ELECTROHYDRODYNAMIC CONTROL OF CONVECTIVE CONDENSATION HEAT TRANSFER AND PRESSURE DROP IN A HORIZONTAL ANNULAR CHANNEL

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By

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To my parents

Abstract

The objective of this research is to investigate the effect of DC, AC and pulse wave applied voltage on two-phase flow patterns, heat transfer and pressure drop during tube side convective condensation of refrigerant HFC-134a in an annular channel. Experiments were performed in a horizontal, single-pass, counter-current heat exchanger with a rod electrode placed along the center of the tube. The electric field was applied across the annular gap formed by the electrode connected to the high-voltage source and the grounded surface of the inner tube of the heat exchanger. The electric field between the two electrodes was established by applying a high voltage to the central electrode. The high voltage was generated by amplifying the voltage output from a function generator. The flow was visualized at the exit of the heat exchanger using a high speed camera through a transparent quartz tube coated with an electrically conductive film of tin oxide.

The effect of a 8 kV DC applied voltage was investigated for mass flux in the range 45 kg/m²s to 160 kg/m²s and average quality of x_{avg} = 45%. The application of the 8 KV DC voltage increased heat transfer and pressure drop by factor 3 and 4.5 respectively at the lowest mass flux of 45 kg/m²s. Increasing the mass flux decreased the effect of electrohydrodynamic forces on the two-phase flow heat transfer and pressure drop.

The effect of different AC and pulse wave applied voltage parameters (e.g.

waveform, amplitude, DC bias, AC frequency, pulse repetition rate and duty cycle) on heat transfer and pressure drop was investigated. Experiments were performed with an applied sine and square waveform over a range of frequencies (2 Hz < f < 2 kHz), peak-to-peak voltages (2 kV $< V_{p-p} <$ 12 kV) and DC bias voltage (-10 kV $< V_{DC} <$ 10 kV), and with an applied pulse voltage of amplitude 12 kV and duty cycle from 10% to 90%. These experiments were performed for a fixed mass flux of 100 kg/m²s, inlet quality of 70%, and heat flux of 10 kW/m². For the same amplitude and DC bias, the pulse wave applied voltage provides a larger range of heat transfer and pressure drop control by varying the pulse repetition rate and duty cycle compared to the sine waveform.

The effect of a step input voltage on two phase flow patterns, heat transfer and pressure drop was examined and analyzed for an initially stratified flow. The flow visualization images showed that the step input voltage caused the liquid to be extracted from the bottom liquid stratum toward the center electrode and then pushed to the bulk flow in the form of twisted liquid cones pointing outward from the central electrode. These transient flow patterns, which are characterized by high heat transfer compared to the DC case, diminish in steady state. The effect of the amplitude of the step input voltage and the initial distance between the electrode and liquid-vapour interface on the liquid extraction was investigated experimentally and numerically. At sufficiently high voltages, the induced EHD forces at the liquid-vapour interface overcame the gravitational forces and caused the liquid to be extracted towards the high voltage electrode. The extraction time decreased with an increase of the applied step voltage and/or decrease of the initial distance between liquid interface and the high voltage electrode. The numerical simulation results were, in general, in agreement with the experimental results.

The effect of pulse repetition rate of pulse applied voltage on two phase flow

patterns, heat transfer and pressure drop can be divided into three regimes. At the low pulse repetition rate range, f < 10 Hz, the two-phase flow responded to the induced EHD forces, and liquid was extracted from the bottom stratum to the center electrode and then pushed back to the bulk flow in the form of twisted liquid cones. Increasing the pulse repetition rate in this range increased the repetition of the extraction cycle and therefore increased heat transfer and pressure drop. In the mid pulse repetition rate range, 10 Hz < f < 80 Hz, the extraction was not completed, which led to lower heat transfer compared to the lower pulse repetition rate range. In this range, the two phase patterns were characterized by liquid-vapour interface oscillations between the center electrode and the bottom stratum and liquid droplet oscillations which increased the momentum transfer and therefore pressure drop. Increasing the pulse repetition rate in this range decreased heat transfer and increased pressure drop. In the high pulse repetition rate range, f > 80 Hz, increasing the pulse repetition rate decreased both the interfacial and droplet oscillations and therefore decreased the heat transfer and pressure drop till the two phase flow patterns resembled that for an applied DC voltage. For the same pulse repetition rate, increasing the mass flux decreased the effect of EHD forces on heat transfer and pressure drop. The heat transfer enhancement ratio and pressure drop ratio increased with an increase of the duty cycle for the same pulse repetition rate of the applied voltage.

Different combinations of pulse repetition rate and duty cycle of applied pulse wave voltage can be used to achieve different values of heat transfer and pressure drop. This can be very beneficial for heat transfer control in industrial applications. An advantage of such control is that it eliminates various measurements devices, control and bypass valves, variable speed pumps, fans and control schemes used in current technology for heat transfer and pressure drop control. The range of control of the ratio of the heat transfer coefficient to the pressure drop is from 8.24 to 20.56 for mass flux of 50 kg/m²s and it decreased with increasing mass flux untill it reached 1.63 to 3.81 at mass flux 150 kg/m²s.

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	between 0 and 8 kV at a mass flux of $G = 55 \ kg/m^2 s$ and quality $x =$	
	45% (pictures equally distributed along one cycle, i.e time between	
	frames is 83.3 msec).	211
E3	Flow regime redistributions for a 6 Hz, 50% duty cycle pulse wave	
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	45% (pictures equally distributed along one cycle, i.e time between	
	frames is 13.9 msec).	212
E4	Flow regime redistributions for a 10 Hz, 50% duty cycle pulse wave	
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	45% (pictures equally distributed along one cycle, i.e time between	
	frames is 8.3 msec).	213
E5	Flow regime redistributions for a 75 Hz, 50% duty cycle pulse wave	
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	frames is 1.1 msec)	214
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	frames is 0.3 msec)	215
E7	Flow regime redistributions for a 400 Hz, 50% duty cycle pulse wave	
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	frames is 0.2 msec).	216

E8	Flow regime redistributions for a 1 Hz, 25% duty cycle pulse wave	
	between 0 and 8 kV at a mass flux of $G = 55 \ kg/m^2 s$ and quality $x =$	
	45% (pictures equally distributed along one cycle, i.e time between	
	frames is 83.3 msec)	217
EO		

Nomenclature

A	flow area, m^2
$ar{A},$	dimensionless area, A/D^2
В	magnetic flux density, Vs/m^2
Bo	boiling number, q''/Gh_{lv}
c_p	specific heat at constant pressure, J/kgK
c_{p-cw}	average specific heat of the water through the chiller, J/kgK
c_{p-pw}	specific heat of the water through the preheater, J/kgK
c_{p-ref}	specific heat of the refrigerant, J/kgK
d,D	diameter, m
D	electric displacement, C/m^2
D_e	electrode diameter, m
D_h	hydraulic diameter, m
e	charge, C
E	electric field strength, V/m
E_n	normal electric field strength, V/m
E_t	tangential electric field strength, V/m
Ehd	EHD number
f	frequency / pulse repetition rate, Hz
f	friction factor
f_{Fr}	friction factor defined by Equation 2.97
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f_i	interfacial surface roughness defined by Equation 2.83
f_e'''	electric body force per unit volume, N/m^3
F_D	dielectrophoretic force per unit volume, $\rm N/m^3$
F_E	electrophoretic force per unit volume, $\rm N/m^3$
F	friction force, N
F	Froude number given by Equation 2.17
F	parameter defined by Equation 2.23
Fr_l	liquid Froude number
Fr_{so}	Soilman's modified Froude number, Soliman [1982, 1983]
g	acceleration due to gravity, m/s^2
G	mass flux, kg/m ² s
G_{bubbly}	bubbly flow transition mass flux, kg/m^2s
G_{mist}	mist flow transition mass flux, kg/m^2s
G_{strat}	stratified flow transition mass flux, kg/m^2s
G_{wavy}	stratified wavy flow transition mass flux, $\rm kg/m^2s$
Ga	Galileo number
Gr	Grashof number
h	heat transfer coefficient, W/m^2K
$h_{ann,J_G=2,5}$	heat transfer coefficient for annular flow at $J_G = 2.5$, W/m ² K
h_c	convective condensation heat transfer coefficient, $\mathrm{W}/\mathrm{m}^2\mathrm{K}$
h_{dc}	heat transfer coefficient for drop wise condensation, $\rm W/m^2 K$
h_f	Nusselt film condensation heat transfer coefficient, $W/m^2 K$
h_i	liquid level above bottom of the electrode, m
h_{id}	dimensionless liquid level above bottom of the electrode, h_i/D
h_l	single phase liquid heat transfer coefficient, W/m^2K
h_{lv}	latent heat of vaporization, kJ/Kg

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h_{strat}	heat transfer coefficient for stratified flow, W/m^2K
h_{tp}	perimeter averaged condensing heat transfer coefficient, $\rm W/m^2 K$
h _{trans}	heat transfer coefficient for annular-stratified flow transition,
	W/m^2K
$\bar{h_L}, h_{ld}$	dimensionless liquid level height, h_l/D
Н	magnetic field strength, A/m
Ι	current, A
j	superficial velocity, m/s
J	current density, A/m^2
J_G	dimensionless vapour mass flux current, $xG/\left[gD\rho_v\left(\rho_l-\rho_v\right)^{0.5}\right]$
Ja	liquid Jakob number, $c_p, l(T_{sat} - T_s)/h_l v$
k	thermal conductivity, W/mK
K	dimensionless parameter defined by Equation 2.18
L	length scale, m
'n	mass flow rate, kg/s
\dot{m}_{w-COND}	water mass flow rate in the condenser, kg/s
\dot{m}_{w-PRE}	water mass flow rate in the water preheater, kg/s
Md	Masuda number
N, N_o	density of the charge, C/m^3
Nu	Nusselt number
Nu_{forced}	Nusselt number
p	pressure, N/m^2
P _i	perimeter of liquid-vapour interface, m
P _{id}	dimensionless perimeter of the interface, P_i/D
P_{red}, Pr	reduced pressure
Pr	Prandtl number

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q	heat transfer rate, W
q_{gain}	heat transfer rate gained from the surroundings, W
q_{PRE}	heat transfer rate from the water and electric preheaters, W
q_w	test section heat transfer rate, W
q''	heat flux, W/m^2
$q_e^{\prime\prime\prime}$	EHD energy generation rate per unit volume, W/m^3
r	radius, m
Ra	Rayleigh number
Re	Reynolds number
S_i	inner surface area of the inner tube, m^2
t	time, sec
T	parameter defined by Equation 2.19
T	temperature, K
T_{n_s}	surface temperature at location n , K
T_s	surface temperature, K
T _{sat}	saturation temperature, K
T^+	dimensionless number defined by Equations 2.58-2.60
u, U	velocity, m/s
u^+	dimensionless velocity, $u/(\sqrt{ au_w/ ho})$
V	voltage, V
V_{DC}	bias voltage, V
V_{p-p}	peak to peak voltage, V
We	Weber number
x	mass vapour quality
x	spatial co-ordinate
x_{IA}	intermittent to annular flow transition quality

.

X	Martinelli parameter
y	spatial co-ordinate
y^+	dimensionless y coordinate, $\sqrt{\tau_w/\rho} y/\nu$
z	spatial co-ordinate
ΔT_{RTD}	RTD temperature difference across the test section, K
ΔP	pressure drop, Pa

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Greek

α	thermal diffusivity, m^2/s
α	void fraction
α_h	homogeneous void fraction
α_{ra}	Rouhani and Axelsson void fraction, given by Equation 2.14
β	fraction of the circumference where film condensation prevails de-
	fined by Equations 2.52, rad
β	volumetric thermal expansion, K^{-1}
γ	ratio between the interfacial velocity and average liquid film veloc-
	ity, $=1.25$ for turbulent flow
δ	liquid film thickness, m
δ^+	dimensionless film thickness, $= \rho_l \delta u_z / \mu_l$
ε	dielectric permittivity, N/V^2
ε_o	dielectric permittivity of free space (8.854.10 ⁻¹²), N/V^2
ε_s	dielectric constant
$ ilde\eta$	dimensionless electric field strength
θ	angle, rad
$ heta_{strat}, heta_{dry}, heta_{l}$	stratified angle around upper perimeter of the tube, rad
κ	Boltzmann's constant, J/K

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.3

μ	dynamic viscosity, kg/ms
μ	ion mobility, m^2/Vs
μ	permeability, N/A^2
μ_o	permeability of the vacuum (4 π .10 ⁻⁷), N/A ²
μ_s	relative permeability
ν	kinematic viscosity, m^2/s
ξ	factor defined by Equation 2.39
ρ	density, kg/m^3
$ ho_e$	charge density, C/m^3
$ ho_{tp}$	two phase density defined by Equation 2.94, $\rm kg/m^3$
σ	surface tension, N/m
σ_e	electric conductivity, S/m
Τ	electrical relaxation time, s
τ	interfacial shear stress, N/m^2
$ au_w$	wall shear stress, N/m^2
ϕ	two phase pressure drop multiplier
$\phi_{l,lt}$	parameter defined by Equation 2.51

Subscripts

annular	annular flow
avg	average
b, bottom	bottom
d	dimensionless
f	friction
i	interface, interfacial
in	inlet

l, L	liquid
ld	liquid and dimensionless
lo	liquid flowing alone through the channel
lt	one phase is laminar and the other phase is turbulent
min	minimum
0	reference
out	outlet
stratified-wavy	stratified-wavy flow
t, top	top
tp	two phase
tt	both phases are turbulent
v, V	vapour
vd	vapour and dimensionless
vo	vapour flowing alone through the channel
w	water

Chapter 1

Introduction

1.1 Heat Transfer Control

The ability to control heat transfer is becoming increasingly important in many industrial and process applications. For example, control of the proper cooling rate during the quenching process of liquid metals is important to control the crystallization and phase transformation and thus the development of new materials. In refrigeration systems, several capacity control schemes (e.g. hot gas by-pass, intermittent running of the compressor, and suction gas throttling) are used to control the load (Yaqub and Zubair [2001]). However, these schemes decrease the coefficient of performance (COP) of the refrigerator significantly due to the corresponding decrease of the compressor efficiency. On the other hand, controlling the heat transfer in the evaporater will control the heat load without affecting the compressor efficiency. In energy storage and transport devices, heat transfer control is important to control the phase change from supercooled water to ice for practical use of ice-slurry as working fluid (Inada et al. [2001]).

Another application that requires heat transfer control is waste heat recovery

using Rankine cycles for internal combustion engines. One of the main challenges in these systems is that there is no control of the thermodynamic states of the working fluid at the inlet or the outlet of the turbine. This is mainly because of the uncontrollable variations of the temperature and mass flow of the exhaust gases which varies the heat input in the boiler. This lack of control of the the heat input leads to either lower performance (if the temperature at the exit of the turbine is higher than the condensing temperature) or low life time of turbine (due to the high moisture at the the exit of the turbine). Control of the heat transfer in the boiler will potentially provide the required control of the superheated vapor which will achieve the optimum performance and maintain the life time of turbine.

Details on additional industrial applications with heat transfer control are reviewed by Aihara [1992]. In most cases, conventional control techniques rely on quasi-steady control of the average temperature, flow rate, bypass, and thermal load of a working fluid (Aihara [1992]). A promising technique for active heat transfer control, without changing the initial flow conditions, is by using electrohydrodynamics (EHD).

Electrohydrodynamics refers to the coupling between the flow field and the high-voltage electric field. This coupling induces additional electric body force in the fluid flow which can induce secondary motions which disturb the boundary layer near the surface and/or cause two-phase redistributions, and therefore changes the heat transfer and pressure drop. The advantages of the EHD technique over other techniques include a rapid and smart control of the heat transfer by varying the applied voltage, it is non mechanical and simple in design and it can be used for gravity free applications (Yabe [1995], Seyed-Yagoobi and Bryan [1999]). Another unique advantage of the EHD technique is that by using AC (or pulse wave) applied voltages, various combinations of heat transfer and pressure drop values can be achieved by
manipulating the applied voltage waveform parameters. The EHD technique strongly depends on the working fluid involved, the heat transfer mode, and the electrode configuration. In gases, corona discharge is the primary force behind the heat transfer enhancement (Ohadi [1990], Chang and Watson [1994]). In two-phase flows, the heat transfer enhancement is due to a combination of different effects. These effects include inducing interfacial instability, inducing droplet entrainment, enhancement of convection and turbulence in both phases and phase redistribution.

1.2 Condensation

The main focus of this research is to investigate the effect of EHD on tube side convective condensation inside an annular channel. Condensation heat transfer plays an important role in many industrial applications including HVAC, power, chemical, waste heat recovery, petroleum processing and aerospace industries. Condensation refers to the change of phase from the vapour state to the liquid state when the vapour temperature is reduced below its saturation temperature. There are two distinct forms of condensation: film condensation and dropwise condensation. In film condensation, the condensate wets the surface and forms a liquid film on the surface that moves down under the influence of gravity. The thickness of the liquid film increases in the flow direction as more vapor condenses on the film. The rate of heat transfer by film condensation is determined primarily by the thickness of the condensate film, which in most cases represents the greatest portion of the thermal resistance. In dropwise condensation, the liquid does not wet the cold surface, and it will form into numerous droplets on the surface. These droplets will coalesce into large drops which are typically removed from the surface due to gravity or the shear forces caused by the vapour flow. Dropwise condensation results in much larger heat transfer coefficients than during filmwise condensation.

For tube side convective condensation, the mechanisms of heat transfer and pressure drop are mainly dependant on the prevailing two-phase flow regime. Different flow patterns may occur depending on the tube orientation, flow rates, and physical properties of the two phases (e.g. stratified flow, stratified-wavy flow, intermittent flow, annular flow, mist flow, and bubbly flow). This investigation focuses on the two phase redistributions due to EHD forces for stratified and annular flow. Annular flow occurs at high mass fluxes and is characterized by a liquid-vapour interface separating a thin annular liquid film at the heat transfer surface area from the vapour flow in the core region. The mode of heat transfer is approximately circumferentially symmetric and is convective condensation. Stratified flow occurs at low mass fluxes in a horizontal tube and the vapour condenses at the upper portion of the tube and flows downward by gravity into the bottom stratum. The heat transfer mode are film condensation for the falling condensate and convective condensation for the bottom liquid stratum in this case.

1.3 Objectives

Applying high voltage can cause phase redistributions and therefore change the heat transfer and pressure drop characteristics during tube side convective condensation (Singh et al. [1997], Gidwani et al. [2002], Sadek et al. [2006a], Laohalertdecha and Wongwises [2007] and Cotton [2009]). The main objective of this research is to investigate the applicability of using electrohydrodynamics as a heat transfer and pressure drop control technique. In particular, the use of AC/pulse electric fields by varying the different waveform parameters (e.g. waveform, frequency, amplitude and duty cycle) were investigated. The specific objectives are:

- 1. To elicit the mechanisms of condensation heat transfer and pressure drop under an applied DC voltage.
- 2. To investigate the effect of a step applied input voltage on two-phase flow pattern redistributions.
- 3. To investigate the effect of high voltage AC and pulsed applied voltage on twophase flow patterns, heat transfer and pressure drop during tube side convective condensation.
- 4. To elicit the mechanisms of condensation heat transfer enhancement and pressure drop penalty for different applied voltage waveform parameters.
- 5. To identify the range of heat transfer and pressure drop that can be achieved by controlling the voltage waveform parameters without changing the flow conditions.

1.4 Scope of Work

The thesis consists of seven chapters including this introductory chapter. An introduction to condensation with an emphasis on tube side convective condensation and a discussion of the advantages of using electrohydrodynamics to control heat transfer and pressure drop are included. The literature review is divided into two chapters. The two-phase flow fundamentals and the key models for predicting the flow patterns, heat transfer and pressure drop for tube side convective condensation are discussed in chapter two. An overview of the theoretical background of electrohydrodynamics and the previous research on the effect of EHD on two-phase flow, heat transfer and pressure drop is reviewed in chapter three with an emphasis on tube side convective condensation in circular and annular channels. The coupling between the electric field equations and the flow field equations is examined and presented in detail in this chapter. The different terms of the induced electric force, the relative importance of each term and associated assumptions and simplifications will be discussed and analyzed. The important dimensionless numbers that characterize the effect of EHD forces on two-phase flow are discussed.

The details of the experimental facilities, methodology and uncertainty analysis are presented in chapter four. The experimental results are presented and discussed in chapter five and six. The effects of DC applied high voltage on two-phase flow regime redistributions, heat transfer and pressure drop for different flow conditions are discussed in chapter five. Chapter six is divided into three sections. The general characteristics of two-phase flow heat transfer and pressure drop due to the application of both AC and pulse wave voltage are examined in section one. The transient dynamics of the two-phase flow due to the application of a step input voltage is analyzed in section two. In section three, the effect of applied pulse voltage on the two-phase flow regime redistributions, heat transfer, and pressure drop are discussed for different flow conditions and different waveform parameters. Based on the experimental results, the potential range of control of heat transfer and pressure drop is identified in chapter six. Chapter seven contains the conclusions and the recommendations for future work.

Chapter 2

Two-Phase Flow

Liquid-vapour phase change processes play an important role in many technological applications. The nearly isothermal heat transfer associated with boiling and condensation processes makes their inclusion in power and refrigeration cycles very advantageous from a thermodynamic efficiency viewpoint. In addition, boiling and condensation can achieve large heat transfer rates with small temperatures differences compared to single phase applications. Processes associated with phase-change phenomena are among the most complex transport processes encountered in engineering applications. These processes include all the complexity of single-phase convective transport (non-linearities, transition to turbulence, instabilities) and additional elements resulting from the motion of the vapour-liquid interface, non-equilibrium effects and interactions between the phases. This chapter consists of two main sections. In the first section, the two-phase fundamentals including the general definitions, nondimensional numbers and the two-phase basic equations are examined in detail. The literature review including previous research on two-phase flow patterns, heat transfer and pressure drop are discussed in the second section.

2.1 **Two-Phase Flow Fundamentals**

2.1.1 General Definitions

The different two-phase flow parameters used throughout this thesis are first reviewed and defined in this section. The total mass flow rate through the tube, \dot{m} , is equal to the sum of the mass flow rates of liquid \dot{m}_l and vapour \dot{m}_v .

$$\dot{m} = \dot{m}_l + \dot{m}_v \tag{2.1}$$

The vapour quality x is the ratio of the vapour mass flow rate to the total mass flow rate:

$$x = \frac{\dot{m}_v}{\dot{m}} \tag{2.2}$$

For a channel with cross-sectional area A, the total mass flux G, liquid mass flux G_l and vapour mass flux G_v are defined as

$$G = \frac{\dot{m}}{A}$$
 , $G_l = G(1-x)$, $G_v = Gx$ (2.3)

The cross sectional averaged void fraction α is defined as the ratio of the vapour flow cross-sectional area to the total cross-sectional area,

$$\alpha = \frac{A_v}{A} \tag{2.4}$$

From continuity, the liquid and vapour mean velocities are defined as:

$$u_{l} = \frac{\dot{m}(1-x)}{\rho_{l}A(1-\alpha)} = \frac{G(1-x)}{\rho_{l}(1-\alpha)}$$
(2.5)

$$u_v = \frac{\dot{m}x}{\rho_v A \alpha} = \frac{Gx}{\rho_v \alpha} \tag{2.6}$$

The superficial velocities of the phases j_l and j_v are defined as the liquid and vapour velocities if they would flow alone in the entire cross section.

$$j_{l} = \frac{G(1-x)}{\rho_{l}} = u_{l}(1-\alpha)$$
(2.7)

$$j_v = \frac{Gx}{\rho_v} = u_v \alpha \tag{2.8}$$

The liquid and vapour Reynolds number in a two-phase flow can be expressed as

$$Re_l = \frac{GD_h}{\mu_l} \frac{1-x}{1-\alpha} \qquad , \qquad Re_v = \frac{GD_h}{\mu_v} \frac{x}{\alpha} \qquad (2.9)$$

while the single phase liquid and vapour Nusselt numbers can be expressed as

$$Nu_l = \frac{hD_h}{k_l} \qquad , \qquad Nu_v = \frac{hD_h}{k_v} \qquad (2.10)$$

The Froude number represents the ratio of the inertia forces to the gravitational forces,

$$Fr = \frac{u^2}{gD_h} \tag{2.11}$$

and the liquid Froude number can be expressed as

$$Fr_l = \frac{G^2}{\rho_l^2 g D_h} \tag{2.12}$$

2.1.2 Void Fraction-Vapour Quality Relations

Accurate prediction of void fraction is necessary for predicting flow pattern transition, pressure drop and heat transfer. An overview of the different empirical, semiempirical, and analytical two-phase flow void fraction models can be found in the literature (e.g. Carey [1992], Jassim et al. [2008] and Dalkilic and Wongwises [2009]). Butterworth [1975] showed that several of the existing void fraction correlations can be represented in the general form

$$\alpha = \left[1 + B_B \left(\frac{1-x}{x}\right)^{n_1} \left(\frac{\rho_v}{\rho_l}\right)^{n_2} \left(\frac{\mu_l}{\mu_v}\right)^{n_3}\right]^{-1}$$
(2.13)

where the constants corresponding to the different correlations are listed in Table 2.1.

 Table 2.1: Values of constants suggested for the various void fraction correlations

Correlation or model	B_B	n_1	n_2	n_3
Homogeneous model	1	1	1	0
Zivi [1964] model	1	1	0.67	0
Wallis [1965] model	1	0.72	0.40	0.08
Lockhart and Martinelli [1949] model	0.28	0.64	0.36	0.07
Thom [1964] model	1	1	0.89	0.18
Baroczy [1965] model	1	0.74	0.65	0.13

More recent studies on developing two phase flow pattern maps for boiling (Steiner [1993], Kattan et al. [1998b]) and condensation (El-Hajal et al. [2003]) used the void fraction model by Rouhani and Axelsson [1970]. Kattan et al. [1998b] used the drift flux model of Rouhani and Axelsson [1970] because it includes the effects of mass flux and surface tension on void fraction. Steiner [1993] modified the vertical tube expression of Rouhani and Axelsson [1970] for horizontal tubes as

$$\alpha_{ra} = \frac{x}{\rho_v} \left(\left[1 + 0.12 \left(1 - x \right) \right] \left[\frac{x}{\rho_v} + \frac{1 - x}{\rho_l} \right] + \frac{1.18 \left(1 - x \right) \left[g\sigma \left(\rho_l - \rho_v \right) \right]^{0.25}}{G\rho_l^{0.5}} \right)^{-1}$$
(2.14)

For tube side convective condensation, El-Hajal et al. [2003] noted that none of the existing models accurately predict the void fraction for the entire range of reduced

pressures (defined as the actual pressure divided by the critical pressure). The authors chose the homogeneous model for the high reduced pressure range and the Rouhani and Axelsson [1970] model for low and medium range and investigated different approaches to interpolate between them. A simple logarithmic mean void fraction ($LM\varepsilon$) was found to give the best results,

$$\alpha = \frac{\alpha_h - \alpha_{ra}}{\ln\left(\frac{\alpha_h}{\alpha_{ra}}\right)} \tag{2.15}$$

2.2 Literature Review on Two-Phase Flow

This section is divided into three subsections; in the first subsection, two-phase tube side condensation flow patterns and the key models for predicting the flow patterns are presented. In the second subsection, the key models for predicting heat transfer during tube side convective condensation are reviewed, with an emphasis on the model by Thome et al. [2003]. Finally, previous theoretical and experimental models for prediction of pressure drop during tube side condensation are presented.

2.2.1 Two-Phase Flow Patterns in Horizontal Tubes

During condensation, the local heat transfer coefficient and pressure gradient can vary appreciably along the axial length of the heat exchanger and depend on how the condensate is distributed within the tube. The main flow patterns during condensation inside horizontal tubes can be classified as: stratified flow(S), stratified-wavy flow(SW), intermittent flow(I) [including plug flow and slug flow], annular flow(A), mist flow(MF) (or mist-annular), and bubbly flow(B) [Collier and Thome [1994], Carey [1992], and Dobson and Chato [1998]].

The development of the different flow patterns during condensation inside



Figure 2.1: Two-phase flow patterns in horizontal tubes, (a) condensation with high liquid fraction, (b) condensation with low liquid fraction (adopted from Collier and Thome [1994]).



Figure 2.2: Schematic diagram of convective condensation flow patterns inside horizontal tubes. horizontal tubes is shown in Figures 2.1 and 2.2. In horizontal tubes, the two factors controlling the flow patterns are gravity and vapour shear. At low mass flux, the two-phase flow pattern is stratified where gravity dominates. Here, the condensate forms primarily on the top portion of the tube and flows downward into a liquid pool which is driven axially by the vapour flow. As the mass flux is increased, the interfacial shear forces become more significant and the liquid-vapour interface becomes unstable and wavy flow is observed. As the vapour velocity increases further, the wavy flow becomes unstable and transitions will occur in two different ways. At low quality (low void fraction), slug flow is observed where the interfacial waves grow and fill the cross-sectional area at some axial locations. As condensation continues the slugs will coalesce and plug flow and bubbly flow occurs. At high quality (high void fraction), the interfacial waves begin to wash up around the tube wall but not enough to create symmetric annular film leading to wavy-annular flow (semi-annular). With further increase in the mass flux the annular film becomes symmetric and annular flow is observed. At very high mass fluxes, the shear forces can cause liquid entrainment in the vapour core leading to mist-annular flow.

Several flow pattern maps have been proposed for predicting the flow pattern transitions in horizontal tubes under both adiabatic (Baker [1954], Mandhane et al. [1974], Taitel and Dukler [1976], Weisman et al. [1979] and Steiner [1993]) and diabatic conditions during evaporation (Kattan et al. [1998a]) or condensation (Breber [1988], Sardesai et al. [1981], Soliman [1982], Soliman and Azer [1971], Tandon et al. [1982] and El-Hajal et al. [2003]). Taitel and Dukler [1976] developed an analytical model to predict the flow pattern transitions for adiabatic flow, which has also been found applicable to diabatic flows (Wattelet et al. [1994]) . The flow pattern map identifies five flow patterns: Stratified, Stratified Wavy, Annular, Intermittent (Plug and Slug), and Dispersed bubble (Figure 2.3). The flow pattern map is presented using the Martinelli parameter

$$X_{tt} = \left[\frac{(dp/dz)_l}{(dp/dz)_v}\right]^{1/2}$$
(2.16)

and a second dimensionless parameter which depends on the specific transition considered. The transition between stratified-wavy and annular flow or intermittent flow was evaluated in terms of X_{tt} and the modified Froude number F, defined as

$$F = \frac{xG}{\left(\rho_v \left(\rho_l - \rho_v\right) Dg\right)^{1/2}}$$
(2.17)

The transition between stratified flow and stratified-wavy flow was specified in terms of the parameter K, defined as

$$K = F R e_l^{0.5} \tag{2.18}$$

The transition between bubbly flow and intermittent flow was specified in terms of the parameter T, defined as

$$T = \left[\frac{-(dp/dz)_{lo}}{(\rho_l - \rho_v)g}\right]^{0.5}$$
(2.19)

where $(dp/dz)_{lo}$ is given by

$$\left(\frac{dp}{dz}\right)_{lo} = \frac{2f_l G^2 \left(1-x\right)^2}{\rho_l D}$$
(2.20)

Soliman [1982, 1983] developed a model to predict the transitions from annular to wavy flow (Soliman [1982]) and from mist to annular flow (Soliman [1983]). The resulting correlations are given by

$$Fr_{so} = 0.024 Re_l^{1.6} \left(\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right)^{1.5} \frac{1}{Ga^{0.5}} \quad \text{for} \quad Re_l \le 1250 \quad (2.21)$$

$$Fr_{so} = 1.28Re_l^{1.04} \left(\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right)^{1.5} \frac{1}{Ga^{0.5}} \quad \text{for} \quad Re_l > 1250 \quad (2.22)$$

where $Ga = gD^3/\mu^2$ is the Galileo number. Based on comparisons with data in tubes 4.8 to 25 mm in diameter, Soliman [1982] concluded that wavy flow occurred for $Fr_{so} < 7$ and annular flow for $Fr_{so} > 7$.

Steiner [1993] modified the Taitel and Dukler [1976] transition boundary from stratified flow to intermittent flow or annular flow by including the effect of surface tension. The transition boundary was modified by introducing the ratio between the Froude number and Weber number as

$$F = \left(\frac{\bar{A}_v^3}{\bar{A}^2} \frac{1}{d\bar{A}_L/d\bar{h}_L} \left[\frac{\pi^2}{25\bar{h}_L^2} \left(\frac{Fr}{We}\right)_l + 1\right]\right)^{1/2}$$
(2.23)

where

$$\left(\frac{Fr}{We}\right)_l = \frac{\sigma_l}{gD^2\rho_l} \tag{2.24}$$

The Taitel and Dukler map and Steiner map are compared in Figure 2.4. For the same Martinelli parameter, the Steiner modified Froude number is higher than that of Taitel and Dukler and hence a higher mass flux is needed to overcome the effect of surface tension and to cause transition from stratified flow to intermittent or annular flow. Based on experimental refrigerant data, Steiner [1993] reported that a Martinelli parameter X = 0.34 represented the transition boundary between intermittent flow and annular flow better than X = 1.6 given by Taitel and Dukler.

Cotton et al. [2005], using visual observations, high speed photography and transient void fraction measurements, compared the flow patterns for refrigerant R-134a to the Taitel and Dukler [1976] and Steiner [1993] flow pattern maps for both tube and annular configurations. For the same annular geometry as that used for the current experiments, Cotton et al. [2005] concluded that the Steiner model predicted



Figure 2.3: Taitel and Dukler [1976] flow pattern map for horizontal two-phase flow.



Figure 2.4: A comparison of Taitel and Dukler [1976] map (--) and Steiner [1993] map (---) (adopted from Cotton [2000]).

the transition from stratified flow to intermittent or annular flow better than the Taitel and Dukler model. Neither model was able to predict the transition between intermittent flow and annular flow. Cotton [2000] explained that for an annulus, when the liquid level reaches the bottom of the inner tube, the surface tension between the liquid and the inner tube acts to pull the interface towards the upper portion of the tube before the trough can reach the bottom of the tube. Based on this argument and the agreement with the experimental data, the author proposed that the transition from intermittent flow to annular flow will occur when the liquid level touches the bottom of the inner tube acts to a Martinelli parameter of X = 0.89.

