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ELECTROSPRAY COOLING AND DROPLET EVAPORATION

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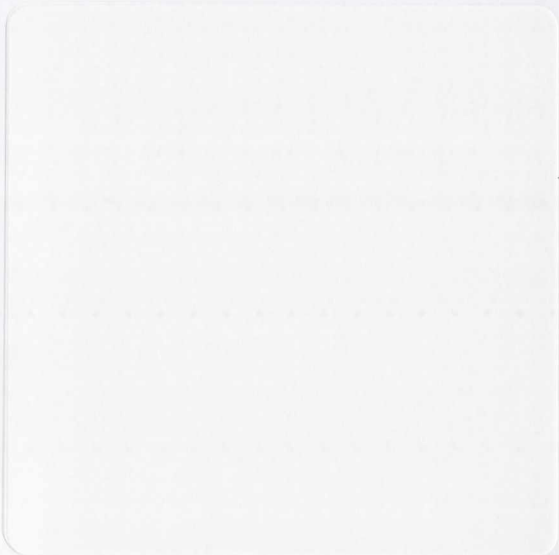
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Abstract

Spray cooling utilises liquid-gas phase transition for highly efficient thermal dissipation. Electro spraying is a method of fluid atomisation by electrostatic means and has applications in colloidal propulsion, mass spectrometry, material production, biomedical sciences, and microcombustion. Of particular interest is electro spray cooling given its generation of monodisperse micron sized charged droplets, self-spray dispersion due to Coulomb repulsions, negation of droplet rebound and increased droplet spreading upon impact due to Coulomb attraction, and minimal energy and space requirements. Despite this, widespread application of electro spray cooling technology has yet to occur. This is in part due to the lack of knowledge of how basic electro spraying parameters interact to impact cooling performance. Most prominently, the mechanisms of heat and mass transfer at the solid-liquid interface are still an underdeveloped topic. This applies in a broader context to spray cooling, as well as electro spray cooling.

This study experimentally investigates electro spray cooling and droplet evaporation on a heated substrate. High speed, high resolution infrared thermography is employed to measure the heat transfer characteristic of an impinging electro spray plume and evaporating droplet. Droplet evaporation is studied to examine the role that the triple line plays in heat transfer from an impacting spray on a heated substrate. Optical imaging and advanced analytical techniques are used to capture droplet dynamics over the course of its evaporation. An experimental set-up has been designed and built to facilitate these measurements, while purpose built code has been developed to analyse the acquired experimental data. Two-phase electro spray cooling is explored for multiple nozzle sizes ($D_i = 0.2 - 1.37$ mm), flow rates ($Q = 2 - 16$ $\mu\text{L min}^{-1}$), separation heights ($H = 2.5 - 17.5$ mm), applied electric fields ($E_{field} = 4 - 8$ kV cm^{-1}), and number of spraying sources ($N_n = 1 - 7$). While, droplet evaporation

has been examined for different heated substrate wettability conditions (partial wetting and partial non-wetting) and applied electric fields ($E_{field} = 0 - 11 \text{ kV cm}^{-1}$).

Analysis of single source and array electrospray cooling reveals two distinct cooling regimes; evaporative and pool electrospray cooling. The evaporative case is denoted by impact and subsequent spreading of individual droplets on the heated substrate resulting in formation of multiple contact lines. Whereas, pool cooling is characterised by coalescence of impinging droplets due to insufficient evaporative heat flux from the heated substrate. This results in a quasi-steady macro sized droplet on the heated substrate. Both cases are defined by evaporation from the contact line. Peak heat transfer was noted at the point just prior to the cooling transition from evaporative to pool cooling. In general, any parameter change that tends to increase the droplet mass flux also increases the convective heat flux in the evaporative cooling regime. A peak heat transfer enhancement of 18.71 times over natural convection was observed. Multiple source cooling was shown to lower wall temperature and temperature standard deviation by 31% and 29% respectively, while increasing heat transfer by 14% when compared to a single nozzle under similar operating conditions.

Examination of droplet evaporation revealed a strong correlation between dissipated thermal energy and contact line length density. Comparing droplet evaporation over similar wetted areas demonstrated a 2.7 times increase in cooling was achieved by replacing a singular large partial non-wetting droplet with several (4.66) smaller partial non-wetting droplets of the same total wetted area. These cases had similar total liquid-gas surface areas and contact angles. The multiple smaller droplets resulted in an increase in contact line length density by 2.15 times in comparison to the larger droplet case. The influence of surface wettability and applied electric field was also shown. Principally, any parameter change that optimised contact line length density increases thermal dissipation. In the context of electrospray cooling this corresponds to the maximum number of individual microscale droplet evaporating on the heated substrate.

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Publications

- M. J. Gibbons, A. J. Robinson. “Heat Transfer Characteristics of Single Cone-Jet Electrospays”. *International Journal of Heat and Mass Transfer*, Under Review (submitted 14/7/2016)
- M. J. Gibbons, P. Di Marco, A. J. Robinson, “Local Heat Transfer to an Evaporating Sessile Droplet in an Electric Field”, *Journal of Physics: Conference Series*, 745, p. 032066, 2016.
- M. J. Gibbons, S. Finnegan, M. Maguire, S. O’Dea, A. J. Robinson. “Heat Transfer Characterisation of a Single Source Ultra-Low Flow Rate Electro spray”. *ExHFT 8: World Congress of Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, Lisbon, Portugal, June 16–20, 2013*.
- M. J. Gibbons, A. J. Robinson. “Plume Characterisation of a Single Source Electro-spray for Micro Electronic Cooling”. *ISEHD 2014: The International Symposium on Electrohydrodynamics, Okinawa, Japan, June 23–25, 2014*
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Nomenclature

Symbol	Description	Units
A	Surface area	$[m^2]$
a	Nozzle spacing - extractor separation height ratio	$[-]$
B	Electric field constant	$[-]$
C	Curvature	$[m^{-1}]$
C_d	Drag coefficient	$[-]$
C_p	Specific heat capacity	$[Jkg^{-1}K^{-1}]$
c	Coverage factor	$[-]$
D_0	Characteristic droplet diameter	$[m]$
D_{acc}	Acceleration zone base diameter	$[m]$
D_b	Droplet base diameter	$[m]$
D_c	Cone-jet base diameter	$[m]$
D_d	Primary droplet diameter	$[m]$
D_{ex}	Extractor orifice diameter	$[m]$
D_i	Inner nozzle diameter	$[m]$
D_j	Jet diameter	$[m]$
D_o	Outer nozzle diameter	$[m]$
dx	Element length	$[m]$
dy	Element width	$[m]$
E_d	Driving electric field strength	$[kVcm^{-1}]$
E_{field}	Electric field strength	$[kVcm^{-1}]$

NOMENCLATURE

E_j	Jet forming region electric field strength	[kV cm ⁻¹]
E_{onset}	Onset electric field for cone-jet spraying	[kV cm ⁻¹]
e_i	Velocity unit vector	[ms ⁻¹]
F_{cp}	Contact pressure force	[N]
F_c	Surface tension force	[N]
F_e	Electrical force	[N]
F_w	Weight force	[N]
$f(\epsilon)$	Dimensionless scaling function for droplet size	[-]
f_e	Droplet production frequency	[Hz]
f_n	Friction factor	[-]
$G(\epsilon)$	Dimensionless scaling function for spray current	[-]
g	Gravitational acceleration	[ms ⁻²]
H	Source – target separation height	[m]
H_1	Source – extractor separation height	[m]
H_2	Extractor – target separation height	[m]
H_D	Droplet height	[m]
h	Heat transfer coefficient	[W m ⁻² K ⁻¹]
h_{fg}	Working fluid latent heat	[kJ kg ⁻¹]
$h_{l,major}$	Major pressure head loss	[Pa]
I_0	Characteristic spray current	[A]
I_{spray}	Spray current	[A]
I_s	Heated substrate current	[A]
j_m	Droplet mass flux	[kg s ⁻¹ m ⁻²]
k	Thermal conductivity	[W m ⁻¹ K ⁻¹]
L_c	Cone length	[m]
L_f	Heated substrate characteristic length	[m]
L_j	Jet length	[m]
L_n	Nozzle length	[m]

NOMENCLATURE

L_s	Length parallel to the current flow of thermal exchange surface	[m]
$\dot{m}$	Mass flow rate	[kg s ⁻¹]
N_n	Number of source nozzles	[-]
n	Number of samples	[-]
P	Pressure	[Pa]
p_s	Heated substrate perimeter	[m]
Q	Flow rate	[m ³ s ⁻¹]
Q_0	Characteristic flow rate	[m ³ s ⁻¹]
q	Heat transfer rate	[W]
q'	Heat transfer per unit length	[W m ⁻¹]
q''	Heat flux	[W m ⁻²]
$q_{e,max}$	Rayleigh limit charge	[C]
q_e	Charge	[C]
$\bar{q}_e$	Surface charge density	[C m ⁻²]
R_1	First principal radius	[m]
R_2	Second principal radius	[m]
R_b	Droplet base radius	[m]
R_{LG}	Liquid-gas thermal resistance	[K W ⁻¹]
R_{net}	Net thermal resistance	[K W ⁻¹]
R_Q	Flow resistance	[m ⁻³]
R_{SL}	Solid-liquid thermal resistance	[K W ⁻¹]
R_T	Droplet apex radius	[m]
r_{cor}	Linear correlation coefficient	[-]
$r_{i,j}$	Displacement vector of the i th and j th droplets	[m]
S_b	Droplet solid - liquid contact area	[m ²]
S_x	X-axis displacement	[mm]
S_y	Y-axis displacement	[mm]
S_z	Z-axis displacement	[mm]

