{eff} \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] \right) + B_i \quad (6.13)$$

$$\text{where } \mu_{eff} = \mu + \mu_t \quad (6.14)$$

It can therefore be seen that modified Reynolds averaged momentum equations remain similar to the laminar equations, though the molecular viscosity μ , is replaced by an effective viscosity μ_{eff} . In addition, the pressure is replaced by a modified pressure p' such that:

$$p' = p + \frac{2}{3} \rho k + \mu_{eff} \frac{2}{3} \mu \frac{\partial u_i}{\partial x_i} \quad (6.15)$$

The turbulent viscosity, μ_t is obtained by assuming that it is proportional to the product of a turbulent velocity scale and length scale according to:

$$\mu_t = \rho \cdot C_\mu \cdot \left(\frac{k^2}{\epsilon} \right) \quad (6.16)$$

The velocity scale is taken to be $k^{1/2}$, and the length scale $k^{3/2}/\epsilon$, where ϵ is the dissipation rate of k . C_μ is an empirical constant.

The values of k and ϵ required for substitution into Equation (6.16) are obtained by the solution of the following two transport equations:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{\sigma_k} \cdot \frac{\partial k}{\partial x_i} \right) + G_k + G_b - \rho \epsilon \quad (6.17)$$

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho u_i \epsilon) = \frac{\partial}{\partial x_i} \left(\frac{\mu_t}{\sigma_\epsilon} \cdot \frac{\partial \epsilon}{\partial x_i} \right) + c_{1\epsilon} \cdot \frac{\epsilon}{k} (G_k + (1 - c_{3\epsilon}) G_b) - c_{2\epsilon} \cdot \rho \frac{\epsilon^2}{k} \quad (6.18)$$

The coefficients $c_{1\epsilon}$, $c_{2\epsilon}$, and $c_{3\epsilon}$ are empirical constants, and σ_k and σ_ϵ are Prandtl numbers governing the turbulent diffusion of k and ϵ . G_k is the rate of production of turbulent kinetic energy, and G_b is the generation of turbulence due to body force.

A number of general features of the k - ϵ model include:

- (a) Simplest turbulence model for which only initial and/or boundary conditions need to be supplied.
- (b) Good performance for many industrially relevant flows.
- (c) Well established, and the most widely validated turbulence model.
- (d) Assumes turbulent viscosity is isotropic, i.e. the ratio between Reynolds Stresses and mean rate of deformation is the same in all directions. This assumption leads to a poor performance in cases such as rotating flows, and flows with extra large strains (e.g. swirling flows).

6.1.2.3 Reynolds Stress Model

The isotropic assumptions made in the k - ϵ model may prove inadequate in flows where the velocity and length scales can vary significantly with direction. The Reynolds Stress Model (RSM) involves solving equations for the individual components of the Reynolds stress. In the Algebraic Stress Model (ASM), these equations are solved algebraically whereas in the Differential Stress Model (DSM), transport equations are solved. Assumptions are made about the six transport equations for the six Reynolds Stresses, and the resulting partial differential equations (PDEs) are solved in conjunction with the transport equation for the rate of dissipation of turbulent kinetic energy, ϵ .

$$\begin{aligned}
& \frac{\partial}{\partial t} (\overline{u'_i u'_j}) + u_k \frac{\partial}{\partial x_k} (\overline{u'_i u'_j}) = && \text{Convective Transport} \\
& - \frac{\partial}{\partial x_k} \left[\overline{(u'_i u'_j u'_k)} + \frac{p}{\rho} (\delta_{kj} u'_i + \delta_{ik} u'_j) - \nu \frac{\partial}{\partial x_k} (\overline{u'_i u'_j}) \right] && \text{Diffusive Transport} \\
& - \left[\overline{u'_i u'_k} \frac{\partial u_j}{\partial x_k} + \overline{u'_j u'_k} \frac{\partial u_i}{\partial x_k} \right] && \text{Production terms} \\
& + \frac{p}{\rho} \left[\frac{\partial u'_i}{\partial x_j} + \frac{\partial u'_j}{\partial x_i} \right] && \text{Pressure Strain term} \\
& - 2\nu \left[\frac{\partial u'_i}{\partial x_k} \frac{\partial u'_j}{\partial x_k} \right] && \epsilon_{ij} = \text{Dissipation term} \\
& - 2\Omega_k \left[\overline{u'_j u'_m} \epsilon_{ikm} + \overline{u'_i u'_m} \epsilon_{jkm} \right] && \text{Rotational Term}
\end{aligned}$$

(6.19)

A number of general features of RSM include:

- (a) Only initial and/or boundary conditions need to be supplied
- (b) A more general turbulence model is produced.
- (c) Accurate calculation of mean flow properties and all Reynolds Stresses for many simple and more complex flows.
- (d) Large CPU resources required to solve the additional PDEs.
- (e) Not as widely validated as the k - ϵ model.

6.1.3 Wall Boundary Conditions for Turbulent Flow

Turbulent flows in the near wall region are affected by the presence of the walls. Firstly, the mean velocity field is affected through the no-slip condition which has to be satisfied at the wall. In addition, turbulence is also changed by the presence of the wall in non-trivial ways. As a result, accurate representation of the flow in the near wall region and successful predictions of wall-bounded turbulent flows go hand in hand.

It is well documented in the literature and backed up by experimental evidence that, in fully developed turbulent flow, the cross-section of the flow can be largely divided into

three layers (Coulson and Richardson, 1996) (Holland and Bragg, 1995) (Douglas *et al.*, 1979). In the innermost layer, called the laminar or viscous sublayer, the flow is almost laminar and molecular transport plays a dominant role in the transfer of momentum, heat and mass. The outer layer is called the fully-turbulent layer, where the contribution of eddy transport is much greater than that of molecular transport, and so the latter can be neglected. Finally, there is an interim region between these two layers called the buffer layer, where the effects of molecular and eddy transport are of comparable importance.

To obtain expressions for the velocity distribution over the whole flow cross-section, dimensional analysis is used together with experimental evidence and a series of assumptions based on the relative importance of certain flow variables. Using these, the structure of the velocity profile near a surface can be approximated by the universal velocity profile. In the fully-turbulent layer, velocity profiles obey a logarithmic law. In the computation of flows near surfaces, there are two models to account for flows in the near wall regions, namely the wall function method and the near-wall approach.

6.1.3.1 Wall Function Method

With the wall function method, semi-empirical formulas are employed to bridge the region between the wall and the fully-turbulent region. In most high-Reynolds-number flows, this method substantially saves computational resources as computations do not have to be performed right to the wall, where the solution variables change most rapidly. In addition, this method is robust and reasonably accurate. However, this method is inadequate in situations where low-Reynolds-number effects are present.

For the k - ϵ model and RSM, boundary conditions are specified using equilibrium wall functions. These functions connect the wall conditions (e.g. wall shear stress) to dependent variables at the near-wall grid node. This grid node is presumed to lie in fully-turbulent fluid, with a y^+ value greater than 11. y^+ is the dimensionless sublayer thickness, the point at which the wall functions are matched to the inner solution. Near the wall, it is found that the wall shear stress, τ is related to the turbulence kinetic

energy by:

$$\tau^2 = C_\mu \cdot \rho^2 \cdot k^2 \quad (6.20)$$

The wall functions for mean velocity yield the following scaled variables:

$$u^+ = \frac{1}{\kappa} \cdot \ln(E \cdot y^+) \quad (6.21)$$

where

$$u^+ = \left(\frac{\rho \cdot C_\mu^{1/4} \cdot k^{1/2}}{\tau} \right) \cdot u \quad (6.22)$$

$$y^+ = \left(\frac{\rho \cdot C_\mu^{1/4} \cdot k^{1/2}}{\mu} \right) \cdot y \quad (6.23)$$

and κ = von Karman's constant (0.42)

E = empirical constant (9.81)

y = distance to the wall

The scaled velocity component parallel to the wall and in the x direction is:

$$u^+ = \begin{cases} y^+, & y^+ < y_0^+ \\ \frac{1}{\kappa} \cdot \ln(E \cdot y^+), & y^+ > y_0^+ \end{cases} \quad (6.24)$$

The cross-over point y_0^+ between the laminar sublayer and the logarithmic region is the upper root of:

$$y_0^+ = \frac{1}{\kappa} \cdot \ln(E \cdot y_0^+) \quad (6.25)$$

6.1.3.2 Near-Wall Approach

For the near-wall approach, a very fine computational mesh is required to resolve the viscosity-affected region all the way to the wall. Although fairly accurate for low Reynolds number flow, this approach is not always very robust. In addition, the required grid causes simulations involved to be highly expensive in terms of computational resources.

6.1.4 Wall Boundary Conditions for Turbulent Heat Transfer

Due to Reynolds' analogy between momentum and energy transfer, the treatment of the heat transfer boundary conditions is similar to that previously outlined for the velocity boundary conditions. As for the law-of-the-wall for velocity, the law-of-the-wall for temperature (or enthalpy) comprises the following two different laws:

- (i) Linear law for the thermal conduction sublayer where conduction is important. The thickness of this layer is, in general, different from the thickness of the (momentum) laminar sublayer, and changes from fluid to fluid.
- (ii) Logarithmic law for the turbulent region where effects of turbulence dominate conduction.

The enthalpy near the wall layer takes the following form:

$$H^+ = \begin{cases} Pr_H \cdot y^+, & y^+ < y_H^+ \\ \frac{Pr_t}{\kappa} \cdot \ln(E_H \cdot y^+), & y^+ > y_H^+ \end{cases} \quad (6.26)$$

where

$$H^+ = \left(\frac{\rho \cdot C_\mu^{1/4} \cdot k^{1/2}}{J_H} \right) \cdot (H_{wall} - H) \quad (6.27)$$

$$Pr_H = \frac{\mu}{\Gamma_H} \quad (6.28)$$

and Pr_t = turbulent Prandtl number

J_H = flux of H at the wall

H_{wall} = value of H at the wall

In CFX, the formula of Jayatilleke (1969) is used for the calculation of E_H , provided that the value is not larger than 5×10^{35} :

$$E_H = E \cdot \exp \left\{ 9\kappa \left(\left[\frac{Pr_H}{Pr_t} \right]^{0.75} - 1 \right) \cdot \left(1 + 0.28 \exp \left[-0.007 \frac{Pr_H}{Pr_t} \right] \right) \right\} \quad (6.29)$$

The quantity, y_H^+ is the larger root of the following non-linear equation:

$$\frac{Pr_t}{\kappa} \cdot \ln(E_H \cdot y_H^+) = Pr_H \cdot y_H^+ \quad (6.30)$$

6.2 Computational Implementation with CFX

CFX is a finite-volume based code using a structured, patched multi-block, nonorthogonal, curvilinear co-ordinate grid with a collocated variable arrangement. The basic solution algorithm is the SIMPLEC pressure correction scheme which uses a variety of linear equation solvers. Spatial discretisation is achieved through the HYBRID scheme, a second-order upwind scheme, and the third order QUICK scheme. CFX has models for multi-phase flow, particle transport, gaseous combustion, chemical species concentration, thermal radiation, compressible and incompressible flows, porous media flow, and conjugate heat transfer. Turbulent flows are modelled with six different methods, i.e., the k - ϵ model, a low Reynolds number k - ϵ model, a RNG k - ϵ model, an algebraic Reynolds stress model, a differential Reynolds stress model, and a differential Reynolds flux model (AEA Technology, 1995) (Freitas, 1995).

CFX is structured around a numerical algorithm that can solve fluid flow problems, in association with user interfaces that allow the input of problem parameters and the examination of results. Three main elements are contained within the CFX code, namely pre-processor, solver and post-processor (AEA Technology, 1995).

(i) Pre-processor

Pre-processing consists of the input of a flow problem, followed by the transformation of this input to a form suitable for use by the solver. CFX-MESHBUILD was used in this work for the definition of the computational domain, grid generation, and specification of some boundary conditions. Multi-block grids are constructed by patching together a number of simple, rectangular grids. Each block of a multi-block grid is uniform in computational space, but may be bent or stretched without tearing or folding to follow the problem boundary geometry in physical space.

(ii) Solver

The solver, CFX-F3D integrates the governing fluid flow equations over the control volumes obtained from the discretisation of the domain. This yields a system of algebraic equations which are solved by iteration.

(iii) Post-processor

The post-processor allows clear data visualisation through the plotting and display of, for example, domain geometry, vector plots, shaded contour plots and particle tracks. CFX-VISUALISE and CFX-VIEW were used in this work.

6.3 The Simulations

The ultimate goal of CFD modelling of PHEs is to be able to predict the local flow and temperature distributions inside any type of plate channel with reasonable accuracy. This local information can then be used to model the corresponding fouling behaviour. Once established, CFD is powerful enough to be used to modify the present design of PHEs to reduce fouling, and even to redesign PHEs completely in order to achieve higher heat transfer performance. However, this goal may only be achievable via the development of better CFD codes with more robust and accurate turbulence models, and faster computers with larger memory.

With the resources available for this work, the aim was to use CFD techniques to simulate the flow patterns and the associated temperature distributions inside a flat PHE channel, and to investigate their implications on fouling. In addition, the impact of distributor designs on the local flow and temperature distributions in such a channel was also studied. In order to make comparisons with the experimental data of this dissertation, CFD simulations were performed with plate geometries similar to those of the Alfa Laval P01 and M3 diagonal flow plates.

The approach taken in this CFD work was to look at the analyses of Bansal (1994) first. Hence, the flow passing through a 2-D channel with a single corrugation on the bottom surface was simulated. However, this was taken one step further by modelling the flow passing through 2-D channels with multiple corrugations on the top and bottom surfaces. In order to investigate the performance of the different turbulent models available, the flow passing over a 2-D backward-facing step was simulated, and the results were compared with data from the literature.

To have some knowledge of the input values of k and ϵ for the turbulence models used, the inlet to the PHE was simulated. These values were then used to model the flow through the following geometries. These geometries are a progression of very simple framework to the actual form of the flat PHE investigated experimentally in this dissertation:

- (i) A double backward-facing step. This can be seen as a very simplified flat plate channel, see Figure 6.21.
- (ii) Another simplified flat plate channel with flat inlet and outlet, see Figure 6.24.
- (iii) Actual flat plate channel with round inlet and outlet, see Figure 6.31.
- (iv) Actual flat plate channels with different distributor designs, see Figures 6.48 to 6.54.
- (v) Modified flat plate channel, see Figure 6.55.