To better identify the flow patterns at different mass velocities and to make the flow pattern map a more useful engineering tool, Kattan et al. [1998a] converted the axes of the Taitel and Dukler and Steiner flow pattern maps to mass flux G and vapour quality x as

$$x = \left\{ X_{tt}^{1.143} \left(\frac{\rho_l}{\rho_v} \right)^{0.571} \left(\frac{\mu_v}{\mu_l} \right)^{0.143} + 1 \right\}^{-1}$$
(2.25)

By extracting the mass flux from the transition equations, the transition boundaries from stratified wavy to annular flow for Taitel and Dukler [1976], Steiner [1993] and Kattan et al. [1998a] maps for horizontal tubes can be represented respectively as

$$G = \left(\frac{\bar{A}_v^3 \rho_v \left(\rho_l - \rho_v\right) Dg \left(1 - \bar{h}_L\right)}{\bar{A}^2 \cdot d\bar{A}_L / d\bar{h}_L}\right)^{1/2} \cdot \frac{1}{x}$$
(2.26)

$$G = \left(\frac{\bar{A}_v^3 \rho_v \left(\rho_l - \rho_v\right) Dg}{\bar{A}^2 \cdot d\bar{A}_L / d\bar{h}_L} \left[\frac{\pi^2}{25\bar{h}_L^2} \left(\frac{Fr}{We}\right)_l + 1\right]\right)^{1/2} \cdot \frac{1}{x}$$
(2.27)

$$G = \left(\frac{\bar{A}_v^3 \rho_v \left(\rho_l - \rho_v\right) Dg}{\bar{A}^2 \cdot d\bar{A}_L / d\bar{h}_L} \left[\frac{\pi^2}{25\bar{h}_L^2} \left(\frac{Fr}{We}\right)_l + 1\right]\right)^{1/2} \cdot \frac{1}{x} + 50$$
(2.28)



Figure 2.5: Flow pattern map for R-134a. Hydraulic diameter 7.75 mm, $T_{sat} = 25$ °C. — Kattan et al. [1998a] transition lines, -- Steiner [1993] transition line.

The Kattan et al. [1998a] map was adopted from Steiner's [1993] map by adding a value of 50 kg/m²s to the mass flux for the transition line between stratified wavy flow and annular flow, Figure 2.5. This was mainly based on the good match with their experimental data.

More recently, El-Hajal et al. [2003] developed a flow pattern map for tube side convective condensation which is shown in Figure 2.6. The authors used the log-mean void fraction model (2.13-2.15) and extended the Kattan et al. [1998a] map for evaporation to condensation and proposed a new transition criterion. An important conclusion was that the transition lines between the different flow patterns are the same for both condensation and evaporation except for the transition boundary between Stratified Wavy flow (SW) and Annular flow (A) at high qualities. The tran-



Figure 2.6: El-Hajal et al. flow pattern transitions for R-134a. Hydraulic diameter= 7.7 mm, $T_{sat} = 25$ °C.

sition boundary between Stratified Wavy and Annular at high qualities represents the onset of dryout which does not occur in condensation. The procedures proposed by El-Hajal et al. [2003] to predict the local flow pattern during condensation start with evaluating the void fraction using the $LM\varepsilon$ method (2.13-2.15). The flow pattern geometrical parameters shown in Figure 2.7 can then be calculated as

$$A_{l} = A(1 - \alpha), \quad A_{v} = A\alpha, \quad A_{ld} = \frac{A_{l}}{D^{2}}, \quad A_{vd} = \frac{A_{v}}{D^{2}}$$
 (2.29)

$$\theta_{strat} = 2\pi - 2 \left\{ \begin{aligned} \pi \left(1 - \alpha\right) + \left(1.5\pi\right)^{1/3} \left[1 - 2\left(1 - \alpha\right) + \left(1 - \alpha\right)^{1/3} - \alpha^{1/3}\right] \\ -\frac{1}{200} \left(1 - \alpha\right) \alpha \left[1 - 2\left(1 - \alpha\right)\right] \left[1 + 4\left(\left(1 - \alpha\right)^2 - \alpha^2\right)\right] \end{aligned} \right\} (2.30)$$

$$h_{ld} = 0.5 \left[1 - \cos\left(\frac{2\pi - \theta_{strat}}{2}\right) \right]$$
(2.31)



Figure 2.7: Geometrical parameters for two-phase flow in a circular tube.

$$P_{id} = \sin\left(\frac{2\pi - \theta_{strat}}{2}\right) \tag{2.32}$$

The different transition lines $(G_{wavy}, G_{strat}, G_{mist}, G_{bubbly}, x_{min} \text{ and } x_{IA})$ are evaluated as

$$G_{wavy} = \left\{ \frac{16A_{vd}^3 Dg\rho_l \rho_v}{x^2 \pi^2 \left[1 - (2h_{ld} - 1)^2\right]^{0.5}} \left[\frac{\pi^2}{25h_{ld}^2} \times \left(\frac{We}{Fr}\right)_l^{-1.023} + 1 \right] \right\}^{0.5} + 50 - 75e^{-\left(x^2 - 0.97\right)^2/x(1-x)}$$
(2.33)

$$G_{strat} = \left\{ \frac{(226.3)^2 A_{ld} A_{vd}^2 \rho_v \left(\rho_l - \rho_v\right) g\mu_l}{x^2 \left(1 - x\right) \pi^3} \right\}^{1/3} + 20x$$
(2.34)

$$G_{mist} = \left\{ \frac{7680A_{vd}^2 Dg\rho_l \rho_v}{x^2 \pi^2 \xi} \left(\frac{Fr}{We}\right)_l \right\}^{0.5}$$
(2.35)

$$G_{bubbly} = \left\{ \frac{256A_{vd}A_{ld}^2 D^{1.25} \rho_l \left(\rho_l - \rho_v\right)g}{0.3164 \left(1 - x\right)^{1.75} \pi^2 P_{id} \mu_l^{0.25}} \right\}^{1/1.75}$$
(2.36)

$$x_{IA} = \left\{ \left[0.2914 \left(\frac{\rho_v}{\rho_l} \right)^{-1/1.75} \left(\frac{\mu_l}{\mu_v} \right)^{-1/7} \right] + 1 \right\}^{-1}$$
(2.37)



Figure 2.8: Geometrical parameters for two-phase flow in a annular channel.

where

$$\left(\frac{We}{Fr}\right)_l = \frac{gD^2\rho_l}{\sigma} \tag{2.38}$$

$$\xi = \left[1.138 + 2\log\left(\frac{\pi}{1.5A_{ld}}\right) \right]^{-2}$$
(2.39)

 x_{min} is the x value at which G_{wavy} is minimum. The flow patterns were identified as: Annular flow if $G > G_{wavy}$, $G < G_{mist}$ and $x > x_{IA}$ Intermittent flow if $G > G_{wavy}$, $G < G_{mist}$ or $G < G_{bubbly}$ and $x < x_{IA}$ Stratified flow if $G_{strat} < G < G_{wavy}$ Fully stratified flow if $G < G_{strat}$ Mist flow if $G > G_{mist}$

In case of annular geometries (Figure 2.8) which is of interest here, the geometrical parameters required for the El-Hajal et al. [2003] model can be derived for a given liquid level (h_L) as (Chang et al. [1985]):

$$h_{id} = 0 \qquad \text{if} \qquad h_{ld} < \frac{D - D_e}{2D}$$

$$h_{id} = \frac{h_i}{D_e} = \left(h_l - \frac{D - D_e}{2}\right) \frac{1}{D_e} \qquad \text{if} \qquad \frac{D - D_e}{2D} \le h_{ld} \le \frac{D + D_e}{2D} \qquad (2.40)$$

$$h_{id} = 1 \qquad \text{if} \qquad h_{ld} > \frac{D + D_e}{2D}$$

$$\theta_{start} = 2\cos^{-1} \left(2h_{ld} - 1\right) \tag{2.41}$$

$$A_{ld} = \frac{A_l}{D^2} = \frac{1}{4} \left[\pi - \cos^{-1} \left(2h_{ld} - 1 \right) + \left(2h_{ld} - 1 \right) \sqrt{1 - \left(2h_{ld} - 1 \right)^2} \right] - \left\{ \frac{1}{4} \left[\pi - \cos^{-1} \left(2h_{id} - 1 \right) + \left(2h_{id} - 1 \right) \sqrt{1 - \left(2h_{id} - 1 \right)^2} \right] \frac{D_e^2}{D^2} \right\}$$
(2.42)

$$A_{vd} = \frac{A_v}{D^2} = \frac{\pi}{4} - \frac{\pi}{4} \frac{D_e^2}{D^2} - A_{ld}$$
(2.43)

$$P_{id} = \frac{P_i}{D} = \sqrt{1 - (2h_{ld} - 1)^2} - \sqrt{1 - (2h_{id} - 1)^2} \left(\frac{D_e}{D}\right)$$
(2.44)

2.2.2 Condensation Heat Transfer Inside Horizontal Tubes

The key studies of tube side convective condensation heat transfer are summarized in Tables 2.2 and 2.2. The condensation heat transfer models are classified into two main categories: (i) gravity dominated flows where the gravity forces dominate over the vapour shear forces and (ii) shear dominated flows where the vapour shear forces overcome the gravity forces.

Gravity driven flows

The gravity-dominated flow patterns include stratified, wavy, and slug flow. These flow patterns were grouped together because the dominant heat transfer mechanism is conduction across the film at the top of the tube. At low vapour velocities, gravitational forces that tend to move condensate down the tube wall are much stronger

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	Hossam
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Reference	Flow	Working fluids	Diameter	Other parameters
	pattern		(mm)	
Carpenter and Colburn [1951]	annular	steam, methanol, ethanol, toleuene, trichloroethylene	12.57	vapour velocity $< 107 \text{ m/s}$ 2 < Pr < 5
Akers and Rosson [1960]	stratified	R12, methanol	15.875	$\begin{array}{l} 0.27 < G_l < 28.5 \\ 0.95 < G_v < 393 \end{array}$
Chato [1962]	stratified	R113,	14.53	$Re_v < 35000$
Rosson and Meyers [1965]	stratified, slug stratified-wavy	acetone, methanol		
Rufer and Kezios [1966]	stratified	R22	$15.875 \\ 9.75$	$\begin{array}{l} 0.3 < x < 0.8 \\ 0.00252 < \dot{m} < 0.0189 \\ T_{sat} = 43 \end{array}$
Boyko and Kruzhilin [1967]	annular	steam	13-20	$\begin{array}{l} 0 < x < 1 \\ T_{sat} = 21 - 310 \end{array}$
Soliman et al. [1968]	annular	steam, methanol, R22 ethanol, toleuene, R113 trichloroethylene,		$\begin{array}{l} 6.1 < \text{vapour velocity} < 305 \\ 0.03 < x < 0.99 \\ 1 < Pr < 10 \end{array}$
Cavallini and Zecchin [1971]	annular	R11, R12, R21 R22, R113, R114		$7000 < Re_l < 53000$
Kosky and Staub [1971]	annular	steam	12.57	
Azer et al. [1972]	annular	R11, R12	12.7	x > 0.18
Traviss et al. [1973]	annular	R12, R22	8	$\begin{array}{l} 161.4 < G < 1532.5 \\ T_{sat} = 25 - 58 \end{array}$
Jaster and Kosky [1976]	stratified	steam	12.5	$\begin{array}{c} 12.6 < G < 45 \\ 0.05 < x < 0.6 \end{array}$
Shah [1979]	annular	refrigerants, methanol benzene, toleuene, ethanol trichloroethylene, water	7-40	$\begin{array}{l} 10.83 < G < 210.55 \\ 0 < x < 1 \\ T_{sat} = 21 - 310 \end{array}$

Table 2.2: The key studies of tube side convective condensation heat transfer

Reference	Flow pattern	Working fluids	Diameter (mm)	Other parameters
Haraguchi et al. [1994]	annular	R22, R134a, R123	12.5	$\begin{array}{c} 2.5 \leq Pr_l \leq 4.5 \\ 3000 < Re_l < 30000 \end{array}$
Tandon et al. [1995]	annular semi annular wavy	R12, R22	10	$\begin{array}{l} 175 < G < 650 \\ 0.1 < x < 0.99 \\ T_{sat} = 20 - 40 \end{array}$
Singh et al. [1996]	stratified-wavy	R134a	12.7	$ \begin{array}{l} 20 < G < 500 \\ 0.06 < x < 0.86 \\ T_{sat} = 35 \end{array} $
Moser et al. [1998]	annular	R11, R12, R22, R 134a R410a, R125	3.14-20	$\begin{vmatrix} 87 < G < 862 \\ 0.01 < x < 1 \\ T_{sat} = 22 - 52 \end{vmatrix}$
Dobson and Chato [1998]	wavy annular mist	R12, R22, R 134a near azeotropic blends	3.14-7.04	$ \begin{array}{l} 75 < G < 650 \\ 0.1 < x < 0.9 \\ T_{sat} = 35 - 60 \end{array} $
Tang et al. [2000]	annular	R22, R 134a, R410a	8.81	$ \begin{array}{l} 250 < G < 810 \\ Fr_{so} > 7 \\ T_{sat} = 35 - 45 \end{array} $
Cavallini et al. [2002]	all	R22, R 134a, R236ea R125, R410a, R32, R407c	8	$ \begin{array}{l} 100 < G < 750 \\ 0 < x < 1 \\ T_{sat} = 30 - 50 \end{array} $
Thome et al. [2003]	all	R11, R113, R12, R22, R236ea R410a, R125, R404a, R32, R134a	3.1-21.4	$\begin{array}{c} 24 < G < 1022 \\ 0.03 < x < 0.97 \end{array}$
Han et al. [2006]	annular	R22, R 134a, R410A	7.92	$ \begin{array}{ c c c c c } 94 < G < 410 \\ 0.15 < x < 0.85 \\ T_{sat} = 30 - 40 \end{array} $

Table 2.2: The key studies of tube side convective condensation heat transfer, continued

McMa Engineering than the liquid-vapour interfacial shear forces that move the condensate in the direction of the flow (Dobson and Chato [1998]). A condensate film forms on the top of the tube and grows in thickness as it flows downward around the circumference. There is typically a condensate liquid stratum at the bottom which is transported in the direction of the mean flow. Chato [1962] studied stratified flows with low vapour velocities, $Re_{vo} < 35000$. The author modified the correlation developed by Chen [1961] for falling film condensation outside a horizontal cylinder for the falling liquid film in the upper portion inside horizontal tubes as

$$Nu = \frac{hD}{k_l} = 0.55 \left[\frac{\rho_l (\rho_l - \rho_v) g h_{lv} D^3}{k_l \mu_l (T_{sat} - T_s)} \right]^{0.25}$$
(2.45)

Chato [1962] concluded that the liquid height of the bottom liquid stratum was relatively constant and had no effect on heat transfer. Later Jaster and Kosky [1976] proposed a similar correlation, but replaced the constant in Chato's correlation (0.55) by a function of void fraction to take into account the variations in the liquid level of the bottom stratum. The proposed correlation is

$$Nu = \frac{hD}{k_l} = 0.725 \left[\frac{\rho_l (\rho_l - \rho_v) g h_{lv} D^3 \alpha^3}{k_l \mu_l (T_{sat} - T_s)} \right]^{0.25}$$
(2.46)

where the void fraction was taken as (Zivi [1964])

$$\alpha = \left[1 + \frac{1 - x}{x} \left(\frac{\rho_l}{\rho_v}\right)^{2/3}\right]^{-1} \tag{2.47}$$

Chato [1962] and Jaster and Kosky [1976] neglected the heat transfer to the bottom liquid stratum compared to the heat transfer to the falling liquid condensate. This assumption is valid only at low mass fluxes (Dobson and Chato [1998]), since the convective heat transfer at the bottom liquid stratum can be significant at high mass fluxes. Rosson and Meyers [1965] developed a correlation to predict heat transfer for stratified, wavy and slug flows by taking into account both heat transfer to the falling liquid condensate and the bottom liquid stratum as

$$Nu = \beta N u_{top} + (1 - \beta) N u_{bottom}$$
(2.48)

where Nu_{top} was evaluated as

$$Nu_{top} = 0.31 Re_v^{0.12} \left[\frac{\rho_l (\rho_l - \rho_v) g h_{lv} D^3}{k_l \mu_l (T_{sat} - T_s)} \right]^{0.25}$$
(2.49)

The heat transfer at the bottom section was modeled using a heat and momentum transfer analogy and expressed as

$$Nu_{bottom} = \frac{\phi_{l,li}\sqrt{8Re_l}}{5\left[1 + \frac{ln\left(1 + 5Pr_l\right)}{Pr_l}\right]}$$
(2.50)

where

$$\phi_{l,lt} = \sqrt{1 + \frac{1}{X_{lt}} + \frac{12}{X_{lt}^2}} \tag{2.51}$$

and X_{tt} is the laminar-turbulent lockhart Martinelli parameter. β is the fraction of the circumference of the tube over which filmwise condensation occurred and can be estimated as

$$\beta = Re_v^{0.1} \qquad \text{if} \qquad \frac{Re_v^{0.6}Re_l^{0.5}}{Ga} < 6.4 \times 10^{-5}$$
$$\beta = \frac{1.74 \times 10^{-5}Ga}{\sqrt{Re_vRe_l}} \qquad \text{if} \qquad \frac{Re_v^{0.6}Re_l^{0.5}}{Ga} > 6.4 \times 10^{-5} \qquad (2.52)$$

Shear dominated flow (Annular)

In fully developed annular flow, there is a thin condensate annulus over the entire tube wall, while the vapour phase flows in the central core, and heat transfer is governed by vapour shear and turbulence. There are three main approaches to predict heat transfer in shear dominated annular flow: (i) shear-based correlations, (ii) boundary layer based correlations, and (iii) two-phase multiplier based correlations. The latter is the most common approach and it consists of multiplying single phase turbulent flow Nusselt numbers by a suitable two-phase multiplier. The single-phase heat transfer coefficients are typically predicted by the Dittus and Boelter [1930] correlation,

$$Nu_l = 0.023 Re_l^{0.8} Pr_l^m \tag{2.53}$$

where m is a constant between 0.3 and 0.4. Several two-phase multiplier-based condensing correlations are available, including those of Akers et al. [1959], Boyko and Kruzhilin [1967], Cavallini and Zecchin [1974], and Shah [1979]. The most common correlation using a two-phase multiplier is that of Shah [1979], where

$$Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[1 + \frac{3.8}{p_{red}^{0.38}} \left(\frac{x}{1-x} \right)^{0.76} \right]$$
(2.54)

and p_{red} is the reduced pressure which is the ratio of the pressure to the critical pressure. The bracketed term is the two-phase multiplier which approaches unity as x approaches 0.

Cavallini and Zecchin [1974] determined the principal dimensionless groups that govern condensation heat transfer. The authors used a regression analysis to justify neglecting many of the groups that did not appear in their final correlation which can be expressed as:

$$Nu = 0.023 Re_l^{0.8} Pr_l^{0.33} \left\{ 2.64 \left[1 + \left(\frac{\rho_l}{\rho_v}\right)^{0.5} \left(\frac{x}{1-x}\right) \right] \right\}^{0.8}$$
(2.55)

Cavallini and Zecchin [1974] compared their correlation with experimental data of six different studies and reported an average standard deviation of 30% between the model predictions and the experimental results.

The frame work of the shear based correlations for annular flow condensation was developed by Carpenter and Colburn [1951]. Their model is based on two assumptions; (a) the condensate layer becomes turbulent at lower values of Reynolds number compared to the case without vapour shear and (b) the dominant thermal resistance to heat transfer in turbulent liquid flow occurs in the laminar sub-layer of the liquid film.

Later, Soliman et al. [1968] refined the framework established by Carpenter and Colburn [1951] and developed a semi-empirical heat transfer correlation for annular flow. The authors implemented improved correlations for calculating the wall shear stress and the frictional pressure drop. Neglecting the gravitational term (which is appropriate for horizontal flow), the correlation can be written as

$$Nu = 0.036 Re_{lo} Pr_l^{0.65} \left(\frac{\rho_l}{\rho_v}\right)^{0.5} \times \sqrt{\frac{2(0.046)x^2}{Re_v^{0.2}}} \phi_v^2 + Bo \sum_{n=1}^5 a_n \left(\frac{\rho_v}{\rho_l}\right)^{n/3}$$
(2.56)

where

$$a_{1} = x (2 - \gamma) - 1$$

$$a_{2} = 2 (1 - x)$$

$$a_{3} = 2 (\gamma - 1) (x - 1)$$

$$a_{4} = \frac{1}{x} - 3 + 2x$$

$$a_5 = \gamma \left[2 - \frac{1}{x} - x \right]$$

and γ is the ratio between the interfacial velocity and average liquid film velocity. Soliman et al. [1968] compared the predictions of their model to the experimental results for steam, R-113, R-22, ethanol, toluene, and tricholoroethylene. The agreement was correct in trend but the deviation were large. No statistical information regarding the deviations was given.

Boundary layer-based correlations are similar to shear-based correlations, except that the thermal resistance throughout the entire liquid film thickness is considered, and not just in the laminar sub-layer. For annular flow, there are many boundary layer-based correlations available, including those of Dukler [1960], Kosky and Staub [1971], Azer et al. [1972], Traviss et al. [1973], and Cavallini and Zecchin [1974]. These correlations are all very similar in approach in that they apply the momentum and heat transfer analogy to an annular flow model. The Kosky and Staub [1971] model is given by

$$h = \frac{c_{pl}(\tau \rho_l)^{0.5}}{T^+} \tag{2.57}$$

where T^+ is a dimensionless number evaluated at the vapour-liquid interface and related to the nondimensional film thickness $\delta^+(=\rho_l \delta u_z/\mu_l)$ by

$$T^+ = \delta P r_l \qquad \qquad \text{for} \qquad \delta^+ \le 5 \qquad (2.58)$$

$$T^{+} = 5Pr_{l} + 5ln \left[1 + Pr_{l} \left(\frac{\delta^{+}}{5} - 1 \right) \right] \qquad \text{for} \qquad 5 \le \delta^{+} < 30 \quad (2.59)$$

$$T^{+} = 5Pr_{l} + 5ln\left[1 + 5Pr_{l}\right] + 2.5ln\left(\frac{\delta^{+}}{30}\right) \qquad \text{for} \qquad \delta^{+} > 30 \qquad (2.60)$$

Kosky and Staub assumed a laminar sub-layer profile $(u^+ = y^+)$ for $Re_l = G(1 - x)D/\mu_l < 1000$ and Prandtl's 1/7th law $(u^+ = 8.74(y^+)^{1/7})$ for $Re_l > 1000$, resulting

in

$$\delta^+ = \frac{Re_l^{0.5}}{2}$$
 for $Re_l < 1000$ (2.61)

$$\delta^+ = 0.0504 R e_l^{7/8} \qquad \text{for} \qquad R e_l > 1000 \qquad (2.62)$$

Neglecting the effects of momentum transfer due to phase-change, the interfacial shear stress (τ) was determined by

~ =

$$\tau = \frac{\left(-dp/dz\right)D}{4} \tag{2.63}$$

where (-dp/dz) is the frictional pressure gradient along the channel evaluated according to the Friedel [1979] correlation (see Equation 2.93).

Traviss et al. [1973] applied the momentum and heat transfer analogy to an annular flow using the Von Karman universal velocity distribution to describe the liquid film, and developed

$$Nu = \frac{F(X_{tt}) Pr_l Re_l^{0.9}}{F_2}$$
(2.64)

where

$$F(X_{tt}) = 0.15 \left[X_{tt}^{-1} + 2.85 X_{tt}^{-0.476} \right]$$
(2.65)

and

$$\begin{split} F_2 &= 0.707 P r_l R e_l^{0.5} & \text{for} \quad R e_l < 50 \\ F_2 &= 5 P r_l + 5 ln \left[1 + P r_l \left(0.09636 R e_l^{0.585} - 1 \right) \right] & \text{for} \quad 50 < R e_l < 1125 \\ F_2 &= 5 P r_l + 5 ln \left[1 + 5 P r_l \right] + 2.5 ln \left(0.00313 R e_l^{0.812} \right) & \text{for} \quad R e_l > 1125 \end{split}$$

The model results were within 7% of the experimental results performed by Traviss et al. [1973].

Cavallini et al. [2002] compared the predictions of Kosky and Staub [1971] model to experimental results of several previous investigations (Cavallini et al. [2002], Dobson and Chato [1998], Tang [1997], Zhang [1998], Chitti and Anand [1996], Wijaya and Spatz [1995], Traviss et al. [1973], and Kim et al. [1996]). Deviations between experiments and predicted values are mostly within $\pm 20\%$. The discrepancies were attributed to the use of the Friedel correlation for the pressure gradient which were developed to cover all flow patterns and not specifically annular flow. Cavallini et al. [2002] developed a new correlation for the frictional pressure drop for annular flow based on the Friedel parameters but with modified numerical constants and exponents. The proposed model showed an average deviation of $\pm 10\%$ against the data bank of HCFC and HFC experimental data points.

Generalized models

Several convective condensation heat transfer models have been published in the past where each correlation is valid over a certain range of parameters. Only few condensation heat transfer models were developed to cover the entire map of two-phase flow patterns. The models by Haraguchi et al. [1994], Dobson and Chato [1998], Cavallini et al. [2002] and Thome et al. [2003] are more general and cover both the shear and gravity controlled condensation phenomena inside smooth tubes. Dobson and Chato [1998] developed two correlations to cover the entire range of two-phase flow patterns, and their validity depended on the mass flux and the modified Froude number (Fr_{so})

$$Fr_{so} = aRe_l^b \left(\frac{1 + X_{tt}^{0.039}}{X_{tt}}\right) \frac{1}{Ga^{0.5}}$$
(2.66)

where

$$a = 0.025$$
 $b = 1.59$ for $Re_l \le 1250$
 $a = 1.26$ $b = 1.04$ for $Re_l > 1250$

For annular flow $(Fr_{so} > 20 \text{ or } G > 500 \text{ kg/m}^2\text{s})$, Dobson and Chato [1998] proposed a two-phase multiplier based correlation

$$Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[1 + \frac{2.22}{X_{tt}^{0.89}} \right]$$
(2.67)

For stratified wavy flow ($Fr_{so} < 20$ and $G < 500 \text{ kg/m}^2\text{s}$), taking into account both the forced convective heat transfer in the bottom stratum and the heat transfer to the falling liquid film, Dobson and Chato [1998] proposed

$$Nu = \frac{0.23Re_v^{0.12}}{1 + 1.11X_{tt}^{0.58}} \left[\frac{GaPr_l}{Ja_l}\right]^{0.25} + \left(1 + \frac{\theta_l}{\pi}\right) Nu_{bottom}$$
(2.68)

where θ_l is the angle subtended from the top of the tube to the liquid level was evaluated by

$$\theta_l = \pi - \arccos\left(2\alpha - 1\right) \tag{2.69}$$

and

$$Nu_{bottom} = 0.0195 Re_l^{0.8} Pr_l^{0.4} \left[1.376 + \frac{c_1}{X_{tt}^{c_2}} \right]$$
(2.70)

where

$$c_1 = 4.172 + 5.48Fr_l - 1.564Fr_l^2 \quad c_2 = 1.773 + 0.169Fr_l \quad \text{for} \quad 0 < Fr_l \le 0.7$$

$$c_1 = 7.242 \qquad c_2 = 1.655 \qquad \text{for} \quad Fr_l > 0.7$$

Cavallini et al. [2002] developed a flow pattern based model, where three groups of correlations were used based on the flow pattern. For annular flow ($J_G > 2.5$), the Kosky and Staub [1971] shear-based model (Equations 2.57-2.63) was used with a modified Friedel [1979] correlation for the frictional pressure drop. For the annularstratified flow transition and stratified flow region ($J_G < 2.5$ and $X_{tt} \leq 1.6$), the heat transfer coefficient was calculated from a linear interpolation between the heat transfer coefficient for annular flow and at $J_G = 2.5$ and the heat transfer coefficient for stratified flow.

$$h_{trans} = h_{strat} + \frac{J_G}{2.5} \left(h_{ann, J_G = 2.5} - h_{strat} \right)$$
(2.71)

where h_{strat} is the heat transfer coefficient for stratified flow as

$$h_{strat} = 0.725 \left\{ 1 + 0.82 \left[\frac{1-x}{x} \right]^{0.268} \right\}^{-1} \left[\frac{k_l^3 \rho_l \left(\rho_l - \rho_v \right) g h_{lv}}{\mu_l D \delta T} \right]^{0.25} + h_l \left(1 - \frac{\theta_l}{\pi} \right)$$
(2.72)

where

$$h_l = \frac{0.023 Re_{lo}^{0.8} Pr_l^{0.4} k_l}{D} (1-x)^{0.8}$$
(2.73)

and

$$1 - \frac{\theta}{\pi} \frac{\arccos\left(2\alpha - 1\right)}{\pi}$$
$$\alpha = \left[1 + \frac{1 - x}{x} \left(\frac{\rho_v}{\rho_l}\right)^{2/3}\right]^{-1}$$

For the stratified-slug transition and the slug flow pattern region $(J_G < 2.5 \text{ and } X_{tt} > 1.6)$, the heat transfer coefficient for the transition region was calculated using (Equations 2.71-2.73), while for the slug flow pattern, the heat transfer coefficient was calculated by

$$h = h_l \left\{ 1 + 2.87 \left[x^{0.9} \left(\frac{\rho_l}{\rho_v} \right)^{0.5} \left(\frac{\mu_v}{\mu_l} \right)^{0.1} \right]^{1.4} \right\}$$
(2.74)

One of the most common used recent models tube side convective condensation is the model by Thome et al. [2003]. It is a flow pattern based model for horizontal tubes. The model predictions were compared to a very broad experimental database obtained in nine different laboratories using for fifteen different working fluids, and was tested for the following range of conditions: mass velocities from 24 to 1022 kg/(m²s), vapor qualities from 0.03 to 0.97, reduced pressures from 0.02 to 0.80 and tube internal diameters from 3.1 to 21.4 mm. Overall, the model predicted 85% of



Figure 2.9: Heat transfer modes in (a) stratified and (b) annular flow.

the refrigerant heat transfer coefficients in the database (1850 points) to within $\pm 20\%$ and 75% of the refrigerant plus hydrocarbon heat transfer coefficients in the database (2771 points) to within $\pm 20\%$. The predictions from the Thome et al. [2003] model modified for annular channels were found to be in a good agreement with the current experimental results to within $\pm 14.7\%$ which are discussed in chapter five.

The convective and film condensation modes are applied to their respective heat transfer surface areas as shown in Figure 2.9. The overall heat transfer coefficient is calculated using a weighted average as

$$h_{tp} = \frac{h_f r \theta + (2\pi - \theta) r h_c}{2\pi r}$$
(2.75)

where h_f and h_c are the film and convective heat transfer coefficients respectively, r is the inner radius of the tube and θ is the falling film angle around the top perimeter of the tube (Figure 2.9). For annular flow, $\theta = 0$ and $h_{tp} = h_c$. The stratified angle θ



Figure 2.10: Graphical representation of the liquid film thickness δ for stratified flow where $A_1 = A_2$ (a and b) and annular flow (c).

is calculated from

$$A_{l} = \frac{d^{2}}{8} \left[(2\pi - \theta) - \sin(2\pi - \theta) \right]$$
(2.76)

where the cross sectional area occupied with the liquid phase A_l is

2

$$A_l = (1 - \alpha) A \tag{2.77}$$

and A is the cross sectional area of the tube. The void fraction α is determined using the logarithmic mean void fraction using the Rouhani and Axelsson [1970] drift flux model and the homogenous model (El-Hajal et al. [2003]).

The Nusselt film (h_f) and the convective (h_c) condensation heat transfer coefficients

are calculated respectively by

$$h_f = 0.728 \left[\frac{\rho_l (\rho_l - \rho_v) g h_{lv} k_l^3}{D \mu_l (T_{sat} - T_s)} \right]^{0.25}$$
(2.78)

$$h_c = 0.003 Re_l^{0.74} Pr_l^{0.5} \frac{k_l}{\delta} f_i \tag{2.79}$$

where T_s is the average temperature of the tube wall wetted with the falling liquid condensate, and the liquid Reynolds number (Re_l) and Prandtl number (Pr_l) are

$$Re_l = \frac{4G(1-x)\delta}{(1-\alpha)\mu_l}$$
(2.80)

$$Pr_l = \frac{c_{pl}\mu_l}{k_l} \tag{2.81}$$

Assuming the area occupied with the liquid stratum, Figure 2.10a, is equivalent to the area occupied with a truncated annular ring, Figure 2.10b, of uniform thickness δ with the same angle θ , the liquid film thickness δ is calculated by

$$A(1 - \alpha) = \frac{(2\pi - \theta)}{8} \left[D^2 - (D - 2\delta)^2 \right]$$
(2.82)

In case of annular flow, 2.10c, the liquid film thickness is calculated using Equation 2.82 where $\theta = 0$. The interfacial surface roughness f_i is obtained from

$$f_i = 1 + \left[\frac{x}{1-x}\frac{\rho_l\left(1-\alpha\right)}{\rho_v\alpha}\right]^{0.5} \left[\frac{\left(\rho_l-\rho_v\right)g\delta^2}{\sigma}\right]^{0.25}$$
(2.83)

Dropwise condensation

Although dropwise condensation is not commonly observed during tube side convective condensation, applying EHD can promote pseudropwise condensation which potentially can increase the heat transfer coefficient significantly. Dropwise condensation may occur when condensation takes place on a surface that is not wetted by the condensate. Dropwise condensation, characterized by large number droplets of varying diameters on the condensing surface instead of a continuous liquid film, is one of the most effective mechanisms of heat transfer. Extremely large heat transfer coefficients can be achieved with this mechanism. In dropwise condensation, the small droplets that form at the nucleation sites on the surface grow as a result of continued condensation, coalesce into large droplets, and slide down when they reach a certain size, clearing the surface and exposing it to vapor. There is no liquid film in this case to resist heat transfer. As a result, with dropwise condensation, heat transfer coefficients can be achieved that are more than 10 times larger than those associated with film condensation (Cengel [2002]).

Dropwise condensation has been studied experimentally for a number of surfacefluid combinations. For example, Griffith [1983] recommended these simple correlations for dropwise condensation of steam on copper surfaces:

$$h_{dc} = 51104 + 2044 T_{sat} \qquad 22^{\circ}C < T_{sat} < 100^{\circ}C \qquad (2.84)$$

$$h_{dc} = 255510 \qquad \qquad 100^{\circ}C < T_{sat} \tag{2.85}$$

2.2.3 Pressure Drop Correlations for Tube Side Condensation

The pressure drop for tube side convective condensation is an important parameter for heat exchanger design and optimization, as it can significantly affect the overall energy consumption of the system. The local two-phase pressure drop gradient consists of three main components; friction, momentum and gravitational. The gravitational term is neglected in horizontal configurations and therefore the total pressure gradient can be written as

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + G^2 \frac{d}{dz} \left[\frac{\left(1-x\right)^2}{\rho_l \left(1-\alpha_v\right)} + \frac{x^2}{\rho_v \alpha_v}\right]$$
(2.86)

The momentum term contributes to the overall pressure drop due to the mass transfer that occurs at the liquid-vapour interface and its importance is dependent on the flow parameters (e.g. mass flux, heat flux and saturation temperature).

The previous research in this area can be categorized into three approaches (Quiben and Thome [2007b]): (i) empirical correlations, (ii) analytical models and (iii) phenomenological models. Quiben and Thome [2007a], Mauro et al. [2007], Ould-Didi et al. [2002], and Tribbe and Mller-Steinhagen [2000] compared some of the most widely used two-phase frictional pressure drop correlations to a large database of experimental results. The comparison showed that the best models were Gronnerud [1979], Friedel [1979], and Muller-Steinhagen and Heck [1986].