NOMENCLATURE

s	Nozzle array spacing	[mm]
s	Standard deviation	[-]
T	Temperature	[K]
T_{∞}	Bulk fluid temperature	[K]
T_b	Working fluid boiling temperature	[K]
T_{spray}	Spray and gaseous co-flow temperature	[K]
T_s	Thermal exchange surface temperature	[K]
T_{wf}	Working fluid reservoir temperature	[K]
t	Time	[s]
t_b	Jet break-up time	[s]
t_{im}	Impact time	[s]
t_v	Viscous time	[s]
u	Velocity	[ms ⁻¹]
u_0	Droplet impact velocity	[ms ⁻¹]
u_d	Droplet velocity	[ms ⁻¹]
u_{js}	Jet surface velocity	[ms ⁻¹]
u_z	Droplet axial velocity	[ms ⁻¹]
V_d	Droplet volume	[m ³]
V_n	Applied nozzle voltage	[V]
V_{onset}	Applied voltage at onset of cone-jet spraying	[V]
V_s	Surface voltage drop	[V]
v	Measurement degree of freedom	[-]
W	Energy	[J]
w	Break-up wavelength constant	[-]
y	Measurand	[-]

Dimensionless Numbers

Bi	Biot number, $Bi = hL/k$	[-]
ER	Enhancement ratio, $ER = h/h_{nc}$	[-]

NOMENCLATURE

Gr	Grashof, $Gr = g\beta(T_s - T_\infty)L^3/\nu$	[-]
Nu	Nusselt number, $Nu = hL/k$	[-]
Oh	Ohnesorge number, $Oh = \mu/(\rho\gamma D_d)^{1/2}$	[-]
Ra	Rayleigh number, $Ra = GrPr$	[-]
We	Weber number, $We = \rho U^2 d_d/\gamma$	[-]

Greek Symbols

α_t	Thermal diffusivity	$[m^2 s^{-1}]$
β	Volumetric thermal expansion coefficient	$[K^{-1}]$
δ	Thickness	$[\mu m]$
δ_{ex}	Extractor electrode thickness	$[\mu m]$
δ_f	Foil thickness	$[\mu m]$
δ_p	Paint layer thickness	$[\mu m]$
ϵ_0	Permittivity of free space	$[F m^{-1}]$
ϵ_r	Relative permittivity	[-]
η	Spray cooling efficiency	[%]
γ	Surface tension	$[N m^{-1}]$
κ	Electrical conductivity	$[S m^{-1}]$
λ	Wavelength	[m]
μ	Dynamic viscosity	$[N s m^{-2}]$
ν	Kinematic viscosity	$[m^2 s^{-1}]$
ϕ_c	Cone semi-angle	$[^\circ]$
ϕ_s	Spray plume angle	$[^\circ]$
ρ	Density	$[kg m^{-3}]$
ρ_{N_n}	Source nozzle density	$[N_n cm^{-2}]$
σ	Stefan-Boltzmann constant	$[W m^{-2} K^{-4}]$
τ	Charge relaxation time	[s]
τ_0	Image force	[N]
τ_{cond}	Conduction time constant	[s]

NOMENCLATURE

τ_e	Maxwell stress	$[\text{Nm}^{-2}]$
τ_r	Interfacial stress	$[\text{Nm}^{-2}]$
θ	Droplet contact angle	$[\circ]$
θ_A	Advancing contact angle	$[\circ]$
θ_R	Receding contact angle	$[\circ]$
ϵ_{CaF_s}	IR window Emissivity	$[-]$
ϵ_p	Paint Emissivity	$[-]$

Subscripts

0	Initial or characteristic
∞	Bulk or infinity
c	Liquid cone meniscus
cap	Capacitive or storage
$cond$	Conduction
$conv$	Convection
d	Droplet
de	Droplet evaporation
ec	Electrospray cooling
$field$	Electric field
gen	Generated
j	Liquid jet
lc	Lateral conduction
max	Maximum
n	Source nozzle
rad	Radiation
$total$	Total
x, y, z	Coordinate direction
f	Foil
g	Gas

xxx

<i>l</i>	Liquid
<i>s</i>	Thermal exchange surface

Acronyms

3D	Three-dimensional
BC	Boundary condition
CCA	Constant contact angle
CCR	Constant contact radius
CHF	Critical heat flux
CLL	Contact line length density
CTP	Cooling transition point
CVD	Crystal vapour deposition
CWF	Constant wall heat flux
CWT	Constant wall temperature
DC	Direct current
DMM	Digital multimeter
DRIE	Deep reactive ion etching
EC	Electrospray cooling
ES	Electrospray
ESDM	Experimental standard deviation of the mean
EU	Expanded uncertainty
HSDR	High speed data recorder
IR	Infrared
IRC	Irish research council
LMIS	Liquid metal ion source
MWIR	Mid-wavelength infrared
NC	Natural Convection
NI	National instrument
NUC	Non-uniformity correction

NOMENCLATURE

PEEK	Polyether ether ketone
PU	Percentage uncertainty
PVD	Physical vapour deposition
RSD	Relative standard deviation
RTD	Resistance temperature detector
SU	Standard uncertainty
TIR	Total internal reflectance
VI	Virtual instrument

Chapter 1

Introduction

Electrospraying is a method of fluid atomisation by electrostatic forces. It can be divided into a number of different operating regimes, some of which were initially identified by Zeleny [1, 2] and later more comprehensively classified by Cloupeau and Prunet-Foch [3]. A widely applied functioning mode is the “cone-jet” characterised by Cloupeau and Prunet-Foch [4]. Cone-jet electrospray electrode design can be divided into two configurations: two electrodes (source - target), and three electrodes (source - extractor - target), these are shown in Figure 1.1. Both designs achieve cone-jet electrospraying in a similar manner but the three electrode configuration increases complexity and is normally applied in multiple source applications [5–9]. In the two electrode configuration (see Figure 1.2a), a suitable

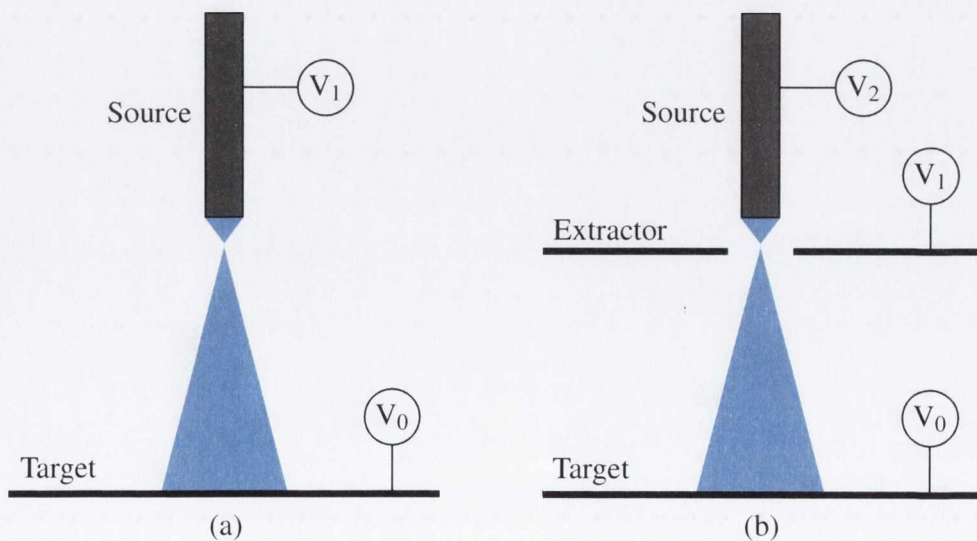


Figure 1.1: Electrospray electrode configuration. (a) Two electrode design (b) Three electrode design.