In the present study, CFD modelling of fluid flow was performed using a commercial package, CFX, on a pentium PC linked by eXceed to a DEC ALPHA 3000 work station, and later on an AlphaServer 8400 5/440 with 8 processors. The fluid under investigation was water, and its properties were obtained from the CFX data base. Generally, inlet conditions such as flow velocity and temperature were specified together with pressure at the outlet. In the double-precision calculations performed, a maximum mass tolerance of 1×10^{-7} kg/s was attained for the sum of the absolute residual of all the cells in the grid. Only incompressible flow with body-fitted co-ordinates were simulated. Most of the CFD results presented in this chapter are grid independent. Results which are not verified to be grid independent due to limitations in computational resources, however have comparable ratios of cell sizes to actual dimensions of their corresponding geometries with those of grid independent simulations. Therefore, it is reasonable to consider that all the CFD results presented are grid independent or almost grid independent. For all the turbulent flows simulated, reasonable y^+ values have been obtained. Copies of typical CFX command files for the different geometries simulated can be found in Appendix B.

The CFD predictions obtained were compared with the results of flow visualisation studies conducted by observing the flow patterns of Acronitrile Butadiene Styrene

(ABS) particles in water via a polycarbonate plate set-up. This visualisation technique is discussed in greater detail in Section 4.7. The CFD predictions for the flat plate channel were found to be in reasonable agreement with the experimental results of flow visualisation and fouling, but discrepancies exist with regard to the reattachment point of the main flow recirculation zone in the flat plate channel. The CFD work undertaken in this dissertation and the implications of the results obtained are discussed in the following sections.

6.3.1 Flow Passing Through a 2-D Channel with a Single Corrugation on the Bottom Surface

Simulations were performed for one of the flow situations investigated by Bansal (1994), namely, a flow passing a 2-D channel with a single corrugation on the bottom surface. The channel geometry and boundary conditions used by Bansal, were adopted in this present work. The height of the corrugation is 2.4 mm which is similar to those in Alfa Laval P01 and M3 plates. The grid was set up in such a way that the grid cells near the corrugation were finer than those further away. Simulations were performed for water with an inlet temperature of 302 K (or 29°C), and an inlet velocity of 0.15 m/s. The flow was 2-dimensional, laminar, incompressible, isothermal and at steady state.

Figure 6.5: Flow characteristics in a 2-D channel with a single corrugation on the bottom surface for 0.15 m/s inlet velocity.

Figure 6.6: Flow characteristics in a 2-D channel with a single corrugation on the bottom surface for 0.10 m/s inlet velocity.

Figure 6.7: Flow characteristics in a 2-D channel with a single corrugation on the bottom surface for 0.05 m/s inlet velocity.

The main difference between the flow characteristics obtained from PHOENICS (Figure 6.1) and that obtained from CFX in this present investigation (Figure 6.5), is that instead of one region of recirculation, two regions of recirculation are observed in the latter case. This may be due to the finer grid used in the present work. To examine the effects of flow velocity on the recirculation zones, the inlet fluid velocity was decreased to 0.10 and 0.05 m/s. The flow characteristics for these flow velocities can be seen in Figures 6.6 and 6.7.

Tailing the first recirculation zone is a second recirculation zone which occurs at the upper surface of the channel, for inlet velocities 0.15 and 0.10 m/s. In these two cases, before the flow reaches the corrugation, the maximum velocity region was at the centre of the channel. As the fluid approaches the minimum cross-sectional area of the corrugation, the maximum velocity moves up towards the top of the channel. Following this, the maximum velocity moves back towards the centre of the channel, and the first recirculation region is created. However, it can be seen from Figures 6.5 and 6.6 that, in moving back towards the channel centre, the fluid maximum velocity overshoots. When this happens, a second smaller recirculation zone is created, after which a steady move towards the centre of the channel is attained by the maximum velocity. As the inlet velocity is lowered to 0.05 m/s, the shift in the maximum velocity zone after passing the corrugation, became smoother. Therefore, no second recirculation zone was observed in Figure 6.7.

Figures 6.5, 6.6 and 6.7 show the results for an abrupt corrugation. Flow passing through a 2-D channel with a single smooth corrugation on the bottom surface was simulated for an inlet flow velocity of 0.15 m/s. The dimensions of this channel are similar to those of the channel with an abrupt corrugation. The results obtained *are* shown in Figure 6.8, which when compared with Figure 6.5 indicates that there is little difference in the flow distribution between the smooth and abrupt corrugations.

Figure 6.8: Flow characteristics in a 2-D channel with a single smooth corrugation on the bottom surface for 0.15 m/s inlet velocity.

6.3.2 Flow Passing Through a 2-D Channel with Multiple Corrugations on the Top and Bottom Surfaces

As stated in Section 2.1.1, the corrugation parameters that affect the performance of PHEs are:

- (a) corrugation angle, β_p
- (b) corrugation wavelength, λ_p
- (c) corrugation amplitude, a_p

Contradictory findings have been reported by some investigators (Gaiser and Kottke, 1990) (Okada *et al.*, 1972) (Saniei and Dini, 1993a) with regard to the effects of λ_p , a_p , and the plate channel gap b_p , on heat transfer and pressure drop. In order to perform a reduced number of experiments, instead of examining each parameter of interest individually, it is usual practice to study the combined effect of λ_p and a_p , as a ratio of λ_p/a_p , and the combined effect of b_p and a_p , as a ratio of b_p/a_p . A summary of the values of these parameters reported in the literature surveyed is given in Table 2.1. A comparison of the range of these parameters and their corresponding heat transfer and pressure drop trends, has led to the theorems proposed in Section 2.1.5. A series of CFD simulations have been performed to analyse the validity of these theorems, which relate the two aforementioned combined effects to heat transfer and pressure drop.

Sinusoidal corrugations with an angle of 90° to the flow direction were simulated on the top and bottom surfaces of 2-D flow channels. The flow through this type of geometry is similar to that passing through a washboard heat exchanger plate channel or a wavy channel as shown in Figure 2.3(b). Although most heat exchanger plates have corrugation angles other than 90° , this angle was chosen so that a 2-D flow geometry with lower CPU requirement can be used instead of a 3-D one. In these simulations, the values of λ_p , a_p and b_p were varied with respect to those of the Alfa Laval M3 plates, which are given in Table 3.1. For these plates, λ_p , a_p and b_p are equivalent to 10.3, 1.2 and 2.4 mm respectively. As a result, the ratios λ_p/a_p and b_p/a_p are 8.58 and 2.0 respectively. The simulated channels all have lengths comparable to that of 10 Alfa Laval M3 type corrugations, which is equivalent to 0.103 m.

The fluid simulated was water, with an inlet flow velocity and temperature of 0.10 m/s and 40°C respectively. The fluid properties were always evaluated at 45°C. However, the results obtained from these simulations were found to be insensitive to the choice of this temperature. For heat transfer purposes, the channel walls were specified to be at an elevated temperature of 70°C. Therefore, heat transfer simulations were performed for a 2-D, laminar, incompressible, steady flow.

The inlet and outlet of the channel were specified as an inlet velocity boundary and an outlet pressure boundary respectively in CFX. Due to the latter specification, it is possible for cases where there is flow recirculation at the channel outlet, to have fluid drawn in through the outlet from outside the flow domain investigated. Unless an outlet temperature is specified, this fluid enters at the ambient temperature (288 K by default). Therefore, whenever necessary, the outlet temperature is set at a guess value similar to the local temperature simulated at the outlet. The correct guess value can be obtained by trial and error.

Run ID	λ_p	a_p	b_p	λ_p/a_p	b_p/a_p	no. of cor.	Re_{in}	Δp	T_{out}
	[mm]	[mm]	[mm]	[/]	[/]		[/]	[Pa]	[°C]
1	10.3	1.2	2.40	8.58	2.0	10	797	89	58
2	34.3	1.2	2.40	28.58	2.0	3	797	20	52
3	4.1	1.2	2.40	3.42	2.0	25	797	1239	68
4	10.3	1.2	0.24	8.58	0.2	10	80	2065	70
5	10.3	1.2	0.72	8.58	0.6	10	239	250	68
6	10.3	1.2	24.0	8.58	20.0	10	7971	5	43

Table 61: Summary of input parameters and results obtained from simulations on the effect of corrugation wavelength λ_p , and plate channel gap b_p on heat transfer and pressure drop.

A summary of the input parameters and results obtained from the simulations, including the number of corrugations (no. of cor.), Reynolds number based on the inlet velocity (Re_{in}), pressure drop (Δp) and outlet temperature (T_{out}) of the channel, are shown in

Table 6.1. As can be seen, the value of a_p was maintained to be constant for all the simulations. Moreover, the heat transfer rate can be inferred from the simulated T_{out} values. Runs 1, 2 and 3 show the effect of λ_p on heat transfer and pressure drop, and are presented in greater detail in Section 6.3.2.1 with reference to Section 2.1.5.2. On the other hand, runs 1, 4, 5 and 6 illustrate the effect of b_p on heat transfer and pressure drop, and are discussed in detail in Section 6.3.2.2 in conjunction to Section 2.1.5.4. However, it would be difficult to draw any implications from these sets of results on the corkscrew flow patterns in chevron plate channels. Modifications to the grids would necessary before such zigzag flow patterns can be simulated.

6.3.2.1 Effect of Corrugation Wavelength, λ_p on Heat Transfer and Pressure Drop

The geometries for runs 2, 1 and 3 with inserts of the actual grids are shown in Figures 6.9(a), (b) and (c) respectively. The grid sizes used for these three 2-D runs in terms of horizontal length and vertical channel gap are 800×20 for both runs 1 and 2, and 1000×20 for run 3. As can be seen in Table 6.1, at a fixed a_p , the ratio of λ_p/a_p decreases with increasing number of corrugations from run 2 to run 1, and finally to run 3.

When the corrugation wavelength is high as in run 2, the fluid flows through this gentle wavy channel as if it were flowing through a flat channel, displaying no flow recirculation zones. The simulated vector distributions of speed (combination of u , v and w velocities) and temperature are shown in Figures 6.10(a) and (b) respectively. Both the direction and length of the speed and temperature vectors indicate the flow direction and the flow magnitude respectively. On the other hand, the colour of the speed and temperature vectors represent the magnitude of the local flow velocity and temperature respectively. As no rigorous mixing is achieved in this geometry, both the heat transfer and pressure drop are low when compared with runs 1 and 3. Runs 1 to 6, all have identical keys for speed and temperature, and therefore these keys are only shown in Figures 6.10(a) and (b).

Figures 6.9(a), (b) and (c): 2-D geometries for runs 2, 1 and 3 respectively, with inserts of the actual grids.

In run 1 where the Alfa Laval M3 plate corrugation dimensions were used, the periodicity of the corrugations are frequent enough to have an impact on the fluid flow pattern. Flow separation occurs and mixing intensifies, thus enhancing the heat transfer process. Since the fluid flows through more corrugations per unit plate length, the corresponding pressure drop increases. Figures 6.11(a) and (b) show the flow and temperature distributions for run 1 respectively. As expected, regions of flow recirculation occur on the lee side of corrugations. It is at these local low-velocity zones that the prevailing temperature is higher. This is because the lower the local flow velocity, the higher the residence time of the fluid in that local region, giving it ample opportunity to be heated up by the hot wall.

Figures 6.10(a) and (b): Flow and temperature distributions for run 2 ($\lambda_p/a_p = 28.58$; $b_p/a_p = 2.0$).

Figures 6.11(a) and (b): Flow and temperature distributions for run 1 ($\lambda_p/a_p = 8.58$; $b_p/a_p = 2.0$).

Figures 6.12(a) and (b): Flow and temperature distributions for run 3 ($\lambda_p/a_p = 3.42$; $b_p/a_p = 2.0$).

In run 3 where the corrugations are very close to one another, the fluid has to flow through the many hills and valleys of the corrugations in order to flow through the channel. As in run 1, intense mixing occurs for run 3, leading to increased heat transfer with a corresponding sharp rise in pressure drop. Figures 6.12(a) and (b) illustrate the flow and temperature distributions for run 3 respectively.

From this set of simulations, it can be concluded that as the corrugation wavelength, λ_p is decreased, heat transfer is augmented with a corresponding rise in pressure drop. This supports the theorem given in Section 2.3.4.

6.3.2.2 Effect of Plate Channel Gap, b_p on Heat Transfer and Pressure Drop

Simulations with increasing plate channel gap were performed with a_p and λ_p being kept constant. The ratio b_p/a_p used in the simulations are 0.2, 0.6, 2.0 and 20.0. The geometries for runs 4, 5 and 6 with inserts of the actual grids are shown in Figures 6.13(a), (b) and (c) respectively. The grid sizes used for these runs are 1000×20 for runs 4 and 5, and 300×75 for run 6. As can be seen from Table 6.1, the ratio b_p/a_p decreases from run 4 to run 5, and then to run 1, and finally to run 6. Therefore, the effect of channel gap, b_p on heat transfer and pressure drop is discussed below in this order (runs 4, 5, 1, and 6), with close reference to Section 2.1.5.4.

Figures 6.14(a) and (b) show the flow and temperature distributions of a narrow corrugated channel with b_p/a_p equal to 0.2 (run 4). As postulated in Section 2.1.5.4, the fluid sees this channel as a straight channel and flows through as if it were flowing through a flat plate channel. Because of this, no mixing occurs but due to the small distance between the two hot walls, the fluid heats up rather rapidly. Unfortunately, this high heat transfer rate is not achieved without a high pressure drop.

Increasing the ratio b_p/a_p to 0.6 in run 5, brought about a decrease in both the outlet temperature and pressure drop when compared with run 4, as shown in Figures 6.15(a) and (b). Nevertheless, the flow patterns for runs 4 and 5 are similar such that the presence of flow recirculation is still undetected. Recirculating flow is finally detected

on the lee sides of the corrugations when b_p/a_p is further increased to 2.0 in run 1, as shown in Figures 6.11(a) and (b).

Figures 6.13(a), (b) and (c): 2-D geometries for runs 4, 5 and 6 with inserts of the actual grids.

However, increasing the plate channel gap further such that b_p/a_p is 20.0 as in run 6, the fluid once again sees the channel as a flat channel. As shown in Figures 6.16(a) and (b), the corrugations are unable to induce mixing of the fluid throughout the cross-section of the channel, leaving pockets created by the corrugations filled with slow recirculating fluid. As a result, heat transfer decreased with a corresponding decrease in pressure drop.