The two-phase flow frictional pressure gradient can be modeled as a function of the single phase liquid or vapor pressure gradient using a two phase multiplier as

$$\frac{dP}{dz} = \phi_l^2 \left(\frac{dP}{dz}\right)_l = \phi_v^2 \left(\frac{dP}{dz}\right)_v = \phi_{lo}^2 \left(\frac{dP}{dz}\right)_{lo} = \phi_{vo}^2 \left(\frac{dP}{dz}\right)_{vo}$$
(2.87)

where the respective single-phase pressure gradients are

$$\left(\frac{dP}{dz}\right)_{l} = \frac{2f_{l}G^{2}\left(1-x\right)^{2}}{\rho_{l}D} \qquad \left(\frac{dP}{dz}\right)_{v} = \frac{2f_{v}G^{2}x^{2}}{\rho_{v}D}$$

$$\left(\frac{dP}{dz}\right)_{lo} = \frac{2f_{lo}G^{2}}{\rho_{l}D} \qquad \left(\frac{dP}{dz}\right)_{vo} = \frac{2f_{vo}G^{2}}{\rho_{v}D}$$

$$(2.88)$$
Lockhart and Martinelli [1949] related the two-phase multipliers ϕ_l and ϕ_v as

$$\phi_l = \left(1 + \frac{C}{X} + \frac{1}{X^2}\right)^{1/2} \tag{2.89}$$

$$\phi_v = \left(1 + CX + X^2\right)^{1/2} \tag{2.90}$$

where X is the Martinelli parameter defined as

$$X = \left[\frac{(dP/dP)_l}{(dP/dP)_v}\right]^{1/2} \tag{2.91}$$

The values of the constant C were found dependent on whether the flow of each phase is laminar or turbulent as shown in Table 2.3.

The single-phase friction factors of the liquid f_l and vapour f_v and Reynolds numbers are calculated using the classical definition with their respective physical properties:

$$f_{l} = BRe_{l}^{-n}, \quad Re_{l} = \frac{G(1-x)D}{\mu_{l}}$$

$$f_{v} = BRe_{v}^{-n}, \quad Re_{v} = \frac{GxD}{\mu_{v}}$$
(2.92)

with B = 16 and n = 1 for laminar flow or B = 0.079 and n = 0.25 for turbulent flow.

The Friedel [1979] model is one of the most-widely used models in the literature, which was developed using a data base of 25000 points. The two-phase friction multiplier ϕ_{lo} in this case is given by

$$\phi_{lo}^2 = C_{F1} + \frac{3.24C_{F2}}{Fr^{0.045}We^{0.035}} \tag{2.93}$$

where

$$C_{F1} = (1-x)^2 + X^2 \left(\frac{\rho_l}{\rho_v}\right) \left(\frac{f_{vo}}{f_{lo}}\right)$$

Liquid	Gas	subscript	С
Turbulent	Turbulent	tt	20
Laminar	Turbulent	\mathbf{lt}	12
Turbulent	Laminar	tl	10
Laminar	Laminar	11	5

Table 2.3: Values of C to fit the empirical curves of Lockhart and Martinelli [1949].

$$C_{F2} = x^{0.78} (1-x)^{0.24} \left(\frac{\rho_l}{\rho_v}\right) \left(\frac{\mu_v}{\mu_l}\right) \left(1-\frac{\mu_v}{\mu_l}\right)^{0.7}$$

$$Fr = \frac{G^2}{gD\rho_{tp}^2}$$

$$We = \frac{G^2D}{\sigma\rho_{tp}^2}$$

$$\rho_{tp} = \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l}\right)^{-1}$$
(2.94)

 f_{vo} and f_{lo} are the friction factors for the total mass flowing as vapour or liquid, respectively. This method is known to work well when the ratio $\mu_l/\mu_v < 1000$, which is the case for most working fluids and operating conditions.

Gronnerud [1979] proposed a two-phase multiplier as

$$\phi_{lo} = 1 + \left(\frac{dp}{dz}\right)_{Fr} \left[\frac{\left(\frac{\rho_l}{\rho_v}\right)}{\left(\frac{\mu_l}{\mu_v}\right)^{0.25}} - 1\right]$$
(2.95)

where

$$\left(\frac{dp}{dz}\right)_{Fr} = f_{Fr} \left[x + 4\left(x^{1.8} - x^{10}f_{Fr}^{0.5}\right)\right]$$
(2.96)

The friction factor f_{Fr} was evaluated as

$$f_{Fr} = 1 for Fr_l \ge 1 (2.97)$$

$$f_{Fr} = Fr_l^{0.3} + 0.0055 \left(\ln \frac{1}{Fr_l} \right)^2 for Fr_l \le 1$$

Muller-Steinhagen and Heck [1986] interpolated between the single phase liquid pressure gradient and the single phase vapour pressure gradient and developed a correlation as

$$\left(\frac{dp}{dz}\right)_{f} = F\left(1-x\right)^{1/3} + Bx^{3}$$
(2.98)

where

$$F = A + 2(B - A)x$$
(2.99)

The factors A and B are the frictional pressure gradients for all liquid $(dp/dz)_{lo}$ and all vapour $(dp/dz)_{vo}$ flow (Equation 2.88). The authors compared their correlation with another fourteen correlations from the literature against an extensive data bank and reported that the model predicted the friction pressure gradient with reasonable accuracy.

Ould-Didi et al. [2002] compared experimental data for evaporation for five refrigerants (R-134a, R-123, R-402A, R-404A and R-502) over mass velocities from 100 to 500 kg/m²s, against seven two-phase frictional pressure drop prediction methods. The authors reported that the method by Muller-Steinhagen and Heck [1986] and that by Gronnerud [1979] were found to provide the most accurate predictions while the widely quoted method of Friedel [1979] gave the third best results.

Quiben and Thome [2007a] compared experimental results to the predictions of the three previous models: Friedel [1979], Gronnerud [1979] and Muller-Steinhagen and Heck [1986]. The comparison showed that although the three models showed reasonable agreement for some particular sets of experimental conditions, they could not predict the pressure drop over the entire range of flow conditions. Quiben and Thome [2007b] developed an analytical model to predict two-phase pressure gradients during evaporation in horizontal tubes. This model can be applied for tube side convective condensation by replacing the Wojtan et al. [2005] flow pattern map which the authors used to predict the flow pattern with another flow pattern map for condensation (e.g. El-Hajal et al. [2003]). The model consists of seven equations for seven different flow patterns; annular, slug+intermittent, stratified-wavy, slug+stratified-wavy, mist, dryout, and stratified. Mist flow and dryout are not reviewed in this section as they are only associated with boiling. The authors modeled annular, stratified-wavy and stratified flow ($x \ge x_{IA}$) as

$$(\Delta p)_{annular} = 4 (f_i)_{annular} \left(\frac{L}{D}\right) \frac{\rho_v u_v^2}{2}$$

$$(f_i)_{annular} = 0.67 \left[\frac{\delta}{D}\right]^{1.2} \left[\frac{(\rho_l - \rho_v) g\delta^2}{\sigma}\right]^{-0.4} \left[\frac{\mu_v}{\mu_l}\right]^{0.08} [We_l]^{-0.034} \qquad (2.100)$$

$$(\Delta p)_{stratified-wavy} = 4 (f_{tp})_{stratified-wavy} \left(\frac{L}{D}\right) \frac{\rho_v u_v^2}{2}$$

$$(f_{tp})_{stratified-wavy} = \frac{\theta_{dry}}{2\pi} f_v + \left(1 - \frac{\theta_{dry}}{2\pi}\right) (f_i)_{annular}$$

$$f_v = \frac{0.079}{Re_v^{0.25}} \qquad (2.101)$$

$$(\Delta p)_{startified, x \ge x_{IA}} = (\Delta p)_{stratified-wavy}$$
(2.102)

For slug+intermittent, slug+stratified-wavy, and stratified $(x < x_{IA})$, the authors used interpolation equations between single phase liquid pressure drop and annular, stratified-wavy and stratified respectively. Using the void fraction in the interpolation procedure insured the smooth transition between flow patterns.

$$(\Delta p)_{slug+intermittent} = \Delta p_{lo} \left(1 - \frac{\alpha}{\alpha_{IA}} \right)^{0.25} + \Delta p_{annular} \left(\frac{\alpha}{\alpha_{IA}} \right)^{0.25}$$
(2.103)

$$(\Delta p)_{slug+startified-wavy} = \Delta p_{lo} \left(1 - \frac{\alpha}{\alpha_{IA}} \right)^{0.25} + \Delta p_{startified-wavy} \left(\frac{\alpha}{\alpha_{IA}} \right)^{0.25}$$
(2.104)

$$(\Delta p)_{startified, x < x_{IA}} = \Delta p_{lo} \left(1 - \frac{\alpha}{\alpha_{IA}} \right)^{0.25} + \Delta p_{startified, x \ge x_{IA}} \left(\frac{\alpha}{\alpha_{IA}} \right)^{0.25}$$
(2.105)

where α_{IA} is the void fraction at the intermittent to annular transition boundary. Comparisons with the data base showed the model was able to predict 82.3% of experimental results within ±30% and 64.71% within ±20% for different flow conditions. Mauro et al. [2007] compared the experimental data, for seven refrigerants (R-134a, R-22, R-407C, R-410A, R-404A, R-417A, and R-507A) over mass velocities from 190 to 1150 kg/m²s, against Quiben and Thome [2007b], Friedel [1979], Muller-Steinhagen and Heck [1986], Gronnerud [1979], and Jung and Radermacher [1989] models. The statistical analysis showed that the method by Quiben and Thome [2007b] is able to give reliable predictions over the entire range of vapour qualities.

Chapter 3

Electrohydrodynamics

The purpose of this chapter is to present a concise introduction to the mechanical equations governing electrohydrodynamics in dielectric fluids. The conservation laws for mass and momentum are presented, followed by a derivation of the interfacial electric force. Finally, the relevant dimensionless numbers are introduced and discussed.

3.1 Governing Equations of Electrohydrodynamics

The conservation equations of a fluid influenced by an electromagnetic field are: Mass Conservation

$$\frac{\partial \rho}{\partial t} + \rho \nabla \cdot (\bar{u}) = 0 \tag{3.1}$$

Momentum Conservation

$$\rho \frac{\partial \bar{u}}{\partial t} + \rho \left(\bar{u} \cdot \nabla \right) \bar{u} = -\rho \bar{g} - \nabla P + \bar{f}_{e}^{\prime\prime\prime} + \mu \nabla^{2} \bar{u}$$
(3.2)

Energy Conservation

$$\frac{\partial T}{\partial t} + \bar{u} \cdot \nabla T = k \nabla^2 T + \frac{q_e''}{\rho c_p}$$
(3.3)

where $\bar{f}_{e}^{\prime\prime\prime}$ and $q_{e}^{\prime\prime\prime}$ are the body force per unit volume and the energy generation rate per unit volume respectively due to the presence of a magnetic or electric field. The additional body force and energy generation due to the presence of the magnetic or electric field [Equations 3.2 and 3.3] were developed by Chu [1959] as

$$\bar{f}_{e}^{\prime\prime\prime} = \bar{f}_{1} + \bar{f}_{2} - \bar{f}_{3} - \bar{f}_{4} + \bar{f}_{5} \tag{3.4}$$

$$\bar{f}_{e}^{\prime\prime\prime} = \rho_{e}\bar{E} + J_{x}\bar{B} - \frac{1}{2}E^{2}\nabla\varepsilon - \frac{1}{2}H^{2}\nabla\mu + \nabla\left[\frac{1}{2}\rho E^{2}\left(\frac{\partial\varepsilon}{\partial\rho}\right)_{T} + \frac{1}{2}\rho H^{2}\left(\frac{\partial\mu}{\partial\rho}\right)_{T}\right] \quad (3.5)$$

where,

B: magnetic flux density

H: magnetic field strength

 f_1 : force per unit volume due to space charges

 f_2 : force per unit volume due to the charge particle motion

 f_3 : force per unit volume due to dielectric property changes

 f_4 : force per unit volume due to the fluid permeability changes

 f_5 : force per unit volume due to the electrostriction and magnetostriction effects and

$$q_e^{\prime\prime\prime} = q_1^{\prime\prime\prime} + q_2^{\prime\prime\prime} + q_3^{\prime\prime\prime} \tag{3.6}$$

$$q_{e}^{\prime\prime\prime} = \left(\bar{J} - \rho_{e}\bar{\mu}\right)\left(\bar{E} + \bar{u} \times \bar{B}\right) + \nabla \cdot \left[\left(\bar{E} + \bar{u} \times \bar{B}\right) \times \left(\bar{H} - \bar{u} \times \bar{D}\right)\right] \\ + \left[\bar{E}\frac{d}{dt}\left(\frac{\bar{D}}{\rho}\right) + \bar{H}\frac{d}{dt}\left(\frac{\bar{B}}{\rho}\right)\right]\rho \qquad (3.7)$$

where,

 q_1''' : heat generation due to flow of charged particles, such as ohmic heating

 $q_2^{\prime\prime\prime}$: energy due to the polarization, such as electromagnetic hysteresis loss

 q_3''' : energy due to the displacement current and time varying magnetic fields, such as energy storage in an electromagnetic field. The dominant terms in Equations 3.5 and 3.7 depend on the working fluid and whether the flow is single phase or two phase as depicted in Table 3.1 (Chang [1998]).

Type of Fluid		Momentum terms due	Energy terms due to	Comments	
		to EMF: $\bar{f}_{c}^{\prime\prime\prime}$	EMF: $q_e^{\prime\prime\prime}$		
	Gas single	$1/2\epsilon_o (\epsilon_s - 1) \nabla E^2 +$	$\nabla \cdot \left(\bar{E} \times \bar{H} \right) + \left[\bar{E} \frac{d \left(\bar{D} / \rho \right)}{dt} + \right]$	$\mu_s = 1, \epsilon_s = 1$	
Non-	phase	$1/2\mu_o\left(\mu_s-1\right)\nabla H^2$	$\frac{\bar{H}}{dt} \frac{d\left(\bar{B}/\rho\right)}{dt}]\rho$	$\nabla \varepsilon$, $\nabla \mu \rightarrow \text{small}$	
conducting	Liquid	$1/2E^2 \left(\partial \varepsilon / \partial T\right)_\rho \bigtriangledown T$	(same as for gas)	ρ∂ε/∂ρ, ρ∂μ/∂ρ	
fluids	single phase			Vµ →small	
I→small	Gas-liquid	$-\frac{1}{2}E^2 \nabla \varepsilon - \frac{1}{2}H^2 \nabla \mu$ (for each phase)	(same as for gas	$\rho_{\psi} = \alpha_v \rho_v + (1 - \alpha_v) \rho_l$	
	two-phase	$+\nabla\left[\frac{1}{2}\rho E^2\left(\frac{\partial\epsilon}{\partial\rho}\right)_T + \frac{1}{2}\rho H^2\left(\frac{\partial\mu}{\partial\rho}\right)_T\right]$	but for each phase)	$ \epsilon_l, \mu_l \rangle > \epsilon_v, \mu_v$	
	Gas single	$ ho_c \vec{E} + \vec{J} imes \vec{B}$	$\overline{J}.\overline{E} + \nabla.(\overline{E} \times \overline{H}) +$	$ \rho_{\epsilon} \neq 0, \ \forall \epsilon, \ \forall \rightarrow 0 $	
	phase	$+\nabla\left[\frac{1}{2}\rho E^{2}\left(\frac{\partial\epsilon}{\partial\rho}\right)_{T}+\frac{1}{2}\rho H^{2}\left(\frac{\partial\mu}{\partial\rho}\right)_{T}\right]$	$\left[\bar{E}\frac{d\left(\bar{D}/\rho\right)}{dt}+\bar{H}\frac{d\left(\bar{B}/\rho\right)}{dt}\right]\rho$	$ ilde{J} = ar{u}n \pm \mu' n ar{E} - D ar{ abla} n$	
}	Liquid	$\bar{J} \times \bar{B}$	$\bar{J}^2/\sigma_e + \nabla - \cdot \left(\bar{E} \times \hat{H}\right) +$	$ ho\partial \varepsilon/\partial ho$, $ ho\partial \mu/\partial ho o 0$	
Conducting	metal single		$ ho ar{H}.d\left(ar{B}/ ho ight)/dt$	$ar{J}=\muar{E}. ho=0$	
fluids	phase			$ abla arepsilon$, $ abla \mu ightarrow 0$	
(I≠0)	Liquid	$\rho_{e}\bar{E} + \bar{J} \times \bar{B} + \frac{1}{2}\rho E^{2} \left(\frac{\partial \epsilon}{\partial \rho}\right)_{T} \nabla T$	(same as for gas	$ abla \mu, \ ho rac{\partial \varepsilon}{\partial ho}, \ ho rac{\partial \mu}{\partial ho} ightarrow 0$	
	single phase			$\rho_e = \rho_+ - \rho$	
	Gas-liquid	$-\frac{1}{2}E^2\nabla\varepsilon-\frac{1}{2}H^2\nabla\mu+$	(same as for gas	$ar{J}^* = \sigma' ar{E}^*$	
	two-phase	$\nabla \left[\frac{1}{2} \rho E^2 \left(\frac{\partial \epsilon}{\partial \rho} \right)_T + \frac{1}{2} \rho H^2 \left(\frac{\partial \mu}{\partial \rho} \right)_T \right]$	but for each phase)	$\bar{E}^* = \bar{E} + \bar{U} \times \bar{B}$	
	L	$\rho_e \vec{E} + \vec{J} \times \vec{B}$ (for each phase)	<u></u>	$\bar{H}^* = \bar{H} + \bar{U} \times \bar{D}$	

 Table 3.1: Relative importance of EHD terms, Chang [1998]

The electric and magnetic field distributions are evaluated using the Maxwell equations which represent the basic fundamental laws governing the electrostatics and electromagnetic phenomena. Maxwell equations consists of four main equations: Gauss's law

$$\varepsilon_o \nabla . D = \rho_e \tag{3.8}$$

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$$\nabla.\bar{B} = 0 \tag{3.9}$$

Faraday's law

$$\nabla \times \bar{E} = -\frac{\partial \bar{B}}{\partial t} \tag{3.10}$$

and Ampere's law

$$\nabla \times \bar{B} = \mu_o \bar{J} + \mu_o \varepsilon_o \frac{\partial E}{\partial t}$$
(3.11)

The conservation of charge states

$$\nabla.\bar{J} + \frac{\partial\rho_e}{\partial t} = 0 \tag{3.12}$$

3.2 The Electrohydrodynamic Body Force

Electrohydrodynamic applications refer to the coupling between the flow field and the high-voltage low-current electric field, where the magnetic field effects generated by the current flux can be neglected compared to the electric fields effects. In this case, the ratio of the electric energy to the magnetic energy must be greater than unity (Castellanos [1998]).

$$\frac{\varepsilon E^2}{B^2/\mu} = \frac{\varepsilon E^2}{c^2 B^2} > 1 \tag{3.13}$$

where c is the velocity of the light in the system, $c = 1/\sqrt{\varepsilon\mu}$. For simplicity, we assume that the constitutive relations, $\overline{D} = \varepsilon \overline{E}$, $\overline{B} = \mu \overline{H}$, and $\overline{J} = \sigma \overline{E}$ are valid with ε , μ , and σ considered constants. The l, τ and u are the characteristic length, time and velocity in the system respectively. Therefore the fourth Maxwell equation (Equation 3.11) can be expressed as

$$\nabla \times \bar{H} = \sigma \bar{E} + \varepsilon \frac{\partial \bar{E}}{\partial t} \tag{3.14}$$

In order to check the validity of the electrohydrodynamic assumption (i.e. Equation 3.13), two possible cases need to be considered; either the ohmic current is dominant or the displacement current is dominant.

The ohmic current is dominant

For the case in which the ohmic current is dominant, scaling of Equation 3.14 leads to (Castellanos [1998])

$$\frac{H_o}{l} \nabla^* \times \bar{H}^* \sim \sigma E_o \bar{E}^* \tag{3.15}$$

where the asterisks denote dimensionless quantities of order one. Therefore,

a

$$H_o \sim \sigma l E_o$$

nd $B_o = \mu H_o \sim \mu \sigma l E_o$ (3.16)

The ratio of the electric energy to the magnetic energy (3.13) can be evaluated by

$$\frac{\varepsilon E^2}{B^2/\mu} \sim \frac{\varepsilon E_o^2}{\left(\mu\sigma l E_o\right)^2/\mu} \\ \sim \frac{\varepsilon/\sigma}{\mu\sigma l^2} = \frac{\tau_e}{\tau_m}$$
(3.17)

where $\tau_e = (\varepsilon/\sigma)$ is the characteristic time of the relaxation of charge due to ohmic conduction and $\tau_m = (\sigma \mu l^2)$ is the characteristic time for the diffusion of the magnetic field. For refrigerants, $\mu \sim \mu_o \sim O(10^{-6})$, $\varepsilon \sim O(10^{-10})$ and $\sigma \sim O(10^{-7})$. Therefore for length scale l of $O(10^{-2})$, the ratio of the electric energy to the magnetic energy will be

$$\frac{\varepsilon E^2}{B^2/\mu} \sim O\left(10^{14}\right) \tag{3.18}$$

The displacement current is dominant

For the case in which the displacement current is dominant, scaling of Equation 3.14

leads to (Castellanos [1998])

$$\frac{H_o}{l} \nabla^* \times \bar{H}^* \sim \frac{\varepsilon E_o}{\tau} \frac{dE_o^*}{dt^*} \tag{3.19}$$

$$H_o \sim \frac{\varepsilon E_o}{\tau} l$$

and $B_o = \mu H_o \sim \frac{\varepsilon E_o \mu l}{\tau} = E_o \frac{u}{c^2}$ (3.20)

where τ is the characteristic time scale for the alternating electric fields. The ratio of the electric energy to the magnetic energy (3.13) can be evaluated by

$$\frac{\varepsilon E^2}{B^2/\mu} \sim \frac{\varepsilon E_o^2}{E_o^2 \left(\frac{u}{c^2}\right)^2/\mu}$$
(3.21)

$$\sim \frac{\tau^2}{\varepsilon \mu l^2}$$
 (3.22)

For an applied sine wave voltage, τ is the time period of the waveform, $\tau \sim O(10^1 - 10^{-3})$. For applied pulse wave voltage, τ is in the same order of magnitude as the rise time of the pulse, $\tau \sim O(10^{-6})$. Therefore the minimum ratio of the electric energy to the magnetic energy will be for the applied pulse wave voltage

$$\frac{\varepsilon E^2}{B^2/\mu} \sim O\left(10^8\right) \tag{3.23}$$

In summary, from Equations 3.18 and 3.23, the terms due to the magnetic fields in Equation 3.5 and 3.7 can be neglected compared to those due to the electric fields. Therefore, the additional terms in the momentum and energy equations can be expressed as

$$\bar{f}_{e}^{\prime\prime\prime} = \rho_{e}\bar{E} - \frac{1}{2}E^{2}\nabla\varepsilon + \frac{1}{2}\nabla\left[\rho E^{2}\left(\frac{\partial\varepsilon}{\partial\rho}\right)_{T}\right]$$
(3.24)

$$q_e^{\prime\prime\prime} = \sigma_e E^2 \tag{3.25}$$

The heat generation term (Equation 3.25) is neglected when applied to a dielectric fluid (Seyed-Yagoobi and Bryan [1999] and Cotton [2000]), as the conductivity of most dielectrics ranges from approximately 10^{-10} to $10^{-16} \ \Omega^{-1}m^{-1}$. In the present investigation using refrigerant R134a, the maximum q_e''' is less than 1 W which is less than 2% of the heat load in the test section.

The three terms on the right hand side of Equation 3.24 represent the electrophoretic, dielectrophoretic, and electrostrictive components of the EHD body force respectively. The second and third terms are occasionally called the polarization forces. The electrophoretic force, also known as the Coulomb force, represents the force acting on the net free charge. For single-phase flows under the application of a direct current, the permittivity is almost constant, unless under high temperature gradient conditions, and the dielectrophoretic term. The secondary motions due to the electrophoretic force are shown schematically in Figure 3.1(a), where a charged body will move along the electric field lines to the opposed charge electrode and impart momentum to the surrounding fluid.

The dielectrophoretic term represents the force acting on a neutral dipole which is induced as a result of an inhomogeneity or spatial change in the permittivity of the dielectric medium due to non-uniform electric fields, temperature gradients or phase differences. The Electrostriction term is caused by the inhomogeneity of the fluid either due to additives or to a variation with density and (or) temperature that create local concentrations or gradients of polarization. Electrostriction term is a gradient force and it is important for force balance calculations at the interface (Panofsky and Phillips [1962], Jones [1978], Peters et al. [1980], Yabe [1995]). The



Figure 3.1: Simple representation of the electric force components on a) charged particles, b) neutral molecule, (c) motion of stratified two phase interface, and(d) bubble/droplet (adopted from Seyed-Yagoobi and Bryan [1999]).

polarization forces will induce translational motion on the neutral polarized body toward the region of higher electric field as depicted in Figure 3.1(b). In two-phase flows, the polarization forces will become significant and will draw the phase of higher permittivity (liquid) toward the region of higher electric field strength as shown in Figure 3.1(c) for an interface and in Figure 3.1(d) for droplets or/and bubbles.

3.3 Dimensional Analysis of the Electrohydrodynamic Body Force

The laminar Navier-Stokes equation for single-phase flow including the EHD forces, free convective terms (approximated by Boussinesq) and incorporating gravity with the pressure term (Chang and Watson [1994]) is

$$\rho \frac{\partial \bar{u}}{\partial t} + \rho \left(\bar{u} \cdot \nabla \right) \bar{u} = -\rho \bar{g} \beta \left(T - T_o \right) - \nabla P + \mu \nabla^2 \bar{u} + \rho_e \bar{E} - \frac{1}{2} E^2 \nabla \varepsilon + \frac{1}{2} \nabla \left[\rho E^2 \left(\frac{\partial \varepsilon}{\partial \rho} \right)_T \right]$$
(3.26)

To evaluate the effect of the different EHD forces on the fluid motion, the nondimensional form of the above equation can be derived by multiplying Equation 3.26 by $L/\rho_o U_o^2$, where L, ρ_o , and U_o are characteristic length, density, and velocity of the system. The following non-dimensional variables are defined:

$$\tilde{u} = \frac{\bar{u}}{U_o} \qquad p = \frac{P}{\rho_o U_o^2} \qquad \tau = \frac{tU_o}{L} \qquad \tilde{\nabla} = L \nabla \qquad \gamma = \frac{\rho}{\rho_o} \qquad \theta = \frac{T}{T_o}$$
$$\tilde{\eta} = \frac{\bar{E}}{E_o} \qquad Re = \frac{U_o L}{\nu} \qquad Pr = \frac{\nu}{\alpha} \qquad Ra = \frac{g\beta T_o L^3}{\nu\alpha} \qquad (3.27)$$

where γ is the density ratio (which is approximately one according to Boussinesq approximation), and the subscript *o* is used to describe a reference value.

The nondimensional form of the electrophoretic component of the electric body force is

$$\frac{L}{\rho_o U_o^2} \rho_e \bar{E} = \left(\frac{eN_o E_o L^3}{\rho_o \nu^2}\right) \left(\frac{1}{Re^2}\right) \tilde{\eta} = \left(\frac{Ehd}{Re^2}\right) \tilde{\eta}$$
(3.28)

where $\rho_e = eN_o(e \text{ is the charge and } N_o \text{ is the charged particle number density})$, and the EHD number *Ehd* is

$$Ehd = \frac{eN_oE_oL^3}{\rho_o\nu^2} \tag{3.29}$$

Assuming that the free charge density within the fluid is equal to the charge injected by the electrodes, $eN_oE_o = \frac{J}{\mu_e} = \frac{I_o}{A\mu_e}$, the EHD number can be written as

(IEEE-DEIS-EHD Technical Committee [2003])

$$Ehd = \frac{I_o L^3}{\rho_o \nu^2 \mu_e A} \tag{3.30}$$

Similarly, the non-dimensional form of the dielectrophoretic term can be represented as

$$\frac{1}{2} \frac{L}{\rho_o U_o^2} E^2 \nabla \varepsilon = \frac{1}{2} \left(\frac{\nu^2}{L^2 U_o^2} \right) \left(\frac{E_o^2 \varepsilon_o L^2}{\rho_o \nu^2} \right) \tilde{\eta}^2 \tilde{\nabla} \varepsilon_s$$
(3.31)

where ε_o is the permittivity of the free space and ε_s is the dielectric constant. The gradient of the dielectric constant can be written as

$$\tilde{\nabla}\varepsilon_s = \left(\frac{\partial\varepsilon_s}{\partial T}\right)\tilde{\nabla}T + \left(\frac{\partial\varepsilon_s}{\partial\rho}\right)\tilde{\nabla}\rho \tag{3.32}$$

Assuming a constant density (single flow), $\tilde{\nabla} \rho = 0$, therefore

$$\frac{1}{2} \left(\frac{1}{Re^2}\right) \left(\frac{T_o}{T_o}\right) \left(\frac{E_o^2 \varepsilon_o L^2 \left(\partial \varepsilon_s / \partial T\right)}{\rho_o \nu^2}\right) \tilde{\eta}^2 \tilde{\nabla} T = \left(\frac{Md}{Re^2}\right) \tilde{\eta}^2 \tilde{\nabla} \theta \qquad (3.33)$$

where the Masuda number, Md is

$$Md = \frac{E^2 \varepsilon_o L^2 T_o \left(\partial \varepsilon_s / \partial T\right)}{2\rho_o \nu^2} \tag{3.34}$$

The electrostrictive term can be represented in dimensionless form using the approximation $\rho \left(\partial \varepsilon / \partial \rho \right)_T \sim T_o \varepsilon_o d\varepsilon_s / dT$, (Landau and Lifshitz [1960]), as

$$\frac{L}{\rho_o U_o^2} \nabla \left(\frac{1}{2} E^2 \rho \left(\frac{\partial \varepsilon}{\partial \rho}\right)_T\right) = \frac{\nu^2}{L^2 U_o^2} L \nabla \left(\frac{E^2 \varepsilon_o L^2}{2\rho_o \nu^2} \frac{d\left(\varepsilon_s - 1\right)}{dT}\right) \tilde{\eta}^2 = \left(\frac{Md}{Re^2}\right) \tilde{\nabla} \left(\theta \tilde{\eta}^2\right)$$
(3.35)

The final form of the dimensionless Navier-Stokes equation can be expressed as

$$\frac{\partial \tilde{u}}{\partial \tau} + \tilde{u}.\tilde{\nabla}\tilde{u} = \frac{Gr}{Re^2}\left(\theta - 1\right) - \tilde{\nabla}p + \frac{1}{Re}\tilde{\nabla}^2\tilde{u} + \frac{Ehd\,\tilde{\eta} - Md\,\tilde{\eta}^2\tilde{\nabla}\theta + Md\,\tilde{\nabla}\left(\theta\tilde{\eta}^2\right)}{Re^2} \quad (3.36)$$

The dimensional analysis indicates that the electrohydrodynamic forces will have a significant effect on the fluid flow if $Md/Re^2 > 1$ and/or $Ehd/Re^2 > 1$. The evaluation of Md and Ehd numbers for different applied DC voltages for single phase liquid, single phase vapour and stratified two-phase flow are presented in Appendix B. The comparison between the vapour EHD number (the high voltage electrode is surrounded by the vapour phase) and the interfacial Masuda number Md_i (based on the interfacial vapour electric field strength) shows that the interfacial Masuda number is four order of magnitude higher than the vapour EHD number which implies that the the dominant force acting on the liquid-vapour interface is the dielectrophoretic force.

In two-phase flows, the dimensional analysis is complicated due to the presence of the liquid-vapour interface. The interfacial jump in the electric field intensity and the singularity of the electric and physical properties creates a problem in evaluation of the proper non-dimensional numbers as the EHD effects at the interface are typically the main subject of interest in the field of EHD in two-phase flows. The evaluation of the relative importance of each term in the EHD body force, (Equation 3.24) is important to understand the effect of EHD on two-phase flow patterns, heat transfer and pressure drop. The ratio between the dielectrophoretic and electrophoretic components of the electric body force can be expressed as

$$\frac{F_D}{F_E} = \frac{\varepsilon_o E^2 \mu_e A}{2I} \nabla \varepsilon_s = \frac{\varepsilon_o E^2 \mu_e A}{2I} \frac{(\varepsilon_l - \varepsilon_v)}{L^*}$$
(3.37)

To evaluate the ratio between the two forces at the interface, L^* must be representative of the interface thickness across which the sudden change in the dielectric constant occurs. It is expected that $L^* \ll L$ (the length scale used in the Masuda and EHD numbers for single phase flow). L^* can be estimated thermodynamically from the classical mean field theory of van der Waals for liquid-vapour interfacial region properties modified by Carey and Wemhoff [2004]. Carey and Wemhoff [2004] found that the model's results are in a good agreement with the experimental results of Beysens and Robert [1987] and reported that the interfacial thickness for refrigerant R134a at 24°C is in the range of 2 to 10 nm.

As an example, the ratio of the dielectrophoretic to the electrophoretic force is estimated for an 8 kV applied voltage and 20 mA current (the maximum current value from the measurements), for a distance of 5 mm which is in the same order as the current geometry. The interfacial electric field intensity is in order of 10^6 assuming a free charge interface. Using a value of 10 nm for the interfacial thickness, the ratio of the dielectrophoretic force to the electrophoretic force is approximately 2000. This indicates that, in the case of no interfacial surface charges, the dielectrophoretic force is three orders of magnitude higher than the electrophoretic force at the interface. Thus, the dielectrophoretic force will be the dominant force that will result in a redistribution of the phases, and the electrophoretic term can be neglected when estimating the interfacial forces due to the applied electric field. The derivation of the dielectrophoretic and electrostrictive surface forces acting on the interface between two immiscible dielectric fluids is shown in detail in Appendix C. The final expression of the normal interfacial force directed from medium 1 toward medium 2 can be expressed as (Stratton [1941] and Scaife [1989]).

$$f_{e}'' = \frac{\varepsilon_{o}}{6} \left\{ \frac{1}{\varepsilon_{s,2}^{2}} \left[4\varepsilon_{s,1}^{2}\varepsilon_{s,2} - 2\varepsilon_{s,1}^{2} - 4\varepsilon_{s,1}\varepsilon_{s,2}^{2} + 2\varepsilon_{s,2}^{2} \right] E_{n,1}^{2} \right\} + \frac{\varepsilon_{o}}{6} \left\{ \left[\varepsilon_{s,2} \left(\varepsilon_{s,2} - 2 \right) - \varepsilon_{s,1} \left(\varepsilon_{s,1} - 2 \right) \right] \left(E_{t,1}^{2} \right) \right\}$$
(3.38)

where $\varepsilon_{s,1}$ and $\varepsilon_{s,2}$ are the dielectric constant of the two mediums.

3.4 Electrohydrodynamic Effects in Two-Phase Flow

The effects of high voltage electric fields on two phase flow can be divided into interfacial effects and bulk effects in the separate phases, and are summarized in Chang and Watson [1994], Yabe et al. [1995], Eames and Sabir [1997], and Laohalertdecha et al. [2007]. The effect of EHD at the interface includes inducing perturbations and waviness into the interface (Melcher [1981], Budov et al. [1987], Penev et al. [1968], and Chang [1989]), causing liquid extraction which causes thinning of the liquid phase thickness (Yabe [1991], Yamashita et al. [1991]), inducing droplet entrainment (Chang [1989]), and promoting pseudropwise condensation (Yabe [1991], Sunada et al. [1991]). For the bulk of the phases, the effect of EHD includes enhancement of convection in both phases (Chang and Watson [1994]), generation of turbulence (Yabe et al. [1995]), and enhancement of vapour bubble/liquid droplet motions (Ogata and Yabe [1993]).