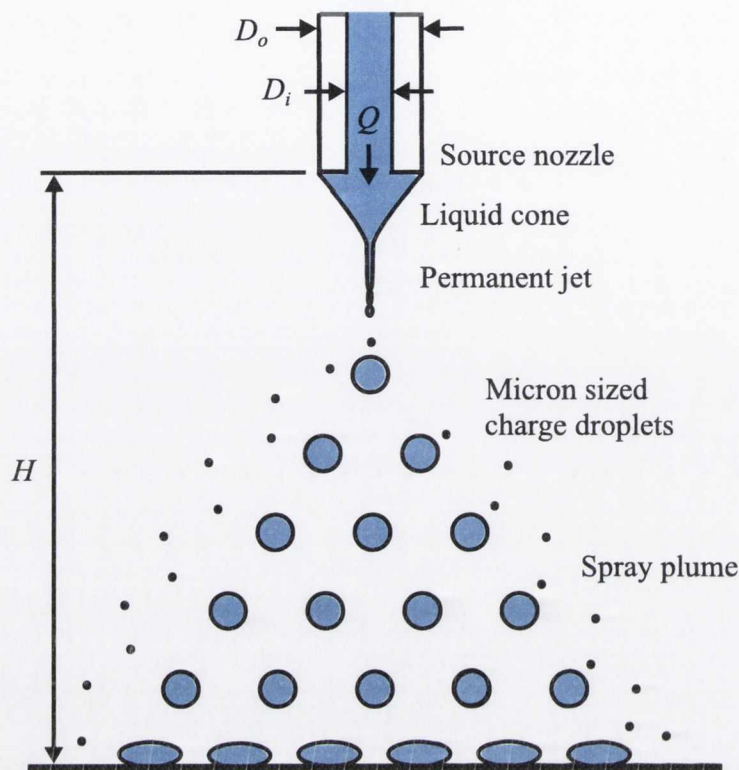


Figure 1.2: Single source electro spraying schematic.

working fluid is supplied to the tip of a source nozzle which is maintained at an electrical potential above that of an adjacent target electrode. This potential difference creates an electric field, and when a sufficient electric field is established, the meniscus at the tip of the source nozzle will deform into the shape of a liquid cone (see Figure 1.3a), often

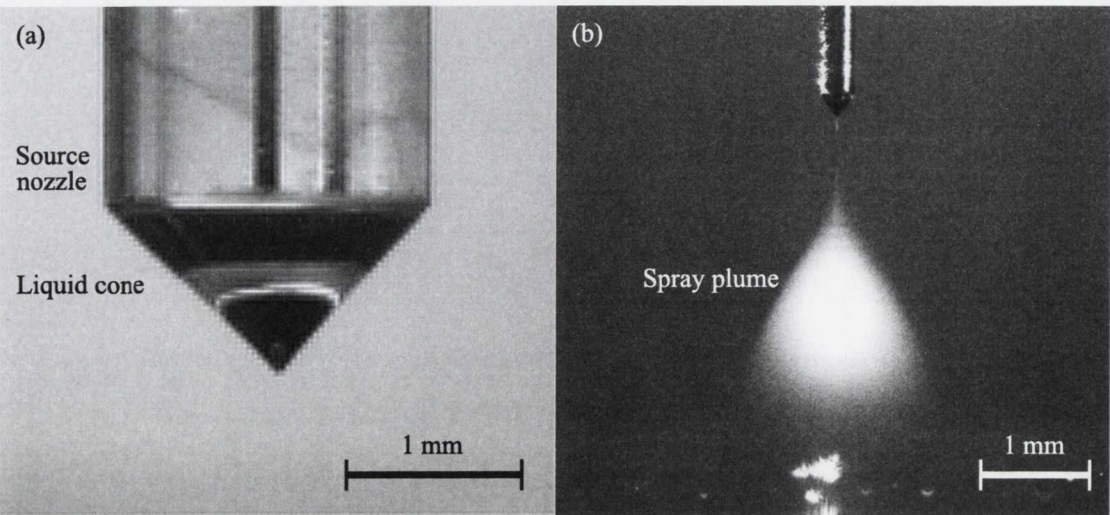


Figure 1.3: Electro spraying. (a) cone-jet and (b) spray plume.

Table 1.1: Single source electrospray order of magnitude operating parameters [3, 12, 13].

Source nozzle inner diameter, D_i	0.108 – 1.9	mm
Source nozzle outer diameter, D_o	0.306 – 2.396	mm
Source voltage, V_n	1 – 15	kV
Source - target separation height, H	2.5 – 30	mm
Flow rate, Q	0.6 – 600	$\mu\text{L min}^{-1} N_n^{-1}$
Jet thickness, D_j	1 – 20	μm
Jet length, L_j	30 – 600	μm
Primary droplet diameter, D_d	2 – 40	μm
Spray current, I_{spray}	4 – 40	nA

referred to as a Taylor cone [10]. This conical shape is as a result of the balance of surface tension, viscous, hydrostatic, gravitational, and electrostatic forces. At the apex of this cone a thin permanent jet is formed. The jet undergoes Rayleigh instability and breaks up into quasi-monodisperse micron-sized charged droplets. Often two families of droplets are produced; primary and satellite. Primary droplets make up $\approx 97\%$ of the flow and $\approx 86\%$ of the current [11]. Coulombic repulsion of the charged droplets enables spray plume dispersion (see Figure 1.3b) and droplet segregation, with primary droplets located in the core of the plume and satellite droplets orientated on the periphery of the spray due to their greater initial charge and reduced inertia in comparison with primary droplets [11]. A list of some of the basic operating parameters of a single source electrospray are listed in Table 1.1.

1.1 Applications

The attractive characteristics of electrospray technology has led to numerous topics of research into the electrospray process and its applications. Detailed reviews have been conducted by Bailey [14] and more recently by Jaworek [15, 16]. The relationship between droplet size and applications is shown in Figure 1.4. One of the great modern catalysts of electrospray technology was the seminal work of John B. Fenn and his colleagues at Yale University in the 1980s [17] leading to his 2002 Nobel Prize in Chemistry. Figure 1.4 illustrates several of the different application areas that electrospraying technology has been applied and their associated droplet size range.

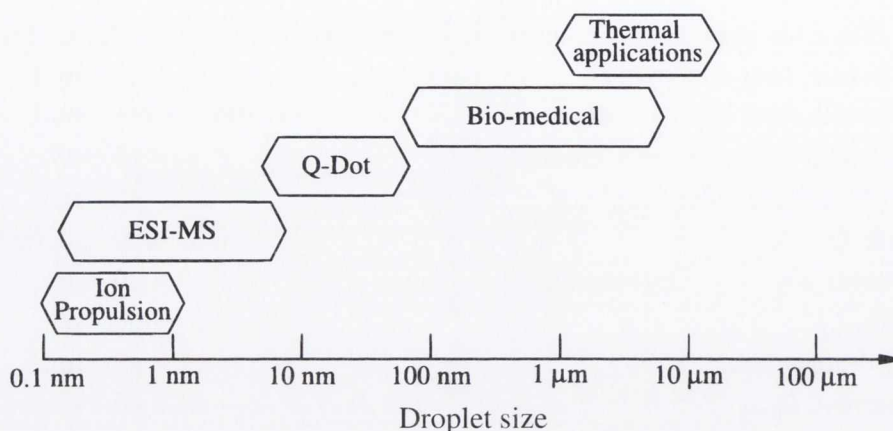


Figure 1.4: Electrospray applications and corresponding droplet size. Reprinted from Gomez and Deng [13].

1.1.1 Colloidal and Ion Propulsion

Extensive research has been carried out on the application of electrospray technology for ion source/microthruster technology [18–21]. Lenguito et al. [21] investigated the applicability of electrospray technology to small scale satellites such as CubeSat (< 10 Kg). They endeavoured to achieve accurate altitude control with $> 50\%$ propulsion efficiency. Their results suggested that microthruster technology can be employed for fine attitude control of microsatellites. They stipulated that further increased scaling by an order of magnitude would allow orbit change and position maintenance for small scale satellites.

1.1.2 Electrospray Ionisation Mass Spectrometry

Fenn developed a technique of soft ionisation of large biomolecules, achieving intact ions in a vacuum from large and complex species in solution. His technique enabled analysis of large and fragile polar molecules that was not at the time possible with conventional soft ionisation methods. Analysis of these molecules is important as they play an inherently important role in biological systems. Fenn's breakthrough owed much to the research of Malcolm Dole [22] in the 1960s. Dole recognised that electrospray technology could be used in the production of gas-phase ions of macromolecules but was hindered by his employment of an ion-drift spectrometer rather than a mass spectrometer in analysing the produced ions. Fenn built on Dole's initial experiments and theories but utilised a mass spectrometer in analysing the generated ions. He demonstrated that electrospray ionisation with mass

spectrometry could by very effectively technique in analysing peptides, proteins, small ions and molecules [17]. Today electrospray ion mass spectrometry is a widely used technique in biological, biochemical, pharmaceutical and medical research. Fenn's work stimulated widespread research into a host of electrospraying applications and fields [3, 4, 17, 23–30].

1.1.3 Material Production and Coating

Electrospray technology has long been applied in the spraying of different materials [31]. As far back as 1966, Hines [32] investigated the process of electrostatic spray painting. He attempted to characterise a single source, defining droplet size and charge for a refinement of the spray painting process. More recently electrospray painting has been refined to thin film deposition and coating. Van Zomeren et al. [33] developed a method of producing thin films of lithium manganese oxide. Their method enabled higher deposition rates and efficiency in comparison to the conventionally used crystal vapour deposition (CVD) and physical vapour deposition (PVD) processes. Electrospray lends itself well to coating applications due to its excellent control of stoichiometry and microstructure as well as large area coverage relative to the source nozzle size. Electrospraying is a powerful tool in coating due to its high material deposition efficiency, simple experimental configuration and large range of starting materials, resulting in numerous compositions and morphologies [31].