As the plate channel gap increases, the heat transfer and the corresponding pressure drop both decrease. This conclusion agrees with the theorem given in Section 2.1.5.4.

Figures 6.14(a) and (b): Flow and temperature distributions for run 4 ($\lambda_p/a_p = 8.58$; $b_p/a_p = 0.2$).

Figures 6.15(a) and (b): Flow and temperature distributions for run 5 ($\lambda_p/a_p = 8.58$; $b_p/a_p = 0.6$).

Figures 6.16(a) and (b): Flow and temperature distributions for run 6 ($\lambda_p/a_p = 8.58$; $b_p/a_p = 20.0$).

6.3.3 Flow Passing Over a 2-D Backward-Facing Step

Flow separation, recirculation and reattachment are features observed in many flows. They occur whenever a fast-flowing fluid is required to bypass an obstacle or whenever a confining wall undergoes a rapid change in orientation, such as the geometry of PHE flow channels.

In this section, a computational study is presented, in which the predictive capabilities of the turbulence models available in CFX were examined, with reference to a separated flow behind a backward-facing step in a 6° expanding channel. The geometry and flow inlet conditions used are similar to the experimental study of Driver and Seegmiller (1985), and the computational study of Lien and Leschziner (1994). All the models and variants pertaining to the latter, were applied to the flow investigated by the former. The results obtained from this study are compared with those from these two publications.

Driver and Seegmiller (1985) conducted their experiments on the tunnel floor of a low-speed wind tunnel facility. Their experimental data showed that the reattachment length is $8.2H_s$, where H_s is the step height. In addition, a secondary corner eddy of length $1H_s$ was also found to exist. Based on the step height H_s , and the maximum inlet velocity, Lien and Leschziner (1994) used a Reynolds number of 36000 in their simulations. The numerical grids used to obtain the high- Re model and the low Re -model solutions contained 110×60 and 110×80 cells respectively. The closest agreement is achieved with the non-linear RNG combination, and the Reynolds stress model (RSM) with isotropic stress-diffusion approximation (which is also known as extra wall reflection terms). Both the standard $k-\varepsilon$ model and the standard RSM similar to those used in CFX, predict early reattachment points of $6.0H_s$ and $6.2H_s$, respectively. The extra wall reflection terms present in the pressure strain tensor for both the algebraic and differential forms of RSM are complex and have been omitted in the standard form of CFX (AEA Technology, 1995).

In the present analysis, both the k - ϵ model and the differential Reynolds stress model (DSM) with default differencing schemes were used to solve for air flow and water flow over a backward-facing step at atmospheric conditions. Air was used because it was used by Driver and Seegmiller (1985) in conducting the experiments. Water was also simulated as this is the fluid used in the present experimental study. The results calculated from these turbulence models would give an indication of the degree of accuracy of the reattachment point in the flat plate channel computed in Section 6.3.7.

6.3.3.1 The k - ϵ Model

The computational mesh is made up of a 2-D, body fitted grid with Cartesian coordinates. The full computational mesh consists of 150×60 cells and is illustrated in Figure 6.17. This mesh was obtained from grid-independence tests with coarser grids. The step height, H , is 0.1 m, and the flow enters as a fully developed flow into the domain. The flow is isothermal, incompressible and at steady state. The physical properties of the fluids were taken at 20°C and at atmospheric pressure.

Figure 6.17: Computational mesh for backward-facing step created using CFX.

From the simulations, the maximum inlet velocities for air and water are 5.6140 m/s and 3.9436×10^{-1} m/s respectively. As their vector and contour plots are similar, only those of water are shown in Figures 6.18(a) and (b) respectively. Both air and water display an early reattachment at $6.3H_s$, which agrees with that predicted by Lien and Leschziner (1994). As can be seen, the k - ϵ model underpredicts the reattachment point by about 23% for this geometry.

Figures 6.18(a) and (b): Vector and contour plots of k - ϵ model showing the reattachment point for water over a backward-facing step.

6.3.3.2 Differential Reynolds Stress Model

The same grid and mass flux input values as for the k - ϵ model were used for DSM. With DSM, it took about 1.68×10^5 and 2.50×10^5 seconds in terms of CPU time for air and water to reach convergence respectively. These CPU times are about 8 to 10 times longer than those required by the k - ϵ model.

For air, the maximum inlet velocity simulated is 5.6389 m/s, while that for water is 3.9610×10^{-1} m/s. As with the k - ϵ model, DSM predicted similar flow patterns for air and water, and therefore only the vector and contour plots for water are shown in Figures 6.19(a) and (b) respectively. Unlike the smooth speed contours of the k - ϵ model at the reattachment region, the DSM speed contours have a nose-like shape. Hence, for DSM, the reattachment point is taken to be vertically below the tip of this 'nose'. Early reattachment at $5.8H_s$ is obtained for both air and water. This agrees

with the predictions of Lien and Leschziner (1994). However, a second recirculation region directly next to the step can also be observed. The simulated length of this recirculation zone is less than the experimental value of $1H_s$, given by Driver and Seegmiller (1985).

Figures 6.19(a) and (b): Vector and contour plots of DSM showing the reattachment point for water over a backward-facing step.

As can be seen, for this geometry, DSM underpredicts the reattachment point by about 29%. As for the k - ϵ model, DSM gives identical reattachment points for both air and water. Although DSM predicts the existence of a second recirculation region as observed in experiments, the predicted reattachment lengths of the main recirculation region are slightly shorter than those using the k - ϵ model. The overall and local flow patterns predicted by DSM bear a closer resemblance to the experimental observations. However, the k - ϵ model is able to capture the expected overall flow pattern successfully in less CPU time.

6.3.4 Inlet Conditions for Flow in Flat Plate Channels

Among the boundary conditions required to simulate flow and temperature distributions for different geometries of flat plate channel in subsequent sections, are those listed in Table 6.2. These parameters are either directly or indirectly obtained from experiments, unless otherwise specified. The wall temperatures were calculated using Equation 2.14. For the same mass flow rate, the flow velocity at different inlet geometries are also given. Although default inlet values for k and ε can be calculated via pertinent equations given in CFX manuals (AEA Technology, 1995), their values can also be obtained from CFD simulations.

<i>Under clean conditions:</i>		<i>Nominal flow velocity in plate channels:</i>			
		0.20 m/s	0.35 m/s	0.65 m/s	1.00 m/s
Inlet temperature	°C	41.5	41.5	41.5	41.5
Outlet temperature	°C	-	47.5	-	-
Average temperature	°C	45.0	45.0	45.0	45.0
Wall temperature (calculated)	°C	-	59.0	-	-
Mass flow rate	kg/s	0.048	0.083	0.155	0.238
Flow velocity at simplified plate channel inlet	m/s	0.500	0.875	1.625	2.500
Flow velocity at real plate channel inlet	m/s	0.060	0.104	0.194	0.298
Pressure at plate channel inlet	Pa	1.0×10^4	1.5×10^4	3.0×10^4	6.0×10^4
Pressure drop across a single plate channel	Pa	0.5×10^3	1.4×10^3	4.3×10^3	9.1×10^3
k at plate channel inlet (simulated)	m^2/s^2	-	1.5×10^{-3}	4.4×10^{-3}	1.5×10^{-2}
k at plate channel inlet (calculated)	m^2/s^2	1.1×10^{-4}	3.5×10^{-4}	1.2×10^{-3}	2.9×10^{-3}
difference between k simulated & calculated	%	-	77	73	81
ε at plate channel inlet (simulated)	m^2/s^3	-	3.5×10^{-3}	1.8×10^{-2}	1.0×10^{-1}
ε at plate channel inlet (calculated)	m^2/s^3	2.5×10^{-4}	1.4×10^{-3}	8.7×10^{-3}	3.2×10^{-2}
difference between ε simulated & calculated	%	-	60	52	68

Table 6.2: Summary of inlet conditions and other relevant information for simulating flow and temperature distributions in flat plate channels.

To obtain more reasonable input values of k and ϵ for flat plate channel simulations, the inlet section to the PHE was itself simulated using the k - ϵ model under isothermal, incompressible and steady state conditions. A diagram of the full model outline of this inlet can be seen in Figure 6.20. With reference to Figure 3.2, this inlet section is between the solution flow control valve and the inlet port of the solution plate channel. This inlet section consists of the following:

- (a) tubing connecting flow control valve to the expanded inlet,
- (b) this expansion where inlet temperature and pressure are measured,
- (c) gasket on the frame plate of the exchanger,
- (d) flat plate with no gasket,
- (e) gasket of water channel, and
- (f) flat plate of water channel.

It is worth mentioning that together (d), (e) and (f) form part of the hot water channel adjacent to the solution channel. The predicted values of k and ϵ given in Table 6.2 are their respective average values at the exit of the simulated inlet section. As can be seen, these values are larger than those calculated using equations given in CFX manuals (AEA Technology, 1995) by as much as 81%.

Figure 6.20: Full model outline of inlet section to flat plate channel.

6.3.5 Double Backward-Facing Step

Flow through a double backward-facing step geometry as the one shown in Figure 6.21, was simulated as a highly simplified case of flow through a diagonal flat plate channel. The flow enters through the bottom of the channel and exits at the top. The

Figure 6.21:
Geometry of double
backward-facing step.

dimensions of this geometry was acquired with modifications from the actual Alfa Laval M3 plates. Water was used as the fluid with relevant boundary conditions from Table 6.2. As the inlet cross-sectional area of the simplified plate channel (presented in the next section), is identical to that for the double backward-facing step geometry, their inlet flow velocities are the same.

Grid independence was confirmed for both the 2-D and 3-D simulations, with 5800 and 30000 cells, respectively. For the 3-D grid, the distance between the two plate walls is subdivided in 5 cells. Isothermal, incompressible, turbulent flow at steady state was simulated using the k - ϵ model. For comparison with available experimental data of the present study, simulations were performed for a nominal flow velocity of 35 cm/s in the plate channel, and at a temperature of 45°C, refer Table 6.2.

For the 2-D simulations, it was found that the results obtained were not sensitive to variations in the input values of k , ϵ and outlet pressure. Flow distributions for a 2-D and 3-D double backward-facing steps are illustrated in Figures 6.22 and 6.23 respectively. The vectors displayed pertain to the middle plane of the channel where z , the perpendicular distance between the two plates is 0.0012 m. In each of these figures, three recirculation zones can be observed. The first is in the corner next to the inlet, the second in the corner adjacent to the outlet, and the third is at the outlet. These three recirculation zones diminish in size as the dimension is changed from 2-D to 3-D. This is reasonable as in the 3-D case, more drag is imposed on the flow from the two plates forming the channel, thus restricting the size of the flow recirculation zones.

Figures 6.22 and 6.23: Flow distribution in 2-D and 3-D double backward-facing steps respectively.

6.3.6 Simplified 3-D Geometry of a Flat Plate Heat Exchanger

Simulations were performed for a simplified geometry of a flat PHE, whose dimensions are comparable with those of the Alfa Laval M3 plates. For the sake of simplicity, the circular inlet and outlet ports of the actual plates have been approximated by flat ends. Bansal (1994) too, had simulated flow through such a simplified geometry, see Figures 6.2 and 6.3, and hence results from this set of simulations are compared with his.

Figure 6.24: Computational grid of simplified flat channel.

Figure 6.24 shows the computational grid used for flow through a simplified flat channel. This grid was generated using body-fitted co-ordinates, and comprised $80 \times 20 \times 24$ cells. This is a 3-D grid while that simulated by Bansal (1994) is a 2-D one with 54×19 cells. The overall length, width and gap of the channel are 0.358, 0.100 and 0.0024 m respectively. Simulations for laminar, incompressible and steady-state flow were performed under both isothermal and heat transfer conditions. On the other hand, simulations for turbulent, incompressible and steady-state flow were only performed for the isothermal case. Grid independence was verified for the turbulent case only, with a finer mesh of $150 \times 50 \times 5$ cells.

6.3.6.1 Isothermal Laminar Flow

Shown in Figure 6.25 are the flow vectors in the simplified flat channel under isothermal condition. Water enters the channel from the left at a velocity of 0.475 m/s and a temperature of 51.5°C . This inlet velocity corresponds to a nominal velocity of 19 cm/s in the plate channel. The simulated pressure drop is about 190 Pa.

Flow recirculation can be observed for the current simulation in the corner adjacent to the inlet. Since the grid used by Bansal (1994) is 2-D, it should give a larger recirculation zone than the present 3-D grid, as discussed previously in Section 6.3.5. However, as can be seen from Figure 6.2, Bansal's results appear not to have zones of recirculation. This may be due to the finer grid used in the current work when compared with that of Bansal.

Figure 6.25: Laminar flow vectors in a simplified 3-D flat channel under isothermal condition.

6.3.6.2 Laminar Flow with Heat Transfer

The inlet flow conditions for the heat transfer runs are similar to those used for the isothermal runs. Instead of adiabatic channel walls in the latter runs, the heat transfer runs have walls maintained at an elevated temperature of 80°C . Nevertheless, the resultant flow patterns obtained from both type of runs are similar.

However, an interesting observation of the temperature distribution between the z planes was made, where z is the perpendicular direction between the two plates. Since the channel gap is 0.0024 m , the middle plane between the two plate walls is $z = 0.0012 \text{ m}$. Away from the ports, the flow patterns should be symmetric about plane $z = 0.0012 \text{ m}$. Therefore, plane $z = 0.0006 \text{ m}$ is equivalent to plane $z = 0.0018 \text{ m}$, and plane $z = 0.0003 \text{ m}$ is equivalent to plane $z = 0.0021 \text{ m}$. Figures 6.26, 6.27 and 6.28 show the shaded contours for temperature at planes $z = 0.0012$, 0.0006 , and 0.0003 m respectively.

$Z = 0.0012 \text{ m}$

Figure 6.26

$Z = 0.0006 \text{ m}$

Figure 6.27

$Z = 0.0003 \text{ m}$

Figure 6.28

Temperature [K]

Figure 6.26: Shaded temperature contours at $Z = 0.0012 \text{ m}$ for laminar flow.

Figure 6.27: Shaded temperature contours at $Z = 0.0006 \text{ m}$ for laminar flow.

Figure 6.28: Shaded temperature contours at $Z = 0.0003 \text{ m}$ for laminar flow.