The electric force acting on the liquid vapour interface can cause interfacial instabilities and force the liquid (higher permittivity) to move to regions of higher electric field which is usually occupied with vapour. This phenomenon is usually referred to as the "Liquid Extraction Phenomenon" (Yabe et al. [1982]), and plays an important role in two-phase flow and heat transfer. In condensation, with the proper electrode configuration, liquid extraction from the condensing surface will



Figure 3.2: Typical liquid extraction phenomena in stratified flow between two parallel plate electrodes (Adopted from Yabe et al. [1995]).

cause thinning of the liquid film and a decrease of the thermal resistance, leading to an increase in heat transfer. Application of a high enough voltage may cause the liquid to be completely extracted from the condensing surface and psuedodropwise condensation can be developed (Sunada et al. [1991]).

The liquid extraction phenomena for a parallel plate electrode configuration containing an initially stationary stratified liquid and vapour is illustrated in Figure 3.2. In the absence of an electric field, a static liquid-vapour interface exists due to surface tension and gravitational restoring forces. Applying an electric field induces a vertical electric forces acting on the interface against the restoring forces. An infinitesimal rise in the liquid level will cause a further increase in the interfacial forces due to the smaller distance between the interface and the high voltage electrode, forcing the liquid level to rise even more. If the electric field is high enough, the EHD forces will overcome the restoring forces and liquid will be extracted towards the upper electrode.

3.5 Review of Studies of Electrohydrodynamic Condensation

There have been several studies on EHD heat transfer enhancement during condensation, as reviewed by Allen and Karayiannis [1995], Eames and Sabir [1997], Seyed-Yagoobi and Bryan [1999] and Laohalertdecha et al. [2007]. The key investigations of EHD condensation are summarized in Table 3.2. Most previous studies on EHD condensation heat transfer have focused on the effect of high voltage electric fields on the free falling liquid film during condensation on vertical surfaces. Velkoff and Miller [1965] studied the effect of EHD on condensation of R113 on a vertical plate using a single wire electrode extended horizontally and parallel to the vertical plate, a flat screen grid electrode, and a plate electrode. The experimental results showed a heat transfer enhancement of factor 1.5 for the single wire electrode and factor 3 for the screen grid electrode. The authors related the enhancement of heat transfer to the destabilization of the interface and what they referred to as pumping of the condensate film away from the condensing surface which leads to thinning of the liquid film. Choi [1968] studied the effect of a DC radial electric field on the condensation heat transfer of Freon-113 inside a vertical tube. The results showed a significant increase of the condensation heat transfer due to the interfacial wave instabilities induced by the electric fields.

Bologa and co-workers (Bologa and Didkovsky [1977], Didkovsky and Bologa [1981], Bologa et al. [1987, 1995]) conducted extensive research on the effect of high voltage electric fields on film condensation heat transfer on vertical plates and tubes. The authors examined the effect of the electric field strength, working fluid, noncondensable gases and physical properties of the two phases on the degree of the heat transfer enhancement. A twenty fold increase in the heat transfer was obtained -

Orientation	Reference	System Description	Enhancement	Notes
			ratio (max)	
	Gerstmann &	Condensation of R 113 on vertical plate using a wire	NA	1
	Choi (1962)	electrode suspended in front of the plate.		
	Velkoff & Miller	Condensation of R113 on vertical plates with	2.5	1,2
	(1965)	different geometric configuration of the electric field.		
	Holmes &	Condensation of Freon 114 on a grounded inclined	10	2,4
	Chapman (1970)	plate, with A.C electric field (60 Hz) by applying a		
		voltage (up to 60 kV) to a second plate.		
Flat	Didkovski &	Film condensation of pure vapour (ether, Freon-113,	20	1,2,5,
Plates	Bologa	hexane) on vertical plates and tubes in an electric		6
	(1981)	field with different strength and frequency.		
	Yabe et. al	Film condensation of R 113 on vertical plate with	2.24	4
	(1982)	wire electrode stretched horizontally and parallel to		
		the plate.		
	Trommelmans	Condensation of R11, R113, and R114 on the	2	1,5
	et al. [1985]	underside of horizontal plate.		
	Bologa et al.	Film condensation of vapour-gas mixture on vertical	7	1,2
	(1987)	plates in presence of non-condensable gases.		
	Choi & Reynolds	Condensation of R113 inside vertical pipes with	2	1,5,6
	(1965)	coaxial electrode.		
	Choi (1968)			
	Didkovski &	See vertical plates		
	Bologa (1981)			
	Smirnov &	Condensation of R113 & diethylether inside pipes	3.6	2,5
	Lunev (1978)	with coaxial electrode.		
Vertical	Joos &	In-tube condensation of R113 with coaxial concentric	1.6	1,3,4
Pipes	Snaddon (1985)	and eccentric rod electrode (0-45 kV d.c).		
	Yabe et al.	Pseudo dropwise external condensation of R113 with	4.5	1,6
	(1986)	helical electrodes.		
	Yamashita et al.	External condensation of R114 & n-perfluorohexane	6	1,2
	(1991)	on a vertical bundle with combined lattice/helical wire		
		electrode system.		
	Wawzyniak &	External condensation of Freon 113 with the	3.5	4
	Seyed-Yagoobi	electrodes soldered to rods attached to the condensing		
	(1996)	surface.		
	Cheung et. al	External condensation of R134a on vertical and	7.2	4
	(1999)	horizontal pipes with helical electrode.		
	Wu et al.	External condensation of R134a for a seven-tube	4.35	4
	(2003)	bundle heat exchanger. Two wire-grid electrodes were		
		used, one surrounded each tube and the other		
		surrounded all seven tubes		

Table 3.2: Review of selected research on EHD condensation

Orientation	Reference	System Description	Enhancement	Notes
			ratio (max)	
	Seth & Lee	External condensation of R 113 (in presence of	1.6	1
	(1974)	noncondensable gas) on a vertical tube		
	Cooper &	In-tube condensation of R12 & R114 using different	2.9	1,6
	Allen (1984)	electrode geometries.		
	Trommelmans &	External condensation of R11,R113 & R114 with	1.1	6,7
	Berghmans (1986)	helical electrode.		
	Damianidis et al.	External condensation of R114 on tubes bundle	1.08	
	(1990)			
	Sunada et al.	Pseudo dropwise external condensation of R113	4	6,7
Horizontal	(1991)	& R123.		
Tubes	Singh et. al	In-tube condensation of R134a using different	7	4,8
	(1995)	coaxial electrodes (up to 15 kV).		
	Cheung et. al	See (vertical pipes)	5.5	
	(1999)			
	Da Silva et. al	External condensation of R134a on enhanced tubes	3	4
	(2000)	with EHD voltage (0-25 kV).		
	Chu et al.	External condensation of steam on smooth and	2.4	4
	(2000)	enhanced tubes.		
	Gidwani et. al	In-tube (smooth and corrugated) condensation of	18.8	
	(2002)	different refrigerants using concentric wire electrode		
1		(up to 18 kV).		
	Butrymowicz et. al	External condensation of R123 on smooth and	2.1	3,4
	(2002)	enhanced tubes .		
1	Wu et. al (2003)	See vertical tubes	1.38	
	Feng & Seyed-	tube side two-phase heat transfer with concentric wire		3
	Yagoobi (2003)	electrode		

Table 3.2: Review of selected research on EHD condensation (continued)

Notes

1	Applying electric fields increases surface instabilities.
2	Electric fields cause liquid spraying from the condensate film.
3	Analytical model to predict heat transfer rates.
4	Liquid extraction from the condensing surface was observed.
5	Results correlated.
6	Dropwise condensation was developed due to the application of electric fields.
7	Effect of EHD on the condensate droplets was discussed.
8	Effect of EHD on the flow patterns was discussed.

with their imposed electric field using a vertical plate electrode. The enhancement in heat transfer was attributed to the EHD effects on the vapour-film interface. This was in the form of condensate spraying into the vapour phase and the appearance of transverse waves that lead to an increase in the condensation surface and reduction of the condensate film thickness. Smirnov and Lunev [1978] experimentally and theoretically studied the enhancement condensation of heat transfer of non-polar fluid (R-113) and weakly polar fluid (diethylether) on a vertical tube using DC and AC electric fields. A correlation based on their data for predicting the heat transfer coefficient was presented. Dyakowski et al. [1982] theoretically investigated the condensate film outside vertical plates subjected to gravitational and EHD forces. A relationship for the heat transfer coefficient ratio under the action of the electric field as a function of film thickness ratio (with and without the electric field) and of the most-unstable wavelength was presented. Joos and Snaddon [1985] developed a model to predict the film condensation heat transfer enhancement due to liquid extraction neglecting the interface waviness and instabilities. It was found that the model results were in good agreement with experiments despite neglecting the instabilities and waviness. The transition from film wise condensation to drop wise condensation due to the application of EHD was studied by Sunada et al. [1991]. The authors investigated the effect of electric fields on condensation of R-123 and R-113 over a vertical smooth plate using a plate electrode. A six fold enhancement for R-123 and four fold for R-113 was reported due to the transition from film condensation to dropwise condensation.

Yabe and co-workers (Yabe et al. [1982, 1986, 1987a,b]) presented a number of experimental and theoretical studies in the field of EHD enhancement of condensation heat transfer. Liquid extraction from the condensate film on a vertical plate was observed by Yabe et al. [1982] by using one pair (or multi-pairs) of twin wire electrodes stretched horizontally and parallel to the plate. The heat transfer was enhanced by a factor of 1.3 for a single pair and 2.5 for multi-pairs. Yabe et al. [1987b] subsequently studied the effect of EHD liquid extraction phenomena on condensation heat transfer around vertical cooled tubes provided with helical wire electrodes. With this electrode configuration, the local heat transfer coefficient increased by a factor of 2.8 and 94.6% of the condensate was removed from the condensation surface.

Yabe et al. [1986] investigated the augmentation of condensation heat transfer created by EHD pseudodropwise condensation. They assumed that EHD enhancement takes place in two steps: EHD liquid extraction phenomena, followed by EHD pseudodropwise condensation on the thinned film, increasing the heat transfer even more. Yamashita et al. [1991] (as referenced by Seyed-Yagoobi and Bryan [1999]) was the first to investigate the application of electric fields in a high pressure, industrial sized condenser. A six fold increase in heat transfer was obtained using a helical electrode configuration with R-114 and C6F14. Wawzyniak and Seyed-Yagoobi [1996] extended the work of Yabe et al. [1987a] by investigating an EHD extraction phenomenon to augment condensation heat transfer using enhanced tubes. Al-Ahmadi and Al-Dadah [2002] proposed a new set of correlations for estimating the heat transfer coefficient of condensation heat transfer using EHD on outside and inside vertical smooth tubes. The test runs were done with various electrode systems and various refrigerants.

A comparison of the enhancement ratio (ratio between the heat transfer coefficient with EHD and that without EHD) versus applied electric field strength for different experiments is shown in Figure 3.3. The main enhancement mechanism was attributed to the EHD extraction phenomenon, except for the studies by Sunada et al. [1991] which dealt with EHD psuedodropwise condensation, and Yabe et al. [1986] and Yamashita et al. [1991] which studied the combination of EHD extraction phenomenon and EHD psuedodropwise condensation. As seen in the figure, the combination of EHD extraction phenomenon with EHD psuedodropwise condensation gives the highest enhancement ratio.



Figure 3.3: Comparison of various studies of EHD enhancement of condensation heat transfer [Wawzyniak et al., 1996]. \circ Velkoff and Miller [1965], \triangle Choi [1968], \Box Cooper and Allen [1984], \times Sunada et al. [1991], \diamond Yabe et al. [1985], - Yabe et al. [1986], + Yamashita et al. [1991], \blacksquare Wawzyniak et al. Turbo CII [1996], \blacklozenge Wawzyniak et al. (Turbo CII w/EHD - smooth w/o EHD) [1996].

There have been relatively fewer investigations on the effect of EHD on condensation heat transfer for horizontal or inclined surfaces. Holmes and Chapman [1970] studied condensation of Freon 114 on a grounded inclined flat plate with a non-uniform A.C. electric field produced by applying a voltage to a second plate placed above the first at various angles. For high voltage electric fields, an increase in the heat transfer coefficients by a factor as high as ten was obtained. From visual observations, the authors speculated that the enhancement was due to surface insta-

bilities and liquid entrainment. Cooper [1992] used different electrode configurations to investigate the EHD enhanced condensation of Freon 114 and Freon12 on a single horizontal tube (either smooth or integrally finned). The authors reported a heat transfer enhancement by a factor up to 2.5, and that the enhancement was mainly due to the thinning of the condensate film and the interface instabilities due to the EHD. The effect of EHD on condensation on the underside of a horizontal plate was investigated by Trommelmans et al. [1985] by applying an electric field perpendicular to the plate using a mesh electrode parallel to the horizontal condensing surface. The authors developed, based on the experimental results, a correlation to predict the heat transfer enhancement in terms of the most unstable wavelength of the electrohydrodynamical instability of the liquid-vapour interface. In their second paper, Trommelmans and Berghmans [1986] extended the theoretical model to analyze the film condensation outside horizontal tubes. They found that the actual form of the interface should be included in the mathematical model. Bologa et al. [1996] experimentally and theoretically investigated the influence of the physical mechanisms of electric field on the condensation heat and mass transfer processes. The heat removal and inter-phase heat transfer in vapour and gas-liquid dispersed systems were discussed. Based on the experimental data, correlations were proposed for predicting the heat transfer coefficient. Yang et al. [1996] studied the EHD coupled heat transfer systems using a double electrode cylinder heat transfer model. Freon-11 was used as the working fluid. The results showed that the electrical field had a significant effect on both boiling and condensation heat transfer.

Using a helical electrode, Cheung et al. [1999] investigated EHD enhancement of external condensation of R-134a over horizontal and vertical smooth tubes. The effects of heat flux, electrode gaps and applied electric field potential were conducted and discussed. A heat transfer enhancement up to 5.5 fold and 7.2 fold for horizon-

tal and vertical tubes was observed, which was attributed to the liquid extraction phenomenon which caused the thinning of the liquid film thickness. Da Silva et al. [2000] presented work on the EHD enhancement of external condensation of refrigerant R-134a on horizontal single enhanced tubes. Experiments were performed on two types of commercially enhanced tubes. The tests were conducted at the EHD voltage ranging between 0 and 25 kV, saturation temperature ranging between 10 and 40°C, and heat flux ranging between 10 to 40 kW/(m^2 K). The result showed that the designed electrode worked well on enhanced tubes, and was able to substantially improve the external condensation heat transfer coefficient. Butrymowicz et al. [2002b] investigated the enhancement of condensation heat transfer of a horizontal finned tube with rod and mesh electrodes placed beneath the tube using HCF-123. As expected, the application of EHD for the new tube electrode configuration increased the heat transfer coefficient (from 27% to 110%), depending on the electrode potential. In their second paper, Butrymowicz et al. [2002a] presented a short stateof-the-art review on the passive and active enhancement of condensation heat transfer techniques. Particular attention was paid to methods concerning the augmentation of the condensate drainage. The method of condensate drainage enhancement by using a drainage strip was presented and a novel EHD technique was presented.

More recently, there have been several studies on the effect of EHD on tube side convective condensation heat transfer for smooth tubes (Table 3.3). Singh et al. [1995] examined different electrode configurations and concluded that a co-axial wire electrode performed more efficiently than a helical electrode. The authors reported that the maximum heat transfer enhancement ratio was 6.4 for the optimum electrode design and decreased with an increase of mass flux and quality. However, it was found that the EHD introduced an additional pressure drop penalty due to the friction losses. An experimental study of EHD-enhanced tube side condensation of alternative refrigerants (R-134a, R-404a, and R-407c) in both smooth and corrugated tubes was conducted by Gidwani et al. [2002]. The authors reported that by using a concentric wire electrode, as recommended by Singh et al. [1995], a maximum heat transfer enhancement of 10.6 fold with a corresponding pressure drop penalty of 13.4 fold was obtained for R404a in smooth tubes.

More recently, Sadek et al. [2006a], using a similar electrode configuration to the one used by Gidwani et al. [2002], reported that heat transfer was enhanced by a factor up to 3.2 with a similar increase in pressure drop for applied voltage of 8 kV for R-134a. The flow patterns with and without the applied electric fields were deduced based on the surface temperature profiles along the top and bottom surfaces of the tube. It was conjectured that the increased heat transfer and pressure drop were due to a flow pattern transition from stratified flow to annular flow, which was caused by the EHD forces on the liquid-vapour interface. Laohalertdecha and Wongwises [2007] studied the effect of EHD on the condensation heat transfer and pressure drop of R134a in the mass flux range of 200-600 kg/m²s using a co-axial 1.47 mm diameter cylindrical electrode. It was found that applying 2.5 kV increased both the heat transfer and pressure drop up to 30% and 25% respectively at the lowest mass flux $(200 \text{ kg/m}^2\text{s})$. An experimental study of the effect of DC and AC (62.5 Hz sinusoidal waveform) applied voltages on tube side condensation heat transfer for R-134a was conducted by Cotton [2009]. It was concluded that application of AC voltage was beneficial compared to DC as different combinations of heat transfer and pressure drop by changing the frequency and duty cycle for the same flow conditions. The differences between DC and AC were attributed to an oscillatory flow pattern, likely due to the continuous transition between flow patterns caused by the approximate "on/off" electric field applied under an AC voltage. The author predicted that a wider range of frequencies will provide greater performance controllability. All the previous

	Working	Test section	Spacing between	Mass flux	Quality	Applied EHD	Maximum	Maximum
	fluid	length (m)	two electrodes (mm)	$\rm kg/m^2s$	%	voltage (kV)	h/h_o	$\Delta P / \Delta P_o$
Singh et al.	R-134a	0.3	3.9	50-300	$x_{avg} =$	15	4.95	
[1997]				(optimum)	10-80			
Gidwani et al.	R-407c	0.3	3.9	20-500	$x_{avg} =$	18	10.6 (R-404a)	13.4
[2002]	R-404a			5	10-90			
Sadek et al.	R-134a	1.5	3.87	50-280	$x_i n =$	8	3	3
[2006]					20-85			
Laohalertdecha	R-134a	2.5	4.025	200-600	$x_{in} =$	2.5	1.3	1.25
and Wongwises [2007]					10-90			
Cotton	R-134a	1.5	3.87	75-400	$x_{in} =$	8 (DC)	3	3
[2009]					50 - 90	10 (AC)		

Table 3.3: Selected experimental investigations of EHD during tube side convective condensation in smooth tubes

investigations concluded that the effect of EHD on heat transfer and pressure drop is more significant in stratified flow (low mass flux) and decreases with an increase of mass flux or quality. The heat transfer enhancement and pressure drop penalty were attributed to the thinning of the condensate film due to liquid extraction towards the center electrode.

Cotton et al. [2003] and Cotton et al. [2005] discussed the interdependence between the applied electric fields and the two-phase flow patterns, and developed an EHD two-phase flow pattern map. The authors incorporated the induced interfacial EHD force into the Kelvin-Helmholtz instability analysis performed by Taitel and Dukler [1976], later extended by Steiner [1993] to predict the flow pattern transition from stratified to annular flow. Based on this approach, they concluded that the additional electric forces would cause transition from stratified to annular or intermittent flow for the same flow conditions.

Previous research shows that applying high voltage electric fields during tube side convective flows cause phase redistribution. With a proper design of the electrode geometry, the phase redistribution will change the heat transfer and pressure drop characteristics in the channel. None of the previous investigations examined the potential of using EHD to control heat transfer and pressure drop. The current investigation examines the potential of using AC and/or pulse wave applied voltage to control heat transfer and pressure drop during tube side convective condensation. Using AC and/or pulse wave applied voltage can provide a better control of the heat transfer compared to an applied DC voltage, because more parameters can be manipulated to achieve the required heat transfer with potentially a minimum pressure drop penalty.

Chapter 4

Experimental Methodology

In this chapter, a detailed description of the test facility, the experimental procedures, data reduction methods and experimental uncertainties are presented and discussed.

4.1 Test Facility

The present investigation utilizes an existing two-phase flow test facility most recently used by Sadek et al. [2006a] for the investigation of electrohydrodynamically augmented tube side convective condensation heat transfer. The test section was modified to study the effect of high voltage AC and pulse electric fields on tube side convective condensation for this investigation. The test facility is a closed loop charged with refrigerant R-134a as shown schematically in Figure 4.1. The main components of the loop are the gear pump, electrically heated sections, water heated section, test section, condenser, chiller, high voltage amplifier and pressurizer along with various measuring devices.



Figure 4.1: Schematic diagram of test facility.

4.1.1 The Refrigerant Loop

Liquid refrigerant, R-134a, is circulated through the loop by a gear pump located at the exit of the condenser to ensure that no vapour passes through the pump. To control the flow rate of the refrigerant, the pump is driven by a variable speed electric motor. After passing through the filter, the liquid refrigerant mass flow rate is measured using one of two turbine flow meters: an Omega FTB 502 for the range between $1.67e^{-6}$ m³/sec and $1.67e^{-5}$ m³/sec, or an Omega FTB 100 for the range between $2.17e^{-5}$ m³/sec and $2.17e^{-4}$ m³/sec. The two-phase refrigerant inlet quality to the test section can be controlled by two processes of heating: direct electric heating and water heating in a heat exchanger. The electric heating system consists of two 1.8 m long, stainless steel tubes having a 12.7 mm outer diameter and 0.25 mm wall thickness. The two electrically heated sections are connected in series to a three phase rectified 5.6 kW power supply (Miller LC261878) with a regulated current output and they are electrically insulated from the rest of the loop by ceramic insulating Swagelok Unions as shown in Figure 4.1. The power input from the electric heater to the refrigerant is determined by measuring the voltage across the tubes together with the current passing through them using digital voltage and current panel meters (Datel-DMS-20PC-2-RS and Datel-DCA5-20PC-6-DC4-RL respectively). The current measurement is facilitated by measuring the resistance across a shunt resistor which is in series with the tubes. The hot water heating system consists of a plate-type heat exchanger. The heating water flow rate and temperatures are controlled through a secondary water loop that will be detailed later. The power input to the refrigerant from the heat exchanger is determined by an energy balance on the water side of the heat exchanger. The energy extracted from the water by the refrigerant requires the measurements of the water temperature at

the inlet and outlet of the heat exchanger as well as the measurement of the water flow rate. The temperatures are measured using T-Type thermocouples and the flow rate is measured with an Omega FTB 100 turbine flow meter. The total amount of heat transferred to the refrigerant is

$$q_{PRE} = [I(V_1 + V_2)] + [\dot{m}_{w-PRE}c_{p-pw}(T_{w-PRE,i} - T_{w-PRE,o})]$$
(4.1)

The refrigerant exiting the heat exchanger is then introduced to the horizontal test section via a straight length of 0.55 m (50 diameters) tubing to achieve fully developed conditions. A 30 kW co-axial single-pass condenser is used to return the refrigerant two-phase mixture exiting the test section to its original liquid state. The condenser water side flow rate is measured using two rotameters, one for low flow rates (up to 3.066 l/min) and one for high flow rates (up to 10.22 l/min). By measuring the condenser water flow rate and the inlet and outlet temperatures, the heat extracted from the refrigerant can be determined by

$$q_{cond} = \dot{m}_{w-COND}c_{p-cw}\left(T_{w-COND,o} - T_{w-COND,i}\right)$$

$$(4.2)$$

To ensure the validity of utilizing equations 4.1 and 4.2, heat losses to the environment were minimized by insulating the piping from the surroundings with 19 mm thick foam insulation (Rubatex: R-180 FS).

A pressurizer is connected to the refrigerant line between the condenser and the gear pump to maintain the pressure of the loop within the range of 0.3 MPa to 0.9 MPa. The pressurizer is constructed from a stainless steel cylinder (15.24 cm diameter and 30.48 cm height), and is equipped with an electric heater and coiled water cooler to increase or decrease the pressure respectively. To avoid over pressurizing the loop the pressurizer is equipped with a safety pressure switch to turn off the heater at a preset pressure and a pressure relief value to release any excessive refrigerant to a recovery container (not shown in the figure).

The refrigerant temperature and pressure are monitored at various locations around the test facility as shown in Figure 4.1. T-Type thermocouples (1.6 mm diameter) are used for temperature measurements and spring-type pressure gauges are used for monitoring the pressure.

4.1.2 Test Sections

The test sections used in the current investigation are horizontally mounted singlepass counter-current shell and tube heat exchanger. The first test section (test section A), previously used by Sadek [2004] and Cotton [2000], is 1.5 m long and was used to study the general characteristics of tube side condensation with applied AC and pulse wave voltage (section 6.1). The second test section (test section B) is 30 cm long and it is used to study the effect of DC applied voltage on two-phase flow redistributions, heat transfer and pressure drop (chapter 5), response of the flow to an applied step input voltage (section 6.2), and the effect of a pulse wave voltage on tube side convective condensation (section 6.3).

The second test section is significantly shorter than the first section, resulting in smaller change of vapour quality across the test section. Thus, the advantage of this test section is that it allows a study of the effect of EHD on condensation with a nearly constant average quality across the test section (e.g. difference between inlet and outlet qualities is less than 7% for heat flux of 6 kW/m²). The details of test section A can be reviewed in Sadek et al. [2006a]. Test section B consists of a horizontally mounted single-pass counter-current shell and tube heat exchanger.

The outer water jacket is constructed from a 30 cm length of clear PVC with inner diameter of 19.05 mm and wall thickness of 3.81 mm. The inner tube containing the two phase refrigerant is a stainless steel tube with an outer diameter of 12.7 mm and an inner diameter of 10.2 mm. A 3.18 mm diameter stainless steel rod electrode was used to apply the electric field across the annular gap formed by the electrode and the surface of the inner stainless steel tube. The electrode is concentric with the inner tube forming an annulus as shown in Figure 4.3. The width of the annular gap is maintained along the entire length of the test section by three non-conducting spacers at intervals of 75 mm. The spacers were machined out of non-conducting material Delrin (dielectric constant of approximately 2.5) and occupy approximately 20% of the cross-sectional flow area. The spacer design is shown in Figures 4.2 and The outside surface temperature of the inner test-section tube is measured with 4.3. 0.5 mm O.D. T-Type thermocouples embedded in the tube at three axial locations. 75 mm apart. Each location contains four thermocouples, at the bottom, top and the two sides. The inlet and outlet temperatures of the refrigerant and the water are measured by 1.6 mm O.D. T-Type thermocouple probes for the refrigerant side and 6 mm O.D. resistance temperatures detectors (RTD) for the water side. The dimensions of the test section and the thermocouple locations are depicted in Figure 4.3. The water flow rate to the test section is measured using one of two turbine flow meters: an Omega FTB 100 for the range between 1.3 l/min and 13 l/min, or a McMillan-104 for the range between 0.2 l/min and 2 l/min.

The pressure drop across the test section is measured with an 8.62 kPa (1.25 psi) Validyne differential pressure transducer, with an accuracy of 0.25% of the full scale connected to the test section via 6.35 mm O.D. tubing. The tubing is passed through a water jacket to condense any vapour in the tubes in order to equalize the hydrostatic head on each side of the transducer.
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Figure 4.3: Schematic diagram of test section B (0.3 m long).

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A viewing section is located at the outlet of the test section for visual observations of the flow. A Fastec Imaging high speed camera is used for visualizing and recording the flow pattern. The viewing section was made of transparent quartz tube coated with an electrically transparent conductive film of tin oxide. The grounded transparent tube ensures the continuation of the electric field across the annular gap formed by the electrode connected to the high-voltage source on one side and the grounded surface of the inner tube on the other side. The electric field was established by connecting the voltage output from a high voltage amplifier (TREK 20/20C) to the center electrode. The two-phase electric field distribution within the quartz tube was compared to that in the stainless steel tube using numerical simulations. The average deviation in the electric field strength between the quartz (flow visualization section) and the stainless steel (heat exchanger test section) was found to be approximately 15%. The different parameters of the applied voltage are controlled using a function generator Tektronix AFG 3021B which is connected to the low voltage side of the high voltage amplifier. A photographic image of the function generator and the high voltage amplifier is shown in Figure 4.4. The current and voltage profiles of the applied signals are measured by using a Agilent oscilloscope.

4.1.3 The Secondary Water Loop

The secondary water loop, needed to provide the heated water to the water preheater, consists of a reservoir, centrifugal pump, submerged heater, turbine flow meter and a control system (Figure 4.1). The heater is constructed from a 15.2 cm diameter horizontal cylinder containing a 4.5 kW, 3-coil, single phase, screw plug-type heater and powered by a 600 VAC power supply. The exit water temperature is maintained at a preset temperature using a Proportional-Integral-Derivative (PID) controller in



Figure 4.4: Photographic image of the signal generator and high voltage amplifier.

conjunction with a 1.6 mm O.D. K-Type thermocouple located at the exit of the heater. The heating water flow rate through the heat exchanger is controlled by diverting a portion of the flow exiting the preheater back to the reservoir via a bypass loop as illustrated in Figure 4.1.

4.1.4 The Chilled Cooling Water Loop

The chilled water loop, needed to supply the cooling water to the test section, consists of a chiller, turbine flow meters and resistance temperatures detectors (RTD). The chiller is a compressor-based recirculating Kodiak chiller with a cooling capacity range from 3.2 kW to 7.3 kW for temperature range from 5°C to 35°C. The inlet and outlet water temperatures to the test section are measured using 6 mm O.D RTDs with uncertainty of ± 0.04 °C. The heat flux extracted from the test section can be controlled

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by controlling either or both the water flow rate and/or the outlet temperature from the chiller. The water flow rate is controlled by diverting a portion of the flow exiting the chiller back to the chiller reservoir via a bypass as illustrated in Figure 4.1. The water outlet temperature from the chiller can be controlled in the range 5° C to 35° C with resolution a of 0.5° C.

4.1.5 Data Acquisition System

The data is acquired using a National Instruments SCXI card (Figure 4.5) interfaced to a dedicated personal computer. The data acquisition card (DAQ) is a 16-bit resolution NI-PCI-6221 with 16-channel analog inputs and maximum sampling rate of 250 KS/s for single channel. For thermocouple measurements, a 32-channel SCXI-1102C module and an 8-channel optically isolated SCXI-1125 module are connected into two input channels on the DAQ board PCI-6221 (Figure 4.5). The surface thermocouples which are embedded in the surface of the grounded tube are connected to the optically isolated module to insure minimization of the electrical noise. The rest of the thermocouples are connected to the 32-channel SCXI-1102C module. Each channel on the two modules includes a gain which is set to 100 for thermocouple measurements. The output of the amplifiers of each module is connected to a multiplexer which passes the signals to the DAQ board, one channel for each module. The input voltage range of thermocouple measurements is from 0 to 2.5mV which allows a minimum resolution, for a 16-bit DAQ board, of 0.038μ V (i.e. the least significant bit (LSB) size). Converting the LSB to temperature using the NIST polynomial calibration curves, gave a resolution of the temperature measurements of 0.001°C.

A SCXI 1000 chassis with two bays (one for each module) is used to connect between the measured signals and the DAQ board. The temperatures signals are

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connected to the modules on the SCXI-1000 via two terminal blocks, SCXI-1303 for the SCXI-1202C module and SCXI-1320 for the SCXI-1125 module. The high voltage signals are connected to the SCXI-1000 via a 1180 feedthrough panel which is a 50-pin cable-extension device to extend signal lines for signals that do not require conditioning from the rear to the front of an SCXI chassis. All the signals are transferred from the chassis to the DAQ board via a SCXI-1349 shielded cable (Figure 4.5).

To avoid hardware and software cold junction compensation inconsistencies, an ice bath was constructed to the specifications outlined in ASTM STP 470 (1970). The bath was assumed isothermal at a temperature of 0.01°C, as the standard specified. The total emf was then converted to a temperature using the individual calibration curve for each thermocouple. The calibrated accuracy was within ± 0.23 °C. The current and voltage traces are recorded by connecting the "output monitors" connectors on the front panel of the high voltage amplifier to an oscilloscope. The voltage output at the voltage connector represents 1/2000th of the high voltage connected to the electrode while 0.5 V at the current connector represents 1 mA of the load current.

4.2 Experimental Procedure

The experimental procedure with and without the application of an electric field was the same except that a few extra steps were required when the high voltage was applied. The main experimental procedure consisted of the following steps:

- The required refrigerant mass flux was set by varying the gear pump speed.
- The condenser water line flow rate was adjusted using a bypass valve to achieve the desired subcooling at the preheater inlet.
- The required inlet quality was set by adjusting the current input to the electric

heaters and the temperature and flow rate of the hot water entering the water preheater.

- The test section heat flux extracted from the two-phase refrigerant mixture was set by adjusting the chiller exit temperature and the water flow rate using a bypass valve at the chiller exit.
- The electric field was applied by connecting the required voltage from the function generator to the central electrode via the high voltage amplifier.
- The system was allowed to reach steady state.

After reaching steady state, all the parameters listed in Table 4.1 are recorded with a sampling rate greater than the Nyquist frequency. The steady state values were obtained by averaging over a time period of ten seconds. All of the manual analog measurements such as rotameter readings and the electric heater current and voltage were documented. In the case of applying an electric field, all the flow parameters were adjusted until the base case (without EHD) conditions were restored. For each test, the two-phase flow pattern at the exit of the test section was recorded using the high speed imaging system.