Electrospray technology can also been applied in the production of nanofibres. Doshi et al. [34] described an electrospinning process, producing extremely thin ($0.05 - 5 \mu\text{m}$ diameter) polymer fibres. This phenomenon occurs if the working fluid is able to solidify in a time shorter than the jet break-up time. They suggested these nanofibres could be applied as reinforcement in composites materials, as a support for very thin polymeric separation, as a wound dressing material as well as in host of other applications. Electrospraying has been investigated in the application of Liquid Metal Ion Source (LMIS) [35, 36]. LMIS is the production of ions using a molten metals source for the process of ion implantation. The generated source ions are accelerated and impacted onto the target surface, altering its material properties. Salata [31] demonstrated the applicability of electrospray technology in the synthesis of GaAs and CdS quantum dots. The produced quantum dots can be used as light generating media for a semiconductor laser that has a zero-threshold current.

1.1.4 Biomedical

Electrospraying has received growing interest for biomedical application in the recent past, primarily in the field of drug and particulate encapsulation [37–39]. It has also been applied in the deposition of ultra-thin films of inorganic, organic and biological materials [31]. A promising application has been found in the treatment of acute and chronic respiratory diseases [40], where the generated spray can target specific areas of the respiratory tract minimizing the required dosage. This may be of particular advantage when cost or adverse side effects are of concern.

1.1.5 Microcombustion

Electrospray technology has been investigated for increasing the efficiency of automobile engines through the delivery of a quasi-monodisperse micron sized droplets for the combustion process [41–43]. Agathou and Kyritsis [43] investigated electrospraying of butanol, ethanol and heptane for automobile engine applications. Their research focused on high speed visualisation of the spray structure under a host of applied voltages and working fluid flow rates. They were unable to produce a quasi-monodisperse spray for any spray conditions or fuels. They stipulated that this was as a result of an oscillating meniscus and Coulombic fission of the primary droplets due to their high Weber number. Deng et al. [44] developed a microcombustion system which had a overall combustor volume of 0.22 cm^3 using JP-8 as a liquid fuel. The microcombustor developed a volumetric heat release rate of 270 MW m^{-3} , which is comparable to that of a conventional gas turbine.

1.1.6 Electrospray cooling

Feng and Bryan [45] were the first to investigate two phase electrospray cooling (EC). Their research studied the average heat transfer enhancement of different nozzle arrays in an enclosed chamber under various spraying regimes with ethanol. They operating primarily outside the cone-jet mode. Their work showed that an optimum average heat transfer enhancement of 1.7 times over the no applied electric field case. This condition corresponded to a single nozzle, inner diameter $D_i = 0.25 \text{ mm}$, at their smallest nozzle - target separation height $H = 5 \text{ mm}$, for a flow rate of $Q = 3000 \text{ } \mu\text{L min}^{-1}$. This fluid was deposited onto

50 mm² surface with a heat flux of 10 Wcm⁻² and a fixed liquid layer appears to have been maintained on the heated substrate. The spraying regime was unspecified but an applied voltage of 7 kV was noted. Due to the enclosed nature of their test section, a higher wall heat flux resulted in greater vapour momentum away from the surface minimizing the effect of some spraying regimes. The multiple source array enabled increased heat transfer enhancement at higher flow rates and lower heat fluxes.

Wang and Mamishev [46] investigated the average heat transfer of enclosed electro-spraying chambers for three different geometry types in the multi-jet regime. They achieved a maximum average heat transfer enhancement of 1.87 fold over natural convection (NC) for their 8 source, 5 mm pitch array at the lowest wall heat flux of 4,384 W m⁻². This case corresponded to a $H = 7.5$ mm, $D_i = 0.210$ mm and $Q = 133.33$ $\mu\text{L min}^{-1}$. Like Feng and Bryan, they noticed an increasing enhancement for decreasing wall heat fluxes. They stipulated that increasing the number of sources and their spacing does not improve the steady state heat transfer. However it does increase the transient average heat transfer performance.

Wang and Mamishev [47] later developed a set of correlations for the average heat transfer enhancement ratio and Nusselt number for an electro-spray operated in the multi-jet regime. They based their model on the traditional impinging liquid jet empirical correlations, modifying them to account for the electric field of the electro-spray. They suggested that the enhancement ratio and Nusselt number was dependent on the Weber number, electric Weber number, Reynold number, the number of electro-spray sources and a characteristic length normalised by the source nozzles inner diameter. Their model for the enhancement ratio showed 83% agreement with their experimental data, within a $\pm 10\%$ deviation. From their results it can be shown that both their models perform well at lower flow rates with erroneous results appearing at higher flow rates. Wang and Mamishev stipulated that this was due to excessive coolant deposition on the target surface.

Deng and Gomez [6] constructed and tested 19 and 37 sourced multiplexed electro-spray arrays with a source density of 253 sources cm⁻². They operated in the cone-jet regime utilising an extractor design. They found that the micron quasi-monodisperse droplets resulted in the onset of two phase cooling at ~ 20 K below the boiling point of the working fluid, while typically this occurs at 3 – 5 K above the boiling point. Deng and Gomez showed that by decreasing the droplet size, while maintaining all other parameters, resulted in an

increase in the critical heat flux by $\sim 20 - 25\%$. They determined that this was as a result of the shorter evaporation time of smaller droplets. An optimal average heat flux removal of 96 W cm^{-2} was noted for the highest flow rate ($1666.67 \mu\text{L min}^{-1}$). By increasing the electrical conductivity of the working fluid they achieved an increase in average heat flux removal of 25% compared to the undoped ethanol case. From these results Deng and Gomez [6] suggested that producing the smallest droplets at the highest rate would optimise both spray efficiency and average heat flux removed. This can be achieved by optimising the packing of source nozzles to achieve small droplets for larger total flow rates.

1.2 Research Motivation

In 1965 George Moore noted that over the previous 6 years “the complexity for minimum component costs has increased at a rate of roughly a factor of two per year”. He predicted that “there is no reason to believe it will not remain constant for at least ten years”. Following Moore’s prediction the number of transistor on an integrated circuit increased linearly for the proceeding 40 years [48, 49]. Since the advent of smartphone technology, with the release of the first iPhone by Apple Inc. in 2007, there has been a growing expectation on small form factor electronics to be more compact while also increasing performance. This trend has resulted in increasing generated heat flux density due to the increasing transistor density, chip resistance, capacitance, and subsequent charge leakage [50].

This progression has driven conventional cooling technologies to a thermal management threshold, where they are unable to remove the generated thermal load. Intel’s latest high performance processor, the i7-6950X, has a thermal design power (TDP) of 140 W operating at “Base Frequency” [51]. Future high speed processor are predicted to achieve heat fluxes of 500 W cm^{-2} ranging up to 1000 W cm^{-2} at hot spots [52–54]. Forced air convection can achieve a heat flux removal of 150 W cm^{-2} [55], while 120 W cm^{-2} can be obtained through pool boiling of water [56]. In order to maintain chip performance, life cycle and avoid component failure it is imperative to keep on-chip surface temperatures below 85°C [57–59]. Clearly a step change in thermal management technology is required to overcome the thermal load generated from the next generation of microelectronics. An emerging solution to this problem is cone-jet electrospray cooling.

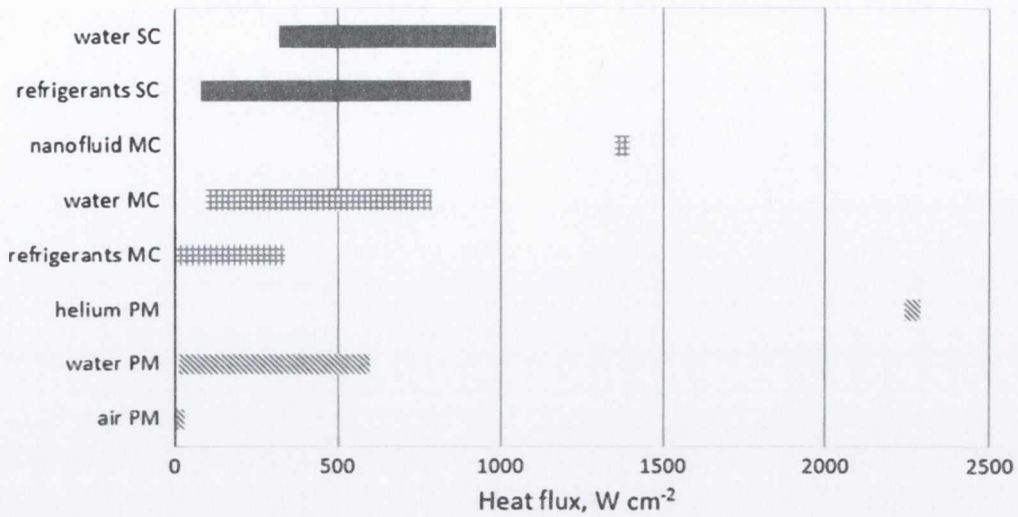


Figure 1.5: State of the art of high heat flux removal technologies: SC - spray cooling; MC - microchannel; PM - porous media. Adapted from Smakulski and Pietrowicz [61].