With the exception of the flow recirculation corner, the temperature profiles are quite uniform along the plane $z = 0.0012$ m. Flow is slow in this corner, and hence gets heated up more than the rest of the flow. Interestingly, a cold region is observed along the corner adjacent to the outlet at $z = 0.0006$ m. This cold region is more pronounced at $z = 0.0003$ m. The fluid immediately adjacent to the hot walls, has higher temperatures when compared with the fluid near the middle of the channel. This is confirmed in Figure 6.29 which shows the speed and temperature profile in the z direction directly in the middle of channel with respect to its length and width. This figure also verifies that speed is slowest near the wall and fastest in the middle of the channel. Therefore, a probable explanation for the cold region near the plate outlet, is that fluid in different z planes travels with little mixing until it reaches the corner near the outlet, where it is mixed and hence attains a lower average temperature than that of the surrounding fluid. However, lower temperatures should cause less deposition to occur in this area, which contradicts with the experimental findings of this dissertation. Nevertheless, these simulations are for a simplified geometry of the actual plate channel used experimentally. Consequently, such a cold region may not appear near the plate outlet for a more realistic geometry of the actual channel.

Figure 6.29: Speed and temperature profiles along the z direction in the middle of the channel.

6.3.6.3 Isothermal Turbulent Flow

Illustrated in Figure 6.30 are the flow vectors in the simplified flat channel for isothermal turbulent flow. Water enters the channel from the left at a temperature of 45°C. The nominal plate channel flow velocity is 35 cm/s, and the simulated pressure drop is approximately 590 Pa. This simulation was performed using the same inlet and outlet conditions as those for the 3-D double backward-facing step run in Section 6.3.5.

Figure 6.30: Turbulent flow vectors in a simplified 3-D flat channel under isothermal condition.

Compared with the flow field of the double backward-facing step in Figure 6.23, this simplified channel has only one recirculation zone which can be found in the corner adjacent to its inlet. Due to the smoothness of this simplified geometry, its recirculation zone is smaller than the main recirculating region in the double backward-facing step.

6.3.7 True Geometry of a Flat Plate Heat Exchanger

The true geometry of the flat PHE used in the experimental investigations of this dissertation (refer Sections 4.9 to 4.14) was modelled. As previously mentioned, the flat plates used have a geometry comparable with that of the Alfa Laval M3 plates. Isothermal and heat transfer simulations were performed for turbulent flow. In addition, simulations were also performed for isothermal laminar flow. The different flow models are compared with each other, and with flow visualisation experiments using ABS particles in Section 4.13. In addition, comparisons are also made with fouling patterns from relevant experiments. This is because fouling occurs in regions where low velocity and high temperature prevail.

6.3.7.1 The Grid

Figure 6.31

Figure 6.32

Figure 6.31:
True geometry of
a flat plate heat exchanger
showing multi-block construction of grid
and positions of inlet and outlet.

Figure 6.32:
Grid of $150 \times 60 \times 5$ cells for
true geometry of
a flat plate heat exchanger.

The geometry is made up of 34 blocks, refer Figures 6.31 and 6.44. Fluid enters at the bottom port hole and leaves through the top port hole. x , y and z are directions from bottom to top, left to right, and into the page respectively. The port holes are at $z =$

0.0000 m, while the flat plate with no port holes is at $z = 0.0024$ m. All simulations were performed using a 3-D, body-fitted grid with Cartesian co-ordinates. Using the k - ϵ model, grid independence was varied under isothermal, incompressible and steady state conditions, and was achieved with $150 \times 60 \times 5$ cells in the $x \times y \times z$ directions for the flows considered. This grid is shown in Figure 6.32.

Laminar flow, and turbulent flow using the k - ϵ model and the Differential Reynolds Stress Model (DSM) were simulated for this geometry. However, convergence was achieved with the above grid size only for the laminar flow and turbulent flow using the k - ϵ model. Because of the complexity of the channel geometry with fluid entering at a high speed through the round port hole and hitting the back plate, a significant amount of movement in the flow is generated. As a result, the mesh is not sufficient to model the flow accurately with only 5 cells in the z -direction. Consequently, 10 cells in this direction were used in order to reach convergence for turbulent flow using DSM and for flow through geometries with different distributors, which is discussed in Section 6.3.8. In the x - y plane as shown in Figure 6.32, the grid with $150 \times 60 \times 10$ cells appears identical to the one with $150 \times 60 \times 5$ cells.

With 5 uniform cells in the z -direction, almost all the y^+ values for turbulent flows were maintained to be above 11. To increase this number of cells, and at the same time maintaining valid y^+ values, the cell distribution had to take the form of a symmetrical geometric progression. For 10 cells, an expansion factor of 1.6 was used in the distribution. The distributions of 5 and 10 cells in the z -direction are shown in Figure 6.33.

Figure 6.33: Distribution of cells in z -direction for plate channel.

6.3.7.2 Isothermal Flow

Both grids of $150 \times 60 \times 5$ and $150 \times 60 \times 10$ cells were used to simulate the flow under isothermal conditions. Water was used as the fluid, and its properties were obtained from the CFX database. The inlet conditions from Table 6.2 for 35 cm/s (or 0.35 m/s) and 100 cm/s (or 1.00 m/s) nominal flow velocities in the plate channel were used, with an average temperature of 45°C. The Reynolds number for these two velocities are 2800 and 8000 respectively.

6.3.7.2.1 Turbulent Flow and the k - ϵ Model

Simulations were performed for an incompressible flow under steady-state conditions. For 35 cm/s, it was found that the results did not vary significantly due to changes in the input values for k , ϵ and outlet pressure. Figures 6.34(a) and (b) show the resulting flow distributions for the two aforementioned grid sizes and a flow velocity of 35 cm/s. Two recirculation regions were observed, one at the corner adjacent to the inlet port and the other at the outlet port. Although flow at the corner adjacent to the outlet port is slow, it is not recirculating. The only difference between the flow distributions for the two grid sizes is that, the recirculation region at the outlet port of the larger grid is smaller. The corresponding pressure drops across the plate channel are 1.3×10^3 Pa and 1.2×10^3 Pa for $150 \times 60 \times 5$ and $150 \times 60 \times 10$ grid sizes respectively. Both are within 15% of the pressure drop measured experimentally which is 1.4×10^3 Pa, refer Table 6.2.

The flow distribution for a nominal channel flow velocity of 100 cm/s is shown in Figure 6.35. As can be seen, the recirculation region adjacent to the inlet port is somewhat smaller than those shown in Figures 6.34(a) and (b) for 35 cm/s. Moreover, the reattachment lengths for 100 and 35 cm/s are 3.4 and 4.6 cm respectively. These lengths are both shorter than their corresponding experimental values, refer Section 4.13.1. However, this is not unreasonable since both the k - ϵ model and DSM underpredicted the reattachment point for the backward-facing step in Section 6.3.3.

The simulated pressure drop across the plate channel is 9.6×10^3 Pa, compared with that of 9.1×10^3 Pa measured experimentally, refer Table 6.2. As the simulated pressure drop is only about 5% higher than its measured value, it can be said that the k - ϵ model predicts the pressure drop across the plate channel with reasonable accuracy.

Figures 6.34(a) and (b): Flow distributions obtained using the k - ϵ model for 35 cm/s nominal channel flow velocity and grids of (a) $150 \times 60 \times 5$ cells and (b) $150 \times 60 \times 10$ cells respectively.

Figure 6.35:
Flow distribution
obtained using the k - ϵ model
for 100 cm/s nominal channel
flow velocity and a grid of
150×60×5 cells.

6.3.7.2.2 Turbulent Flow and Differential Reynolds Stress Model

With DSM, a fully converged solution could not be obtained under steady state conditions using the 150×60×5-cell grid. Increasing the number of cells in the z direction resulted in the steady-state calculation being neither converging nor diverging. As steady-state solutions were achieved using the k - ϵ model, they should exist for the runs with DSM. However, a steady-state calculation may be considered as a transient calculation run for an infinite amount of time. If a flow simulation, from its initial guess to its steady-state solution is such that the flow approaches the steady-state solution in a very complex manner, then it is possible that an attempt to reach the steady-state solution in a single huge time step will overshoot the mark and never recover (AEA Technology, 1995). Consequently, the cells in the z direction were increased to 10 which gave a grid of 150×60×10 cells, and simulations were performed under transient conditions.

Figure 6.36: Variation of speed along x axis for transient runs using DSM
($y = 0.01$ m; $z = 0.0012$ m).

Figure 6.37: Variation of speed along y axis for transient runs using DSM
($x = 0.08$ m; $z = 0.0012$ m).

Figure 6.38: Variation of speed along z axis for transient runs using DSM ($x = 0.08$ m; $y = 0.01$ m).

The DSM runs were completed for a total of 5 seconds in real time. These runs were stopped at various time intervals in order to verify if steady state had been attained, and then restarted. This verification involved examining specific variables along a number of line sections. As can be seen in Figures 6.36, 6.37, and 6.38, for each line section parallel to the x , y and z axes respectively, the variable profiles tend to merge as a steady state is attained.

The flow pattern at time = 5 seconds is shown in Figure 6.39. As can be seen, the recirculating region and the low flow velocity region in the corners adjacent to the inlet and outlet ports respectively, are larger than those obtained using the $k-\epsilon$ model. These two regions have almost the same lengths in the main flow direction (along the x axis) for both DSM and the $k-\epsilon$ model. However, the former model predicted larger widths in the transverse flow direction (along the y axis) for these two regions. Although DSM is able to predict the recirculation region near the inlet port with closer resemblance to that observed experimentally, it still underpredicts the corresponding point of reattachment. In addition, the simulated pressure drop across the channel was about 1.0×10^3 Pa which is approximately 29% lower than that measured experimentally. Therefore, in terms of pressure drop, the $k-\epsilon$ model is more accurate than DSM.

Figure 6.39:
Flow distribution
obtained using DSM
for 35 cm/s nominal
channel flow velocity and
a grid of 150×60×10 cells
at time = 5 seconds.

6.3.7.2.3 Laminar Flow

Simulations were also performed to investigate the laminar flow distribution. Figure 6.40 shows the laminar flow pattern for a nominal channel flow velocity of 35 cm/s. Although the low-velocity zone adjacent to the inlet port is larger when compared with those obtained using the k - ϵ model and DSM, the actual laminar flow region that is recirculating, is very small. Furthermore, the simulated pressure drop across the channel is approximately 2.8×10^3 Pa, which is about 100% higher than its experimental counterpart. It is therefore apparent that, the laminar flow model grossly overpredicts the pressure drop for this geometry.

Figure 6.40:

Flow distribution obtained using the laminar flow model for 35 cm/s nominal channel flow velocity and a grid of $150 \times 60 \times 10$ cells.

6.3.7.2.4 Comparison of CFD Flow Model Predictions with Experimental Results

A summary of the CFD flow model predictions and experimental results is given in Table 6.3. With reference to Section 4.13.1, the experimental reattachment lengths (R.L.) were obtained from ABS flow patterns. Due to flow fluctuations, these reattachment lengths varied within $\pm 0.005 \text{ m}$. On the other hand, reattachment lengths obtained via simulations are consistent to within $\pm 0.001 \text{ m}$. As can be seen, all the flow models underpredict the reattachment length. The $k-\epsilon$ model gives the best prediction in terms of reattachment length and pressure drop. Although DSM is able to predict a larger area of recirculation in the corner adjacent to the inlet port, its required CPU time is about two orders of magnitude higher than those of the other flow models used. In other words, the DSM run took approximately 35 days to complete, while a typical run using any of the other models required only about a day to complete.

Flow Model	Steady State or Transient	Velocity [m/s]	Grid	ΔP [Pa]	R. L. [m]	CPU Time [s]	[day]
Experiment	-	0.35	-	1.4×10^3	0.10	-	-
Experiment	-	1.00	-	9.1×10^3	0.11	-	-
k - ϵ	Steady State	0.35	$150 \times 60 \times 5$	1.3×10^3	0.046	3.011×10^4	0.3
k - ϵ	Steady State	0.35	$150 \times 60 \times 10$	1.2×10^3	0.046	4.564×10^4	0.5
k - ϵ	Steady State	1.00	$150 \times 60 \times 5$	9.6×10^3	0.034	1.658×10^4	0.2
DSM	Transient (t=5s)	0.35	$150 \times 60 \times 10$	1.0×10^3	0.040	3.049×10^6	35.3
Laminar	Steady State	0.35	$150 \times 60 \times 10$	2.8×10^3	0.010	5.758×10^4	0.7

Table 6.3.: Summary of CFD flow model predictions and experimental results.

The similarities and/or differences between the three flow models used are further displayed using line sections parallel to the x , y and z axes in Figures 6.41, 6.42 and 6.43 respectively, for a grid of $150 \times 60 \times 10$ cells. As expected, the predictions of the k - ϵ model and DSM are more compatible with each other, than with those pertaining to the laminar flow model. Due to the enormous CPU time required by DSM, the k - ϵ model was used in all heat transfer simulations with and without distributor designs.

Figure 6.41: Variation of speed along x axis for different flow models with a grid of $150 \times 60 \times 10$ cells ($y = 0.01$ m; $z = 0.0012$ m).

Figure 6.42: Variation of speed along y axis for different flow models with a grid of $150 \times 60 \times 10$ cells ($x = 0.08$ m; $z = 0.0012$ m).

Figure 6.43: Variation of speed along z axis for different flow models with a grid of $150 \times 60 \times 10$ cells ($x = 0.08$ m; $y = 0.01$ m).

6.3.7.3 *Turbulent Flow with Heat Transfer*

In actual PHEs, the temperature distribution of a plate wall is by no means uniform. However, in order to simplify the heat transfer simulations, parts of the channel walls were specified at higher constant temperature than the incoming fluid. This is discussed in greater detail in Sections 6.3.7.3.1 and 6.3.7.3.2. Since the fluid outlet temperature is known from experimental results in Chapter 4, the wall temperature was raised by trial and error until the simulated outlet temperature reaches that achieved experimentally. Due the design of a PHE, a typical plate channel is not heated or cooled via all its bounding walls. For instance, the gasket in contact with the fluid does not heat or cool the fluid. Therefore, two sets of simulations, namely heat transfer area #1 and #2 were performed, with the less heated wall area pertaining to the former. The grids used are of $150 \times 60 \times 10$ cells.