4.3 Data Reduction

Excluding the thermal resistance of the stainless tube, which is one order of magnitude lower than that for the inner convective thermal resistance, the cross sectional thermal resistance circuit can be represented by Figure 4.6. The thermal resistances can be expressed as

$$R_{w-top} = \frac{1}{A_{top}h_{w-top}}$$
, $R_{w-bottom} = \frac{1}{A_{bottom}h_{w-bottom}}$

Symbol	Measured parameter
T_{R-Sub}	Refrigerant inlet temperature (Subcooled)
T_{R-i}	Refrigerant test section inlet temperature
T_{R-o}	Refrigerant test section outlet temperature
$T_{R-COND-i}$	Refrigerant condenser outlet temperature
$T_{R-COND-o}$	Refrigerant condenser outlet temperature
$T_{w-COND-i}$	Condenser water inlet temperature
$T_{w-COND-o}$	Condenser water outlet temperature
$T_{w-PRE-i}$	Preheat water inlet temperature
$T_{w-PRE-o}$	Preheat water outlet temperature
$T1_{s-b}$	Test section bottom surface temperature at location 1
$T1_{s-t}$	Test section top surface temperature at location 1
$T1_{s-s1}$	Test section side1 surface temperature at location 1
$T1_{s-s2}$	Test section side2 surface temperature at location 1
$T2_{s-b}$	Test section bottom surface temperature at location 2
$T2_{s-t}$	Test section top surface temperature at location 2
$T2_{s-s1}$	Test section sidel surface temperature at location 2
$T2_{s-s2}$	Test section side2 surface temperature at location 2
$T3_{s-b}$	Test section bottom surface temperature at location 3
$T3_{s-t}$	Test section top surface temperature at location 3
$T3_{s-s1}$	Test section side1 surface temperature at location 3
$T3_{s-s2}$	Test section side2 surface temperature at location 3
ΔT_{RTD}	Temperature difference of water across the test section
ΔP	Differential pressure across the test section
Q_R	Volumetric flow rate of single-phase subcooled refrigerant
Q_w	Volumetric flow rate of test section water
Q_{w-PRE}	Volumetric flow rate of preheater water
Q_{w-COND}	Volumetric flow rate of condenser water
V_i	Direct electric heating preheater voltages
I_i	Direct electric heating preheater current
	Current and voltage profiles

Table 4.1: Experimental measured parameters

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Figure 4.6: Cross sectional thermal circuit for the test section.

$$R_{ref-top} = \frac{1}{A_{top}h_{ref-top}} \quad , \quad R_{ref-bottom} = \frac{1}{A_{bottom}h_{ref-bottom}} \tag{4.3}$$

The water side heat transfer coefficient is assumed to be uniform, therefore

$$h_{w-top} = h_{w-bottom} \tag{4.4}$$

Therefore, the ratio between the overall heat input from the top part to the bottom part can be calculated by

$$\frac{q_t}{q_b} = \frac{A_{top}h_{w-top}\left(T_{s-top} - T_w\right)}{A_{bottom}h_{w-bottom}\left(T_{s-bottom} - T_w\right)} = \frac{A_{top}\left(T_{s-top} - T_w\right)}{A_{bottom}\left(T_{s-bottom} - T_w\right)}$$
(4.5)

where T_w is the average water temperature in the water jacket. The total overall heat extracted by the cooling water from the test section is determined by

$$q_w = q_t + q_b = Ah \left(T_{sat} - T_{S_{avg}} \right) = \dot{m}_w c_{pw} \left(\Delta T_{RTD} \right)$$

$$\tag{4.6}$$

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The arithmetic average of the surface temperatures, $T_{S_{avg}}$ is obtained according to

$$T_{S_{avg}} = \frac{1}{12} \left[\sum_{1}^{3} Ti_{s-t} + \sum_{1}^{3} Ti_{s-b} + \sum_{1}^{3} Ti_{s-s1} + \sum_{1}^{3} Ti_{s-s2} \right]$$
(4.7)

For annular flow and stratified flow with the liquid height less than half the tube diameter, the arithmetic average of the surface temperatures, T_{s-top} and $T_{s-bottom}$ are obtained according to

$$T_{s-top} = \frac{1}{9} \left[\sum_{1}^{3} Ti_{s-t} + \sum_{1}^{3} Ti_{s-s1} + \sum_{1}^{3} Ti_{s-s2} \right]$$
(4.8)

$$T_{s-bottom} = \frac{1}{3} \left[\sum_{1}^{3} T i_{s-b} \right]$$

$$\tag{4.9}$$

Substituting by Equation 4.5 in Equation 4.6 results in

$$q_b \left[1 + \frac{A_{top} \left(T_{s-top} - T_w \right)}{A_{bottom} \left(T_{s-bottom} - T_w \right)} \right] = Ah \left(T_{sat} - T_{S_{avg}} \right)$$
(4.10)

or

$$q_b = \frac{Ah \left(T_{sat} - T_{S_{avg}}\right)}{1 + \frac{A_{top} \left(T_{s-top} - T_w\right)}{A_{bottom} \left(T_{s-bottom} - T_w\right)}}$$
(4.11)

and

$$q_t = q_w - q_b \tag{4.12}$$

Therefore the overall top, bottom and total heat transfer coefficients are calculated as

$$h_t = \frac{q_t}{A_{top} \left(T_{sat} - T_{s-top}\right)}, \ h_b = \frac{q_b}{A_{bottom} \left(T_{sat} - T_{s-bottom}\right)}, \ h = \frac{q_w}{A \left(T_{sat} - T_{S_{avg}}\right)}$$
(4.13)

where

$$\frac{A_{top}}{A_{bottom}} = \frac{\theta}{2\pi - \theta} \tag{4.14}$$

 θ can be calculated using the correlation from El-Hajal et al. [2003] as

$$\theta_{strat} = 2\pi - 2 \left\{ \begin{aligned} \pi \left(1 - \alpha\right) + \left(1.5\pi\right)^{1/3} \left[1 - 2\left(1 - \alpha\right) + \left(1 - \alpha\right)^{1/3} - \alpha^{1/3}\right] \\ -\frac{1}{200} \left(1 - \alpha\right) \alpha \left[1 - 2\left(1 - \alpha\right)\right] \left[1 + 4\left(\left(1 - \alpha\right)^2 - \alpha^2\right)\right] \end{aligned} \right\}$$
(4.15)

The refrigerant inlet quality is obtained from the energy balance across the electric and water preheaters such that

$$x_{in} = \frac{\frac{q_{PRE}}{\dot{m}_{ref}} - c_{p-ref} \left(T_{sat} - T_{R-sub} \right)}{h_{lv,in}}$$
(4.16)

where $h_{lv,in}$ is the latent heat of evaporation at the inlet saturation temperature, c_{p_l} is the liquid specific heat, T_{R-sub} is the subcooled refrigerant temperature at the inlet of the preheaters, and q_{PRE} is the net heat input by the electric and water preheaters as determined by Equation 4.1.

The maximum pressure drop across the test section is less than 0.4% of the absolute pressure, therefore the effect of the change in the pressure between the inlet and the exit of the test section on the two-phase refrigerant properties can be neglected. The refrigerant outlet quality is calculated by applying energy balance across the test section such that

$$x_{out} = x_{in} - \frac{q_w}{\dot{m}_{ref} h_{lv}} \tag{4.17}$$



Figure 4.7: Energy balance between the power extracted from the refrigerant by the condenser and the test section condenser and the power input to the refrigerant by the preheaters and the surroundings

The refrigerant average quality across the test section can be estimated as

$$x_{avg} = \frac{x_{in} + x_{out}}{2} \tag{4.18}$$

To confirm the accuracy of the experimental measurements, the power extracted from the refrigerant by the condenser and the test section condenser was compared with the power input by the preheaters and the surroundings as

$$q_{out} = q_w + q_{COND} \tag{4.19}$$

$$q_{in} = q_{PRE} + q_{gain} \tag{4.20}$$

where q_{gain} is the heat gain from the surroundings between the condenser outlet and

the electric heater inlet, and is given by

$$q_{gain} = \dot{m}_{ref} c_{p-ref} \left(T_{R-Sub} - T_{R-COND-o} \right) \tag{4.21}$$

As shown in Figure 4.7, the energy balance was found to be within $\pm 15\%$ for all the tested conditions. Additional energy balance tests were also performed for the condenser, test section, and the preheaters separately under single phase flow conditions. The energy balance was found to be within $\pm 5\%$, $\pm 7\%$, and $\pm 20\%$ for the test section, preheaters, and condenser respectively. This indicates that the error in the overall energy balance is mainly due to that in the condenser. This result provides confidence in the efficacy of the measurements in the test section and the preheaters and in the analysis techniques used in this investigation. To confirm the repeatability of the experiments, selected tests were repeated for the same flow conditions at different times and the results were found to be within $\pm 5\%$.

4.4 Uncertainty Analysis

An uncertainty analysis of the experimental results was performed as outlined by Kline and McClintock [1953]. The uncertainty of any measured variable consists of two main components, a fixed error (Bias error) and a random error (precision error). The total uncertainty, δx_i , can be expressed as

$$\delta x_i = \left[(B_i)^2 + (P_i)^2 \right]^{0.5} \tag{4.22}$$

where B_i and P_i are the bias and precision errors respectively. The bias errors are obtained either from the suppliers for flow meters, RTD and pressure transducers or from the calibration curves for thermocouples. The various measurement devices and their associated bias uncertainties are listed in Table 4.2. The precision errors are evaluated from the experimental measurements as twice the standard deviation of 100,000 readings captured with a sampling rate of 10000 s/sec for the same condition. The maximum relative bias, precision and total uncertainties for the current experiments are listed in Table 4.3. For any dependent parameter, f, the uncertainty can be approximated from the root of the sum of the squares of the total uncertainties of each individual element which contributes to the measurement uncertainty. In this way, the total uncertainty associated with f is given by the expression,

$$\delta f = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial f}{\partial x} \delta x_i\right)^2} \tag{4.23}$$

where x_i are the independent variables on which the parameter f depends.

$$f = f(x_1, x_2, \dots, x_n) \tag{4.24}$$

The experimental uncertainties for the calculated parameters are summarized in Table 4.4. The calculations for the uncertainties associated with the main calculated parameters are explained in detail in Appendix F.

Measurement	Device	Bias uncertainty		
	(Make, type)	absolute	relative % of the reading	
Temperature	type-T	±0.23 °C	$\max ((a) 5^{\circ}C) = \pm 4.6\%$	
(calibrated)	Omega thermocouple		min (@ 60°C)= ± 0.38 %	
test section water	(RTD, T100-250-1)	± 0.057 °C	$\max ((0.9^{\circ}C)) = \pm 6.3\%$	
temperature difference	Automatic Systems Labs		min (@ $2.2^{\circ}C$) = $\pm 2.6\%$	
Refrigerant volume flow rate	FTB-502, Omega	± 0.00047 l/min	$\max (a) 0.2 l/min = \pm 0.24\%$	
(low)	turbine flow meter		min (@ 2 L/min)= $\pm 2.4\%$	
Refrigerant volume flow rate (high)	FTB-101, Omega		$\pm 0.51 \%$	
Test section-water volume flow rate (high)	turbine flow meter			
Preheater-water volume flow rate				
Test section volume water flow rate	104, McMillan		± 4 %	
	turbine flow meters			
Condenser water flow rate	F. & P. Rotameters	3.07 ± 0.06 l/min (low)		
		$10.2 \pm 0.2 \; l/min \; (high)$		
Pressure Drop	DP-15, Validyne	\pm 0.0215 kPa	max ($(a) 0.097 \text{ kPa}) = \pm 20\%$	
	pressure transducer		min (@ 2.3 kPa)= $\pm 0.9\%$	
Voltage to the electrical heater	DMS-20PC-2-RS Series		$\pm 0.05\%$	
	digital panel voltmeters			
Current to the electrical heater	DCA5-20PC-6-DC4-RL		$\pm 0.1\%$	
	digital current meter			

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Measurement	Maximum relative uncertainty		
(value at max uncertainty)	precision	bias	total
Refrigerant saturation temperature, T_{sat} (25°C)	± 0.36 %	± 0.92 %	\pm 0.99 %
Surface temperature, T_s (10°C)	$\pm 0.9 \%$	$\pm 2.3 \%$	$\pm 2.47 \%$
Refrigerant subcooled temperature, T_{R-Sub} (20°C)	$\pm 0.45 \%$	± 1.15 %	\pm 1.23 %
Test section water temp. difference, ΔT_{RTD} (0.9°C)	$\pm 4.5 \%$	± 6.33 %	\pm 7.74 %
Refrigerant volume flow rate (0.2315 l/min)	$\pm 5 \%$	± 0.2 %	± 5 %
Test section water volume flow rate (0.46 l/min)	$\pm 1.22 \%$	± 4 %	\pm 4.18 %

Table 4.3: Precision, bias and total uncertainty of various measurements

 Table 4.4:
 Uncertainties in calculated parameters

Parameter	Maximum Percentage Error		
Heat Flux (W/m^2)	$\pm 9.21~\%$		
Heat Transfer Coefficient (W/m^2K)	±12 %		
Reynolds Number	±3 %		
Average quality	±10.38 %		

Chapter 5

Tube Side Convective Condensation with an Applied DC Voltage

The main objective of this thesis is to investigate the use of AC or pulse wave voltage to control heat transfer and pressure drop during convective condensation. However, this requires a fundamental understanding of the two-phase electrohydrodynamics under the application of a DC high voltage. In this chapter, the effect of a DC applied voltage on two-phase flow redistributions, heat transfer and pressure drop during tube side convective condensation is presented. The mechanisms of heat transfer with and without EHD forces are elicited and discussed in detail. Sadek et al. [2006a] studied the effect of DC applied voltage on heat transfer and pressure drop for tube side convective condensation using a relatively long test section (1.5 m long). Due to the large changes of vapour quality along the test section and therefore two-phase flow patterns, it was not possible to elicit the effect of EHD forces on distinct flow patterns. The results presented in this chapter were obtained using the shorter test section (0.3 m). Thus, the change in quality across the test section was relatively small allowing the study the effect of EHD on condensation for specific flow patterns.

5.1 Two-Phase Flow Patterns in Horizontal Channels

Identifying the two-phase flow patterns is an essential and necessary step to predict heat transfer in two phase flow. Each flow pattern flow is associated with specific heat transfer characteristics which depend on the distribution of the two phases. The two-phase flow patterns in the current experiments are identified by using the twophase flow pattern maps and from both the flow visualization images and the tube side surface circumferential temperature distributions.

5.1.1 Without EHD Forces

In this study, a modified version of the flow map proposed by El-Hajal et al. [2003] is used to identify the flow patterns. The flow patterns deduced from flow visualization and the surface temperature measurements of the current experiments are observed to be in a good agreement with the flow pattern transitions lines of El-Hajal et al. [2003] as shown in Figure 5.1. The mass flux for the current experiments is in the range 45 kg/m²s to 160 kg/m²s, and inlet and outlet vapour qualities are 48% and 42% respectively. The relatively small change of vapour quality across the test section has no significant effect on the two-phase flow patterns. The void fraction estimated using El-Hajal et al. [2003] model is in the range 0.9 to 0.92. The two-phase flow pattern for these flow conditions is stratified wavy for G < 147 kg/m²s or annular flow for G > 147 kg/m²s (Figure 5.1). At low mass flux, the flow pattern is close to the



Figure 5.1: Flow pattern map for R-134a adopted for annular geometries. $D_o=10.92$ mm, $D_i=3.18$ mm, $T_{sat}=25$ °C. — El-Hajal et al. [2003] transition lines, \triangle stratified wavy experimental data. × electrode boundaries.

stratified transition line and moves toward the transition line between stratified wavy and annular flow as the mass flux is increased. The flow patterns are corroborated from the tube side circumferential surface temperature distribution. The average of the three axial locations of the time averaged top, side and bottom surface subcooled temperatures for the different mass fluxes are shown in Figure 5.2 with representative flow images and schematic diagrams of the flow patterns. In general, the higher the surface subcooled temperature the higher the thermal resistance at the heat transfer surface. The bottom surface subcooled temperatures are higher than those for the top and side temperatures which indicates that the flow is either stratified or stratified wavy as shown in the flow images. The side surface subcooled temperatures are higher than the top surface subcooled temperatures due to the thicker liquid condensate film at the sides compared to the top. Increasing the mass flux increases the inertial forces



Figure 5.2: Time-averaged subcooled surface temperatures without EHD forces for a constant heat flux of $q''=5.7 \text{ kW/m}^2$ and average quality of $x_{avg}=45\%$. \Box top, \diamond bottom, \triangle side.

which increases the interfacial instabilities and causes liquid droplet entrainment, and therefore the bottom liquid stratum get thinner and the top, bottom and side subcooled temperatures get closer to each other.

5.1.2 EHD Two-Phase Flow Patterns

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Cotton et al. [2005] developed a methodology to predict two-phase flow pattern transitions under the effect of an applied DC voltage. This is shown schematically in Figure 5.3. The transition line from stratified wavy to annular flow presented by Steiner [1993] was modified to include the induced interfacial EHD force. The transition line proposed by Cotton et al. [2005] can be represented in terms of mass flux



Figure 5.3: Flow chart of Cotton et al. [2005] model to predict two-phase flow pattern redistributions due to the application of DC voltage.

and vapor quality as

$$G = \left\{ \frac{A_{vd}^{3}\rho_{v}\left[Dg\left(\rho_{l}-\rho_{v}\right)-f_{e}^{\prime\prime}\right]}{x^{2}A^{2}\left[1-\left(2h_{ld}-1\right)^{2}\right]^{0.5}} \left[\frac{\pi^{2}}{25h_{ld}^{2}}\times\left(\frac{Fr}{We}\right)_{l}+1\right] \right\}^{0.5}$$
(5.1)

The normal interfacial force, f''_e , directed from the liquid towards the vapour was calculated, assuming a free charge interface, using (Melcher [1963])

$$f_e'' = \frac{\varepsilon_{s,l} \left(\varepsilon_{s,l} - \varepsilon_{s,v}\right)^2}{\varepsilon_{s,v} \left(\varepsilon_{s,l} + \varepsilon_{s,v}\right)} \varepsilon_o E_{n,l}^2 \tag{5.2}$$

For the same annular geometry as the current investigation, the results show that

applying a 8 kV DC voltage will cause two-phase flow pattern transition from stratified wavy flow (with the liquid level below the electrode) to annular flow at any mass flux for quality in the range 20% < x < 40%. For x > 40%, the transition from stratified wavy flow to annular flow occurs at a lower mass flux than that without EHD forces. The experimental data of Cotton et al. [2005] and Sadek et al. [2006a], deduced from flow visualizations at the exit of the heat exchanger and surface temperature measurements, were in good agreement with the EHD flow map predictions.

The EHD flow pattern map, however, can not be used to identify the appropriate heat transfer mechanisms under the application of EHD forces. The EHD flow pattern map can not differentiate between annular flow with liquid film flowing axially with the flow which occurs at high mass flux, and the uniform annular film flow pattern which occurs at low mass flux due to liquid extraction by the EHD forces. The main heat transfer mechanism in annular flow without the applied voltage is convective condensation, while the heat transfer mechanism in the EHD induced uniform annular film flow pattern is likely to be film condensation which is principally a function of the liquid film thickness.

The average top, side and bottom surface subcooled temperatures for different mass fluxes and fixed average quality of 45% with 8 kV applied voltage are shown in Figure 5.4. All three temperatures are similar for a given mass flux for the entire range of mass flux. This implies that the two-phase flow pattern consists of a relatively uniform annular liquid film along the circumference as shown in the flow images (Figure 5.4). The bottom and side surface subcooled temperatures at 8 kV are less than those without EHD force (Figure 5.2) for the entire range of mass flux. This implies that the EHD forces cause liquid extraction from the bottom stratum and from the side film condensate which diminish the thermal resistance and decreases the subcooled temperatures. For mass flux less than 120 kg/m²s, the top surface



Figure 5.4: Time-averaged subcooled surface temperatures with 8 kV applied voltage, for a constant heat flux of q''=5.7 kW/m² and average quality of $x_{avg}=45\%$. \Box top, \diamond bottom , Δ side.

subcooled temperatures at 8 kV are less than those at 0 kV which can be attributed to liquid extraction from the top film condensate as well as the bottom stratum and the side liquid film to the core of the two-phase flow. For high mass flux (more than $120 \text{ kg/m}^2\text{s}$), the top surface subcooled temperatures at 8 kV are greater than those at 0 kV. This is because the vapour inertial forces drive the extracted liquid towards the heat transfer surface area and therefore the thin top liquid condensate get thicker and the subcooled surface temperatures get slightly higher.

5.2 Heat Transfer and Pressure Drop

During convective condensation inside tubes, various flow patterns can occur, depending on the orientation and length of the tube, the heat flux along the tube axis, and

	Experimental		Experimental Model		Deviation(%)	
$G(kg/m^2s)$	Bottom	Top	Bottom	Тор	Bottom	Top
55	889	3116	943	2970	6.1	4.7
78	1301	3439	1219	3061	6.3	11.0
104	1589	3659	1533	3122	3.5	14.7
133	1807	3586	1867	3117	3.3	13.1
158	2004	3611	2132	3119	6.4	13.6

Table 5.1: Time averaged heat transfer coefficients (W/m^2K) and deviations between model (El-Hajal et al. [2003]) and experiments

relevant fluid properties. For mass flux in the range from $45 \text{ kg/m}^2\text{s}$ to $160 \text{ kg/m}^2\text{s}$, two types of heat transfer mechanisms occur within the tube without EHD forces (Thome et al. [2003]): (i) convective condensation associated with the axial flow of the bottom liquid stratum and (ii) film condensation associated with the thin liquid condensate falling around the periphery of the tube. The experimental results for the top and bottom heat transfer coefficients along with error bars for average quality of 45% are compared to the model predictions of El-Hajal et al. [2003] in Figure 5.5. The two-phase flow pattern along the entire test section is stratified wavy and therefore the top heat transfer coefficient is calculated using Equation 2.78 for film condensation, while the bottom heat transfer coefficient is calculated using Equation 2.79 for convective condensation. The deviations between the experimental results and the model for different flow conditions are shown in Table 5.1. The model predictions of the bottom and top heat transfer coefficients are within 6.5% and 14.7% respectively. The model under predicts the top heat transfer, which can be attributed to neglecting the effect of the axial vapour shear on the film condensation heat transfer. However, these deviations are comparable to the measurement uncertainties and therefore the models can be used for the current investigation.



Figure 5.5: Heat transfer coefficient without EHD forces for \blacksquare experimental-bottom, \blacktriangle experimental-top, \square model-bottom (Equation 2.79), \triangle model-top-(Equation 2.78).

The effect of mass flux on the time averaged top, bottom and overall heat transfer coefficient without an applied voltage at fixed average quality of 45% is shown in Figure 5.6. Increasing the mass flux from 47 kg/m²s to 160 kg/m²s increases the overall heat transfer coefficient by factor 1.3 from approximately 2500 W/m²K to 3300 W/m²K. The top heat transfer coefficient is higher than that at the bottom over the entire mass flux range. This indicates that the heat transfer of the falling film condensate is greater than the convective condensation of the liquid stratum for this range of mass flux. Increasing the mass flux increases the top heat transfer coefficient till a mass flux of 100 kg/m²s and then it remains almost constant for higher mass fluxes. Film condensation heat transfer is mainly a function of the temperature difference between the wall temperature and the saturation temperature which is dependent on the liquid film thickness. Increasing the mass flux has two opposing



Figure 5.6: The effect of mass flux on heat transfer coefficient without applied voltage (V = 0 kV). \Box top, \diamond bottom , \times overall. Heat flux of $q''=5.7 \text{ kW/m}^2$ and average quality of $x_{avg}=45\%$

effects on the top heat transfer coefficient. Firstly, it increases the axial vapour shear perpendicular to the direction of the falling condensate. This leads to a thinner film thickness and therefore higher heat transfer. On the other hand, increasing the mass flux sweeps part of the liquid from the bottom stratum towards the top and sides of the tube. This causes the liquid film at the top to thicken and therefore lowers the top heat transfer. For mass flux less than 100 kg/m²s, it is likely that increasing the mass flux, increases the vapour shear, leading to thinner film thickness and therefore higher heat transfer coefficients. For mass flux greater than 100 kg/m²s, the thickening of the film thickness due to the liquid being swept from the bottom stratum towards the top and sides of the tube offsets the thinning due to the vapour shear and therefore the heat transfer remains nearly constant. For the heat transfer at the bottom, increasing the mass flux increases the heat transfer for the entire range tested. The convective heat transfer mechanism is a function of Reynolds number and the liquid film thickness. Increasing the mass flux increases Reynolds number of the bottom stratum and decreases the height of the liquid stratum which increases heat transfer in this case.

The effect of a 8 kV DC applied voltage on heat transfer and pressure drop for different mass fluxes and fixed average quality of 45% is shown in Figure 5.7. The heat transfer ratio, which is the ratio of the heat transfer coefficient with applied voltage to the heat transfer coefficient without applied voltage, is maximum (h/h_o) =2.8) at the lowest mass flux ($G = 55 \text{ kg/m}^2 \text{s}$) and decreases with an increase of the mass flux till it reaches a value of 1.2 at a mass flux of 156 kg/m²s (Figure 5.7(a)). The corresponding pressure drop ratios are 4.5 and 2.8 at mass fluxes $G = 55 \text{ kg/m}^2 \text{s}$ and $G = 156 \text{ kg/m}^2 \text{s}$ respectively (Figure 5.7(b)). These results are consistent with the dimensionless analysis which shows that the effect of EHD will be significant if the interfacial Masuda number (Md) is in the same order of the square of liquid Reynolds number (Re_l^2) . The effect of mass flux on the ratio (Md/Re_l^2) and heat transfer ratio is illustrated in Figure 5.8. The ratio (Md/Re_l^2) is calculated for the liquid stratum based on the maximum interfacial electric field for the corresponding liquid height. At mass flux = 57 kg/m²s, the dimensionless ratio (Md/Re_l^2) is approximately 0.7 which corresponds to heat transfer ratio and pressure drop ratio of 2.8 and 4.5 respectively. Increasing the mass flux decreases the ratio (Md/Re_l^2) till it reaches a value of 0.04 with a corresponding minimal heat transfer and pressure drop ratios of 1.2 and 2.8 respectively at mass flux of $156 \text{ kg/m}^2\text{s}$.

The EHD top and bottom heat transfer coefficients are compared to those predicted using dropwise condensation, film condensation and convective condensation models in Figure 5.9. The results indicate that the most applicable conventional heat transfer mechanism for the EHD condensation is film condensation. The deviation



Figure 5.7: The effect of mass flux on (a) heat transfer and (b) pressure drop, $\Box 0$ kV, $\triangle EHD-8$ kV, and \times ratio. Heat flux of q''=5.7 kW/m² and average quality of $x_{avg}=45\%$.



Figure 5.8: The effect of mass flux on $\diamond Md/Re_l^2$, and \triangle heat transfer ratio (8 kV). Heat flux of q''=5.7 kW/m² and average quality of $x_{avg}=45\%$

between the film condensation model predictions and the experimental values can be attributed to the EHD convection effects in the liquid film and mixing/exchange of liquid between the vapour core and the liquid film which can increase heat transfer and is not accounted for in the model.

The top and bottom heat transfer coefficients without EHD (0 kV) and with 8 kV DC applied voltage for different mass fluxes and fixed average quality of 45% are presented in Figure 5.10. With EHD, the top and the bottom heat transfer coefficients are almost equal for the entire range of mass flux. This indicates that the applied voltage minimizes the effect of the gravitational forces and causes the heat transfer to be more uniform circumferentially. At low mass fluxes, both the top and the bottom heat transfer coefficients at 8 kV are higher than those at 0 kV. The an-



Figure 5.9: Heat transfer coefficient for \triangle 8 kV DC-top, \square 8 kV DC-bottom, $\blacktriangle --\blacktriangle$ film condensation (Equation 2.78), $\blacksquare --\blacksquare$ convective condensation (Equation 2.79), $\blacklozenge --\blacklozenge$ dropwise condensation (Equation 2.84).

nular film thickness with and without EHD forces and the amount of liquid extraction due to EHD forces can be estimated based on the experimental heat transfer results using the film condensation model (Thome et al. [2003]) [see Appendix D for details]. Before the application of the high voltage, the film thickness is 1.55 mm and 0.03 mm for the bottom stratum and the top and side liquid condensate respectively. After applying the high voltage, the film thickness reduces to 0.01 mm along the entire circumference. Applying 8 kV induces additional EHD interfacial forces which cause liquid extraction [see Appendix D for details] of almost 94% from both the bottom liquid stratum and the top and side film condensate, Figure 5.11. The liquid extraction ratio is defined as the percentage ratio of the area of the liquid in the vicinity of the heat transfer surface without EHD force to that with EHD force [see Appendix



Figure 5.10: The effect of mass flux on the overall heat transfer coefficient. $\Box 0$ kV-top, $\blacksquare 0$ kV-bottom, $\triangle 8$ kV-top, $\blacktriangle 8$ kV-bottom. Heat flux of q''=5.7 kW/m² and $x_{avg}=45\%$



Figure 5.11: The effect of mass flux on extraction ratio due to 8 kV applied voltage. Measurements taken for a constant heat flux of $q''=9.6 \ kW/m^2$ and $x_{avg}=45\%$.

D for details. This diminishes the thermal resistances for both the bottom stratum and the film condensate at the top and sides of the tube and therefore increases heat transfer. Increasing the mass flux at 8 kV DC applied voltage decreases the ratio (Md/Re_l^2) which decreases the effect of EHD on liquid extraction and therefore decreases the top and bottom heat transfer coefficients. The liquid extraction decreases from 94% at the mass flux 55 kg/m²s to 87% at mass flux 156 kg /m²s. For mass flux greater than approximately 130 kg/m²s, the effect of EHD on the top heat transfer is negligible and cause enhancement only for the bottom heat transfer. This can be attributed to the higher (Md/Re_l^2) interfacial EHD forces for the bottom liquid stratum compared to the top and sides liquid film. The closer the interface to the high voltage electrode, the higher the values of the interfacial electric field intensity and therefore the induced interfacial EHD forces and interfacial Md number. Although the extraction ratio is 87% at mass flux 156 kg $/m^2$ s, the heat transfer ratio is only 1.2. This can be attributed to the relatively higher heat transfer coefficient at the bottom stratum without EHD compared to that at lower mass fluxes. Thus, the heat transfer enhancement due to the change of the mode of heat transfer at the bottom section of the tube from convective condensation (without EHD) will not be significant.

Summary

For stratified flow with liquid-vapour interface below the high voltage center electrode, applying a DC voltage induces EHD forces which primarily acts at the liquid-vapour interface. These forces increase the interfacial instabilities and may cause liquid to be extracted away from the heat transfer surfaces towards the vapour core and causes an increase in heat transfer and pressure drop. The liquid extraction changes the mode of heat transfer at the bottom of the tube from convective condensation into film condensation. At low mass fluxes (low Reynolds number), film condensation heat transfer is higher than that for convective condensation and therefore the overall heat transfer with 8 kV applied voltage is higher than that without applied voltage. The effect of EHD forces on two-phase flow decreases with an increase in mass flux, ,which is reflected in a decrease of the ratio Md/Re_l^2 .

Chapter 6

Tube Side Convective Condensation under AC and Pulse Wave Applied Voltage

This chapter is divided into three sections. The effect of different AC and pulse wave applied voltage parameters on heat transfer and pressure drop during convective condensation are presented in the first section. In the second section, the transient characteristics of the two-phase flow due to the application of a step voltage are investigated. In the last section, the effect of the pulse repetition rate and duty cycles of applied pulse wave on two-phase flow patterns, heat transfer and pressure drop at different flow conditions are presented and discussed.

6.1 AC/Pulse Electrohydrodynamics

The use of AC and/or pulse wave applied high voltage provides an opportunity for a better control of the heat transfer and pressure drop compared to an applied DC voltage because more parameters can be manipulated to achieve the required heat transfer with potentially a minimum pressure drop penalty. The results of using alternating high voltage electric fields on heat transfer and pressure drop for tube side convective condensation are presented and discussed in this section (Sadek et al. [2009]). Tests were performed with sine, square and pulse applied voltage waveforms (Figure 6.1) over a range of frequencies (2 Hz < f < 2 kHz), peak-to-peak voltages (2 kV < V_{p-p} < 12 kV) and DC bias voltage (-10 kV < V_{DC} < 10 kV), for a fixed mass flux of 100 kg/m²s, inlet quality of 70%, outlet quality of 30% and heat flux of 10 kW/m². The experiments in this section were performed using test section A (1.5 m long).

To examine the characteristic differences between AC and pulse waveforms and DC high voltage electric fields, experiments were performed with a pulse wave and a sine wave with relatively small peak-to-peak voltage of 2 kV. The effect of varying the DC applied voltage on the overall heat transfer ratio and the pressure drop ratio is shown in Figure 6.2, while that of varying the average voltage of the sine wave and pulse wave for two frequencies/pulse repetition rates of 60 Hz and 2 kHz are presented in Figures 6.3 and 6.4 respectively. As shown in Figure 6.2, increasing the magnitude of the applied voltage increased both the heat transfer and the pressure drop, which agrees with the previous experimental results of Sadek et al. [2006a] and Singh et al. [1997]. The increase is very similar for either a positive or negative applied voltage of the same magnitude. There is no significant effect of the polarity in this case. This can be attributed to the negligible effect of the free charges (electrophoretic term), and therefore the dominant terms of the EHD electric body force (Equation 3.24) in two-phase flow for the DC case are the dielectrophoretic and electrostrictive terms (Singh et al. [1997]). These terms are function of the the square of the electric field strength and therefore independent of the polarity of the applied


Time period (T)= 1/ff: frequency or pulse repetition rate V_{DC} : DC bias voltage V_{avg} : average voltage V_{p-p} : peak to peak voltageDuty ratio (cycle)= D/T × 100

Figure 6.1: Schematic diagram of applied voltage waveforms. (a) Square wave, (b) sine wave, and (c) pulse wave (pulse train).



Figure 6.2: Effect of varying the DC applied voltage on a) heat transfer ratio and b) pressure drop ratio.



Figure 6.3: Effect of average voltage, V_{avg} , on a) heat transfer ratio and b) pressure drop ratio. Measurements taken for constant peak-to-peak voltage, $V_{p-p} = 2$ kV and constant frequency/pulse repetition rate of 60 Hz. \Box pulse wave (50 % duty cycle), \diamond sine wave.



Figure 6.4: Effect of average voltage, V_{avg} , on a) heat transfer ratio and b) pressure drop ratio. Measurements taken for constant peak-to-peak voltage, $V_{p-p} = 2$ kV and constant frequency/pulse repetition rate of 2 kHz. \Box pulse wave (50 % duty cycle), \diamond sine wave.

voltage. Increasing the DC average voltage, for both sine and pulse waveforms at 60 Hz and 2 kHz, had a similar effect on heat transfer and pressure drop compared to the DC case with the same applied voltage. At the minimum average voltage of 0 kV, the maximum deviations between the heat transfer and the pressure drop ratios and those for the DC case at applied voltage of 0 kV are 7% and 6% respectively. At 8 kV average voltage, the maximum deviations from the DC case are 12% and 5%. For this relatively small peak-to-peak voltage, the amplitude of the oscillations of the applied voltage is small, and the heat transfer and pressure drop are mainly function of the average voltage. Therefore the effects of the frequency and waveform of the oscillations are not discernible as shown in Figures 6.3 and 6.4.

The effect of the frequency on the overall heat transfer and pressure drop is shown in Figure 6.5. The results are presented for both a sine wave and square wave with duty cycle of 50% with no DC bias and a fixed peak-to-peak voltage of 12 kV. In this range of frequency (5 Hz < f < 2 kHz), the frequency has an insignificant effect on the heat transfer ratio for both waveforms and the heat transfer ratio is approximately 1.5 for the square wave and approximately 1.3 for the sine wave. The main mechanism for the heat transfer enhancement is liquid extraction due to the induced EHD forces. Since liquid extraction is independent of the polarity of the applied DC signal as shown from the DC results (Figure 6.2), no significant change in heat transfer is expected when the applied voltage oscillates between -6 kV and 6 kV. The pressure drop ratio $(\Delta p / \Delta p_o)$, however, initially increases with the frequency and reaches a maximum value of about 2.6 for the square wave and 2.1 for the sine wave at about 20 Hz (Figure 6.5(b)). The pressure drop ratio then decreases with a further increase of frequency till it reaches a value of 1.3 for both applied waveforms at frequency of 2 kHz. The initial increase of the frequency from 5 Hz to 20 Hz likely causes an increase in the oscillations of the liquid droplets in the vapour core



Figure 6.5: Effect of frequency on a) heat transfer ratio and b) pressure drop ratio. Measurements taken for constant average voltage, $V_{avg}=0$ kV and peak-to-peak voltage, $V_{p-p} = 12$ kV. \Box square wave (50 % duty cycle), \diamond sine wave.

and consequently the interaction between the droplets and the high voltage electrode, leading to the increase in the pressure drop. As the frequency is increased beyond 20 Hz, it is likely that the time response of the liquid droplet becomes greater than the time period of the applied voltage. In this case, the liquid droplets cannot respond completely to the high voltage AC signal and therefore the effect of frequency on the pressure drop starts to diminish. More discussion about the response of the liquid droplets to the applied voltages is presented in section 6.3.