1.3 Next Generation Cooling

Direct liquid impingement is an attractive method of high heat flux removal for several reasons: first, removal of the relatively high thermal resistance between heat spreaders and the microelectronic chip; second, enhanced heat transfer due to velocity gradients at the stagnation zone; and third, potential for phase change cooling utilising the latent heat of the cooling fluid. This enables high heat flux capacity at reduced mass fluxes [60, 61]. The state of the art of high heat flux cooling technologies has recently been reviewed by Smakulski and Pietrowicz [61]. Among the most promising are: heat removal technique with porous media, microchannel heat sinks, and spray cooling [61]. The primary findings of their review are shown in Figure 1.5.

Spray cooling is the parent field of electrospray cooling incorporating atomisation of a liquid by means other than by electrostatic forces. Spray cooling is an attractive proposition as it is one of the most efficient heat flux removal mechanisms per unit flow [62, 63]. Spray cooling electronic devices can reduce power consumption by 35% in comparison to the more conventional air cooling systems [64]. Yang et al. [65] achieved heat flux removal of 820 W cm^{-2} for an air-atomised nozzle with water operating at a flow rate of 2 L min^{-1} . Tilton [66] achieved a heat flux removal of 1000 W cm^{-2} with water for a calculated film thickness of $1.4 \text{ }\mu\text{m}$ at a superheat of 20°C . While Chen et al. [56] reported a heat flux

removal of 945.7 W cm^{-2} using Bete#3 nozzle with water as the working fluid.

1.3.1 Electrospray Cooling

A promising candidate of spray cooling is cone-jet electrospraying which is characterised by its:

1. Near uniform mass flux in the spray core [11, 67]. Monodispersity of the droplets coupled with near uniform mass flux in the spray core minimises hot spots and helps maintain a uniform temperature across the cooled surface;
2. Quasi-monodisperse micron sized droplets [11, 30, 68]. This enables the formation of a near uniform liquid film. Spray cooling has been shown to produce films an order of magnitude smaller than liquid jets [69]. The thinner the film, the lower the thermal resistance, resulting in higher heat flux removal;
3. Droplet size control. By varying the liquid flow rate, applied voltage and liquid electrical conductivity one can produce quasi-monodisperse droplets in the nanometer up to hundred of micron size range [13];
4. Self spray dispersion by Coulomb repulsion, which also prevents droplet coagulation and enhances mixing with surrounding gas [11];
5. Efficient cooling. Impacting charged droplets have been shown to negate droplet rebound and increase droplet spreading as a result of an image force ¹ between the droplet and target surface [70, 71]. This results in a more effective heat transfer process than other traditional sprays.
6. Minimal energy requirements. Droplet production is facilitated primarily by electrostatic forces. A typical electrical energy requirement of $\sim 35 \mu\text{W}$ for a single source using ethanol at a working distance of 7.5 mm from the target surface and flow rate of $\sim 6 \mu\text{L min}^{-1}$ [5]. The spray can be generated from a source nozzle of relatively large bore in comparison to the generated droplets. This results in a ~ 100 fold decrease in

¹Image force - The electric force between a charged droplet and induced image charges on the conductive substrates [60, 70].

the required pressure in comparison with conventional fluid atomisers [6, 56]. This reduces space requirements as large sources of pressure head are not required for operation.

7. Low liquid flow. Electrospray cooling facilitates high heat flux removal for minimal working fluid flow rate ($\sim \mu\text{L}$) resulting in low total liquid volume requirements. This nominal space requirement is importable for application requiring mobility such as space and microelectronic cooling.

Development and optimization of two phase electrospray cooling is an underdeveloped topic, in part due to insufficient knowledge of how the various electrical, geometrical, thermal and hydraulic parameters influence both average and local heat transfer. On a broader scale, the mechanisms governing heat and mass transfer from a heated substrate to the deposited working fluid is still an underdeveloped topic. This applies to spray cooling, as well as that of electrospray cooling. Strong correlation of heat transfer intensity and triple contact line length for spray cooling has been observed [72, 73].

1.4 Dissertation Organisation

Chapter 2 presents a review of the literature on electrospraying cooling and droplet evaporation. Chapter 3 outlines the fundamentals of the experimental apparatus, the experimental procedure and discusses the methodology of processing the experimental data and its associated uncertainty. Chapter 4 presents a parametric investigation into heat flux removal of single source electrospray by varying the source – target separation height, working fluid flow rate and source nozzle size. Chapter 5 describes the development of an electrospray array with a $115\text{ N}_n\text{ cm}^{-2}$ packing density and demonstrates its cooling capability for varied flow rate and applied driving electric field. Chapter 6 and 7 presents an analysis of the impact both surface wettability and applied electric field have on the local heat transfer to an evaporating droplet. Finally Chapter 8 and 9 summarises the main findings and recommends the future direction of this research.

Chapter 2

Literature Review

This chapter will provide an comprehensive review of the literature surrounding electrospray cooling and heat and mass transfer to an evaporating fluid. First, the electrospray process will be deconstructed. Next, a detailed review of heat and mass transfer to an evaporating droplet and impinging spray will be reviewed. Then, multi-source electrospraying will be considered. Finally, the underdeveloped field of electrospray cooling will be examined in detail.

2.1 Electrospraying

John Zeleny [1, 2] was the first to study the effect that an applied electric field has on the meniscus of a working fluid. He observed the production of charged droplets emitted from the apex of a liquid meniscus under various functioning modes, each with very different characteristics. Electrospraying can be divided into a number of different spraying regimes. Each regime, or functioning mode, has its own advantages and subsequent drawbacks. Using various working fluids, Cloupeau and Prunet Foch [74] classified 11 spraying regimes, 5 of which were previously unidentified. Grace and Marijnissen [35] created a flow chart highlighting 12 different modes of spraying. They showed that for a constant source – target separation height H , and electrode configuration, the established spraying regime was a function of the applied source voltage V_n , working fluid flow rate Q , and working fluid surface tension γ . Figure 2.1 is adapted from their work, updated here to provide a more accurate representation of the various spraying regimes. The spraying regimes themselves are often split into two groups, distinguished by their flow pattern; pulsating and continuous.

2.1. ELECTROSPRAYING

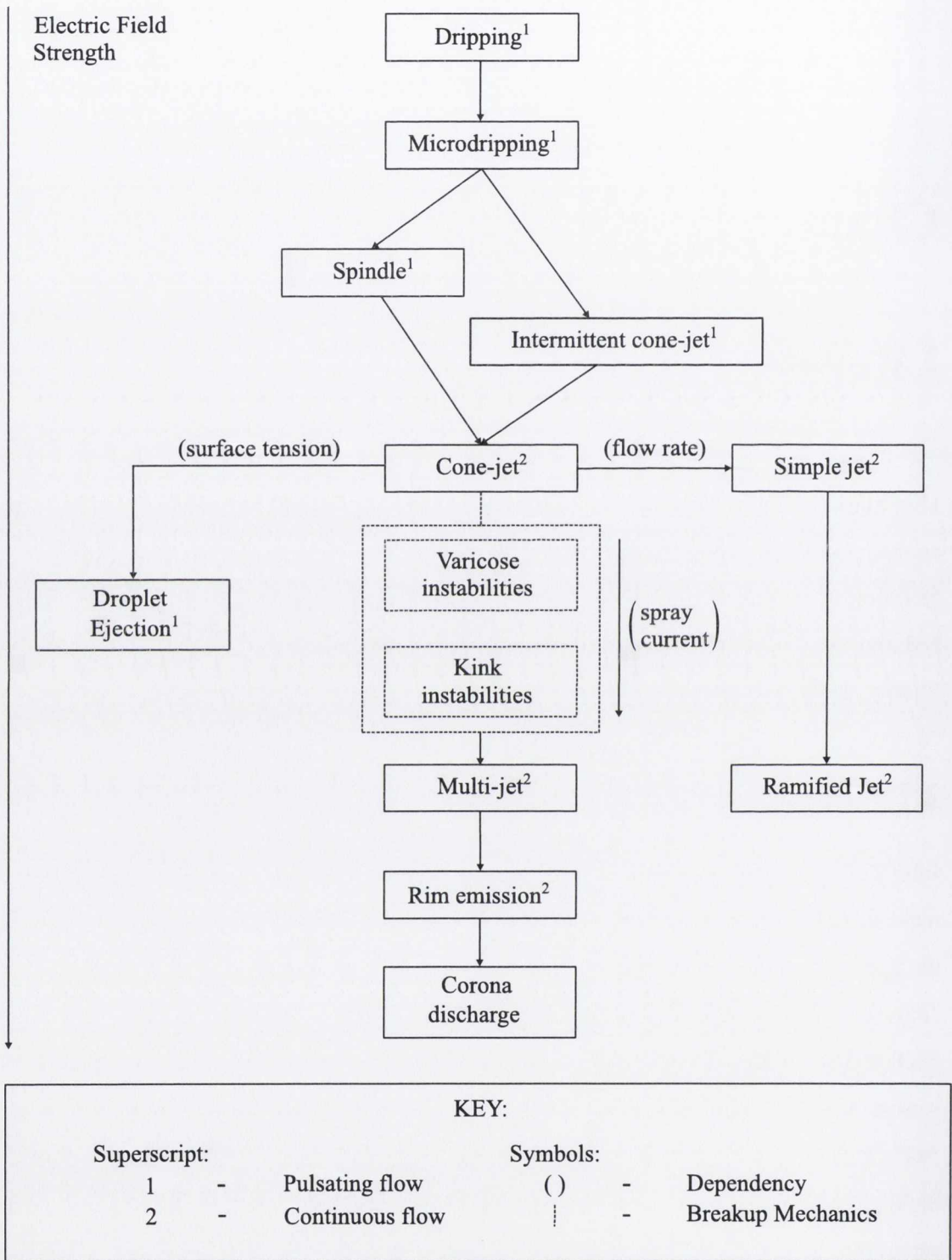


Figure 2.1: Electrospay spraying regimes overview. Adapted from Grace and Marijnissen [35].