6.3.7.3.1 Heat Transfer Area #1

Simulations with heat transfer were performed for a nominal flow velocity of 35 cm/s in the plate channel, with fluid entering and leaving at 41.5 and 47.5°C respectively. These conditions are similar to those obtained experimentally, see Table 6.2. A wall temperature of 70°C was specified for the shaded area shown in Figure 6.44.

For this set of heat transfer simulations, similar flow and pressure drop profiles to those of the corresponding isothermal case were obtained. Temperature distribution for $z = 0.0012$ m is shown in Figure 6.45. As a result of their prevailing low flow velocities, the two corners adjacent to the inlet and outlet ports have higher temperatures. A gradual temperature increase from the centre of the plate at $z = 0.0012$ m to either sides of the wall was predicted by the simulations. The difference between this temperature trend and that observed in the simplified plate geometry of Section 6.3.6 may be attributed to the different flow models used (k - ϵ model and laminar respectively), and the different inlet and outlet geometries.

Figure 6.44

Figure 6.45

Figure 6.44: Heat transfer area #1.

Figure 6.45: Temperature distribution at $z = 0.0012$ m for heat transfer area #1 obtained using the k - ϵ model.

6.3.7.3.2 Heat Transfer Area #2

Incorporation of the distributors in the next section requires some modifications to the heated wall area, leading to heat transfer area #2 as shown in Figure 6.46. With an increase in the heated area than heat transfer area #1, a lower wall temperature of 65°C was imposed. Once again, for this set of heat transfer simulations, the flow and pressure drop profiles obtained are similar to those of the corresponding isothermal case. The temperature distribution for $z = 0.0012$ m shown in Figure 6.47, is very similar to that for heat transfer area #1.

Figure 6.46

Figure 6.47

Figure 6.46: Heat transfer area #2.

Figure 6.47: Temperature distribution at $z = 0.0012$ m for heat transfer area #2 obtained using the k - ϵ model.

6.3.7.3.3 Comparisons with Experiments

Comparing the flow and temperature distributions simulated using the k - ϵ model in Figures 6.34(a) and 6.47 respectively, with the fouling pattern obtained under similar conditions in Figures 4.49(a) and (b), it is apparent that fouling occurs in regions where the prevailing flow velocity is low and where high temperature prevails. In addition, the recirculating, slow and fast flow zones in the plate inlet region observed using ABS particles, refer Figure 4.54(c), are also evident in the simulated flow patterns. The fouling profiles near the inlet port as shown in Figure 4.49(c), correspond well to these CFD predicted regions of low flow velocity.

6.3.8 Distributor Designs in Plate Heat Exchangers

The aim of having different distributor designs on heat exchanger plates is to eliminate the presence of low-flow-velocity regions, especially regions of recirculating flows, and hence promote a more even flow distribution in the plate channels. This is an attempt to increase heat transfer, without having to pay too high a penalty on the corresponding pressure drop.

In all the simulations performed for different plate distributor designs, the $150 \times 60 \times 10$ -cell grid and the k - ϵ model were used with a nominal channel flow velocity of 35 cm/s. The different geometries of the distributors were created in the command files by blocking certain cells. Designs A to E have been investigated experimentally, refer Section 4.14. The predicted flow and temperature distributions for these designs can therefore be compared with their respective fouling patterns obtained experimentally. In addition, the experimental pressure drops across the channel obtained at 20°C, are also compared with those simulated at an average temperature of 45°C in Table 6.4. As can be seen, all the pressure drops were underpredicted by CFD by as much as 70%. This is because unlike the experimental measurements, the CFD results do not include port pressure losses. However, the CFD results show qualitatively which distributor designs would give the highest and lowest pressure drops.

<i>Distributor Design:</i>	<i>Pressure Drop from:</i>		
	Experiment [Pa]	CFX [Pa]	Difference [%]
none	1.4×10^3	1.2×10^3	14
A	15.5×10^3	4.7×10^3	70
B	4.4×10^3	2.3×10^3	48
C	3.1×10^3	1.6×10^3	48
D	2.1×10^3	1.5×10^3	29
E	1.6×10^3	1.4×10^3	13
F	-	1.6×10^3	-
G	-	3.4×10^3	-
H	-	1.2×10^3	-

Table 6.4:
Comparison of
experimental and
predicted pressure
drops for different
distributor designs.

For this set of simulations involving plate distributors, heat transfer area #2 shown in Figure 6.46, and the walls of all the distributors were maintained at an elevated temperature of 65°C. This specification was adopted to maintain simplicity in the simulations, even though the distributors behave adiabatically in the experimental runs. However, the area of the distributor walls in contact with the fluid is reasonably small, such that their effects should be negligible when compared with those of the heated channel walls.

Figures 6.48 to 6.55 show the flow and temperature distributions of distributor designs A to H respectively. These figures all have identical keys for speed and temperature, which range from 0.00 to 1.29 m/s, and from 314.5 K (41.5°C) to 320.5 K (47.5°C) respectively. Designs A to D are made up of blockages in the distributor sections only, while designs E and F comprise blockages in the heat transfer area only. Design G has blockages in both the distributor and main heat transfer areas. Conversely, design H is a modified flat plate geometry without any blockages.

As previously mentioned in Section 4.14, compared with the bare plates without any distributors, more fouling occurred on all the plates with distributors. The simulated flow distributions for plate distributor designs A to D confirm that, in attempting to increase heat transfer by eliminating the recirculating and/or low-velocity regions in the corners adjacent to the inlet and outlet ports, these distributors create additional regions of flow recirculation and low flow velocity. These additional undesirable flow regions are formed downstream of all the distributors, especially the longer ones. This verifies the disadvantage of having long distributors instead of shorter ones, as previously discussed in Section 4.14.

Unlike the flow in commercial corrugated plate channels, in designs A to D where flow is distributed using blockages in the distributor sections only, there is nothing to break up the low-flow-velocity zones created in the main heat transfer section of the channels. Even for design E which has blockages in the main heat transfer section of the plate, additional regions of low flow velocity still occur in the vicinity of the blockages.

Figures 6.48(a), (b), (c) and (d): Flow and temperature distributions for distributor design A.

(a) Flow

(b) Temperature

Figures 6.49(a) and (b): Flow and temperature distributions for distributor design B.

(a) Flow

(b) Temperature

Figures 6.50(a) and (b): Flow and temperature distributions for distributor design C.

(a) Flow

(b) Temperature

Figures 6.51(a) and (b): Flow and temperature distributions for distributor design D.

(a) Flow

(b) Temperature

Figures 6.52(a) and (b): Flow and temperature distributions for distributor design E.

(a) Flow

(b) Temperature

Figures 6.53(a) and (b): Flow and temperature distributions for distributor design F.

(a) Flow

(b) Temperature

Figures 6.54(a) and (b): Flow and temperature distributions for distributor design G.

(a) Flow

(b) Temperature

Figures 6.55(a) and (b): Flow and temperature distributions for distributor design H.

Fluid heats up more in regions of low flow velocity, as previously mentioned. Therefore, in the simulations performed, regions of low flow velocity coincide with regions of high temperature as can be seen from Figures 6.48 to 6.55. It is worth mentioning that these temperature distributions are more exaggerated than those in actual PHEs. This is because for the simulations, the walls are maintained at a constant elevated temperature. On the other hand, as the hot fluid flows counter-current to the cold fluid in PHE experiments, instead of being constant, the prevailing wall temperature in the cold channels should gradually decrease from the outlet port to the inlet port. Nevertheless, the predicted regions of low flow velocity and high temperature for designs A to E, agree reasonably well with the corresponding deposit locations shown in Section 4.14.

As shown in Figure 6.53, design F is made up of 5 columns of square blockages arranged in such a way that they formed a line of 60° to the main flow direction, similar to the corrugation angle of the Alfa Laval plates investigated, refer Table 3.1. Although successful in reducing the original regions of low flow velocity, this design introduces additional regions favourable for fouling to occur as well. The same applies to design G which has blockages not only in the main heat transfer plate area, but also in the distributor section. However, with more blockages in the plate channel, design G has a higher simulated pressure drop than design F.

Interestingly, the most even distribution of flow occurs for the plate design without any distributors, as shown in Figure 6.34(b). This plate geometry can be further improved by design H, where regions of low flow velocity, i.e. the two corners opposite the inlet and outlet ports have been blocked off. It is because the Alfa Laval plates have such a deep concave curve at these corners that recirculating and slow flows are induced. As fouling patterns and flow patterns in flat plate channels are asymmetric, design H also has an asymmetric plate geometry. Although no recirculation is observed in the plate corners for design H, see Figure 6.55, the surrounding flow velocity is slow. Thus, design H can be further modified by blocking the plate corners even more, making the plates more streamline. In practice, heat exchanger plates and gaskets can be redesigned to block off flow in the two corners mentioned above.

It is apparent from the simulations performed that the plate geometry together with any internal blockages in the distributor and/or main heat transfer sections, have profound effects on the resulting flow and temperature distributions. These distributions in turn affect the fouling behaviour in the plate channel.

7. CONCLUSIONS & RECOMMENDATIONS FOR FUTURE WORK

7.1 Conclusions

Under clean conditions, commercial Alfa Laval M3 diagonal and side flow plates exhibit similar heat transfer characteristics. This contradicts the theory that diagonal flow plates should have higher heat transfer than their side flow counterparts. In addition, both the pressure drop and the corresponding friction factor are slightly lower for diagonal flow plates than side flow ones. These trends could be due to the different flow patterns in their respective plate channels. In conjunction with a corrugated polycarbonate plate, visual observation of flow patterns in these channels were performed with a suspension of Acronitrile Butadiene Styrene (ABS) particles in water at room temperature. For both the diagonal and side flow set-ups, the flow velocity was observed to be fastest in the main flow region between their inlet and outlet ports, and slowest in the plate corners adjacent to these ports. With different flow paths, M3 diagonal and side flow channels have different local flow distributions, which contribute to their unique heat transfer and pressure drop characteristics.

Investigation of calcium sulphate (CaSO_4) precipitation fouling was carried out with Alfa Laval P01 and M3 plates, and custom-made flat plates with good repeatability of data. Effects of solution velocity, temperature, concentration, and port position were investigated using P01 and M3 diagonal flow plates. The fouling rate was found to increase with decreasing solution flow velocity, and increasing solution temperature and initial solution concentration. The rate of deposition is therefore, likely to be reaction-controlled rather than diffusion-controlled. These trends are similar to those previously published by Bansal (1994), who additionally found solution concentration to be important only in the initial stages of fouling. However, in the present investigation, fouling was observed to be strongly influenced by solution concentration top-ups during the course of an experiment. Furthermore, the time and amount of top-up also affect the resulting fouling behaviour. The topping-up times are marked by a sharper increase in pressure drop, but no such corresponding increase in fouling

resistance was registered. This is because the pressure drop increase was most likely caused by deposit formation near the outlet region of the plate which does not contribute much to heat transfer. In addition, M3 diagonal flow plates display slightly lower fouling resistances, and pressure drop values than their side flow counterparts. A similar overall change in pressure drop is however, observed for both plate types.

To enhance the understanding of deposit growth, visual observation of fouling pattern development was performed using M3 diagonal and side flow plates in conjunction with a corrugated polycarbonate plate. For both plate types, deposition first occurs in the plate corner adjacent to the outlet port, and an asymmetric fouling pattern was eventually formed. Hence, it was confirmed that high temperature and low flow velocity are the two criteria necessary for deposition to occur. When only one of the two is present, deposition may still occur but its rate would be very much dependent on the local conditions, such as surface roughness, surface material and degree of supersaturation. Furthermore, fouling was observed to commence at the contact points between adjacent plates. X-ray diffraction (XRD) analysis confirmed that gypsum (calcium sulphate dihydrate, $\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$) was the sole crystallising deposit. Despite inconsistent trends between set-ups with and without the polycarbonate plate, the former set-up reveals fouling patterns in M3 plate heat exchangers, and hence provides new insights into the effects of flow distribution on fouling. Results from the polycarbonate plate experiments were confirmed with a series of fouling experiments performed with the normal set-up, which were terminated prematurely to examine the fouling pattern at different run-times.

Several methods to mitigate fouling were investigated. Successful mitigation methods involved the addition of sodium nitrate (NaNO_3) into the process solution, solution velocity surges, and air flushes of the solution flow. Although the addition of NaNO_3 into CaSO_4 solution reduced fouling significantly, the amount of NaNO_3 used (30.0 g/l) was ten times more than the initial concentration of CaSO_4 (3.0 g/l). It can therefore be inferred that this method may neither be a cost effective nor an environmentally friendly way of mitigating fouling. As a method of removing pre-existing deposits on untreated stainless steel surfaces, velocity surge seems to be more effective in the presence of

more abundant deposits. Air flush or the introduction of a high air flow into the plate heat exchanger solution flow proved to be capable of removing a pre-existing fouling layer, and hence can be considered as an option for mitigating fouling. Conversely, continuous air injection into the solution flow, ion-sputtered plates and magnetic devices failed to reduce fouling. Continuous air injection promoted more deposition than that of single phase flow, such that even the fouling deposit patterns obtained were different. This may be a result of complex interactions between two phase flow and fouling. There seemed to be no apparent difference in the fouling behaviour between the untreated stainless steel plates and plates which are ion-sputtered with diamond-like-carbon (DLC), diamond-like-carbon and a silicon polymer (DLC & HMDSO), diamond-like-carbon and C_2F_2 (DLC & C_2F_2), and tetrahedral-amorphous carbon (ta-C). It is rather disappointing that not only ion-sputtering has not proved itself capable of mitigating fouling in plate heat exchangers, but also flushing of deposits attached to such plates has not been successful in terms of fouling reduction. Furthermore, it is inconclusive whether ta-C sputtered plates and a magnetic device could reduce the formation of $CaSO_4$ fouling in plate heat exchangers.

In order to simplify the analysis of the relationship between plate design and fouling, experiments were also carried out using 2-mm-thick, custom-made flat plates. As there are no contact points to maintain the plate channel gaps, non-uniform inter-channel gaps can occur in a flat plate heat exchanger. This has been substantiated by heat transfer and pressure drop analyses under clean conditions. Moreover, in the absence of contact points, thicker flat plates were required to counterbalance their lack of mechanical stability. This and the absence of corrugations which results in less mixing and lower turbulence level, cause a lower thermal efficiency to prevail for the flat plate heat exchanger when compared to that of a corrugated one. For a nominal solution flow velocity of 35 cm/s, corrugations bring about a heat transfer enhancement of about 5 times higher in the Alfa Laval M3 plates than that of the flat plates. Conversely, for the same solution flow velocity, the pressure drop for the Alfa Laval M3 plates is approximately 8 times higher than that of the flat plates.