The effect of frequency (sine wave) or pulse repetition rate (pulse wave) on the overall heat transfer and pressure drop at an average voltage of 6 KV and a peakto-peak voltage of 12 kV is shown in Figures 6.6. In this case, the signal oscillates between 0 kV and 12 kV. The relatively large difference between the minimum (0 kV)and the maximum (12 kV) of the applied voltage leads to different EHD characteristics compared to the case with no bias voltage. At low frequency/pulse repetition rate, increasing the frequency/pulse repetition rate from 5 Hz to 10 Hz increases both the heat transfer enhancement and the pressure drop penalty. In this frequency/pulse repetition rate range, the time period of the applied waveform is likely larger than the time response of the liquid stratum and liquid is extracted to the vapour core during the voltage on period (12 kV) and returns back to the bottom stratum during the voltage off period (0 kV). Therefore, both heat transfer and pressure drop increase because of the repetitive flow pattern oscillations. For the frequency/pulse repetition rate range between 10 Hz and 100 Hz, the heat transfer enhancement ratio decreases while the pressure drop penalty continues to increase. The difference between the effect of frequency/pulse repetition rate on heat transfer and pressure drop is likely due to the different response times of the liquid stratum and the liquid droplets in the vapour core. The time response of the thicker liquid stratum at the bottom of the tube is higher than that for the liquid droplets in the vapour core. Consequently,



Figure 6.6: Effect of frequency/pulse repetition rate on a) heat transfer ratio and b) pressure drop ratio. Measurements taken for constant average voltage, $V_{avg}=6$ kV and peak-to-peak voltage, $V_{p-p}=12$ kV. \Box pulse wave (50 % duty cycle), \diamond sine wave.

the stratified flow cannot completely respond to the high frequency/pulse repetition rate of the applied voltage and therefore the heat transfer enhancement ratio starts to decrease. However, the liquid droplets can follow the frequency/pulse repetition rate to a higher value which increases the pressure drop penalty with an increase of frequency/pulse repetition rate due to the increase of the momentum transfer between the central electrode and the liquid droplets. For frequency/pulse repetition rate higher than 100 Hz, there is no effect of frequency on heat transfer for both applied voltage waveforms. In this range of frequency/pulse repetition rate, the time period of the applied voltage waveforms becomes much smaller than the time response of the liquid droplets and therefore the droplet oscillations and the pressure drop penalty decreases with an increase in the frequency/pulse repetition rate. A more detailed analysis on the effect of frequency/pulse repetition rate on heat transfer and pressure drop is presented in the following sections. Both heat transfer and pressure drop are higher for the applied pulse wave voltage compared to the sine wave applied voltage for the same average voltage (V_{avg}) and peak-to-peak voltage (V_{p-p}) . This can be attributed to a higher effective voltage of the applied pulse wave voltage. One measure of the effective voltage would be to use the rms value. The rms values for the two waveforms are 6 kV and 4.2 kV in case of $V_{avg} = 0$ and 8.5 kV and 7.4 kV in case of $V_{avg} = 6$ kV for the pulse and the sine wavefroms respectively.

The effect of duty cycle for a fixed DC bias voltage of 6 kV, peak-to-peak voltage of 12 kV and pulse repetition rate of 60 Hz is presented in Figure 6.7. The heat transfer enhancement ratio increases from 1.15 at 10% duty cycle to 2.54 at 90% duty cycle. Increasing the duty cycle directly increases the effective voltage of the applied signal, leading to an increase in the heat transfer enhancement. The pressure drop penalty, however, increases from 1.8 at 10% duty cycle to 3 around 40% duty cycle and then is almost constant with any further increase in the duty cycle. A more



Figure 6.7: Effect of duty cycle of a pulsed applied voltage of pulse repetition rate of 60 Hz, DC bias voltage, $V_{DC}=0$ kV, and peak-to-peak voltage, $V_{p-p}=12$ kV on a) heat transfer ratio and b) pressure drop ratio.

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detailed analysis on the effect of duty cycle on heat transfer and pressure drop is presented in the section 6.3.

The results here show that applying an AC or pulse wave voltage provides a promising tool for heat transfer and pressure drop control by manipulating the different parameters of the applied voltage. A pulse waveform is more advantageous than a sine waveform because of the additional control provided by changing the duty cycle.

6.2 Dynamics of Applied Step Voltage

To get better insight into the effect of an applied pulse wave voltage on tube side convective condensation, the effect of a step input voltage on the transient characteristics of two-phase flow patterns, surface temperatures, and pressure drop were investigated.

6.2.1 Transient Two Phase Flow Patterns

The transient response of the flow to a step input voltage of 8 kV was visualized at the exit of the heat exchanger using a high speed imaging system at a record rate of 1000 frames per second through a transparent quartz tube coated with an electrically conductive film of tin oxide (Sadek et al. [2006b], Sadek et al. [2008]). The time response of the flow starting from 0.005 sec prior to applying the voltage for mass flux of 55 kg/m²s and quality of 35% is shown in Figure 6.8. Before the voltage is applied, the two-phase flow pattern is stratified wavy with the liquid-vapour interface just below the electrode (Figure 6.8(a)). At the instance of application of the voltage (t=0 sec), liquid is extracted from the liquid stratum toward the electrode due to the induced interfacial electric forces (Figure 6.8(b)). At 0.005 sec, the liquid extraction is more intense, forming high-intensity liquid-vapour interaction regions below the electrode (Figure 6.8(c) and 6.8(d)). At a time of around 0.02 sec, the liquid can be seen pushed in the form of twisted liquid cones pointing radially outward from the central electrode, Figure 6.8(e). For simplicity, these flow patterns will be referred to as "twisted cone" patterns hereafter in the thesis. The amount of liquid moved to the upper section increases as time increases from 0.02 to 0.04 sec (Figure 6.8(e))6.8(g)). As time is further increased, the liquid-vapour interactions below the electrode stay intense while the intensity of the liquid "twisted cone" patterns starts to decrease

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Figure 6.8: Transient time response of two-phase flow due to an applied DC voltage of 8 kV.

and eventually diminish at steady state (Figure 6.8(h) and 6.8(k)). These transient two-phase flow patterns are expected to result in different heat transfer and pressure drop characteristics compared to the application of DC electric fields. Understanding and controlling these transient flow patterns using pulsed electric fields will provide a mean of using electrohydrodynamics for controlling heat transfer and pressure drop in thermal systems.

The time characteristics of the transient two-phase flow patterns due to the application of a step input voltage can be represented schematically as in Figure 6.9. Prior the application of the applied voltage (t < 0), the flow pattern is stratified with the liquid-vapour interface below the electrode. At time $t > \tau_1$ after the onset of the applied voltage (t = 0), the transient flow patterns (liquid "twisted cone" patterns) start to appear and stay active for a time duration τ_2 (represented in Figure 6.9 as a decaying amplitude sine wave). For time $t > \tau_2$, the intensity of these transient flow patterns diminish with time and relax to the flow pattern of the applied DC voltage. The onset of the transient flow patterns is shown schematically in an expanded view in Figure 6.9(b) to illustrate the time duration from the onset of the step voltage to the start of the transient flow patterns (τ_1).

To be able to control the transient flow patterns and therefore heat transfer and pressure drop, there are three time scales that need to be examined and identified.

Time needed to initiate the transient flow patterns, τ_1

This is the time required to initiate the transient flow patterns from the time of the applied step input voltage. This time parameter identifies the frequency limit beyond which the transient flow patterns will not appear. The flow visualization experiment shows this time parameter to be approximately 0.025 sec for the applied step input voltage of 8 kV.



Figure 6.9: Schematic diagram of the different transient time parameters and flow patterns with step input voltage of 8 kV.

Duration time of the transient flow patterns ("twisted cone" patterns), au_2

This time parameter is important to evaluate the effect of the transient flow patterns on the average heat transfer and pressure drop for different pulse repetition rates and duty cycles. One way to identify τ_1 experimentally is by using flow visualization. For example, for an applied step input voltage of 8 kV, mass flux of 55 kg/m²s and average quality of 45%, the flow visualization experiment (Figure 6.8)shows this time parameter to be approximately 0.55 sec.

Liquid extraction time, τ_3

This is the time needed for the liquid-vapour interface to touch the bottom of the center electrode starting from the onset of the applied step input voltage.

The predictions of the three time parameters, τ_1, τ_2 , and τ_3 , is essential to optimize the pulse repetition rate and the duty cycle of the pulse wave applied voltage to control the transient two phase flow patterns. The effect of the transient two phase flow patterns on heat transfer and pressure drop for different pulse repetition rates and duty cycles is investigated in section 6.3. The liquid extraction time, τ_3 is examined experimentally and numerically in the next section for different applied voltages and distances between the electrode and the liquid-vapour interface.

6.2.2 Liquid Extraction

The liquid extraction time depends on the magnitude of the applied voltage, as well as the initial location of the liquid-vapour interface relative to the high voltage electrode. The objective of this section is to investigate, experimentally and numerically, the liquid extraction for a stratified liquid-vapour zone due to an applied DC step input voltage. In particular, experiments and numerical simulations are performed to estimate the liquid extraction time for different applied step input voltages and



Figure 6.10: Typical flow image showing (a) liquid level and (b) reference line 2 pixels (0.088 mm) above the liquid level that is used for the image processing.

distances between the liquid-vapour interface and the high voltage electrode.

Liquid Extraction experiments

The flow was visualized using a high speed camera at a record rate of 2000 fps. To determine the liquid extraction times, two sets of experiments were performed; convective experiments with the two-phase flowing with mass flux $G=55 \text{ kg/m}^2\text{s}$ and stationary experiments with the refrigerant initially stationary within the test section. The convective experiments were conducted adiabatically for different vapour qualities, while the stationary experiments were performed with the loop charged to different levels so that the stratified liquid layer was at different locations relative to the central high voltage electrode. For this study, the voltage was applied as a step input that was synchronized with the recording trigger of the high speed camera.

Liquid extraction will only occur when the electric interfacial forces are sufficiently large to overcome the gravity forces, and depends on both the applied voltage and initial liquid level (i.e. the distance between the interface and the high voltage electrode). For the convective experiments, the accurate identification of the initial interface location using any image processing technique is very difficult due to the presence of the interfacial waves. Therefore, the initial interface location was identified



Figure 6.11: Plot of (a) mean gray value and (b) slope of the mean gray value of the interface for different frames. -- represents ± 2 standard deviations of the signal.

visually from the high speed flow images. However, for the stationary experiments, an image processing routine using a commercial software package was developed to accurately estimate the interface location and therefore the liquid extraction time. The image processing technique is illustrated in Figure 6.10. First, the initial liquid level is identified (Figure 6.10a), and a reference line 2 pixels (corresponding to 0.088) mm) above the liquid level is marked on the image frame (Figure 6.10b). The mean gray value along the reference line once the voltage is applied is calculated for each image frame, and shown in Figure 6.11(a). It is clear that there is a change in the gray value as the liquid level passes the reference line. In this case, the gray value changes from about 120 corresponding to the vapour, to about 137 as the liquid crosses the reference line. A magnitude and slope threshold method was used to estimate the start of the liquid extraction. An initial estimate of the start of the extraction was obtained when the grav level exceeded the magnitude threshold defined as two standard deviations above the average reference value. The change of the gray value between two successive frames (i.e the slope of the gray value of the reference line) was also plotted as shown in Figure 6.11(b). The start time was also determined from this plot defined as the time when the magnitude of the slope exceeded the threshold defined as two standard deviations above the average value. The start time used here was defined when the estimates from both the magnitude and slope threshold matched, to ensure no false crossings were obtained. The end of the liquid extraction is defined when the interface touches the electrode, and it is identified by visual observation of the high speed flow images. For the case shown in Figure 6.11, the liquid extraction starts at frame 40 and ends at frame 53, which corresponds to an extraction time of 6.5 msec for the given conditions. The experimental uncertainty in estimating the start and the end times from the flow images is ± 0.5 msec, which corresponds to the temporal resolution of the imaging system. The total uncertainty in estimating the

extraction time is ± 1 msec.

Numerical methodology

A two-dimensional, numerical model was developed to predict the response of the liquid stratum in a stratified liquid-vapour zone to an applied DC step input voltage for different interface-electrode gap distances. A quasi-steady approach is adopted, where the extraction is divided into a number of steps. The velocity of the liquid-vapour interface is assumed constant through each step. The numerical model consists of two modules which are solved for each step. In the first module, the electrostatic equations for the two-phase domain are solved, using a finite element commercial software COMSOL. The induced interfacial EHD forces due to the application of the high voltage are estimated from the resulting electric field, and incorporated as a negative pressure boundary condition at the interface in the Navier-Stokes equations for the liquid stratum. The liquid transport equations are solved using the finite element commercial software ANSYS-CFX. The resulting interfacial velocity values are used to estimate the new profile of the liquid-vapour interface and the procedure is repeated for the next step as illustrated in Figure 6.12. The procedure and results are illustrated for the case where the interface was 1.5 mm below the electrode and the applied voltage was 3 kV. The numerical simulations were performed for the same cross sectional geometry as the experimental facility (Figure 6.13). The working fluid for the simulations was refrigerant R-134a with liquid and vapour dielectric constants of 9.5 and 1, respectively. The electrostatic equations governing the electric field are presented in equations (6.1-6.7). Starting with Gauss's law

$$\nabla . \dot{D} = \rho_e \tag{6.1}$$

Assuming the dielectric fluid is linear, isotropic and homogenous, the electric dis-



Figure 6.12: The algorithm for the numerical scheme to estimate the liquid extraction time for a stratified liquid-vapour zone due to electrohydrodynamics.

placement can be expressed as,

$$\bar{D} = \varepsilon \bar{E} \tag{6.2}$$

substituting in Equation 6.1 gives

$$\nabla_{\cdot}\bar{E} = \frac{\rho_e}{\varepsilon} \tag{6.3}$$

In the absence of a change in the magnetic field, Faraday's law of induction gives

$$\nabla \times \bar{E} = -\frac{\partial \bar{B}}{\partial t} = 0 \tag{6.4}$$



Figure 6.13: Electrode geometry and the liquid and vapour domain for the numerical simulations.

Since the curl of the electric field is zero, the electric field can be defined by a scalar electric potential field, V

$$\bar{E} = -\nabla V \tag{6.5}$$

substituting in Equation 6.3 gives

$$\nabla^2 V = -\frac{\rho_e}{\varepsilon} \tag{6.6}$$

For the current simulations, it was assumed that there is no space and surface charges



Figure 6.14: Numerical mesh at (a) t=0 and (b) t=5.98 msec for an applied voltage of 3 kV and an initial interface-electrode gap distance of 1.5 mm.

within the fluid and therefore Equation 6.20(e) can be simplified to the Laplace equation

$$\nabla^2 V = 0 \tag{6.7}$$

The validity of neglecting the space and surface charges are discussed in the results.

The mesh corresponding to the initial condition and at the last step, t=5.98 msec, is shown in Figure 6.14. A grid independence study was performed for the first and last time steps in the simulation to determine the appropriate grid size. The number of elements in the mesh was increased till the average discrepancy in the electric field results is within $\pm 1\%$ when compared to the finer grid results. A mesh of 14615 triangular elements (Figure 6.14(a)) at t=0 msec and a mesh of 87092 triangular elements (Figure 6.14(b)) at t=5.98 msec was used. In each case, a non-uniform mesh with a finer distribution of elements near the interface was used because of the larger electric field gradients in this region. A greater number of elements are required for grid independence in the later time steps as the interface becomes more deformed due to the electric field.



Figure 6.15: Two-dimensional electric field distribution at (a) t=0 msec and (b) t=5.98 msec for an applied voltage of 3 kV and an initial interface-electrode gap distance of 1.5 mm.

The corresponding two dimensional electric field is shown in Figures 6.15. It is clear that the electric field is much stronger within the vapour domain, and there is a significant change in the electric field at the vapour-liquid interface. This jump condition results in a large electrically induced interfacial force (dielectrophoretic and electrostrictive forces from Equation 3.24), which in this case acts at the liquid interface against gravity. Assuming a free charge interface, the interfacial electric force is estimated, from Stratton [1941] and Scaife [1989], as

$$f_{e}'' = \frac{\varepsilon_{o}}{6} \left\{ \frac{1}{\varepsilon_{s,2}^{2}} \left[4\varepsilon_{s,1}^{2}\varepsilon_{s,2} - 2\varepsilon_{s,1}^{2} - 4\varepsilon_{s,1}\varepsilon_{s,2}^{2} + 2\varepsilon_{s,2}^{2} \right] E_{n,1}^{2} \right\}$$
$$+ \frac{\varepsilon_{o}}{6} \left\{ \left[\varepsilon_{s,2} \left(\varepsilon_{s,2} - 2 \right) - \varepsilon_{s,1} \left(\varepsilon_{s,1} - 2 \right) \right] \left(E_{t,1}^{2} \right) \right\}$$
(6.8)

where $\varepsilon_{s,1}$ and $\varepsilon_{s,2}$ are the dielectric constant of the two mediums.

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The distribution of the interfacial force for the initial condition where the voltage is first applied is shown in Figure 6.16. As expected from the electric field

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Figure 6.16: Distribution of the interfacial electric force at the initial condition (t=0) of applying the step input voltage for the case of an applied voltage of 3 kV and an electrode-interface distance of 1.5 mm.

distribution, the interfacial force is a maximum at the centerline, and acts against gravity. The interfacial force is used as the boundary condition in a commercial finite element code (ANSYS-CFX) to solve for the flow field within the liquid domain using the Navier-Stokes equations for laminar flow. In these simulations, both the surface tension forces at the interface and the weight of the vapour are assumed negligible compared to the electric forces and the weight of the liquid. The interfacial electric forces are incorporated as a negative pressure boundary condition acting normal to the interface. A no-slip boundary condition is used at the wall. A non uniform numerical mesh consisting of 24330 triangular elements with a finer mesh distribution close to the surface was used for the liquid flow simulations at t=0 msec as shown in Figure 6.17(a). A more refined mesh is used for each subsequent step to take into account the movement of the interface. A maximum of 241500 triangular elements was used at



Figure 6.17: (a) Numerical mesh and (b) velocity field for the liquid phase at the first instance (t=0) of applying the step voltage for the case of an applied voltage of 3 kV and an electrode-interface distance of 1.5 mm.

the final time step. As for the electric field, a grid independence study was performed on the first and last time steps for the liquid flow simulations. The mesh was refined till the change in the solution was less than 0.5%. The velocity field within the liquid domain for an applied voltage of 3 kV at the first instance (t=0) of applying the step voltage is shown in Figure 6.17(b). The corresponding interfacial velocity at this instance is shown in Figure 6.18. In the central region, the interfacial velocity is upwards against gravity with a maximum at the centreline of approximately 0.096 m/s, while it is downwards away from the central region. The numerical results show that the interfacial electric force is sufficient to overcome gravity and extract the liquid in the central region upwards towards the electrode. Two criteria could be used for progressing the simulations in time: (1) time based, where the iterations are performed for fixed time intervals of Δt or (2) spatial based, where the next iteration is



Figure 6.18: Interfacial normal velocity for the liquid phase at the first instance (t=0) of applying the step voltage for the case of an applied voltage of 3 kV and an electrode-interface distance of 1.5 mm.

performed after the interface at the centreline has moved by a fixed distance Δy . For the present simulations, a spatial based time step criterion was used for progressing the simulations in time. To determine the time step, the total distance for liquid extraction, which is defined as the distance between the liquid-vapour interface and the electrode surface at the centerline, was first divided into equally spaced lengths, Δy . The length step is computed as

$$\Delta y = \frac{L}{N} \tag{6.9}$$

where L is the total distance for liquid extraction and N is the total number of time steps. Assuming a constant interfacial velocity through each step, the time required for the liquid-vapour interface to move one length step is computed as

$$\Delta t = \frac{\Delta y}{V_{x=0}} \tag{6.10}$$

where $V_{x=0}$ is the interfacial velocity at the centerline as determined from the numerical simulation. An explicit scheme where the interfacial velocities are determined at the beginning of each time step is used to compute the time step. This method for computing the time steps was compared to a central differencing scheme and it was determined that the choice of scheme had a negligible effect on the solution. To determine an appropriate number of steps (N), N was increased till the extraction time was within $\pm 5\%$ of that computed from a smaller number of steps. The comparison between the extraction time for two different number of steps is shown in Figure 6.19. Increasing the number steps from N=20 to N=30 had an insignificant effect on the extraction time (less than 3%). A length step computed using N=20 was determined to be acceptable for the conditions of this study.

The displacement of the interface at the end of the time step is estimated by assuming the interfacial velocity remains constant over Δt . The process is repeated until either liquid extraction occurs or when the interface stabilizes, in which there is no extraction. Liquid extraction is deemed complete when the liquid level at the centerline reaches the electrode.

The evolution of the liquid-vapour interface with time is shown in Figure 6.20. For this condition, the applied voltage was sufficient to extract the liquid towards the electrode. As the liquid is extracted, the thickness of the liquid column being pulled towards the electrode begins to narrow. The numerical simulations were stopped prior to the formation of an unrealistic narrow liquid peak at the centerline of the liquid-vapour interface which results in non physical results when additional



Figure 6.19: Comparison between the elapsed time for liquid extraction for an initial electrode-liquid interface gap distance of 1.8 mm and an applied voltage of 8 kV using \triangle 20 length steps and \blacktriangle 30 length steps.

iterations were performed. The percentage of the additional time added to the total extraction time per each step $(\Delta t_n / \sum_{i=1}^n \Delta t_i)$ is shown in Figure 6.21. Although the simulations were not performed till the liquid reached the electrode, the results in Figure 6.21 show that the additional time that is required for the interface to reach the electrode after the simulations was terminated would be very small compared to the time already progressed. The additional time added per time step is estimated by extrapolating the existing profile (Figure 6.21) till step 20 at which the liquid interface touches the electrode. The additional time is less than 1.5% of the total time. The latter steps contribute very little to the overall extraction time since the interfacial forces are increased significantly as the interface approaches the electrode.



Figure 6.20: Evolution of the liquid--vapour interface for an applied voltage of 3 kV and an initial interface-electrode distance of 1.5 mm.



Figure 6.21: Time contributed to the overall liquid extraction time for each step for an applied voltage of 3 kV and an initial interface-electrode distance of 1.5 mm.

for the liquid to travel may be considered to occur, relative to the early stages of extraction, instantaneously. The increase in the interfacial velocity at the centerline as the liquid-vapour interface is brought closer to the electrode is shown in Figure 6.22. At the last step in the simulation, the velocity at the center of the interface increases by factor 3 and exceeds 1.0 m/s.

The flow images of the liquid extraction of the stationary two phase zone for the representative case of an applied voltage of 3 kV with an initial interface to electrode gap distance of 1.5 mm is shown in Figure 6.23. The stationary two phase zone with the liquid-vapour interface below the electrode before the application of the voltage is shown in Figure 6.23a. At the time of application of the voltage (t = 0msec), the liquid can be seen to respond to the electric forces (Figures 6.23(a)-(d)). As time progresses, the liquid is extracted from the liquid stratum towards the electrode due to the induced interfacial electric forces (Figure 6.23 (e)-(h)). As the distance



Figure 6.22: Centerline velocity at various distances from the electrode for an applied voltage of 3 kV and an initial interface-electrode distance of 1.5 mm.

between the interface and the charged electrode decreases, a more rapid movement of the interface is observed due to the increased electric field distribution in this region. As time progresses further, high intensity liquid-vapour interactions can be observed below the electrode (Figures 6.23(i)).

A comparison between the experimental liquid extraction time for convective flow with mass flux of $G = 50 \text{ kg/m}^2\text{s}$ and quality of x = 60% and stationary liquid stratum for different applied voltages is shown in Figure 6.24. For convective flow, the presence of the interfacial waves causes difficulty in identifying the location of the liquid-vapour interface (i.e initial gap distance) and thus there is a larger uncertainty in these results. The initial gap distance for the convective experiments is estimated to be between 1.5 and 1.8 mm while for the stationary experiments using the image processing routine as 1.55 mm, 1.7 mm, and 1.8 mm. The comparison shows that



Figure 6.23: Transient time response of the liquid-vapour interface due to an applied DC voltage of +3 kV. The time between frames is 1 msec.

the extraction time for both the convective and the stationary experiments are similar for the same range of the initial gap distance which implies that the effect of the axial convection of the two-phase flow on the extraction time is small. Due to the uncertainties associated with identifying the liquid level in the convective experiments and the difficulty to control the initial gap distance, the effect of the applied voltage and initial gap distance on liquid extraction time was studied for a stationary liquid stratum. The stationary experimental results showing the effect of the applied voltage and initial distance between the electrode and liquid-vapour interface on the liquid extraction time are shown in Figure 6.25. Increasing the applied voltage and/or decreasing the initial distance between the electrode and liquid-vapour interface decreases the liquid extraction time. For example, at a fixed voltage of 3 kV, the extraction time reduced from 10 msec to 1.5 msec as the distance between the liquid-vapour interface and the electrode was reduced from 1.6 mm to 1.26 mm. For a

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Figure 6.24: Effect of applied voltage on liquid extraction times for × convective flow ($G = 50 \text{ kg/m}^2\text{s}$, x = 60%) with initial gap distance between 1.5 and 1.8 mm and stationary liquid stratum with initial gap distance of \diamond 1.55 mm, \triangle 1.7 mm, \circ 1.8 mm.

fixed distance of 1.5 mm between the liquid interface and the electrode, increasing the applied voltage from 3 kV to 7 kV decreases the extraction time from approximately 9 msec to 0.5 msec.

The numerical results for the elapsed time at the end of each step for different applied voltages for an initial distance between the electrode and liquid-vapour interface of 1.8 mm is shown in Figure 6.26. As seen in the experiments, the numerical results show that increasing the voltage from 4 kV to 8 kV, at a fixed distance between the liquid interface and the electrode of 1.8 mm, reduced the extraction time from 4.95 msec to 2.23 msec. The results for a fixed applied voltage of 5 kV with different gap distances are shown in Figure 6.27. These results clearly show that as the gap distance decreases, the extraction time decreases. The numerical results are



Figure 6.25: Experimental liquid extraction times with initial gap distance between electrode and liquid-vapour interface at applied voltage of \diamond 3 kV, \Box 4 kV, \triangle 5 kV, \times 6 kV, \circ 7 kV, and \blacksquare 8 kV.

consistent with the physics, as an increase in the electric field strength in the vicinity of the interface either by increasing the voltage or decreasing the gap distance, will result in an increase in the electric body force responsible for liquid extraction.

In addition to predicting the extraction time for different applied voltages at different liquid levels, it is important to be able to predict the minimum applied voltage which will induce sufficiently strong interfacial electric forces to overcome gravity and cause liquid extraction for a given interface to electrode distances. The numerical results for the interfacial liquid velocity at the centerline for applied DC step input voltages of 0.5 kV, 2 kV and 3 kV with the initial liquid level 1.5 mm below the charged electrode are presented in Figure 6.28. For 3 kV, the velocity increases as the liquid level increases, which indicates the liquid is being extracted



Figure 6.26: Numerical elapsed time for liquid extraction for initial electrode-liquid interface gap distance of 1.8 mm at \circ 4 kV, \Box 5 kV, \triangle 6 kV and \diamond 8 kV.

towards the electrode. For the applied voltage of 2 kV, there is a small interfacial velocity at the instance the voltage is applied, which subsequently decreases as the gravity overcomes the electric forces, and there is no net liquid extraction. At 0.5 kV, the electric forces at the interface are too small to overcome gravity and there is relatively no movement of the liquid-vapour interface. These results are consistent with the experimental findings that liquid extraction was not observed below applied voltages of 2 kV.

The numerical results for the extraction time are compared to the experimental results in Figure 6.29. In general, there is reasonable agreement between the experimental results and the numerical results. For all cases, the experimentally determined extraction time is lower than those determined numerically. The deviation between the experimental and the numerical extraction times vary from an average


Figure 6.27: Numerical elapsed time for liquid extraction for applied voltage of 5kV and initial electrode-liquid interface gap distances of \circ 1.5 mm, \Box 1.8 mm, and \triangle 2.1 mm.

factor of 1.1 at the highest extraction time to an average factor of 3.25 at the lowest extraction time. These discrepancies can be attributed to the following reasons:

1- Neglecting surface tension: The surface tension forces are stabilizing forces which act against the interfacial electric forces. If the surface tension forces are significant, they will slow down the liquid extraction and increase the extraction time. The importance of the surface tension relative to the electrical forces can be evaluated using the electric bond number, which is the ratio between the electric force to the capillary force (Castellanos and Gonzalez [1998]), and can be defined as

$$Be = \frac{(\varepsilon E^2) R}{\gamma} \tag{6.11}$$

For an applied voltage of 5 kV and an initial gap distance between the liquid surface



Figure 6.28: Maximum interface velocity (numerical results) at applied voltage of $\diamond 0.5 \text{ kV}$, $\triangle 2 \text{ kV}$ and $\Box 3 \text{ kV}$.

and electrode surface of 1.5 mm, the bond number is approximately 10. In this case, the surface tension forces can be neglected. However, for an applied voltage of 3 kV, the bond number is approximately 0.5 at t=5.98 msec. In this case, the surface tension effects become more relevant and improvement in the numerical liquid extraction predictions would benefit from their inclusion.

2- Neglecting the electrophoretic component: The effect of the electrophoretic force on liquid extraction depends on the magnitude and the polarity of the free charges. The influence of the electrophoretic force on the liquid extraction phenomenon can be evaluated in two steps. In the first step, the liquid extraction time is compared to the relaxation time. The relaxation time (ε/σ) is defined as the time needed for the bulk space charges to accumulate at the liquid-vapour interface. If the relaxation time is less than the liquid extraction time, some of the bulk free space



Figure 6.29: Comparison between numerical and experimental liquid extraction times for different applied voltages at gap distances between electrode and liquid interface of \triangle , \blacktriangle 1.5 mm, \circ , \bullet 1.8 mm and \Box , \blacksquare 2.1 mm. The open and filled symbols correspond to the numerical and stationary experimental results, respectively.

charges will accumulate at the interface before the liquid extraction is complete. The second step is to compare the electrophoretic term based on the accumulated surface charges to the dielectrophoretic and electrostrictive terms. For the current working fluid, the relaxation time is estimated as 0.5 msec which is less than the extraction time for all the cases and therefore the effect of the electrophoretic term should be examined relative to the dielectrophoretic and electrostrictive terms. This can be done by comparing the EHD number (Ehd) representing the electrophoretic term to the Masauda number (Md) representing the dielectrophoretic and electrostrictive terms. Assuming that all the space charges will accumulate at the interface, the (Ehd) and

(Md) can be expressed as, IEEE-DEIS-EHD Technical Committee [2003]

$$Ehd = \frac{I_o L^3}{\rho_o \nu^2 \mu_e A} \qquad \qquad Md = \frac{E^2 \varepsilon_o L^2 T_o \left(\partial \varepsilon_s / \partial T\right)}{2\rho_o \nu^2} \tag{6.12}$$

where I_o is the conduction current which can be determined from the experimental current measurements [see Appendix B for details]. The preliminary comparison between *Ehd* and *Md* numbers shows that, for an 8 kV applied voltage, the electrophoretic forces will have a significant effect if the conduction current is more than 0.3 mA. The accumulation of the surface charges at the interface and the role of the electrophoretic forces, however, need to be further investigated.

3- 2-D simulations: The flow visualization images show interfacial instabilities in the axial direction (Figures 6.23(d) to 6.23(f)). This is not accounted for in the numerical simulations as the present simulations are performed only for a 2-D cross section. These axial interfacial instabilities would create a higher electric field at the interface compared to the 2-D case, and result in lower values of extraction time compared to the 2-D case.

4- **Experimental uncertainties**: The experimental extraction times were estimated from the flow images which were captured using a high speed camera with 2000 frame per second. The total uncertainty is 1 msec, which can be significant compared to the extraction times.

In summary, the experimental results showed that the liquid extraction times are in a good agreement for both the stationary and convective cases and there is no significant effect of the two-phase axial flow motion on the extraction time. The experimental and numerical results are consistent with the physical mechanism of the liquid extraction due to the application of a high voltage electric field. At sufficiently high voltages, the induced EHD forces at the liquid-vapour interface can overcome

the gravitational forces and causes the liquid to be extracted towards the charged electrode. The extraction time decreases with an increase of the applied step voltage and/or decrease of the initial distance between liquid interface and the electrode. The numerical model predictions for the minimum voltage required for liquid extraction agree with the experimental results. For the current geometry, for applied voltages greater than 3 kV at a gap distance of 1.49 mm between the liquid-vapour interface and the electrode, the induced interfacial forces were strong enough to overcome gravity and cause liquid extraction towards the electrode. For lower applied voltages, the gravitational forces are greater than the electric forces and no liquid extraction occurred. The closer the interface was to the electrode, the faster was the liquid extraction. The discrepancies in the numerical and experimental extraction times varied from a factor 0.05 at the highest extraction time to a factor of 2.5 at the lowest extraction time. These discrepancies could be attributed to both the experimental uncertainties and the simplifications in the numerical simulations. In general, however, the numerical model gave reasonable results as a first approximation and can be modified to take into consideration the factors that have been neglected in the current simulations. The future research will focus on expanding the numerical model to predict the time response of the two phase flow patterns after the liquid touches the central electrode (τ_1 and τ_2).

6.2.3 Transient Characteristics of Surface Temperatures and Pressure Drop

The response of the pressure drop, bottom and top surface temperatures to the applied step input voltage is shown in Figure 6.30. The bottom surface temperature starts to increase with the onset of the step input voltage from 22 °C till it reaches

24.8 °C after 2.5 sec [Figure 6.30(b)]. The increase in the bottom surface temperature reflects an increase in the heat transfer coefficient at the refrigerant side due to the liquid extraction from the bottom stratum which is the main thermal resistance in the system. The top surface temperature profile shows negligible change in the temperature which is expected due to the fact that the EHD has insignificant effect on the thin liquid condensate at the top of the tube [Figure 6.30(c)]. The pressure drop temporal profile [Figure 6.30(a)] shows a decaying amplitude high frequency oscillations between the onset of the step voltage which the pressure drop value is(≈ 0.2 kPa) and the time at which the pressure drop stabilizes at value of ≈ 0.6 kPa. This profile is a typical time response for a diaphragm-type differential pressure transducers. The duration, frequency, and the amplitude of these oscillations are functions of how the pressure sensor is connected to the test section (e.g. size and length of the connection tubes).

Summary

Applying a step input high voltage during tube side condensation causes the appearance of transient two-phase flow patterns that are not present in the steady state under the application of DC applied voltage case. The goal of this research is to use applied pulse wave voltage to control these transient two phase flow patterns and therefore control heat transfer and pressure drop. To achieve this goal, the time characteristics of these transient two phase flow patterns must be predicted and controlled. The numerical prediction of the liquid extraction time, τ_3 , is the first step to achieve this goal. In the future, the numerical model will be expanded to predict the time characteristics of the two phase flow patterns after the liquid touches the central electrode (i.e. τ_1 and τ_2).



Figure 6.30: Temporal profile for (a) overall pressure drop, (b) bottom surface temperature and (c) top surface temperature for an applied DC step input voltage of 8 kV.

6.3 Effect of Pulse Wave Voltage on Tube Side Convective Condensation

The previous sections showed that applying pulse wave high voltages are advantageous for heat transfer and pressure drop control compared to DC applied voltages. This is due to the presence of the transient two phase flow patterns induced as a result of the application of a step input voltage. These transient flow patterns can be controlled by manipulating the pulse repetition rate and/or duty cycle of the applied pulse wave. In this section, the effect of pulse repetition rate and duty cycle of a pulse wave applied voltage on two phase flow patterns, heat transfer and pressure drop for different flow conditions for tube side convective condensation are presented.