The pulsating modes consist of: dripping, microdripping, spindle and intermittent cone-jet. While the cone-jet, simple jet, ramified jet and multi-jet are defined as the continuous regimes

Table 2.1: Single source cone-jet electrospray working fluid property spectrum [3].

Electrical conductivity, κ	$10^{-9} - 10^{-1}$	S m^{-1}
Surface tension, γ	$0.022 - 0.073$	Nm^{-1}
Viscosity, μ	$0.5 \times 10^{-3} - 80 \times 10^{-3}$	Nsm^{-2}

of spraying. Cone-jet electrospaying is the only regime that can produce stable micron sized quasi-monodisperse droplets that lends itself well to a multiple source scalable design. As such it the only operating mode employed in this research. Table 2.1 highlights the spectrum of working fluid properties that cone-jet electrospray can be achieved with [3].

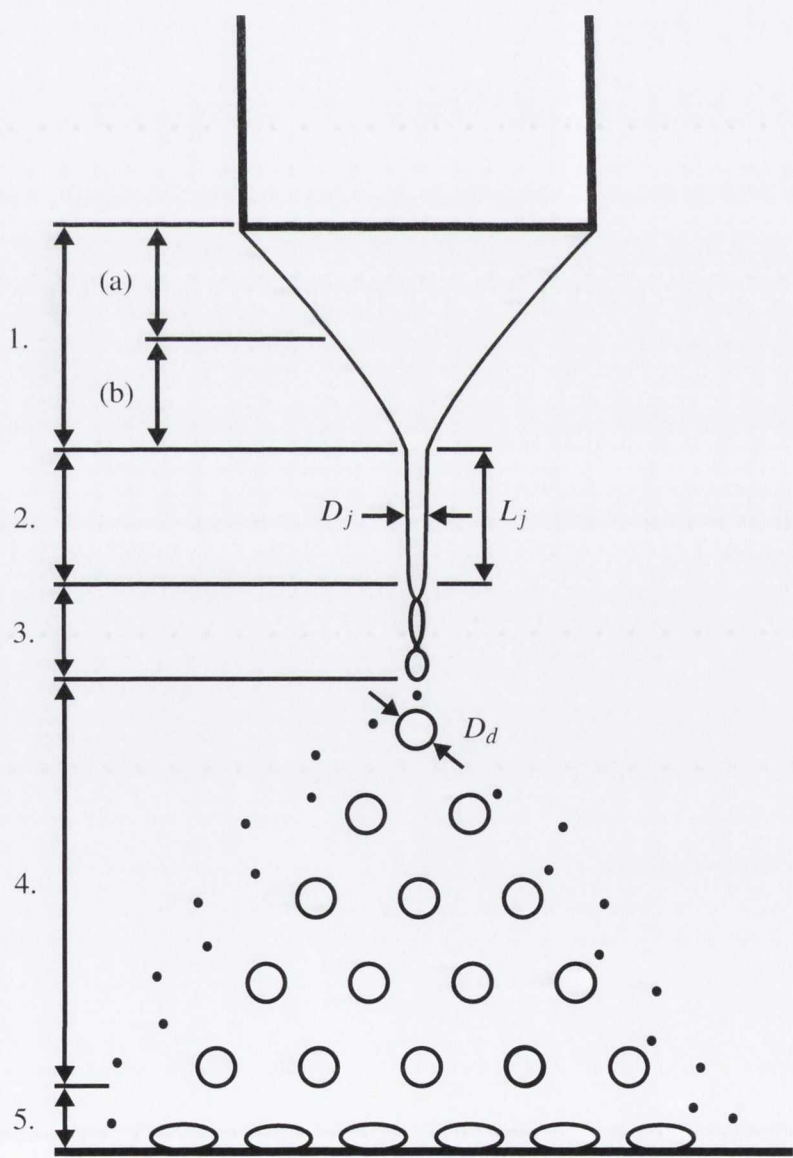


Figure 2.2: Electrospaying process.

2.2 Cone-jet Electrospray Cooling

Cone-jet electrospray cooling can be divided into five distinct phases; 1. cone formation, 2. jet formation, 3. droplet creation, 4. droplet transit, and 5. droplet impact and heat transfer. These processes are shown in Figure 2.2.

2.2.1 Cone Formation

The applied electric field distorts the morphology of the liquid meniscus into that of a cone. This cone is defined by a balance of surface tension, hydrostatic, gravitational, viscous and electrical forces [76, 77]. This force balance is shown in Figure 2.3. The cone itself can be split into two distinct regions [78], defined by the primary means of charge transportation. The first (1(a)) through conduction and the second (1(b)) through convection. This second zone is often termed the acceleration zone. The size of this acceleration region is dependent on the electrical conductivity of the working fluid. If the electrical conductivity is high the acceleration zone is localised to the tip of the cone with the rest of the cone (1(a)) in a state

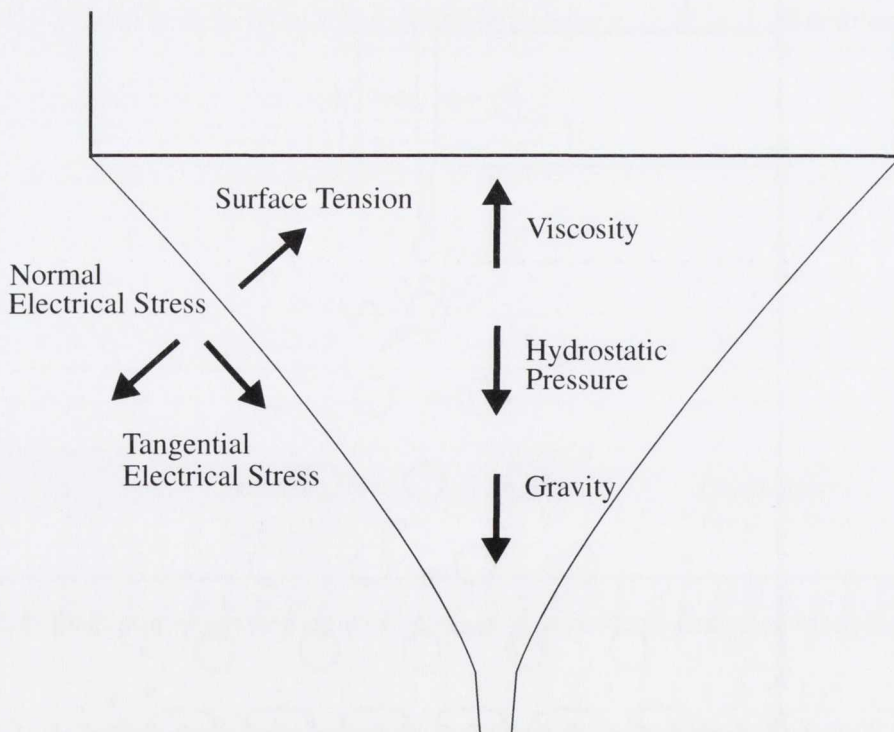


Figure 2.3: Cone-jet force balance.