Experimentation with flat plates revealed increased fouling with increase surface roughness and solution concentration. The latter trend is in accordance with that of the Alfa Laval plates. Unlike the unsanded flat plates which exhibit a clear roughness delay time, the sanded plates which are smoother, have negative fouling resistances throughout the entire duration of the experiments. It therefore can be inferred that for the period the flat plates were allowed to foul, the increased turbulence brought about by the deposits penetrating into the laminar sublayer, outweighs their additional thermal resistance.

Figures 7.1(a), (b) and (c): (a) ABS particle flow pattern, (b) CaSO_4 fouling pattern, (c) CFD predicted flow distribution, for 35 cm/s solution velocity.

Experimentation using flat diagonal flow plates and a flat polycarbonate plate revealed similar flow and fouling patterns to those of the Alfa Laval plates. Figures 7.1(a), (b) and (c) serve to compare the flow pattern of the ABS particles obtained at room temperature for a solution flow velocity of 35 cm/s, with its corresponding CaSO_4 fouling pattern and flow distribution with heat transfer predicted using CFD. In these figures, the flow enters and leaves via the lower and upper ports respectively. Figure

7.1(a) shows a large zone of low flow velocity in the lower plate corner, and a smaller one in the upper corner. Since the upper part of the plate is hotter than the rest of the plate, deposition starts at the upper plate corner before proceeding to occur at the lower plate corner. This further confirms that low flow velocity and high temperature are the two criteria necessary for fouling to occur. In addition, the fouling deposits formed a wavy front at the plate inlet region, where local areas with less fouling correspond to areas where the prevailing local flow velocity is higher, and vice versa. Furthermore, with a constant volumetric flow rate, fouling was observed to decrease with decreasing plate channel gap. This is because as the channel gap decreases, the corresponding flow velocity increases. Thus, it can be concluded that the prevailing flow velocity has a profound influence on the resulting fouling behaviour.

Five distributor designs, namely designs A to E were examined experimentally. Designs A to D have distributors in the flat plate inlet and outlet regions only. In order to prevent the occurrence of fouling, these distributors aim at directing flow to the low-flow-velocity zones in the plate corners adjacent to the ports. However, additional regions of low flow velocity are created in the wake of these distributors, and with nothing to break up these unfavourable regions, more deposition occurred on these plates when compared to (bare) flat plates without any distributors. Furthermore, larger plumes of deposits tend to form in the wake of longer distributors, thus making them inferior to shorter ones. As fouling develops, the fluid flow characteristics in a plate channel may alter completely, for instance, deposit growth can reduce the effective width of a plate, thus reducing the cross-sectional flow area. In addition, the presence of the distributors also raises the clean pressure drop of the channel. As inlet distributors distribute inflowing fluid across the plate channel, while outlet distributors only 'collect' flow into the outlet port, the former are more influential on the flow distribution, and hence the resulting fouling pattern. Therefore, in order to avoid the creation of large low-flow velocity zones and high pressure drops, extra care should be taken in designing these distributors. Emphasis should not only be given to the designs of the inlet distributors, but should also be given to ways of breaking up any additional low-flow-velocity zones generated. In addition, preference should be given to short distributors instead of longer ones.

Using an approximate fouling model, the fouling behaviour of plate heat exchangers which are 'overdesigned' with additional parallel plates, larger plates of different aspect ratios and additional plates in a second pass, was assessed. For adhesion-controlled fouling, the flow velocity effect is more profound than that of the temperature. As the addition of parallel plates results in a reduced flow velocity in the plate channels for constant flow rates, this traditional method of excess heat transfer area incorporation is unsuitable if adhesion-controlled fouling is expected. In the presence of this fouling mechanism, the excess area is best incorporated by having a two pass arrangement, or by using plates with half the standard plate width. These methods result in higher flow velocities capable of increasing the removal rates of fouling deposits. With a higher flow velocity however, a higher pressure drop and higher pumping cost are also unavoidable.

On the other hand, the prevailing temperature has a dominant effect on the rate of reaction fouling. With the exception of the two pass arrangement and plates with half the standard plate width, the average wall temperatures of all the 'overdesigned' heat exchangers are comparable. Consequently, there is not much difference in their heat transfer performance in the presence of reaction fouling.

As flow characteristics play a major role in fouling in plate heat exchangers, flow simulations have been performed using a commercial CFD package, CFX. For flow through a 2-D channel with a single corrugation on the bottom surface, two areas of flow recirculation were observed. In addition, flow passing through 2-D channels with multiple corrugations on the top and bottom surfaces was also simulated. For plates with a 90° corrugation angle, both the simulated heat transfer and pressure drop increase with decreasing corrugation wavelength. On the other hand, increasing the plate channel gap causes the heat transfer and pressure drop to decrease.

The k - ϵ model and the Differential Reynolds Stress Model (DSM) underpredict the point of reattachment for air and water flowing past a 2-D backward-facing step by 23% and 29% respectively. Although the overall and local flow patterns predicted by DSM bear a closer resemblance to that observed experimentally, the k - ϵ model is able

to capture the expected overall flow pattern successfully in less Central Processing Unit (CPU) time.

In addition to the actual flat plate channel with geometry similar to that of the Alfa Laval M3 plate heat exchanger, two simplified channels, namely a double backward-facing step, and a flat plate channel with flat inlet and outlet, were also simulated. Moreover, simulations were performed to obtain more reasonable input values of k and ϵ for these plate channels. The simulated values are larger than those calculated using equations given in AEA Technology (1995) by approximately 50 to 80%. For a steady, isothermal, incompressible, turbulent flow past a double backward-facing step, the simulated recirculation zones obtained using the k - ϵ model diminish in size as the dimension is changed from 2-D to 3-D. On the other hand, for the other simplified plate geometry, an interesting cold region in the plate corner adjacent to the outlet is observed for laminar flow with heat transfer. This cold region is however, not observed for turbulent flow in the actual flat plate channels.

Fluid flow in the true geometry of a flat plate heat exchanger was simulated under conditions similar to the fouling experiments performed. Steady laminar flow, steady turbulent flow using the k - ϵ model, and transient turbulent flow using DSM were examined. These models all underpredict the reattachment length, with laminar flow giving the most unrealistic results. Although DSM is able to predict a larger area of recirculation in the plate corner adjacent to the inlet port, the k - ϵ model gave a better estimation of the reattachment length and the pressure drop across the channel. Moreover, the required CPU time for DSM under transient conditions, is about two orders of magnitude higher than that for the other flow models. The flow distribution predicted using the k - ϵ model is shown in Figure 7.1(c). As can be seen, the k - ϵ model is capable of predicting the overall flow characteristics with reasonable accuracy and in relatively short CPU time. Thus, CFD can be inferred as a validated tool for predicting the overall flow characteristics in plate heat exchangers.

Eight distributor designs, of which five were examined experimentally, were simulated using the k - ϵ model. The simulated regions of low flow velocity and high temperature

agree well with the locations of fouling deposits obtained in the corresponding experiments. The additional designs simulated further confirm that, in eliminating the original regions of low flow velocity present in the flat plate channels without distributors, blockages in the distributor and/or main heat transfer areas create extra regions of low flow velocity favourable for fouling to occur. Moreover, these blockages also bring about a corresponding increase in pressure drop. It is however, difficult to compare the flow in the main heat transfer area of all these designs with the zigzag fluid motion in commercial chevron plates.

One way of achieving a more even distribution of flow in flat plates with the present Alfa Laval M3 geometry is to redesign the plate geometry such that the plate corners adjacent to the ports are smoother. Simulations show that this redesigned geometry has no recirculating flows in the plate corners, and hence should be less susceptible to fouling.

7.2 Recommendations for Future Work

Based on the encouraging findings mentioned above, several recommendations are put forward for future investigations:

- (i) Precipitation fouling of another salt, e.g. calcium carbonate (CaCO_3) or calcium phosphate ($\text{Ca}_3(\text{PO}_4)_2$), and multicomponent fouling would be most interesting to investigate. The difference between the deposit structure of single and multiple salt precipitation is worth examining. This is because single-salt solutions tend to form hard, tenacious deposits, while solutions containing salt mixtures tend to precipitate in an irregular pattern on the surface. The latter type of deposit is softer and less adherent, moreover its irregular surface promotes erosion by the fluid flowing over it (Garrett-Price *et al.*, 1985). Hence, it might be possible to reduce the overall fouling rate by promoting salt mixtures to precipitate. Even if the resultant softer deposit remains attached to the heat transfer surface, it would be easily removable.

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- (ii) In order to visualise flow pattern and fouling pattern development better in commercial plate heat exchangers, e.g. Alfa Laval M3 plates, the corrugated polycarbonate plate used should have sinusoidal grooves instead of rectangular ones. Moreover, other plate materials with greater transparency are worth investigating. A highly transparent plate with sinusoidal imprints can be used to visualise the zigzag motion of fluid flowing in different plate channel designs more clearly and accurately.
 - (iii) The effect of added chemicals has only been studied for one concentration of NaNO_3 which reduced fouling dramatically. For this fouling mitigation method to be more cost effective and environmentally friendly, it is essential to investigate if lower concentrations of NaNO_3 would have the same effect on the precipitation fouling of CaSO_4 . The addition of different chemicals which have either a salt-in or salt-out effect, or a combination of different chemicals into the process solution would certainly give some interesting results.
 - (iv) Anomalous results were obtained in the presence of two phase flow and fouling. Therefore, more experiments with different air flow rates, and visualisation of two phase flow will be required.
 - (v) Although the ion-sputtered plates investigated have not reduced fouling, other different ion-sputtered plates, especially ta-C may be able to mitigate fouling successfully. In addition, due to the success of ion-implanted heater rods in mitigating fouling (Müller-Steinhagen and Zhao, 1997), experimentation with ion-implanted plates should be pursued. Successful application of ion-sputtering and/or ion-implantation on heat exchanger plates may bring about significant economic advantages.
 - (vi) To enhance the understanding of the relationship between plate design and fouling further, visual observation of flow characteristics and fouling pattern development using custom-made plates with blockages which resemble corrugations in the

commercial plates, in both the plate distributor and main heat transfer areas should be carried out.

- (vii) The present work on CFD needs to be extended to model 3-D flow channels with geometry similar to those of the commercial chevron plates. This is so that the zigzag motion of fluid observed experimentally can be simulated. Once the results for these flow simulations are validated, improvements in the present plate design can be predicted with confidence via CFD simulations. In addition, instead of imposing a fixed wall temperature for the plate channel, heat transfer simulations with a three channel set-up similar to that used experimentally, should give results which better resemble those obtained experimentally. Although these CFD implementations would give more insights into the flow, heat transfer and pressure drop characteristics in a commercial plate heat exchanger, greater computational resources would be required.

APPENDIX A: FURTHER EXPERIMENTAL PROCEDURES

A.1 Preparation of Titration Solutions

The solutions used for the determination of CaSO_4 concentration were prepared as follows:

- (a) *EDTA Solution (0.01 M)*. 7.44 g of disodium dihydrogen ethylene-diamine-tetraacetate ($\text{C}_{10}\text{H}_{14}\text{N}_2\text{O}_8\text{Na}_2 \cdot 2\text{H}_2\text{O}$) was dissolved in distilled water, and diluted to a total volume of 2000 ml to make 0.01 M of EDTA ($[\text{CH}_2\text{N}(\text{CH}_2\text{COOH})_2]_2$) solution. This solution was prepared in large quantities to minimise any measuring error. As EDTA solutions are not very stable, they were kept in dark bottles and the same solution was not used for more than 3 weeks.
- (b) *Eriochrome Black T and Calmagite*. Initially, Eriochrome Black T was used as the indicator for EDTA titration. It was diluted with ethanol to improve its fluidity. However, it decomposes slowly on standing and is therefore less stable than Calmagite indicator solution (Fritz and Schenk, 1987). Consequently, the titration indicator was changed from Eriochrome Black T to Calmagite. Both indicators may be used interchangeably.
- (c) *Magnesium Sulphate Solution*. For the preparation of a 0.01 M solution of magnesium sulphate, 1.23 g of magnesium sulphate hepta-hydrate ($\text{MgSO}_4 \cdot 7\text{H}_2\text{O}$) was dissolved in 500 ml of distilled water.
- (d) *Ammonia-Ammonium Chloride Buffer Solution (pH = 10)*. 142 ml of concentrated ammonia solution (specific gravity 0.88-0.90) was added to 17.50 g of ammonium chloride (NH_4Cl), and then diluted to a volume of 250 ml with distilled water. The pH of the buffer solution was checked with a calibrated pH meter to make sure that it reached the desired pH value.