6.3.1 Effect of Pulse Repetition Rate and Duty Cycle

The effect of pulse repetition rate on two-phase flow patterns, heat transfer and pressure drop is studied for three different duty cycle values of 25%, 50%, and 75%. The duty cycle is defined as the percentage ratio between the voltage on duration time of the pulse to the time period of a complete cycle. The effect of EHD on two-phase flow convective condensation can be attributed to three factors: (1) Liquid extraction from the bottom stratum to the core of the test section which reduces the thermal resistance due to the stratified liquid layer (Chang and Watson [1994], Cotton et al. [2005], Bryan and Seyed-Yagoobi [2001]), (2) destabilization of the thermal boundary layer at the heat transfer surface for both the liquid stratum and the annular liquid film due to the applied electric field (Jones [1978]) and (3) thinning of the circumferential liquid film which might lead to pseudo drop-wise condensation (Sunada et al. [1991], ?). The increase in pressure drop due to EHD can, in addition

to the above mechanisms, be attributed to the momentum transfer at the central electrode. This can be significant due the intensive electrically induced liquid-vapour interactions at the electrode surface, which will not be reflected in the heat transfer.

Two-phase flow patterns

The effect of 1 Hz, 50% duty cycle pulse wave voltage on the two-phase flow patterns at mass flux of $G = 55 \ kg/m^2 s$ and average quality x = 45% is shown in Figure 6.31 with 12 images equally distributed over one cycle of the applied pulse wave voltage. At this pulse repetition rate, the time period of the applied voltage is large enough for the two-phase flow patterns to exhibit similar transient patterns observed in the 8 kV step input voltage case (Figure 6.8). The two phase flow pattern oscillates between stratified wavy flow observed at 0 kV (Figures 6.31 (a,b,l)), the transient flow patterns (liquid "twisted cone" patterns, Figure 6.31 (d-h))and the 8 kV DC applied voltage flow patterns (Figure 6.31 (i)).

Similar two-phase flow patterns are observed for a pulse repetition rate less than 10 Hz. Increasing the pulse repetition rate in this range decreases the time duration of the liquid "twisted cone" patterns but sustains the liquid vapour interaction below the electrode. The flow images for a pulse repetition rate of 10 Hz, are shown in Figure 6.32. The flow images show liquid columns (liquid extraction) between the bottom stratum and the central electrode, but no liquid above the electrode. This is likely because the time period of the applied voltage is less than the time needed to initiate the transient flow patterns (estimated as 0.025 sec from the step input voltage experiments). Therefore, there is no formation of liquid "twisted cone" patterns above the electrode, but the liquid vapour interactions below the electrode are sustained.

For a pulse repetition rate greater than 10 Hz, the two-phase flow patterns are shown in Figure 6.33. There is no significant difference in the two-phase flow patterns



Figure 6.31: Flow pattern redistributions for a 1 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 83.3 msec).



Figure 6.32: Flow pattern redistributions for a 10 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 8.3 msec).



Figure 6.33: Flow pattern redistributions for 50% duty cycle pulse wave between 0 and 8 kV for (a) 10 Hz, (b) 75 Hz, (c) 250 Hz, (d) 400 Hz. Mass flux of $G = 55 kg/m^2s$ and quality x = 45%.

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throughout the cycle and therefore, each case is represented by a flow image and a schematic diagram (Figure 6.33(b)-(d)). The liquid-vapour interactions in this range cause the appearance of liquid droplets which oscillate with the same frequency as the pulse repetition rate of the applied voltage. In the pulse repetition rate, 10 Hz < f < 100 Hz (Figure 6.33 (b)), the liquid droplets are able to respond to the applied voltage and therefore increasing the pulse repetition rate greater than 100 Hz (Figure 6.33 (c)-(d)), the liquid droplets cannot completely respond to the applied voltage and the amplitude of the liquid droplet oscillations decreases with an increase of pulse repetition rate. The response continues to decrease till there is no significant effect of the pulse repetition rate on the two-phase flow patterns at 400 Hz (Figure 6.33(d)). The detailed high speed flow images for the different frequencies are shown in Appendix E.

To get more insight on the effect of pulse repetition rate and duty cycle of the pulse wave applied voltage on two-phase flow redistributions, heat transfer and pressure drop, the temporal profiles of applied voltage, pressure drop, bottom and top subcooled surface temperatures for the low frequency range of 0.5 Hz to 10 Hz, are examined as shown in Figures 6.34, 6.35 and 6.36. At 0.5 Hz applied pulse wave voltage (Figure 6.34), both the pressure drop and the bottom surface temperature sensors respond to the oscillations (on/off) of the applied voltage, while the top surface temperature shows no response to the applied pulsed voltage. The time response of the pressure sensor is small enough to capture the on/off effect of the applied pulse wave voltage and the pressure drop oscillates between its value at 0 kV and that at 8 kV DC applied voltage. The amplitude of the bottom surface temperature oscillations are smaller than the temperature difference between the corresponding values at 0 kV and 8 kV DC applied voltage. This is because it takes approximately 2.5 sec for the



Figure 6.34: Temporal profiles of (a) voltage, (b) overall pressure drop at 0 kV, 8 kV and 0.5 Hz (50 % duty cycle) pulse wave applied voltage, (c) bottom surface temperature at 0 kV, 8 kV and 0.5 Hz (50 % duty cycle) pulse wave applied voltage and (d) top surface temperature at 0 kV, 8 kV and 0.5 Hz (50 % duty cycle) pulse wave applied voltage.



Figure 6.35: Temporal profiles of (a) voltage, (b) overall pressure drop at 0 kV, 8 kV and 4 Hz (50 % duty cycle) pulse wave applied voltage, (c) bottom surface temperature at 0 kV, 8 kV and 4 Hz (50 % duty cycle) pulse wave applied voltage and (d) top surface temperature at 0 kV, 8 kV and 4 Hz (50 % duty cycle) pulse wave applied voltage.



Figure 6.36: Temporal profiles of (a) voltage, (b) overall pressure drop at 0 kV, 8 kV and 10 Hz (50 % duty cycle) pulse wave applied voltage, (c) bottom surface temperature at 0 kV, 8 kV and 10 Hz (50 % duty cycle) pulse wave applied voltage and (d) top surface temperature at 0 kV, 8 kV and 10 Hz (50 % duty cycle) pulse wave applied voltage.

bottom surface thermocouple to respond from its value at 0 kV to that at 8 kV (Figure 6.30(b)). At 4 Hz (Figure 6.35), the temperature measurements show no response to the oscillations of the applied voltage while the pressure drop oscillates between the 0 kV value and the 8 kV value with the same frequency as the applied voltage (Figure 6.35 (b)). At 10 Hz (Figure 6.36), the time response of the pressure drop and the temperature sensors within the system are larger than the time period of the applied voltage, and therefore cannot respond to the applied voltage. To determine the frequency response of the pressure drop and heat transfer to the applied pulsed voltage for the low frequency range (f < 10 Hz), the Fast Fourier transform (FFT) spectra of the temperature and the pressure fluctuations are examined in Figures 6.37 and 6.38 respectively for a 50% duty cycle pulsed voltage. The frequencies of the applied signals are shown within the Figures. The results show that the temperature and pressure sensors respond to the applied pulse wave voltage up to 6 Hz and 10 Hz respectively. Beyond these frequencies, the response of the sensors is not fast enough to capture the oscillations of the applied voltage. To investigate the effect of the applied pulse wave voltage for frequencies more than 10 Hz, the flow images were analyzed using a commercial image processing software (ProAnalyst) to track the droplets and estimate the FFT spectra of the droplet oscillations. Figure 6.39 shows a flow image with a trace curve of a single droplet for a representative case of pulse applied voltage of 40 Hz and 25% duty cycle. Figure 6.40 shows the FFT spectra of the droplet oscillations for an 25% duty cycle and 8 kV amplitude pulse wave voltage for the same flow conditions (mass flux of $G = 55 \ kg/m^2 s$ and average quality of x = 45%). A distinct peak is observed at the same frequency as the pulse repetition rate of the applied voltage in the frequency range from 20 Hz till 100 Hz. The magnitude of the peak decreases with an increase of the pulse repetition rate till it diminishes at pulse repetition rate of 200 Hz. The results show that the two-phase



Figure 6.37: FFT spectra of pressure fluctuations for 50% duty cycle. $G = 55 kg/m^2s$, $x_{avg} = 45\%$.



Figure 6.38: FFT spectra of average bottom surface temperature fluctuations for 50% duty cycle. $G = 55 \text{ kg/m}^2 \text{s}$, $x_{avg} = 45\%$.



Figure 6.39: Typical trace of a single liquid droplet for 40 Hz, 25% duty cycle applied pulsed voltage.



Figure 6.40: FFT spectra for typical droplets oscillations for 25% duty cycle applied pulsed voltage.

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flow is able to respond to the applied signal up to 200 Hz, beyond which the droplet oscillations are diminished.

Heat transfer and pressure drop

The effect of pulse repetition rate on heat transfer and pressure drop for duty cycle values of 25%, 50% and 75% at mass flux of 55 kg/m²s and average quality of 45%is shown in Figures 6.41. The horizontal lines represent the weighted average of the heat transfer coefficient and pressure drop assuming that the flow oscillates between 0 kV and 8 kV with no transient effects. The heat transfer and pressure drop ratios for the 8 kV DC case are 2.5 and 4.2 respectively (Figure 5.7). The effect of pulse repetition rate on heat transfer can be divided into three regions. In the first region, or pulse repetition rate less than 5 Hz, the heat transfer coefficient is higher than the DC weighted average heat transfer coefficient for all three duty cycles. This higher heat transfer values compared to the DC case can be attributed to the higher heat transfer associated with the transient "liquid twisted cone" flow patterns which is not accounted for in the weighted average heat transfer values. The greater the number of liquid "twisted cone" patterns indicates that more liquid is being extracted away from the heat transfer surface and therefore the higher the heat transfer. Increasing the pulse repetition rate in this range (less than 5 Hz), decreases the time duration of the liquid "twisted cone" patterns and therefore decreases the heat transfer ratio. In the second region, for the pulse repetition rate in the range 5 Hz < f < 40 Hz, the heat transfer is less than that for the weighted average. Increasing the pulse repetition rate in this range diminishes the liquid "twisted cone" patterns until they completely disappear for f > 10 Hz. The liquid vapour interactions below the electrode are decreased and therefore the heat transfer ratio is decreased. For the 25% duty cycle, the variation of the heat transfer ratio with frequency is not very prominent. There is a small decrease initially with pulse repetition rate and then it remains nearly



Figure 6.41: The effect of pulse repetition rate on (a) heat transfer and (b) pressure drop, $\triangle 25\%$ duty cycle, $\Box 50\%$ duty cycle, $\diamond 75\%$ duty cycle, - weighted average 25\%, -- weighted average 50\%, -- weighted average 75\%. Heat flux of q''=5.7 kW/m² and average quality of $x_{avg}=45\%$.

constant (1.4) for pulse repetition rates greater than 10 Hz. This is likely because the fraction of time which the flow is subjected to the 8 kV (25% of the time period) is not sufficient to cause significant liquid extraction. In this case, the main reason for the heat transfer enhancement is the electrically induced liquid vapour interactions which destabilize the thermal boundary layer. For the 75% duty cycle, the fraction of time at which the two-phase flow is subjected to EHD (75% of the time period) is greater than that for the 25% and 50% duty cycles. This leads to higher heat transfer ratios (2.4 at 0.5 Hz and 1.65 at 40 Hz) because of a greater amount of liquid extraction from the bottom stratum.

Measuring heat transfer associated with the transient flow patterns is impractical due to the short time duration of these flow patterns compared to the time response of the temperature and pressure drop sensors. A simple model to estimate the heat transfer coefficients associated with the transient flow patterns is developed. The overall heat transfer coefficient for a typical pulse wave voltage with a time period T and duty cycle D (Figure 6.42) can be expressed as

$$h = \left(1 - \frac{D}{100}\right) h_{0\,kV} + \left(\tau \frac{D}{100}\right) h_{transient} + \left((1 - \tau) \frac{D}{100}\right) h_{8\,kV} \tag{6.13}$$

where (D/100)T is the time duration of the high voltage and $\tau(D/100)T$ is the time duration of the transient flow patterns, h_{0kv} , h_{8kV} and $h_{transient}$ are the heat transfer coefficients corresponding to 0 kV (without EHD), 8 kV DC and transient flow patterns respectively. Knowing h, h_{0kV} , and h_{8kV} from experimental measurements, the heat transfer coefficient associated with the transient flow patterns $(h_{transient})$ is estimated using Equation 6.13. The time duration of the transient flow patterns is estimated from the flow visualization experiments. The time duration of the transient flow patterns for mass flux of 55 kg/m²s and average quality of 45% is estimated to be 0.55 sec. For 50% duty cycle and 1 Hz frequency applied voltage, h is 4972 W/m²K, $h_{0\,kV}$ is 2673 W/m²K, and $h_{8\,kV}$ is 6461 W/m²K. Substituting in Equation 6.13 gives $h_{transient} = 7271$ W/m²K. The $h_{transient}$ values for the different applied voltage are shown in Table 6.1. The average value of the heat transfer coefficient associated with the transient flow patterns ($h_{transient}$) is 7241 W/m²K which is equivalent to heat transfer ratio ($h_{transient}/h_{0\,kV}$) of 2.7 compared to 2.4 for the 8 kV DC.

In the third region, for pulse repetition rate greater than 40 Hz, the heat transfer ratio is nearly constant for the three duty cycles; however, the ratio increases with an increase in the duty cycle. The flow visualization results (Figure 6.33 (b)-(d)) for this pulse repetition rate range show no liquid extraction above the electrode. This is due to the voltage on duration period in this range of pulse repetition rate is less than the time needed for the liquid to be extracted to the electrode and pushed to the vapour core ($\tau_1 = 0.025$ sec). The flow visualization results, however, show a continuous formation and destruction of liquid droplets which oscillate with the same frequency as the applied voltage signals up to 100 Hz which agree with Cotton [2009]. Although the frequency of the droplet oscillations follows the pulse repetition rate of the applied voltage voltage, the value of the heat transfer ratio remains relatively unchanged in this pulse repetition rate range. This suggests that the heat transfer enhancement in this range is mainly due to the destabilization of the thermal boundary which is not significantly affected by the oscillations of the droplets. The heat transfer ratio in this range (pulse repetition rate greater than 40 Hz) remains relatively constant at a value of 1.3, 1.4 and 1.6 for 25%, 50% and 75% duty cycle respectively. These values lie between the values at applied DC voltage of 0 kV (heat transfer ratio of 1) and 8 kV (heat transfer ratio of 2.7). The heat transfer values, for pulse repetition rate greater than 40 Hz, are less than those for the weighted average. This is mainly attributed to the non-linear relationship between the EHD force and



Figure 6.42: Graphical representation of one pulse including the transient effects.

	Time duration of the transient flow patterns=0.55 sec		
	25% duty cycle	50% duty cycle	75% duty cycle
0.5 Hz	7377	7850	8592
1 Hz	7271	7271	7277
2 Hz	6901	6929	7349
4 Hz	6745	6469	6868
	Average value	7241	
	Standard deviation	7.7 %	

Table 6.1: Heat transfer coefficient associated with the transient flow patterns $(h_{transient})$

the applied voltage (electric field), Equation 3.24. Thus, the average heat transfer value obtained from applied voltages at 0 kV and 8 kV will be greater than the value at the average voltage. This suggests that there could be an "effective" voltage that determines heat transfer when using AC/pulse voltages.

The effect of pulse repetition rate on pressure drop shows different characteristics than that on heat transfer (Figure 6.41(b)). This is mainly due to the presence of the central high voltage electrode which contributes significantly to the pressure drop across the test section. The pressure drop for the current flow configuration is due to (i) momentum transfer at the wall and (ii) to the momentum transfer at the electrode surface. The latter will not contribute to any heat transfer enhancement. For the 25% duty cycle, the pressure drop ratio increases from 1.5 at pulse repetition rate of 0.5 Hz to approximately 2.58 at pulse repetition rate of 100 Hz. In the low pulse repetition rate range (f < 10 Hz), where liquid extraction from the bottom stratum occurs in shape of liquid "twisted cone" patterns above the electrode, the pressure drop is less than that for the DC weighted average. This indicates that the increase in the pressure drop due to these transient flow patterns is less than the increase in pressure drop for the 8 kV DC case. Increasing the pulse repetition rate increases the oscillations of the liquid vapour interactions below the electrode and the liquid droplets and therefore increases the pressure drop ratio till it exceeds that for the DC weighted average at pulse repetition rate 8 Hz and reaches a maximum value of 2.58 at 100 Hz. For pulse repetition rate greater than 100 Hz, the pressure drop decreases with an increase of pulse repetition rate. This is mainly because the response of the two-phase flow (and the liquid droplets) to the applied signals decreases with a resultant decrease in intensity of the liquid-vapour interactions and therefore the pressure drop. The spectral analysis of the liquid droplet oscillations shows no response of the liquid droplets to the applied signals for frequencies greater than 100 Hz.



Figure 6.43: Effect of duty cycle on (a) heat transfer and (b) pressure drop, $\triangle 1$ Hz, $\Box 50$ Hz, $\diamond 800$ Hz. Heat flux of q''=5.7 kW/m² and average quality of $x_{avg}=45\%$.

For the 50% and 75% duty cycles, the pressure drop characteristics are qualitatively similar to the 25% duty cycle. There are two main differences between the trends for the different duty cycles; the first difference is that the initial rate of the increase and the decrease of the pressure drop ratio are higher for the 25% duty cycle compared to the 50% and 75% duty cycles. The effect of the pulse wave applied voltage on pressure drop can be divided into two factors: (i) the increase in pressure drop during the voltage-on period (the flow is subjected to the applied 8 kV), which increases with the duty cycle and (ii) the increase in pressure drop due to the oscillations of the liquid vapour interface and the liquid droplets, which increases with the pulse repetition rate. At lower duty cycles (25%), the latter effect is more significant than that due to the voltage-on period. Therefore the effect of pulse repetition rate will be more significant causing higher rates of change in pressure drop compared to the higher duty cycles. Increasing the duty cycle decreases the impact of the change in the pulse repetition rate and therefore decreases the rate of change of pressure drop with pulse repetition rate.

The effect of duty cycle on heat transfer ratio and pressure drop ratio for pulse repetition rate of 1 Hz, 50 Hz and 800 Hz is shown in Figure 6.43. At 0% duty cycle the heat transfer and pressure drop values correspond to those for 0 kV DC applied voltage. For the same pulse repetition rate, increasing the duty cycle increases the effective value of the applied voltage and therefore increases heat transfer and pressure drop till they reach the corresponding values of 8 kV DC applied voltage at 100% duty cycle. The results of Figure 6.43 agree with the results of the previous section, where the heat transfer ratio is higher at 1 Hz compared to 50 Hz and 800 Hz as there is enough time to complete the liquid extraction cycle. The pressure drop ratio will be higher at 50 Hz compared to 1 Hz and 800 Hz due to the droplet oscillations which increase the momentum transfer between the two-phase flow and the center electrode.

The effect of pulse repetition rate for a 50% duty cycle pulse wave applied voltage on heat transfer and pressure drop for three different mass fluxes (57 kg/m²s, 100 kg/m²s and 150 kg/m²s) are shown in Figure 6.44. The effect of pulse repetition rate on heat transfer and pressure drop ratios at different mass fluxes has similar trends qualitatively but are different quantitatively. The heat transfer ratio decreases from 1.9 at pulse repetition rate of 1 Hz to 1.4 at pulse repetition rate of 800 Hz for mass flux of 57 kg/m²s. For mass fluxes 100 kg/m²s and 150 kg/m²s, the heat transfer ratio decreases from 1.4 and 1.2 at 1 Hz to 1.1 at 800 Hz respectively. Similarly, the maximum pressure drop decreases from 3.5 at 57 kg/m²s to 2.4 at 100 kg/m²s to 2.1 at 150 kg/m²s for the same pulse repetition rate of 80 Hz. For the same amplitude and duty cycle, increasing the mass flux decreases the ratio (Md/Re_l^2) , as discussed in chapter 5, and therefore decreases the effect of EHD on heat transfer and pressure drop for the whole range of pulse repetition rate. At mass flux = 57 kg/m²s, the dimensionless ratio (Md/Re^2) is approximately 0.7 and decreases to 0.12 and 0.05 at mass fluxes of 100 kg/m²s and 150 kg/m²s respectively.

To examine the benefits of using a pulse wave applied voltage compared to DC voltages at different mass fluxes, the effect of 8 kV DC applied voltage on the ratio of heat transfer coefficient to pressure drop is compared to that for 8 kV amplitude pulse wave applied voltage for pulse repetition range from 0.5 Hz to 200 Hz and duty cycle range from 5% to 95% in Figure 6.45. At mass flux of 50 kg/m²s, changing the pulse repetition rate and the duty cycle of the pulse wave applied voltage provides a control range of the ratio of heat transfer to pressure drop from 8.24 to 20.56, compared to a fixed value of 11.92 for the DC case. The control range decreases by increasing the mass flux till it reaches to from 1.63 to 3.81 for mass flux 150 kg/m²s compared to a single value of 1.64 for the DC case.



Figure 6.44: Effect of pulse repetition rate on (a) heat transfer ratio and (b) pressure drop ratio for 50% duty cycle, 0 to 8 kV applied pulse wave ($\Box G = 57 \text{ kg/m}^2 \text{s}, \diamond G = 100 \text{ kg/m}^2 \text{s}, \Delta G = 150 \text{ kg/m}^2 \text{s}$).



Figure 6.45: Effect of mass flux on heat transfer to pressure drop ratio at constant average quality x = 45% for $\triangle 8$ kV DC applied voltage and $\bullet - \bullet 8$ kV amplitude pulse wave applied voltage with different pulse repetition rates and duty cycles.

Summary

The results show that applying pulse wave signals provide a wide range of control of heat transfer and pressure drop compared to DC applied voltage case. Different combinations of heat transfer and pressure drop values can be achieved by varying the pulse repetition rate and the duty cycle of the applied pulse wave voltage. The different characteristics of the pulse waveform compared to the DC case are due to the appearance of new two-phase flow patterns which have not been observed in the DC case. Theses two-phase flow patterns can be controlled by manipulating the pulse repetition rate and the duty cycle of the applied voltage.

Chapter 7

Conclusions and Recommendations

7.1 Summary and Conclusions

The effect of high voltage DC, AC and pulse wave electric fields on heat transfer and pressure drop for tube side convective condensation was investigated. The study was performed in four stages: (1) Experimental study of the effect of DC applied voltage on two-phase flow patterns, heat transfer and pressure drop, (2) Experimental study of the characteristics of tube side convective condensation under the effect of pulse wave and AC applied voltage with different waveform parameters, (3) Experimental and numerical studies of the dynamics of two-phase flow under the effect of a step input high voltage and (4) Experimental study of pulse repetition rate and duty cycle of pulse wave voltage on two-phase flow patterns, heat transfer and pressure drop for different flow conditions.

The experiments were performed in a horizontal, single-pass, counter-current heat exchanger with a rod electrode placed along the center of the tube. Two test sections were used in the current investigation. One (test section B) was 1.5 m long and the second test section (test section B) was 30 cm long. The shorter test section was used to maintain the quality and hence the flow pattern relatively constant over the test section. The outer water jacket was constructed from a PVC tube with inner diameter of 19.05 mm and wall thickness of 3.81 mm. The inner tube containing the two phase refrigerant is a stainless steel tube with an outer diameter of 12.7 mm and an inner diameter of 10.2 mm. A central 3.18 mm diameter stainless steel rod electrode was used to apply the electric field across the annular gap formed by the electrode and the surface of the inner stainless steel tube. The flow was visualized at the exit of the heat exchanger using a high speed camera through a transparent quartz tube coated with an electrically conductive film of tin oxide. The electric field was established by amplifying low voltage signals from a function generator using a high-voltage amplifier.

The application of high voltage electric fields induces interfacial electric forces at the liquid–vapour interface. These forces, when sufficiently high, cause the liquid to be extracted away from the proximity of the heat transfer surface to the bulk of the two-phase flow. Liquid extraction changes the mode of heat transfer from convective condensation to a mode similar to the film condensation for the given flow conditions (mass flux and vapour quality). This decreases the thermal resistance of the system and therefore increases the heat transfer. The pressure drop increases under the effect of EHD due to the destabilization of the boundary layer and the increase of the momentum transfer between the two-phase flow and the center electrode. For an applied DC voltage of 8 kV and average quality of 45%, the heat transfer and pressure drop ratios (defined as the ratios of the heat transfer coefficient and pressure drop with applied voltage to those without the applied voltage) are 2.8 and 4.5 at the lowest mass flux ($G=57 \text{ kg/m}^2\text{s}$) respectively. The heat transfer and pressure drop ratios decrease with an increase in mass flux with a value of 1.2 and 2.8 at the highest mass flux investigated of 156 kg/m²s. The dimensionless analysis of the momentum equation including the EHD forces showed that the effect of EHD will be significant if $Md/Re_l^2 > 1$. Increasing the mass flux decreases the ratio Md/Re^2 and therefore decreases the effect of EHD forces on the two-phase flow heat transfer and pressure drop.

Experiments were performed using sine, square and pulse wave applied voltages. The effect of frequency, pulse repetition rate, duty cycle, amplitude, and DC bias was investigated. The effect of EHD on heat transfer and pressure drop is higher for the case of a 50% duty cycle square/pulse wave compared to a sine wave for the same frequency, amplitude and DC offset. This is attributed to the higher rms voltage of the square/pulse wave compared to the sine wave. The use of pulse wave voltages is more advantageous than a sine wave voltage because better control can be accomplished by controlling the duty cycle. The effect of pulse repetition rate and duty cycle on heat transfer has different characteristics compared to that on the pressure drop. This can be utilized to obtain different combinations of heat transfer and pressure drop by changing the pulse repetition rate and duty cycle for the same flow conditions.

To get better insight into the effect of pulse wave applied voltage on tube side convective condensation, the effect of a 8 kV step input voltage on the transient characteristics of two-phase flow patterns, surface temperatures and pressure drop were investigated. Flow visualization showed that the step input voltage caused the liquid to be extracted from the bottom liquid stratum to the center electrode and then pushed back to the bulk flow in the form of liquid "twisted-cone" patterns. These transient flow patterns diminish in steady state. These flow patterns are characterized by higher heat transfer values due to less liquid in touch with the heat transfer surface compared to the DC case. Numerical simulation were performed to predict the liquid extraction time from a stratified liquid-vapour zone. The simulation results were found to be in a good agreement with the experimental results. The extraction time increases with a decrease of the applied voltage and/or an increases of the initial gap distance between the electrode and the liquid-vapour interface due to the decrease of the induced interfacial EHD forces.

For the current electrode geometry, the effect of pulse repetition rate of a 8 kV amplitude pulse wave applied voltage on two phase flow regime, heat transfer and pressure drop can be divided into three ranges. At the low pulse repetition rate range, the two-phase flow responds to the induced EHD forces, and liquid is extracted from the bottom stratum to the center electrode and then pushed back to the bulk flow in the form of liquid "twisted-cone" patterns. Increasing the pulse repetition rate in this range increases the repetition of the extraction cycle and therefore increases heat transfer and pressure drop. In the mid pulse repetition rate range, the two phase flow does not have enough time to complete the extraction cycle which leads to lower heat transfer values compared to the lower pulse repetition rate range. In this range, the two phase patterns are characterized by liquid-vapour interface oscillations between the center electrode and the bottom stratum and liquid droplet oscillations which increases the momentum transfer and therefore pressure drop. Increasing the pulse repetition rate in this range decreases heat transfer and increases pressure drop. In the high pulse repetition rate range, increasing the pulse repetition rate decreases both the interfacial and droplet oscillations and therefore decreases the heat transfer and pressure drop till the two phase flow patterns resembles that for a DC case where pulse repetition rate has no effect on two-phase flow, heat transfer and pressure drop. For the same frequency, increasing the mass flux decreases the ratio Md/Re_l^2 and therefore decreases the effect of EHD on heat transfer and pressure drop for the whole range of frequency.

The conclusions of the current investigation can be summarized as:

- Applying high voltage pulse wave signals with different frequencies and duty cycles provides a greater range of control of heat transfer and pressure drop during tube side convective condensation compared to DC signals.
- The power consumption using pulse/AC applied voltage is approximately the same magnitude as that using DC applied voltage (details of power consumption analysis can be reviewed in Appendix B).
- Manipulating the pulse repetition rate and duty cycle leads to different twophase flow patterns and therefore different values of heat transfer to pressure drop ratios which is very beneficial in various industrial applications.
- The transient two phase flow patterns ("twisted-cone") due to the application of a step input voltage are associated with higher heat transfer compared to the steady 8 kV DC case.
- Numerical simulations predictions of the liquid extraction time for a stationary stratum are in a good agreement with the experimental results .
- At low mass flux (stratified flow), applying EHD changes the heat transfer mechanism from convective condensation for the bottom stratum into film condensation which increases the overall heat transfer.

7.2 Research Contributions

The key research contributions from this thesis are

1. Performed a parametric experimental study to investigate the effect of high voltage DC, AC and pulse wave applied voltage on two phase flow regimes, heat transfer and pressure drop during tube side convective condensation. ("In-tube convective condensation under AC high voltage electric fields", Journal of Enhanced Heat Transfer, accepted for publication, 2009)

- Elicited the mechanisms of heat transfer enhancement and pressure drop penalty due to the application of EHD for DC and pulse wave voltage.
 ("Effect of Frequency on Two-Phase Flow Regimes under High Voltage AC Electric Fields", Journal of Electrostatics, Vol. 6, p. 25-31, 2008)
- Evaluated the transient time needed for liquid extraction from a bottom stratum to the vapour core for different applied voltages and different liquid levels.
 ("Visualization of Flow Regime Transitions in Two-Phase Flow Under High Voltage Electric Fields", Proceedings of ASME Fluids Engineering Division Summer Meeting 2006)
- 4. Developed a numerical algorithm to estimate liquid extraction time and the minimum voltage required for liquid extraction.

("Numerical Simulation of the Liquid Extraction from a Stratified Liquid-Vapor Zone using Electrohydrodynamics", ICHMT International Symposium on Advances in Computational Heat Transfer, Marrakech, Morocco, 2008)

5. Examined the advantages of using pulse wave applied voltage for potential control of heat transfer and pressure drop for tube side convective condensation compared to DC applied voltage (Manuscript in progress for submission to the International journal of Heat and Mass Transfer).
7.3 Recommendations For Future Work

This study showed that there is good potential for using AC/pulse wave voltage for heat transfer and pressure drop control in heat exchangers using dielectric fluids. However, further research is required before EHD can be used as a control tool in industrial applications. Recommendations for future research includes experimental, numerical and analytical studies to completely understand the effect of pulse wave applied voltage on two-phase heat transfer and pressure drop for industrial heat exchangers such as a plate type heat exchanger.

- Numerical simulations of the electric fields can be used to optimize the electrode geometry for the practical implementation in heat exchangers.
- Update the numerical model developed in the current research by incorporating the surface tension and surface charges and by extending the model from two dimensional to a three dimensional model. This is necessary to precisely predict liquid extraction for two phase flow due to EHD forces.
- Extend the numerical model to predict the transient two phase flow patterns after the liquid touches the bottom surface of the electrode (i.e. τ_1 and τ_2). This is essential to be able to optimize the pulse repetition rate and duty cycle of the applied pulse wave to control the two phase flow patterns, and therefore heat transfer and pressure drop.
- Although the experiments showed that there is no effect of polarity on liquid extraction, it can influence the liquid-vapour interactions after the liquid touches the electrode. The effect of polarity of the applied voltage needs to be studied numerically and experimentally for different flow condition and applied waveforms.

- More research is required to understand the physical mechanisms of the formation and destruction of the transient liquid-vapour interactions (liquid "twisted cone" patterns) and liquid droplets entrainment. This allows for a better analysis of the effect of these patterns on heat transfer and pressure drop.
- Due to the limitations of the current visualization test section, it was not possible to view the cross sectional view of the two-phase flow pattern. Additional flow visualization and void fraction techniques are required to observe cross sectional two phase flow distributions.

Appendix A: Transport and Electrical Properties of R-134A

Transport Properties

The refrigerant R-134a transport properties are adopted from ASHRAE Handbook-Fundamentals (ASHRAE [1997]). Table B.1 lists the transport property equations correlated for the range of refrigerant temperatures measured in this study (10 °C to 40 °C).

Property	State	Correlation	R ²
Pressure	Saturated	$P_s = -184.53 + 474.73 \exp\left(T_s/42.96\right)$	0.9999968
Density	Liquid	$\rho_L = 1290.68 - 2.317 \ T^{1.1173}$	0.999979
	(subcooled)		
Specific heat	Liquid	$c_{pL} = 1.1527 + 0.1824 \ exp(-T/62.569)$	0.9999528
	(subcooled)		
Viscosity Liquid		$\mu_L = 18.382 + 268.55 \ exp(-T/77.425)$	0.9999945
	(subcooled)		
Thermal	Liquid	$k_L = 93.443 - 0.4648 \ T^{0.99816}$	0.9999426
conductivity	(subcooled)		

 Table A1:
 Transport property correlations

Property	State	Correlation R ²		
	Liquid	$\rho_{Ls}^{0.5} = 38.378 - 0.1414 \ P^{0.5}$	0.9999959	
Density	(saturated)			
	Vapour	$ \rho_{Vs} = -371.336 + 372.316 \ exp(P/8230.56) $	0.9999995	
	(saturated)			
	Liquid	$c_{pLs}^2 = 1.5364 + [0.00482 \ P/ln(P)]$	0.9999293	
Specific heat	(saturated)			
	Vapour	$ln(c_{pVs}) = -0.304973 + 0.0018896 P^{0.5} ln(P)$	0.9999316	
	(saturated)			
	Liquid	$\mu_{Ls}^{0.5} = 32.299 - 2.87848 \ln{(P)}$	0.9999916	
Viscosity	(saturated)			
	Vapour	$\mu_{Vs}^{0.5} = 2.9554 - 0.02081 \ P^{0.5}$	0.9999128	
	(saturated)			
	Liquid	$k_{Ls}^{-1} = 0.008661 + 2.11E - 5 P^{0.5} ln(P)$	0.9999232	
Thermal	(saturated)			
conductivity	Vapour	$k_{Vs}^{-1} = 0.17917 - 0.01662 \ ln\left(P ight)$	0.9999789	
	(saturated)			
Units: T (°C), P (kPa), ρ (kg/m ³), c_p (kJ/kgK), μ (μ Pas), k (mW/mK)				

 Table A1: Transport property correlations (continued)

Electrical Properties

For better analysis of the EHD phenomena, accurate measurement of the electrical properties of the working fluid is necessary. Several studies have explored the electrical properties of different refrigerants including R134a. Cotton [2000] presented a comparison between the most relevant studies with the measurements of this experiment and discussed the reasons of discrepancies, if they exist, between different studies. Due to the similarity of the working fluid and the electrode configuration between Cotton's investigation and the current research, the electrical properties used

in the present study are the same as Cotton [2000].