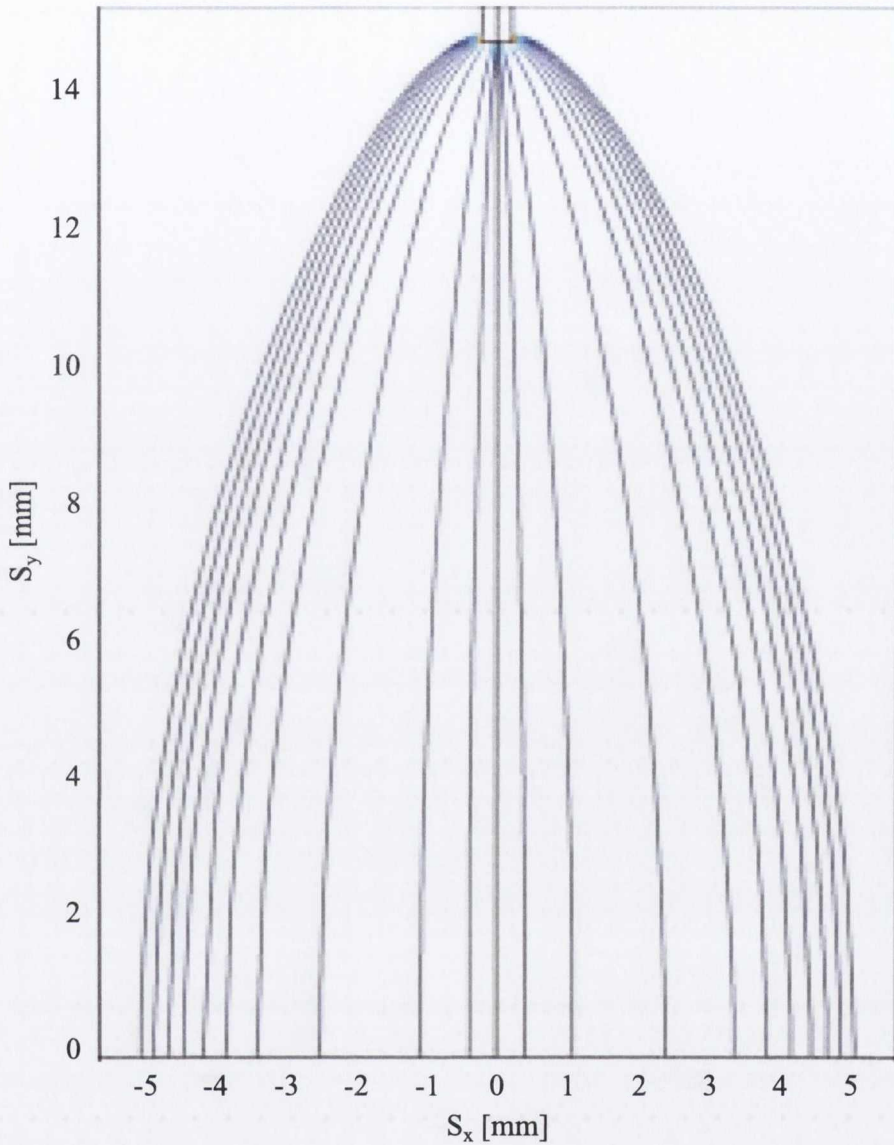


Figure 2.4: Electric field lines between a grounded plate and source nozzle. No working fluid, $V_n = 5000$ V, $H = 15$ mm, $D_o = 0.4$ mm and nozzle length $L_n = 30$ mm. Reprinted from Zhong et al. [75].

of static equilibrium. Conversely, if the electrical conductivity is low the acceleration zone grows and extends towards the base of the cone. If the acceleration zone extends beyond the base of the cone then the cone may behave asymmetrically [74, 76]. The cone length L_{cj} is the distance between the nozzle edge and the cone apex and is a function of the applied voltage and is typically of the same order as the source nozzle radius [76].

Figure 2.4 shows the electric field lines between a source nozzle and target ground electrode in the absence of working fluid. Zhong et al. [75] computed these results using

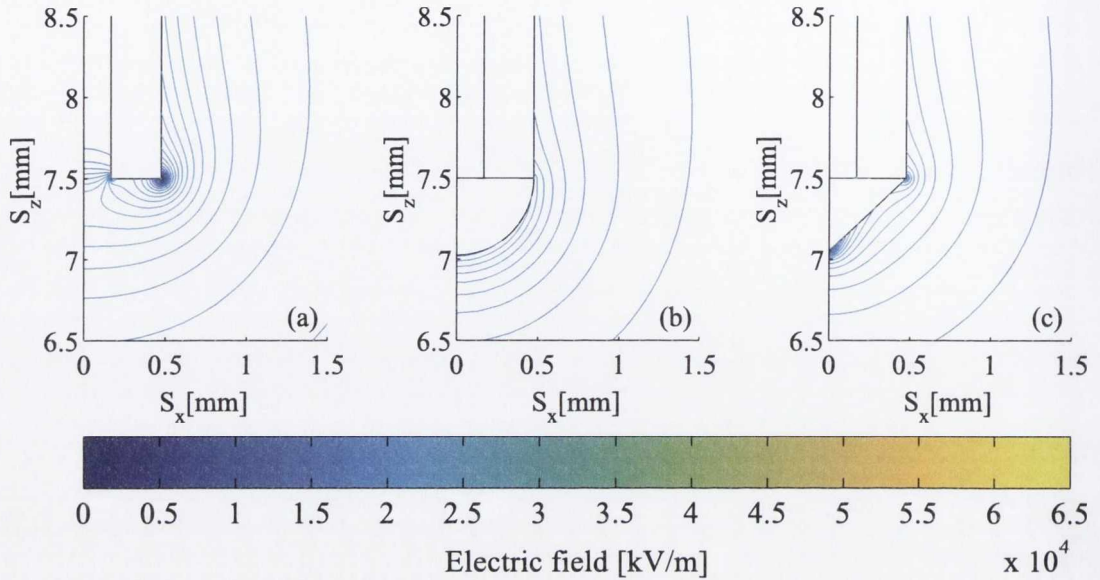


Figure 2.5: Electric field shape and intensity for static cases.

COMSOL multiphysics (No working fluid, $V_n = 5000$ V, $H = 15$ mm, $D_o = 0.4$ mm and nozzle length $L_n = 30$ mm.). A peak electric field was noted at the outside edge of the source nozzle. A hyperbola shape was observed for the plotted electric field lines. Figure 2.5 qualitatively illustrates an electric field contour plot of the cone-jet electrospray at different stages of spray formation; nozzle with no liquid, nozzle with hemispherical liquid cap and nozzle with liquid cone. The electric field was modelled in 2D space using the PDE electrostatic toolbox in Matlab 2016. The geometry in 2D space corresponds to the experimental case outlined in Figure 4.2 ($D_i = 0.330$ mm, $D_o = 0.63$ mm, and a flat semi-infinite ground electrode placed 7.5 mm directly below the nozzle for all cases). A 4 kV potential is applied to the boundary of the source nozzle and working fluid, 0 kV to the ground electrode with all other boundaries left open. Figure 4.2a corresponds to the most basic case of no working fluid. A higher electric field is qualitatively observed at the inside and outside edge of the nozzle. Figure 4.2b demonstrates the resultant electric field with a liquid hemispherical cap on the nozzle. The absence of sharp geometries results in more dispersed electric field. Figure 4.2c illustrates the electric field in the case of an ideal liquid cone with no jet. The highest electric field is noted at the jet forming region and at the base of the cone. Penetration of the electric field lines on the liquid cones establishes a tangential electrical field component. This component acts on the cone surface charge facilitating liquid acceleration of the jet downstream [3, 12, 80].

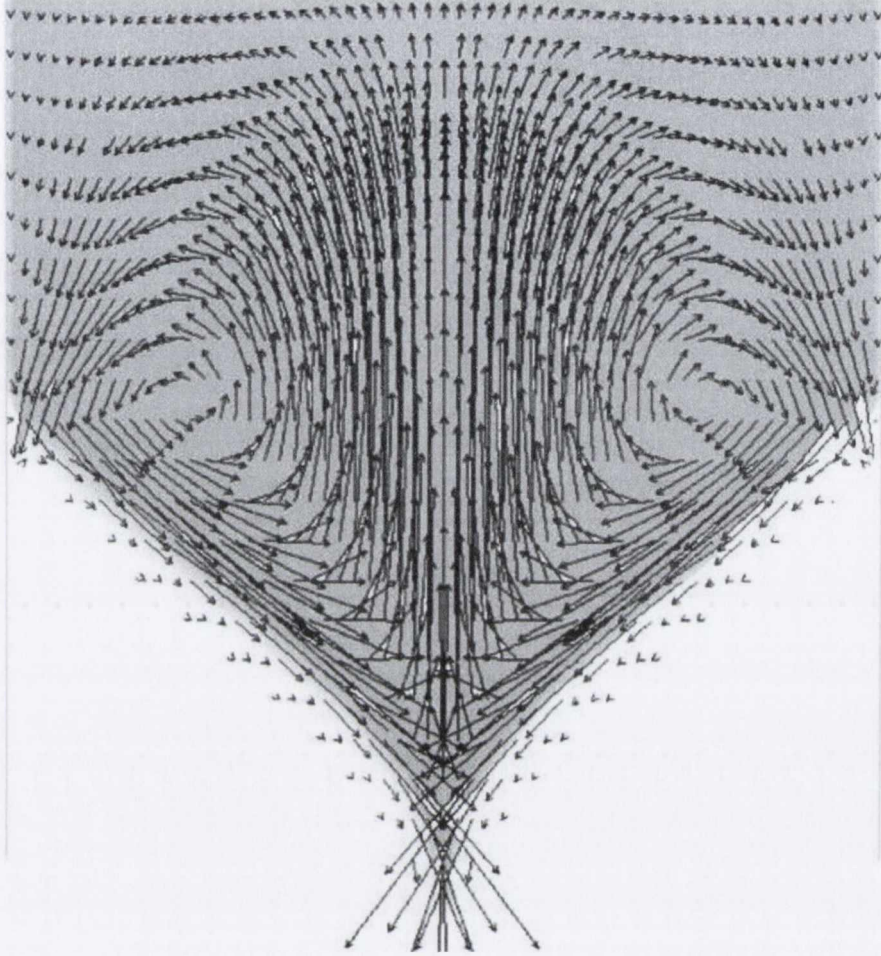


Figure 2.6: Velocity field in an ethanol liquid cone. $Q = 1 \mu\text{Lmin}^{-1}$ and $E_{field} = 10^4 \text{ kV m}^{-1}$. Reprinted from Lastow and Balachandran [79].