A.2 Start-Up & Shut-Down Procedures

START-UP PROCEDURE	
1	<ul style="list-style-type: none"> switch on computer load & run data acquisition program, FOUL
2	<ul style="list-style-type: none"> set temperature controller of hot H₂O tank to 86°C pour 25 litres of distilled water into hot H₂O tank
3	<ul style="list-style-type: none"> pour 60 litres of distilled water into solution tank switch on solution tank temperature controller & set controller to manual with cooling H₂O fully on turn on cooling water to solution tank (at 27 units) set solution tank temperature controller to 20°C
4	<ul style="list-style-type: none"> switch on solution pump for distilled water to recycle back to tank at high flow rate add required amounts of Na₂SO₄ and Ca(NO₃)₂·4H₂O into solution tank let solution recycle back to tank for about 15 minutes open solution-tank-to-PHE valve and solution-tank-to-preheater-and-DPHE valve to let solution circulate rig at high flow rate for about 15 minutes
5	<ul style="list-style-type: none"> switch on hot H₂O pump (recycle only) switch on air supply to PHE hot H₂O flow rate control valve switch on PHE hot H₂O flow rate controller and set flow rate to 70 ml/s fill all hot water lines (to PHE & DPHE) with cold water close hot-H₂O-to-DPHE valve and hot-H₂O-to-PHE valve switch off PHE hot H₂O flow rate controller switch off hot H₂O pump switch on tap water supply to hot H₂O tank switch on hot H₂O tank heaters
6	<ul style="list-style-type: none"> take solution sample & titrate sample with EDTA to determine solution concentration repeat steps 4 and 6 until desired solution concentration is reached
7	<ul style="list-style-type: none"> open all 3 pressure tapings on solution side when hot water in tank has reached 86°C, switch on hot H₂O pump (recycle only) set PHE hot H₂O flow rate to 40 ml/s (valve to DPHE is closed) solution is circulated via PHE only (valve to preheater is closed) at high flow rate for PHE solution inlet temperature (T1) to reach 52°C set solution tank temperature controller to 51.5°C maintain PHE hot H₂O inlet temperature (T3) at 87°C by increasing PHE hot H₂O flow rate slowly to desired value (and setting temperature controller of hot H₂O tank to >86°C) insert conductivity probe into solution tank & switch on conductivity meter once temperatures T1 & T3 have stabilise, set hot H₂O temperature controller to 86°C

8	<ul style="list-style-type: none"> ☆ switch on air supply for PHE solution flow rate control valve ☆ switch on PHE solution flow rate controller and set flow rate to desired value • ☆ wait for temperatures T1 & T3 and hot H₂O & solution flow rates to stabilise 🖨 start data acquisition program
---	--

SHUT-DOWN PROCEDURE

1	☆ take sample
2	<ul style="list-style-type: none"> 🖨 stop data acquisition program 🖨 retrieve data from hard disk onto diskette
3	<ul style="list-style-type: none"> ☆ close solution-tank-to-PHE valve ☆ close tappings of all 3 pressure transducers (flow and pressure drop) ☆ set solution tank temperature controller to manual with cooling H₂O fully on ☆ switch off solution pump & drain solution from tank ☆ switch off solution flow rate controller & switch off air supply to solution flow rate control valve ☆ remove conductivity probe
4	<ul style="list-style-type: none"> • switch off heaters in hot H₂O tank • switch off water supply to hot H₂O tank • switch off hot H₂O pump & drain hot H₂O tank • switch off hot H₂O flow rate controller & switch off air supply to hot H₂O flow rate control valve
5	<ul style="list-style-type: none"> ☆ disassemble PHE to take fouled plates off ☆ replace fouled plates taken from PHE with clean plates and reassemble PHE
6	<ul style="list-style-type: none"> ☆ fill solution tank with tap water ☆ switch on solution pump to flush solution lines with tap water at high flow rate for 1 hour ☆ switch off solution pump & drain solution tank
7	<ul style="list-style-type: none"> ☆ clean solution tank lid ☆ repeat step 6 ☆ repeat step 6 again to let it run overnight ☆ switch off solution tank temperature controller & switch off cooling water to solution tank
8	<ul style="list-style-type: none"> • fill hot H₂O tank with tap water • switch on hot H₂O pump to flush all hot water lines for about 2 hours • switch off hot H₂O pump & drain tank
9	<ul style="list-style-type: none"> ☆ take pictures of fouled plates ☆ take crystal sample(s) from fouled plates ☆ clean fouled plates ☆ flush air compressor and all (five) air filters
10	<ul style="list-style-type: none"> ☆ disassemble PHE to check if everything is clean ☆ assemble PHE with clean plates ready for next run ☆ remove filter cartridge & clean & reassemble filter cartridge

11	<ul style="list-style-type: none">☆ set solution tank temperature controller to manual with cooling H₂O fully on & turn on cooling H₂O☆ pour 60 litres of distilled water into solution tank☆ switch on solution pump to flush solution lines with distilled water at high flow rate for 1 hour☆ flush tappings of (three) pressure transducers & both sample points🖨 use data acquisition program to check flow rates and pressure drop☆ switch off solution pump & drain solution tank
12	<ul style="list-style-type: none">☆ repeat step 8 with distilled H₂O

APPENDIX B: CFX COMMAND FILES

B.1 Flow Passing Through a 2-D Channel with a Single Corrugation on the Bottom Surface

```
>FLOW3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 5000000
  TOTAL CHARACTER WORK SPACE 1000
  TOTAL REAL WORK SPACE 10000000
  MAXIMUM NUMBER OF BLOCKS 10
  MAXIMUM NUMBER OF PATCHES 30
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 10
>>OPTIONS
  TWO DIMENSIONS
  BODY FITTED GRID
  CARTESIAN COORDINATES
  LAMINAR FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  STEADY STATE
>>MODEL TOPOLOGY
>>INPUT TOPOLOGY
  READ GEOMETRY FILE
>>MODEL DATA
>>TITLE
  PROBLEM TITLE '2-DIMENSION FLOW PASS A CORRUGATION'
>>PHYSICAL PROPERTIES
>>STANDARD FLUID
  FLUID 'WATER'
  STANDARD FLUID REFERENCE TEMPERATURE 3.0200E+02
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 10000
  OUTPUT MONITOR POINT 10 50 1
  MASS SOURCE TOLERANCE 1.0000E-10
>>CREATE GRID
>>INPUT GRID
  READ GRID FILE
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
  PATCH NAME 'INLET PLANE'
  U VELOCITY 1.5000E-01
  V VELOCITY 0.0000E+00
>>SET VARIABLES
  PATCH NAME 'OUTLET PLANE'
  PRESSURE 0.0000E+00
>>STOP
```

B.2 Flow Passing Through a 2-D Channel with Multiple Corrugations on the Top and Bottom Surfaces

```
>>CFXF3D
>>SET LIMITS
TOTAL INTEGER WORK SPACE 6000000
TOTAL CHARACTER WORK SPACE 5000
TOTAL REAL WORK SPACE 13500000
MAXIMUM NUMBER OF BLOCKS 100
MAXIMUM NUMBER OF PATCHES 700
MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 200
>>OPTIONS
TWO DIMENSIONS
BODY FITTED GRID
CARTESIAN COORDINATES
LAMINAR FLOW
HEAT TRANSFER
INCOMPRESSIBLE FLOW
STEADY STATE
USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
>>SOURCE OF DATA
PCP
>>FLUID DATA
FLUID 'WATER'
MATERIAL TEMPERATURE 3.1800E+02
MATERIAL PHASE 'LIQUID'
>>SET INITIAL GUESS
>>INPUT FROM FILE
READ DUMP FILE
UNFORMATTED
>>SELECT VARIABLES FROM FILE
ALL RELEVANT DATA
>>TITLE
PROBLEM TITLE '10 CORRUGATIONS B/A=2.0'
>>SOLVER DATA
>>PROGRAM CONTROL
MAXIMUM NUMBER OF ITERATIONS 10000
MASS SOURCE TOLERANCE 1.0000E-12
>>EQUATION SOLVERS
U VELOCITY 'STONE'
V VELOCITY 'STONE'
PRESSURE 'AMG'
ENTHALPY 'AMG'
>>REDUCTION FACTORS
U VELOCITY 1.0000E-03
V VELOCITY 1.0000E-03
PRESSURE 1.0000E-03
ENTHALPY 1.0000E-03
>>SWEEPS INFORMATION
>>MAXIMUM NUMBER
U VELOCITY 500
V VELOCITY 500
PRESSURE 1000
ENTHALPY 500
```



```

>>UNDER RELAXATION FACTORS
  U VELOCITY 3.0000E-01
  V VELOCITY 3.0000E-01
  PRESSURE 3.0000E-01
  DENSITY 3.0000E-01
  VISCOSITY 3.0000E-01
  TEMPERATURE 1.0000E+00
  ENTHALPY 7.0000E-01
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
  PATCH NAME 'INLET'
  U VELOCITY 1.0000E-01
  V VELOCITY 0.0000E+00
  TEMPERATURE 3.1300E+02
>>SET VARIABLES
  PATCH NAME 'OUTLET'
  PRESSURE 0.0000E+00
  TEMPERATURE 3.3200E+02
>>WALL BOUNDARY CONDITIONS
  PATCH NAME 'WALLTOP'
  TEMPERATURE 3.4300E+02
>>WALL BOUNDARY CONDITIONS
  PATCH NAME 'WALLBOT'
  TEMPERATURE 3.4300E+02
>>STOP

```

B.3 Flow Passing Over a 2-D Backward-Facing Step

```

>>CFXF3D
>>OPTIONS
  TWO DIMENSIONS
  BODY FITTED GRID
  CARTESIAN COORDINATES
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  STEADY STATE
  USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
>>SOURCE OF DATA
  PCP
>>FLUID DATA
  FLUID 'WATER'
  MATERIAL TEMPERATURE 2.9300E+02
  MATERIAL PHASE 'LIQUID'
>>SET INITIAL GUESS
>>INPUT FROM FILE
  READ DUMP FILE
  UNFORMATTED
>>SELECT VARIABLES FROM FILE
  ALL RELEVANT DATA
>>TITLE
  PROBLEM TITLE 'BACKWARD-FACING STEP'
>>PHYSICAL PROPERTIES

```

```
>>TURBULENCE PARAMETERS
  >>TURBULENCE MODEL
    TURBULENCE MODEL 'DIFFERENTIAL STRESS'
>>SOLVER DATA
  >>PROGRAM CONTROL
    MAXIMUM NUMBER OF ITERATIONS 10000
    OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-2'
    OUTPUT MONITOR POINT 30 15 1
    MASS SOURCE TOLERANCE 1.0000E-07
  >>DEFERRED CORRECTION
    K START 1
    K END 2000
    EPSILON START 1
    EPSILON END 2000
  >>EQUATION SOLVERS
    U VELOCITY 'AMG'
    V VELOCITY 'AMG'
    PRESSURE 'AMG'
    K 'LINE SOLVER'
    EPSILON 'LINE SOLVER'
    UU 'LINE SOLVER'
    VV 'LINE SOLVER'
    WW 'LINE SOLVER'
    UV 'LINE SOLVER'
  >>REDUCTION FACTORS
    U VELOCITY 1.5000E-01
    V VELOCITY 1.5000E-01
    PRESSURE 3.0000E-02
    K 1.5000E-01
    EPSILON 1.5000E-01
    UU 1.5000E-01
    VV 1.5000E-01
    WW 1.5000E-01
    UV 1.5000E-01
  >>SWEEPS INFORMATION
    >>MAXIMUM NUMBER
      U VELOCITY 100
      V VELOCITY 100
      PRESSURE 500
      K 100
      EPSILON 100
      UU 100
      VV 100
      WW 100
      UV 100
    >>MINIMUM NUMBER
      U VELOCITY 10
      V VELOCITY 10
      PRESSURE 50
      K 10
      EPSILON 10
      UU 10
      VV 10
      WW 10
      UV 10
  >>UNDER RELAXATION FACTORS
    U VELOCITY 4.0000E-01
    V VELOCITY 4.0000E-01
```

```

PRESSURE 4.0000E-01
DENSITY 2.0000E-01
VISCOSITY 5.0000E-01
K 4.0000E-01
EPSILON 4.0000E-01
UU 3.0000E-01
VV 3.0000E-01
WW 3.0000E-01
UV 3.0000E-01
>>MODEL BOUNDARY CONDITIONS
>>MASS FLOW BOUNDARY CONDITIONS
  FLUXES -2.3000E+02
  MASS FLOW SPECIFIED
>>SET VARIABLES
  PATCH NAME 'OUTLET'
  PRESSURE 0.0000E+00
>>STOP

```

B.4 Inlet Conditions for Flow in Flat Plate Channels

```

>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 6000000
  TOTAL CHARACTER WORK SPACE 5000
  TOTAL REAL WORK SPACE 13500000
  MAXIMUM NUMBER OF BLOCKS 100
  MAXIMUM NUMBER OF PATCHES 700
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 200
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  CARTESIAN COORDINATES
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  STEADY STATE
  USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
  >>SOURCE OF DATA
    PCP
  >>FLUID DATA
    FLUID 'WATER'
    MATERIAL TEMPERATURE 3.1450E+02
    MATERIAL PHASE 'LIQUID'
>>TITLE
  PROBLEM TITLE 'INLET TO PHE'
>>PHYSICAL PROPERTIES
  >>TURBULENCE PARAMETERS
    >>TURBULENCE MODEL
      TURBULENCE MODEL 'K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 3000
  OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-11'

```

```

OUTPUT MONITOR POINT 5 5 10
MASS SOURCE TOLERANCE 1.0000E-10
>>EQUATION SOLVERS
U VELOCITY 'AMG'
V VELOCITY 'AMG'
W VELOCITY 'AMG'
PRESSURE 'AMG'
K 'LINE SOLVER'
EPSILON 'LINE SOLVER'
>>MODEL BOUNDARY CONDITIONS
>>MASS FLOW BOUNDARY CONDITIONS
FLUXES -8.340000E-02
MASS FLOW SPECIFIED
>>SET VARIABLES
PATCH NAME 'OUTPIPE1'
PRESSURE 1.5000E+04
>>STOP

```

B.5 Double Backward-Facing Step

```

>>CFXF3D
>>SET LIMITS
TOTAL INTEGER WORK SPACE 6000000
TOTAL CHARACTER WORK SPACE 5000
TOTAL REAL WORK SPACE 13500000
MAXIMUM NUMBER OF BLOCKS 50
MAXIMUM NUMBER OF PATCHES 500
MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 50
>>OPTIONS
THREE DIMENSIONS
BODY FITTED GRID
CARTESIAN COORDINATES
TURBULENT FLOW
ISOTHERMAL FLOW
INCOMPRESSIBLE FLOW
STEADY STATE
USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
>>SOURCE OF DATA
PCP
>>FLUID DATA
FLUID 'WATER'
MATERIAL TEMPERATURE 3.1800E+02
MATERIAL PHASE 'LIQUID'
>>SET INITIAL GUESS
>>INPUT FROM FILE
READ DUMP FILE
UNFORMATTED
>>SELECT VARIABLES FROM FILE
ALL RELEVANT DATA
>>TITLE
PROBLEM TITLE '3D TWO BACKWARD-FACING STEP'
>>PHYSICAL PROPERTIES
>>TURBULENCE PARAMETERS