Based on the comparison between Fellows et al. [1991], Gurova et al. [1994] and Barao et al. [1996] and the experimental measurements, Cotton [2000] recommended using Gurova et al. [1994] correlation (Equation A1) for the liquid specific dielectric constant along the saturation line.

$$\varepsilon_L = \sum_{i=0}^3 c_i T^i$$
 , $\varepsilon_L = \sum_{i=0}^3 d_i \rho^i$ (A1)

where

$$c_{0} = 11.3597 \pm 0.0104 \qquad d_{0} = -203.655 \pm 0.011$$

$$c_{1} = -83.5 \pm 1.7E - 3 \qquad d_{1} = 51.7pm3.3E - 2$$

$$c_{2} = 30 \pm 1E - 5 \qquad d_{2} = 43.00 \pm 0.25E - 4$$

$$c_{3} = -26 \pm 1E - 7 \qquad d_{3} = 1.27 \pm 0.06E - 7$$

with T in °C and ρ in kg/m³. (Gurova et al. [1994] stated that T in Kelvin).

As reviewed by Cotton [2000], the measurements of gaseous specific dielectric constant for refrigerant appears in two previous studies (Meyer and Morrison [1991] and Barao et al. [1995]). The comparison between the two studies shows that the approximate range of the saturated vapor specific dielectric constant for the temperature range of the current investigation (20.0 °C $< T_{sat} < 26.0$ °C) is between 1.085 and 1.100. Cotton [2000] suggested using the average value of $\varepsilon_V = 1.09$ as the saturated vapor specific dielectric constant for the temperature range used in the current study.

Appendix B: Comparison of Masuda and EHD numbers

In this Appendix, a study of the relative importance of the electrophoretic force which is represented by the EHD number and the dielectrophoretic and electrostrictive forces represented by the Masuda number is presented. In the first section, the single phase liquid, vapour, and two-phase EHD and Masuada dimensionless numbers are evaluated for different values of current and voltages. The appropriate current values for the estimation of the EHD number are examined in the second section.

Calculations

The (Ehd) and (Md) can be expressed as, IEEE-DEIS-EHD Technical Committee [2003]

$$Ehd = \frac{I_o L^3}{\rho_o \nu^2 \mu_e A} \qquad \qquad Md = \frac{E^2 \varepsilon_o L^2 T_o \left(\partial \varepsilon_s / \partial T\right)}{2\rho_o \nu^2} \tag{B1}$$

where

A (area of the electrode)= $\pi D_e l = 0.003 \text{ m}^2$ L= Hydraulic diameter = $D_o - D_e = 0.01092 - 0.00318 = 0.00774 \text{ m}$

For refrigerant R134a:

 $\rho_o|_{liquid} = 1200 \text{ kg/m}^3$ $\rho_o|_{vapour} = 32 \text{ kg/m}^3$



Figure B1: Electrode geometry and the liquid and vapour domain.

$\mu _{liquid}$ (viscosity)= 0.00022 Pas	$\mu _{vapour}(viscosity) = 0.000012 $ Pas
$\mu_e _{liquid}$ (ion mobility)= 9e-8	$\mu_e _{vapour}$ (ion mobility)= 0.001
$\frac{\partial \varepsilon_s}{\partial T} _{liquid} = 0.074$	$\frac{\partial \varepsilon_s}{\partial T} _{vapour} = 0.0025$
$\varepsilon_o = 8.854 \text{e}{-12} \text{ A}^2 \text{ s}^4 \text{ kg}^{-1} \text{ m}^3$	$T_o = 24$ °C

The EHD number and the Masuda number for both single phase liquid, single phase vapour and two-phase flow are plotted against current and applied voltage in Figures B2-B5. The Masuda numbers in case of single phase liquid or vapour are evaluated using the electric field values at the electrode surface (E_o) , while in twophase flow the electric field values in the vapour phase at the liquid-vapour interface (Figure B1) are used. The electric field strength values for single phase and two phase (Figure B1) case are obtained numerically using commercial software (Comsol). The following conclusions can be drawn from the figures:

1. The EHD number for the milliamp range of current is four orders of magnitude higher than that for the microamp range for both liquid (Figures B2a and B3a) and vapour (Figures B2b and B3b).

2. The EHD and Masuda numbers for single-phase liquid (Figures B2a, B3a and B4a) are higher than that of single phase vapour (Figures B2b, B3b and B4b).

3. The liquid and vapour Masuda number values in case of two-phase flow are almost2.2 times higher than the Masuda number values in case of single phase flow.

In summary, the figures indicate that the EHD number is higher than the Masuda number in both the liquid phase and the vapour phase for current in the milliamp range. However, this does not reflect the relative importance of the electrophoretic force and the dielectrophoretic force in two-phase flows. The Masuda number under estimates the effect of the dielectrophoretic force in two-phase flow. This is mainly because that Masuda number does not take into account the interfacial discontinuity in the dielectric constant in two phase flow. The dielectrophoretic force due to this discontinuity can be very significant. Furthermore, the EHD number is derived based on the assumption that the free charge density within the fluid is equal to the charge injected by the electrodes. To compare between the electrophoretic force and the dielectrophoretic force in two-phase flows, the two-phase Masuda number should be compared to the vapour EHD number, as the electrode (source of injected charges) is surrounded by the vapour phase. This means that the two-phase Masuda number (Figure B5) is in the same order of magnitude of the EHD number for current values in the milliamp range (Figure B3b) and three order of magnitude higher than the EHD number for current values in the microamp range (Figure B3b).

Current and Voltage Profiles

To investigate the right appropriate current values for the estimation of the EHD number, the voltage and current profiles for a step applied voltage with rise time of 0.1 ms, 1 ms, 10 ms and 100 ms are shown in Figures B6-B9 respectively. The total current measured is a summation of conduction current (mainly a function of σ and E) and displacement current (mainly a function of dE/dt). The current value before the application of the step voltage is approximately 0 mA and increases sharply at the application of the sudden step voltage. The current reaches a maximum value before falling back to its initial value. The current profile indicates that the current spike is mainly a displacement current and that the conduction current is the same value as the initial and final values. The maximum current value is approximately 24 mA for the 0.1 ms rise time, 3 mA and 0.3 mA for 1 ms and 10 ms rise time respectively. The diminishing of the current spike with an increase of the rise time implies that the measured current values are primarily displacement current and that the conduction current is in the order of microamp.

The power consumption due to the application of the pulse wave voltage is evaluated by

$$P = I \times V_{rms} \tag{B2}$$

where V_{rms} is the square root mean value of applied pulse voltage waveform, and I is the conduction current which is in the microamp range as shown from figures B6-B9. The effect of duty cycle on power consumption for an 8 kV applied pulse wave is shown in Figure B10. Increasing the duty cycle increases the V_{rms} of the applied pulse wave and therefore increases the power consumption. The maximum power consumption is less than 1 W which corresponds to 2% of the heat load in the

test section.

-



Figure B2: a) Single phase liquid EHD number vs. current in microAmp range. b) Single phase vapour EHD number vs. current in microAmp range.



Figure B3: a) Single phase liquid EHD number vs. current in milliAmp range. b) Single phase vapour EHD number vs. current in milliAmp range.



Figure B4: a) Single phase liquid Masuda number vs. voltage. b) Single phase vapour Masuda number vs. voltage.



Figure B5: a) Two-phase liquid Masuda number vs. voltage. b) Two-phase vapour Masuda number vs. voltage.



Figure B6: Temporal profile for voltage and current and for an applied DC step voltage of 8 kV with rise time of 0.1 msec.



Figure B7: Temporal profile for voltage and current and for an applied DC step voltage of 8 kV with rise time of 1 msec.



Figure B8: Temporal profile for voltage and current and for an applied DC step voltage of 8 kV with rise time of 10 msec.



Figure B9: Temporal profile for voltage and current and for an applied DC step voltage of 8 kV with rise time of 100 msec.



Figure B10: Power consumption vs. duty cycle for 8 kV amplitude pulse wave.

Appendix C: EHD Interfacial Force

In this section, the derivation of the surface force acting on the interface between two dielectric mediums is represented (Scaife [1989]). Neglecting the electrophoretic term, the electrohydrodynamic body force acting on a dielectric medium can be expressed as [Panofsky and Phillips [1962], Stratton [1941] and Landau and Lifshitz [1960]]

$$\bar{f}_{e}^{\prime\prime\prime} = -\frac{\varepsilon_{o}}{2}E^{2}\nabla\varepsilon_{s} + \frac{\varepsilon_{o}}{2}\nabla\left[\rho E^{2}\left(\frac{\partial\varepsilon_{s}}{\partial\rho}\right)_{T}\right]$$
(C1)

In index notation, Equation C1 reads

$$f_{e,i} = -\frac{1}{2}\varepsilon_o E^2 \partial_i \varepsilon_s + \frac{1}{2}\varepsilon_o \partial_i \left(E^2 \rho \partial_\rho \varepsilon_s\right) \tag{C2}$$

Assuming no free charges,

$$\nabla \cdot \bar{D} = \nabla \cdot \left(\varepsilon_o \varepsilon_s \bar{E} \right) = \varepsilon_o \partial_j \varepsilon_s E_j = 0 \tag{C3}$$

For a static electric field,

$$curl\,\bar{E} = 0 \tag{C4}$$

which implies that

$$\partial_i E_j = \partial_j E_i \tag{C5}$$

Therefore

$$\partial_{j} \left(\varepsilon_{s} E_{i} E_{j} \right) = E_{i} \partial_{j} \left(\varepsilon_{s} E_{j} \right) + \varepsilon_{s} E_{j} \partial_{j} E_{i}$$
$$= \varepsilon_{s} E_{j} \partial_{j} E_{i}$$
$$= \varepsilon_{s} E_{j} \partial_{i} E_{j}$$
(C6)

The first term in Equation C2 can be expressed as

$$-\frac{1}{2}\varepsilon_{o}\left(E^{2}\partial_{i}\varepsilon_{s}\right) = -\frac{1}{2}\varepsilon_{o}\left(E_{j}E_{j}\partial_{i}\varepsilon_{s}\right) = -\frac{1}{2}\varepsilon_{o}\left[\partial_{i}\left(\varepsilon_{s}E_{j}E_{j}\right) - 2\varepsilon_{s}E_{j}\partial_{i}E_{j}\right]$$
$$= -\frac{1}{2}\varepsilon_{o}\left[\partial_{i}\left(\varepsilon_{s}E_{j}E_{j}\right) - 2\partial_{j}\left(\varepsilon_{s}E_{i}E_{j}\right)\right]$$
(C7)

Equation C2 can be written in the form

$$f_{e,i} = -\partial_i \left[\frac{1}{2} \varepsilon_o E^2 \left(\varepsilon_s - \rho \partial_\rho \varepsilon_s \right) \right] + \partial_j \left(\varepsilon_o \varepsilon_s E_i E_j \right)$$
$$= -\partial_j \left[\frac{1}{2} \varepsilon_o E^2 \left(\varepsilon_s - \rho \partial_\rho \varepsilon_s \right) \right] \delta_{i,j} + \partial_j \left(\varepsilon_o \varepsilon_s E_i E_j \right)$$
(C8)

Any volume force, $\bar{f},$ can be expressed in terms of a stress tensor as

$$f_i = \partial_j S_{i,j} \tag{C9}$$

From Equations C8 and C9, we find that

$$S_{i,j} = -\frac{1}{2} \varepsilon_o E^2 \left(\varepsilon_s - \rho \partial_\rho \varepsilon_s \right) \delta_{i,j} + \varepsilon_o \varepsilon_s E_i E_j \tag{C10}$$

In matrix notation

$$\begin{bmatrix} S_{i,j} \end{bmatrix} = \begin{bmatrix} S_{1,1} & S_{1,2} & S_{1,3} \\ S_{2,1} & S_{2,2} & S_{2,3} \\ S_{3,1} & S_{3,2} & S_{3,3} \end{bmatrix}$$
(C11)

where

$$S_{i,j} = -\frac{1}{2} \varepsilon_o E^2 \left(\varepsilon_s - \rho \partial_\rho \varepsilon_s \right) + \varepsilon_o \varepsilon_s E_i^2 \quad \text{for} \quad i = j$$
$$= \varepsilon_o \varepsilon_s E_i E_j \quad \text{for} \quad i \neq j \tag{C12}$$

Consider a boundary between two dielectric media, labeled 1 and 2, as illustrated in Figure C1. The unit normal vector, \hat{n}_1 , directed from medium 1 to medium 2, is opposite to \hat{n}_2 , the unit normal vector pointing from medium 2 to medium 1. Neglecting surface tension forces, the net stress, $\Pi_{1\to 2,i}$, acting in the direction of $n_{1,i}$ is given by

$$\Pi_{1 \to 2,i} = (S_{1;i,j} - S_{2;i,j}) n_{1,j} \tag{C13}$$

where



Figure C1: The interface separating fluid 1 from fluid 2 has a unit normal vector $\hat{n}_1 = -\hat{n}_2$ directed from medium 1 to medium 2.

$$S_{k;i,j} = -\frac{1}{2} \varepsilon_o E_k^2 \left(\varepsilon_{k,s} - \rho_k \partial_{\rho_k} \varepsilon_{k,s} \right) \delta_{i,j} + \varepsilon_o \varepsilon_{k,s} E_{k,i} E_{k,j} \quad k = 1 \text{ or } 2$$
(C14)

The choice of the Cartesian coordinates is arbitrary so we take the unit normal vector \hat{n}_1 , to be equal to \hat{i}_3 , the unit vector in the 3 direction, so that

$$n_{1,1} = n_{1,2} = 0, \qquad n_{1,3} = 1$$
 (C15)

From Equation C13 we obtain

$$\Pi_{1\to 2,i} = S_{1;i,3} - S_{2;i,3} \tag{C16}$$

For $i \neq 3$,

$$S_{1;i,3} - S_{2;i,3} = \frac{1}{2} \varepsilon_o \left(\varepsilon_{1,s} E_{1,i} E_{1,3} - \varepsilon_{2,s} E_{2,i} E_{2,3} \right)$$
(C17)

Since direction 3 is normal to the boundary, directions 1 and 2 must be tangential. Therefore Equation C17 becomes

$$S_{1;i,3} - S_{2;i,3} = \frac{1}{2} \varepsilon_o \left(\varepsilon_{1,s} E_{1,t} E_{1,n} - \varepsilon_{2,s} E_{2,t} E_{2,n} \right), \quad i \neq 3$$
(C18)

where the subscripts n and t indicate the normal and tangential components respectively. For charged-free interface, the boundary condition at the interface are

$$E_{1,t} = E_{2,t} \tag{C19a}$$

$$\varepsilon_o \varepsilon_{1,s} E_{1,n} = \varepsilon_o \varepsilon_{2,s} E_{2,n}$$
 (C19b)

Substituting Equation C19a and C18 in Equation C16 shows that the net tangential stress at the interface is equal to zero

$$\Pi_{1\to 2,1} = \Pi_{1\to 2,2} = 0 \tag{C20}$$

and the normal net stress at the interface is

$$\Pi_{1\to2,3} = S_{1;3,3} - S_{2;3,3} \tag{C21}$$

Using Equation C14, Equation C21 can written as

$$\Pi_{1\to2,3} = -\frac{1}{2} \varepsilon_o \left(E_1^2 \rho_1 \partial_{\rho_1} \varepsilon_{1,s} - E_2^2 \rho_2 \partial_{\rho_2} \varepsilon_{2,s} \right) + \frac{1}{2} \varepsilon_o \varepsilon_{1,s} E_1^2 - \frac{1}{2} \varepsilon_o \varepsilon_{2,s} E_2^2 - \varepsilon_o \varepsilon_{1,s} E_{1,n}^2 + \varepsilon_o \varepsilon_{2,s} E_{2,n}^2$$
(C22)

where in each medium,

$$E^2 = E_t^2 + E_n^2$$
 (C23)

Using Equation C19b, the normal net stress at the interface can be expressed as

$$\Pi_{1\to2,3} = -\frac{1}{2} \varepsilon_o \left(E_1^2 \rho_1 \partial_{\rho_1} \varepsilon_{1,s} - E_2^2 \rho_2 \partial_{\rho_2} \varepsilon_{2,s} \right) + \frac{1}{2} \varepsilon_o \left(\varepsilon_{1,s} - \varepsilon_{2,s} \right) \left[\left(\varepsilon_{1,s} / \varepsilon_{2,s} \right) E_{1,n}^2 + E_{1,t}^2 \right]$$
(C24)

Using Clausius-Mossotti law for non dipolar fluids

$$\rho \frac{\partial \varepsilon_s}{\partial \rho} = \frac{(\varepsilon_s - 1)(\varepsilon_s + 2)}{3} \tag{C25}$$

Substituting by Equation C25 in Equation C24, the final form of the normal net stress at the interface can be expressed as

$$\Pi_{1\to2,3} = \frac{\varepsilon_o}{6} \left\{ \frac{1}{\varepsilon_{s,2}^2} \left[4\varepsilon_{s,1}^2 \varepsilon_{s,2} - 2\varepsilon_{s,1}^2 - 4\varepsilon_{s,1} \varepsilon_{s,2}^2 + 2\varepsilon_{s,2}^2 \right] E_{n,1}^2 \right\} + \frac{\varepsilon_o}{6} \left\{ \left[\varepsilon_{s,2} \left(\varepsilon_{s,2} - 2 \right) - \varepsilon_{s,1} \left(\varepsilon_{s,1} - 2 \right) \right] \left(E_{t,1}^2 \right) \right\}$$
(C26)

The final forms of the dielectrophoretic and electrostrictive terms separately can be expressed as

$$\Pi_{1 \to 2,3}(dielectrophoretic) = \frac{\varepsilon_o}{6} \left\{ \frac{1}{\varepsilon_{s,2}^2} \left[3\varepsilon_{s,1}^2 \varepsilon_{s,2} - 3\varepsilon_{s,1} \varepsilon_{s,2}^2 \right] E_{n,1}^2 \right\} + \frac{\varepsilon_o}{6} \left\{ \left[-3\varepsilon_{s,2} + 3\varepsilon_{s,1} \right] \left(E_{t,1}^2 \right) \right\}$$
(C27)
$$\Pi_{1 \to 2,3}(electrostrictive) = \frac{\varepsilon_o}{6} \left\{ \frac{1}{\varepsilon_{s,2}^2} \left[\varepsilon_{s,1}^2 \varepsilon_{s,2} - 2\varepsilon_{s,1}^2 - \varepsilon_{s,1} \varepsilon_{s,2}^2 + 2\varepsilon_{s,2}^2 \right] E_{n,1}^2 \right\} + \frac{\varepsilon_o}{6} \left\{ \left[\varepsilon_{s,2}^2 + \varepsilon_{s,2} - 2 - \varepsilon_{s,1}^2 - \varepsilon_{s,1} + 2 \right] \left(E_{t,1}^2 \right) \right\}$$
(C28)

The total, dielectrophoretic and electrostrictive forces acting at a flat horizontal interface for an applied voltage of 3 kV and an initial interface-electrode gap distance of 1.5 mm are shown in Figure C2. The dielectrophoretic and electrostrictive forces have opposing effects on the liquid vapour interface. The dielectrophoretic force is an extraction force acting against the gravitational and surface tension forces while the electrostrictive force is a restoring force.



Figure C2: Interfacial forces for a flat horizontal interface at an applied voltage of 3 kV and an initial interface-electrode gap distance of 1.5 mm. — total force, — dielectrophoretic force and — \cdot — electrostrictive force.

Appendix D: Liquid Extraction and Heat Transfer EHD Modeling

D1: Equivalent film thickness

The heat transfer under the application of high voltage electric fields is modeled using the film condensation model. During film condensation, the heat transfer across the liquid film is assumed to be due to conduction only, Nusselt [1916]. Therefore, the liquid film thickness can be evaluated by

$$\delta = \frac{k_l}{h} \tag{D1}$$

where h is the experimental heat transfer coefficient. The top and bottom equivalent film thickness without applied voltage and with 8 kV DC applied voltage is shown in Figure D1. The annular film thickness is 0.01 mm with EHD while it is 1.55 mm and 0.03 mm for the bottom stratum and the top and side condensate respectively in case without EHD force.

During stratified flow condensation, the liquid condensate in contact with the heat transfer surface is the main thermal resistance and therefore liquid extraction increases heat transfer and pressure drop. Estimation of the amount of liquid extracted due to EHD is important to understand the influence of the EHD on heat transfer



Figure D1: Effect of mass flux on liquid film thickness for $\Box 0$ kV-top, $\blacksquare 0$ kV-bottom, $\triangle 8$ kV-top, $\blacktriangle 8$ kV-bottom. Heat flux of q''=5.7 kW/m² and $x_{avg}=45\%$.



Figure D2: Schematic diagram of the two phase flow pattern with initial stratified flow before the onset of the applied voltage (a) without applied voltage, (b) with applied voltage

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and pressure drop.

The liquid extraction ratio Ex can be defined as the percentage ratio of the area of the liquid in vicinity of the heat transfer surface without EHD force to that with EHD force (Figure D2).

$$Ex = \frac{A_l - A_{EHD}}{A_l} \times 100 \tag{D2}$$

$$A_l = A \left(1 - \alpha \right) \tag{D3}$$

$$A_{EHD} = \frac{\pi}{4} \left[D^2 - (D - 2\delta_{EHD})^2 \right]$$
(D4)

where A_l , A_{EHD} and δ_{EHD} are the area occupied with the liquid in contact with the heat transfer surface area with and without EHD and the liquid film thickness with EHD respectively.

Appendix E: Flow Visualization

The effect of applied pulse wave voltage, listed in Table E1, on two-phase flow patterns at a mass flux of $G = 55 \ kg/m^2 s$ and quality x = 45% is illustrated in Figures E1 to E9. Each figure consists of a maximum of 12 images equally distributed over one cycle of the applied pulse wave voltage. The effect of pulse repetition rate of 50% duty cycle pulse wave voltage on two phase patterns is shown in Figures E1 to E7. For pulse repetition rate , $f \leq 10$ Hz (Figures E1 to E4), applying the high voltage causes the liquid to be extracted from the bottom stratum to the center electrode and then being pushed to the bulk of the annulus in the form of liquid twisted cones. Increasing the pulse repetition rate, f > 10 Hz, decreases the time available for liquid extraction and therefore diminishes the amount of liquid observed on the top of the center electrode.

The effect of duty cycle of a 1 Hz pulse wave voltage is shown in Figures E8, E2 and E9 for 25%, 50% and 75% duty cycles respectively. Increasing the duty cycle increases the time the two phase flow is subjected to the high voltage and therefore increases the intensity of the liquid vapour interactions. This agrees with the heat transfer and pressure drop results presented in chapter six which showed an increase of heat transfer and pressure drop with the duty cycle for the same frequency.

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	25% duty cycle	50% duty cycle	75% duty cycle
0.5 Hz		\checkmark	
1 Hz	1	\checkmark	\checkmark
6 Hz		\checkmark	
10 Hz		\checkmark	
75 Hz		\checkmark	
250 Hz		\checkmark	
400 Hz		1	

Table E1: Flow visualization tests



Figure E1: Flow regime redistributions for a 0.5 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 166.7 msec).

2



Figure E2: Flow regime redistributions for a 1 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 83.3 msec).



Figure E3: Flow regime redistributions for a 6 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 13.9 msec).

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Figure E4: Flow regime redistributions for a 10 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 8.3 msec).

1



Figure E5: Flow regime redistributions for a 75 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 1.1 msec).

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Figure E6: Flow regime redistributions for a 250 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 0.3 msec).

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Figure E7: Flow regime redistributions for a 400 Hz, 50% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 0.2 msec).



Figure E8: Flow regime redistributions for a 1 Hz, 25% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 83.3 msec).

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Figure E9: Flow regime redistributions for a 1 Hz, 75% duty cycle pulse wave between 0 and 8 kV at a mass flux of $G = 55 kg/m^2s$ and quality x = 45% (pictures equally distributed along one cycle, i.e time between frames is 83.3 msec).

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Appendix F: Uncertainty Analysis

A-Uncertainty in temperature measurements

The uncertainty of the temperature measurement associated with the temperature calibration was estimated as outlined by Drnovsek et al. [1998]. The uncertainty associated with the thermocouple ice bath cold junction and the data accusation system were included in the analysis. The total uncertainty in the temperature measurements is estimated by

$$\delta T = \sqrt{\sum_{i=1}^{7} (\delta_i)^2} \tag{F1}$$

where

 δ_1 : The uncertainty associated with the reference temperature (RTD).

 δ_2 : The uncertainty associated with the drift of the reference temperature (RTD).

 δ_3 : The uncertainty associated with the RTD measuring device.

 δ_4 : The uncertainty associated with the calibration instrument.

 δ_5 : The uncertainty associated with the ice bath.

 δ_6 : The uncertainty associated with the data acquisition system.

- δ_7 : The uncertainty associated with curve fitting the calibration data.
- δ_8 : The uncertainty associated with the reproducibility of the measurements.

Uncertainty associated with the reference temperature (RTD)

The reference thermometer used for the calibration is a platinum resistance thermometer (RTD) with reported uncertainty from the manufacture of ± 0.01 °C for temperature range above 40 °C.

Uncertainty associated with drift of the reference temperature (RTD)

There is no documents from the manufacture about the annual drift of the RTD sensor. The annual drift of the RTD is estimated from Drnovsek et al. [1998] as ± 0.0075 °C/year. Therefore the total uncertainty of the drift over 5 years is ± 0.0375 °C.

Uncertainty associated with RTD measuring device

The uncertainty of the DP251 precision RTD thermometer is reported by the manufacture as ± 0.01 °C.

Uncertainty associated with the calibration instrument

The uncertainty associated with the assumption of constant temperature medium in the dry well temperature calibrator is determined by measuring the temperature in different locations in the constant temperature block (copper block) using the same RTD sensor. The results show variations in the temperature less than the minimum resolution of the RTD reader (0.01 $^{\circ}$ C). Therefore the total uncertainty of the calibration instrument is the accumulation of the uncertainty of the RTD and the minimum resolution.

$$\delta_4 = \pm \sqrt{(0.001)^2 + (0.001)^2} = \pm 0.014 \tag{F2}$$

Uncertainty associated with the ice bath

The uncertainty of the ice bath was determined by measuring the temperature of the ice bath using the RTD sensor. The results show no deviation from the 0 $^{\circ}$ C within the minimum resolution of the RTD reader (0.01 $^{\circ}$ C). Therefore the total uncertainty of the the ice bath can be estimated as

$$\delta_5 = \pm \sqrt{(0.01)^2 + (0.01)^2} = \pm 0.014$$
 (F3)

Uncertainty associated with data acquisition system

The data acquisition system consists of two temperature boards and one high voltage board. The maximum total uncertainty of the data acquisition system is calculated neglecting the offset error as it is incorporated in the calibration correlations. The maximum uncertainty is ± 0.195 °C.

Uncertainty associated with curve fitting the calibration data

A comparison between linear, quadratic, cubic and power correlations was performed to fit the calibration data for each thermocouple. The maximum uncertainty associated with the curve fitting was found to be within $\pm 0.04^{\circ}$ C.

Uncertainty associated with reproducibility of the measurements

The reproducibility uncertainty is estimated as twice the standard deviation for 1000 readings. This corresponds to 95% confidence level. The calculated uncertainty is found to be less than $\pm 0.1^{\circ}$ C.

Therefore the total uncertainty of the temperature measurements is given by

$$\delta T = \pm \sqrt{(0.01)^2 + (0.0375)^2 + (0.01)^2 + (0.014)^2 + (0.014)^2 + (0.195)^2 + (0.04)^2 + (0.1)^2}$$

= ±0.23

(F4)

Other uncertainties such as those caused by the deterioration of the thermocouples with time, work/cold/heat hardening of the internal wires, thermal emf in connection wires, and variable offset drift of the data acquisition system cannot be evaluated and they assumed negligible in the current analysis.

B-Uncertainty in calculated parameters

Heat flux

The heat flux, q'', extracted by the coolant water from the test section is determined by

$$q'' = \frac{\dot{m}_w c_{pw} \left(\Delta T_{RTD}\right)}{\pi D_i L} \tag{F5}$$

Neglecting the uncertainty associated with the diameter of the inner tube and the length of the test section, the total uncertainty of the heat flux Q'' is a function of the uncertainty of the water mass flow rate, $\delta \dot{m}_w$, and the RTD temperature difference, $\delta \Delta T_{RTD}$

$$\delta q'' = \sqrt{\left(\frac{\partial q''}{\partial m_w}\delta m_w\right)^2 + \left(\frac{\partial q''}{\partial \Delta_{RTD}}\delta \Delta T_{RTD}\right)^2}$$
(F6)

The total relative uncertainty of the heat flux q'' can be calculated by dividing equation F6 by equation F5, the final expression of the relative uncertainty is

$$\frac{\delta q''}{q''} = \sqrt{\left(\frac{\delta m_w}{m_w}\right)^2 + \left(\frac{\delta \Delta T_{RTD}}{\Delta T_{RTD}}\right)^2} \tag{F7}$$

The maximum total relative uncertainties of the water mass flow rate and the RTD temperature difference are 5% and 7.74% of the reading respectively. Therefore, the total relative uncertainty of the heat flux q'' is

$$\frac{\delta q''}{q''} \times 100 = \sqrt{(5)^2 + (6.65)^2} \cong 8.32\%$$
 (F8)

Average vapour quality

Using Equations 4.16, 4.17 and 4.18 and 4.23, the uncertainty of the average vapour quality of the test section, δx_{avg} , is a function of the uncertainty of q_{PRE} , q'', m_w , T_{sat}

and T_{R-sub} .

$$\delta x_{avg} = f\left(\delta q'', \delta q_{PRE}, \dot{m_w}, T_{sat}, T_{R-sub}\right) \tag{F9}$$

Using equations 4.23, and 4.1, the uncertainty of the amount of heat added to the working fluid before the inlet to the test section, q_{PRE} , can be expressed as

$$\delta q_{PRE} = \sqrt{ \left(\frac{\partial q_{PRE}}{\partial I} \delta I \right)^2 + \left(\frac{\partial q_{PRE}}{\partial V} \delta V \right)^2 + \left(\frac{\partial q_{PRE}}{\partial \dot{m}_{w-PRE}} \delta \dot{m}_{w-PRE} \right)^2 + \left(\frac{\partial q_{PRE}}{\partial T_{w-PRE-i}} \delta T_{w-PRE-i} \right)^2 + \left(\frac{\partial q_{PRE}}{\partial T_{w-PRE-o}} \delta T_{w-PRE-o} \right)^2$$
(F10)

where

$$\frac{\partial q_{PRE}}{\partial I} = 2V \tag{F11}$$

$$\frac{\partial q_{PRE}}{\partial V} = I \tag{F12}$$

$$\frac{\partial q_{PRE}}{\partial \dot{m}_{w-PRE}} = c_{pw} \left(T_{w-PRE-i} - T_{w-PRE-o} \right) \tag{F13}$$

$$\frac{\partial q_{PRE}}{\partial T_{w-PRE-i}} = -\dot{m}_{w-PRE} c_{pw} T_{w-PRE-o}$$
(F14)

$$\frac{\partial q_{PRE}}{\partial T_{w-PRE-o}} = \dot{m}_{w-PRE} c_{pw} T_{w-PRE-i}$$
(F15)

In the current experimental matrix, only the electric heater is used to control the inlet quality to the test section, therefore δq_{PRE} is only function of $(\frac{\partial q_{PRE}}{\partial I}\delta I)$ and $(\frac{\partial q_{PRE}}{\partial V}\delta V)$. Substituting with the total uncertainty values of I and V in equation F10, the maximum total relative uncertainty of q_{PRE} is $\delta q_{PRE} = 0.112\%$.

Neglecting the uncertainty associated with the refrigerant properties, the final form of the total relative uncertainty of the inlet quality x_{in} can be expressed as (using equation 4.23 and 4.16)

$$\delta x_{in} = \sqrt{\left(\frac{\partial x_{in}}{\partial q_{PRE}}\delta q_{PRE}\right)^2 + \left(\frac{\partial x_{in}}{\partial \dot{m}_R}\delta \dot{m}_R\right)^2 + \left(\frac{\partial x_{in}}{\partial T_{sat}}\delta T_{sat}\right)^2 + \left(\frac{\partial x_{in}}{\partial T_{R-sub}}\delta T_{R-sub}\right)^2}$$
(F16)

where

$$\frac{\partial x_{in}}{\partial q_{PRE}} = \frac{1}{\dot{m}_{ref} h_{lv}} \tag{F17}$$

$$\frac{\partial x_{in}}{\partial \dot{m}_R} = -\frac{q_{PRE}}{\dot{m}_R^2 h_{lv}} \tag{F18}$$

$$\frac{\partial x_{in}}{\partial T_{sat}} = -\frac{c_{p_l}}{h_{lv}} \tag{F19}$$

$$\frac{\partial x_{in}}{\partial T_{R-sub}} = \frac{c_{p_l}}{h_{lv}} \tag{F20}$$

(F21)

Substituting with the total uncertainty values for q_{PRE} , \dot{m}_R , T_{sat} and T_{R-sub} , the maximum total relative uncertainty of x_{in} is $\delta x_{in} = 14.65\%$. Similarly, using equation 4.23 and 4.17, the maximum total relative uncertainty of x_{out} is $\delta x_{out} = 14.68\%$. Therefore, the final form of the total relative uncertainty of the average quality x_{avg} can be expressed as (using equation 4.23 and 4.18)

$$\delta x_{avg} = \sqrt{\left(\frac{\partial x_{avg}}{\partial x_{in}}\delta x_{in}\right)^2 + \left(\frac{\partial x_{avg}}{\partial x_{out}}\delta x_{out}\right)^2}$$
(F22)

where

$$\frac{\partial x_{avg}}{\partial x_{in}} = \frac{\partial x_{avg}}{\partial x_{out}} = \frac{1}{2}$$
(F23)

Substituting with the total uncertainty values for x_{in} and x_{out} , the maximum total relative uncertainty of x_{avg} is $\delta x_{avg} = 10.38\%$.

Heat transfer

The inner heat transfer coefficient can be expressed as

$$h_i = \frac{q''}{\left(T_{sat} - T_{S_{avg}}\right)} \tag{F24}$$

The total relative uncertainty of the average heat transfer coefficient can be calculated using

$$\frac{\delta h}{h} = \sqrt{\left(\frac{\delta q''}{q''}\right)^2 + \left(\frac{\delta \left(T_{sat} - T_{S,avg}\right)}{\left(T_{sat} - T_{S,avg}\right)}\right)^2}$$
(F25)

where

$$\delta \left(T_{sat} - T_{s,avg} \right) = \sqrt{\left(\delta T_{sat} \right)^2 + \left(\delta T_{s,avg} \right)^2}$$
(F26)

and
$$\delta T_{S,avg} = \frac{1}{\sqrt{12}} \delta T_s$$
 (F27)

Substituting with the total uncertainty values for q'', T_{sat} and T_s , the maximum total relative uncertainty of h is $\delta h = 8.37\%$.

A sensitivity analysis of the effect of the uncertainty of the temperature measurement on the heat transfer coefficient uncertainty was performed. The results showed that increasing the temperature uncertainty from $\pm 0.23^{\circ}$ C to $\pm 0.5^{\circ}$ C (typical uncertainty values in industrial applications), increases the heat transfer coefficient uncertainty from $\pm 12\%$ to $\pm 19\%$.

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