```

```
>>TURBULENCE MODEL
  TURBULENCE MODEL 'K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 5000
  OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-2'
  OUTPUT MONITOR POINT 20 10 3
  MASS SOURCE TOLERANCE 1.0000E-10
>>EQUATION SOLVERS
  U VELOCITY 'AMG'
  V VELOCITY 'AMG'
  W VELOCITY 'AMG'
  PRESSURE 'AMG'
  K 'LINE SOLVER'
  EPSILON 'LINE SOLVER'
>>REDUCTION FACTORS
  U VELOCITY 2.0000E-01
  V VELOCITY 2.0000E-01
  W VELOCITY 2.0000E-01
  PRESSURE 1.0000E-02
  K 2.0000E-01
  EPSILON 2.0000E-01
>>SWEEPS INFORMATION
>>MAXIMUM NUMBER
  U VELOCITY 50
  V VELOCITY 50
  W VELOCITY 50
  PRESSURE 500
  K 50
  EPSILON 50
>>MINIMUM NUMBER
  U VELOCITY 5
  V VELOCITY 5
  W VELOCITY 5
  PRESSURE 50
  K 5
  EPSILON 5
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
  PATCH NAME 'INLET PORT'
  U VELOCITY 8.7500E-01
  V VELOCITY 0.0000E+00
  W VELOCITY 0.0000E+00
  K 1.5000E-03
  EPSILON 3.5000E-03
>>SET VARIABLES
  PATCH NAME 'OUTLET PORT'
  PRESSURE 1.4000E+04
>>STOP
```

B.6 Simplified 3-D Geometry of a Flat Plate Heat Exchanger

```
>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 6000000
```

TOTAL CHARACTER WORK SPACE 5000
TOTAL REAL WORK SPACE 13500000
MAXIMUM NUMBER OF BLOCKS 50
MAXIMUM NUMBER OF PATCHES 500
MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 50
>>OPTIONS
THREE DIMENSIONS
BODY FITTED GRID
CARTESIAN COORDINATES
TURBULENT FLOW
ISOTHERMAL FLOW
INCOMPRESSIBLE FLOW
STEADY STATE
USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
>>SOURCE OF DATA
PCP
>>FLUID DATA
FLUID 'WATER'
MATERIAL TEMPERATURE 3.1800E+02
MATERIAL PHASE 'LIQUID'
>>TITLE
PROBLEM TITLE '3D SIMPLIFIED GRID'
>>PHYSICAL PROPERTIES
>>TURBULENCE PARAMETERS
>>TURBULENCE MODEL
TURBULENCE MODEL 'K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
MAXIMUM NUMBER OF ITERATIONS 5000
OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-4'
OUTPUT MONITOR POINT 20 20 3
MASS SOURCE TOLERANCE 1.0000E-10
>>EQUATION SOLVERS
U VELOCITY 'AMG'
V VELOCITY 'AMG'
W VELOCITY 'AMG'
PRESSURE 'AMG'
K 'LINE SOLVER'
EPSILON 'LINE SOLVER'
>>REDUCTION FACTORS
U VELOCITY 2.0000E-01
V VELOCITY 2.0000E-01
W VELOCITY 2.0000E-01
PRESSURE 1.0000E-02
K 2.0000E-01
EPSILON 2.0000E-01
>>SWEEPS INFORMATION
>>MAXIMUM NUMBER
U VELOCITY 50
V VELOCITY 50
W VELOCITY 50
PRESSURE 500
K 50
EPSILON 50
>>MINIMUM NUMBER
U VELOCITY 5


```
V VELOCITY 5
W VELOCITY 5
PRESSURE 50
K 5
EPSILON 5
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
  PATCH NAME 'INLET PORT'
  U VELOCITY 8.7500E-01
  V VELOCITY 0.0000E+00
  K 1.5000E-03
  EPSILON 3.5000E-03
>>SET VARIABLES
  PATCH NAME 'OUTLET PORT'
  PRESSURE 1.4000E+04
>>STOP
```

B.7 True Geometry of a Flat Plate Heat Exchanger

```
>>CFXF3D
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  CARTESIAN COORDINATES
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  TRANSIENT FLOW
  USE DATABASE
>>MODEL DATA
>>MATERIALS DATABASE
  >>SOURCE OF DATA
    PCP
  >>FLUID DATA
    FLUID 'WATER'
    MATERIAL TEMPERATURE 3.1800E+02
    MATERIAL PHASE 'LIQUID'
>>SET INITIAL GUESS
  >>INPUT FROM FILE
    READ DUMP FILE
    UNFORMATTED
  >>SELECT VARIABLES FROM FILE
    ALL RELEVANT DATA
>>TITLE
  PROBLEM TITLE 'NEW FLAT PLATE 3D'
>>PHYSICAL PROPERTIES
  >>TRANSIENT PARAMETERS
    >>ADAPTIVE TIME STEPPING
      NUMBER OF TIME STEPS 200
      INITIAL TIME STEP 1.0000E-02
      MINIMUM TIME STEP 1.0000E-03
      MAXIMUM TIME STEP 1.0000E+05
      MULTIPLY TIME STEP BY 2.0000E+00
      DIVIDE TIME STEP BY 2.0000E+00
      MINIMUM INTERVAL BETWEEN INCREMENTS 3
```

```
MAXIMUM NUMBER OF CONTIGUOUS DECREMENTS 5
BACKWARD DIFFERENCE
LINEAR TIME DIFFERENCING
/* INITIAL TIME 0.0000E+00 */
MAXIMUM TIME 4.9850E+00
>>TURBULENCE PARAMETERS
>>TURBULENCE MODEL
TURBULENCE MODEL 'DIFFERENTIAL STRESS'
>>SOLVER DATA
>>PROGRAM CONTROL
MAXIMUM NUMBER OF ITERATIONS 200
OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-16'
OUTPUT MONITOR POINT 15 15 5
MASS SOURCE TOLERANCE 1.0000E-07
>>DEFERRED CORRECTION
K START 10001
K END 10002
EPSILON START 10001
EPSILON END 10002
>>EQUATION SOLVERS
PRESSURE 'AMG'
K 'AMG'
EPSILON 'AMG'
>>REDUCTION FACTORS
U VELOCITY 1.0000E-03
V VELOCITY 1.0000E-03
W VELOCITY 1.0000E-03
PRESSURE 1.0000E-03
K 1.0000E-03
EPSILON 1.0000E-03
UU 1.0000E-03
VV 1.0000E-03
WW 1.0000E-03
UV 1.0000E-03
VW 1.0000E-03
WU 1.0000E-03
>>SWEEPS INFORMATION
>>MAXIMUM NUMBER
U VELOCITY 500
V VELOCITY 500
W VELOCITY 500
PRESSURE 1000
K 500
EPSILON 500
UU 500
VV 500
WW 500
UV 500
VW 500
WU 500
>>MINIMUM NUMBER
U VELOCITY 1
V VELOCITY 1
W VELOCITY 1
PRESSURE 1
K 1
EPSILON 1
UU 1
```

```
VV 1
WW 1
UV 1
VW 1
WU 1
>>TRANSIENT CONTROL
>>CONVERGENCE TESTING ON VARIABLE
PRESSURE
>>CONTROL PARAMETERS
MINIMUM RESIDUAL VALUE 1.0000E-05
MAXIMUM RESIDUAL VALUE 1.0000E+25
REDUCTION FACTOR 1.0000E+03
DIVERGENCE RATIO 1.0000E+05
>>UNDER RELAXATION FACTORS
U VELOCITY 5.0000E-01
V VELOCITY 5.0000E-01
W VELOCITY 5.0000E-01
PRESSURE 7.0000E-01
DENSITY 3.0000E-01
VISCOSITY 7.0000E-01
K 5.0000E-01
EPSILON 5.0000E-01
UU 5.0000E-01
VV 5.0000E-01
WW 5.0000E-01
UV 5.0000E-01
VW 5.0000E-01
WU 5.0000E-01
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
PATCH NAME 'INLET PORT'
U VELOCITY 0.0000E+00
V VELOCITY 0.0000E+00
W VELOCITY 1.0440E-01
K 1.5000E-03
EPSILON 3.5000E-03
>>SET VARIABLES
PATCH NAME 'OUTLET PORT'
PRESSURE 0.0000E+00
>>STOP
```

B.8 Distributor Designs in Plate Heat Exchangers

```
>>CFXF3D
>>SET LIMITS
MAXIMUM NUMBER OF PATCHES 1000
>>OPTIONS
THREE DIMENSIONS
BODY FITTED GRID
CARTESIAN COORDINATES
TURBULENT FLOW
HEAT TRANSFER
INCOMPRESSIBLE FLOW
STEADY STATE
USE DATABASE
```

```
>>MODEL TOPOLOGY
>>CREATE PATCH
  PATCH NAME 'AIN1A'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 1 1 20 23 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1B'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 2 2 17 19 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1C'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 3 3 14 16 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1D'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 4 4 12 13 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1E'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 5 5 10 11 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1F'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 6 6 9 9 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1G'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 7 7 8 8 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1H'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 8 9 7 7 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1I'
  BLOCK NAME 'BLOCK-NUMBER-11'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 10 10 6 6 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1J'
  BLOCK NAME 'BLOCK-NUMBER-13'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 1 1 6 6 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1K'
  BLOCK NAME 'BLOCK-NUMBER-13'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 2 5 5 5 1 10
>>CREATE PATCH
  PATCH NAME 'AIN1L'
```

```
BLOCK NAME 'BLOCK-NUMBER-13'
PATCH TYPE 'SOLID'
PATCH LOCATION 6 8 4 4 1 10
>>CREATE PATCH
PATCH NAME 'AIN2A'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 5 14 15 1 10
>>CREATE PATCH
PATCH NAME 'AIN2B'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 6 6 15 16 1 10
>>CREATE PATCH
PATCH NAME 'AIN2C'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 7 7 16 17 1 10
>>CREATE PATCH
PATCH NAME 'AIN2D'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 8 8 17 18 1 10
>>CREATE PATCH
PATCH NAME 'AIN2E'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 9 9 19 20 1 10
>>CREATE PATCH
PATCH NAME 'AIN2F'
BLOCK NAME 'BLOCK-NUMBER-10'
PATCH TYPE 'SOLID'
PATCH LOCATION 10 10 20 21 1 10
>>CREATE PATCH
PATCH NAME 'AIN2G'
BLOCK NAME 'BLOCK-NUMBER-12'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 8 21 21 1 10
>>CREATE PATCH
PATCH NAME 'AOUT1A'
BLOCK NAME 'BLOCK-NUMBER-29'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 8 19 19 1 10
>>CREATE PATCH
PATCH NAME 'AOUT1B'
BLOCK NAME 'BLOCK-NUMBER-31'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 6 19 19 1 10
>>CREATE PATCH
PATCH NAME 'AOUT2A'
BLOCK NAME 'BLOCK-NUMBER-27'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 10 30 30 1 10
>>CREATE PATCH
PATCH NAME 'AOUT2B'
BLOCK NAME 'BLOCK-NUMBER-29'
PATCH TYPE 'SOLID'
PATCH LOCATION 1 8 30 30 1 10
```

```

>>CREATE PATCH
  PATCH NAME 'AOUT2C'
  BLOCK NAME 'BLOCK-NUMBER-31'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 1 6 30 30 1 10
>>CREATE PATCH
  PATCH NAME 'AOUT3A'
  BLOCK NAME 'BLOCK-NUMBER-30'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 1 8 11 11 1 10
>>CREATE PATCH
  PATCH NAME 'AOUT3B'
  BLOCK NAME 'BLOCK-NUMBER-32'
  PATCH TYPE 'SOLID'
  PATCH LOCATION 1 6 11 11 1 10
>>MODEL DATA
>>DIFFERENCING SCHEME
  U VELOCITY 'QUICK'
  V VELOCITY 'QUICK'
  W VELOCITY 'QUICK'
  K 'CCCT'
  EPSILON 'CCCT'
>>MATERIALS DATABASE
>>SOURCE OF DATA
  PCP
>>FLUID DATA
  FLUID 'WATER'
  MATERIAL TEMPERATURE 3.1800E+02
  MATERIAL PHASE 'LIQUID'
/* >>SET INITIAL GUESS
>>INPUT FROM FILE
  READ DUMP FILE
  UNFORMATTED
>>SELECT VARIABLES FROM FILE
  ALL RELEVANT DATA */
>>TITLE
  PROBLEM TITLE 'DISTRIBUTOR DESIGN B'
>>PHYSICAL PROPERTIES
>>TURBULENCE PARAMETERS
>>TURBULENCE MODEL
  TURBULENCE MODEL 'K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 10000
  OUTPUT MONITOR BLOCK 'BLOCK-NUMBER-16'
  OUTPUT MONITOR POINT 10 15 5
  MASS SOURCE TOLERANCE 1.0000E-10
>>DEFERRED CORRECTION
  K START 10001
  K END 10002
  EPSILON START 10001
  EPSILON END 10002
>>EQUATION SOLVERS
  U VELOCITY 'AMG'
  V VELOCITY 'AMG'
  W VELOCITY 'AMG'
  PRESSURE 'AMG'
  K 'AMG'

```



```
EPSILON 'AMG'
ENTHALPY 'AMG'
>>REDUCTION FACTORS
U VELOCITY 2.0000E-01
V VELOCITY 2.0000E-01
W VELOCITY 2.0000E-01
PRESSURE 7.0000E-02
K 2.0000E-01
EPSILON 2.0000E-01
ENTHALPY 1.0000E-01
>>SWEEPS INFORMATION
>>MAXIMUM NUMBER
U VELOCITY 50
V VELOCITY 50
W VELOCITY 50
PRESSURE 300
K 50
EPSILON 50
ENTHALPY 300
>>MINIMUM NUMBER
U VELOCITY 1
V VELOCITY 1
W VELOCITY 1
PRESSURE 1
K 1
EPSILON 1
ENTHALPY 1
>>UNDER RELAXATION FACTORS
U VELOCITY 5.0000E-01
V VELOCITY 5.0000E-01
W VELOCITY 5.0000E-01
PRESSURE 7.0000E-01
DENSITY 3.0000E-01
VISCOSITY 7.0000E-01
K 5.0000E-01
EPSILON 5.0000E-01
TEMPERATURE 1.0000E+00
ENTHALPY 7.0000E-01
>>MODEL BOUNDARY CONDITIONS
>>SET VARIABLES
PATCH NAME 'INLET PORT'
U VELOCITY 0.0000E+00
V VELOCITY 0.0000E+00
W VELOCITY 1.0440E-01
K 1.5000E-03
EPSILON 3.5000E-03
TEMPERATURE 3.1450E+02
>>SET VARIABLES
PATCH NAME 'OUTLET PORT'
PRESSURE 0.0000E+00
TEMPERATURE 3.2050E+02
>>WALL BOUNDARY CONDITIONS
PATCH NAME 'WALL'
TEMPERATURE 3.3800E+02
>>STOP
```

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