Figure 2.6 highlights the the velocity field in an ethanol liquid cone ($Q = 1 \mu\text{Lmin}^{-1}$ and $E_{field} = 10^4 \text{ kV m}^{-1}$). This solution was computed by Lastow and Balachandran [79] by solving a modified heat transfer equation with CFX 4.4 (ANSYS Inc.) to solved the electric field equations.

Geoffrey Taylor [10] was the first to give a mathematical explanation of the cone geometry observed by Zeleny. Taylor assumed no external applied hydrostatic pressure on the working fluid, gravitation forces are neglected, the cone is perfectly straight and the working fluid is at equipotential. He showed that equilibrium between capillary and electrostatic forces can only be achieved when the semi-angle of the liquid cone ϕ_c is equal to 49.29° . Taylor's model was unable to account for spray current or working fluid flow through the cone due to the absence of a liquid jet. Hayati et al. [12, 20] observed a cone angle range of $17.7^\circ - 53.4^\circ$ for an isopar

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m and n-butanol mixture for $Q = 200 - 1000 \mu\text{L min}^{-1}$, $V_n = 9 - 14 \text{ kV}$, $H = 22 \text{ mm}$ and $D_o = 2 \text{ mm}$. Many authors have attempted to build on the early work of Taylor. Analytical, numerical and experimental investigation on the cone morphology, stability, semi-angle have been carried out by Mutoh et al. [81], Smith [80], Joffre and Cloupeau [82], Hayati et al. [12, 20], Cloupeau and Prunet-Foch [3, 4, 77], Rosell-Llompart and De La Mora [83], Pantano et al. [84], among others. Eyring et al. [85] defined the electric field between a point source and a ground plate:

$$E_{field} = \frac{BV_n}{D_o \ln(8H/D_o)} \quad (2.1)$$

where D_o is the outer diameter of the source nozzle and B is an experimentally defined constant, often taken as 4 in the literature [45, 86]. Smith [80] defined the onset electric field E_{onset} to induce cone-jet electrospraying as:

$$E_{onset} \approx \left(\frac{4\gamma \cos \phi_c}{\epsilon_0 D_o} \right)^{1/2} \quad (2.2)$$

where ϵ_0 is the permittivity of free space. The onset electric field is the field required to overcome the surface tension of the working fluid and cause instability of the static Taylor cone leading to formation of a charged jet [75]. Equation 2.1 and 2.2 can be combined and solved to estimate the required source nozzle voltage for a given single source configuration. For example taking ethanol as the working fluid, $H = 7.5 \text{ mm}$, $D_o = 0.7 \text{ mm}$, and $\phi = 49.3^\circ$, a theoretical onset voltage of 2,126 V is calculated.

2.2.2 Jet Formation

Once a sufficient electric field is applied to overcome the surface tension of the working fluid a permanent jet forms at the apex of the cone. The length L_j and width D_j of this jet is dependent on the surface tension, viscosity μ , electrical conductivity κ , applied electric field, and flow rate of the working fluid [4, 40]. Penetration of the applied electric field into the liquid cone leads a tangential electrical field component. This tangential component acts on the cone surface charge facilitating liquid acceleration of the jet downstream [3, 12, 80]. While the acceleration zone base diameter D_{acc} is much smaller than the cone-jet base

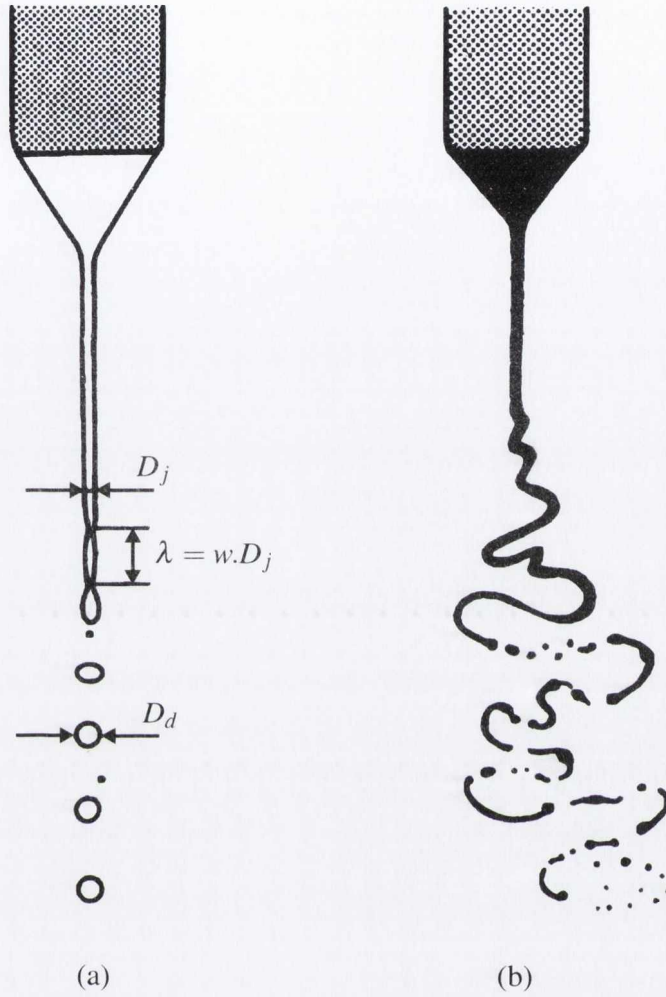


Figure 2.7: Cone-jet break-up (a) varicose instabilities and (b) kink instabilities. Reprinted from Cloupeau and Prunet-Foch [74].

diameter D_c the formed jet would be of near identical characteristics regardless of nozzle size [74]. The average break-up position of the jet is approximately 30 – 50 times D_j in the direction of the spray [87]. The length of the jet increases with the viscosity, surface tension and flow rate of the working fluid and decreases with increasing electrical conductivity [74, 87].

2.2.3 Droplet Creation

The formed jet undergoes Rayleigh instability [88] and breaks up into charged droplets. Droplets produced in the cone-jet regime can be mono or polydispersed in nature depending on their break-up condition. Satellite droplets can accompany primary droplet production [40,

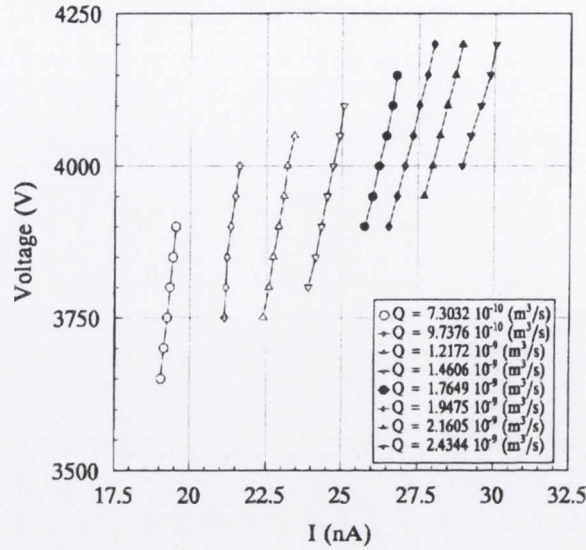


Figure 2.8: Dependence of spray current on the source voltage for different values of the flow rate. Working fluid was heptane doped with a small amount of Stadis 450, $D_0 = 1$ mm, $D_i = 0.7$ mm, with a separation height of “a few centimeters”. Reprinted from Gañán-Calvo et al. [89].

74]. These break-up conditions can be separated into two regimes; varicose and kink instabilities, shown in Figure 2.7, with the former producing monodisperse and the latter polydisperse droplets.

2.2.3.1 Spray Current

For highly conducting liquids, the spray current is transported primarily by conduction in the cone and convection in the jet [13]. It has been shown by numerous authors [27, 89–91] that the spray current of an electrospray is dependent on the source nozzle size, permittivity of free space, relative permittivity of the working fluid, working fluid flow rate, and applied voltage. Figure 2.8 shows the spray current dependency on working fluid flow rate and source nozzle voltage as observed by Gañán-Calvo et al. [89]. Their working fluid was heptane doped with a small amount of Stadis 450, $D_0 = 1$ mm, $D_i = 0.7$ mm, with a separation height of “a few centimeters”.

2.2.3.2 Varicose Instability

Vonnegut and Neubauer [92] were the first to note the quasi-monodisperse nature of an aerosol produced when an electric field is applied to a liquid meniscus. Since their initial

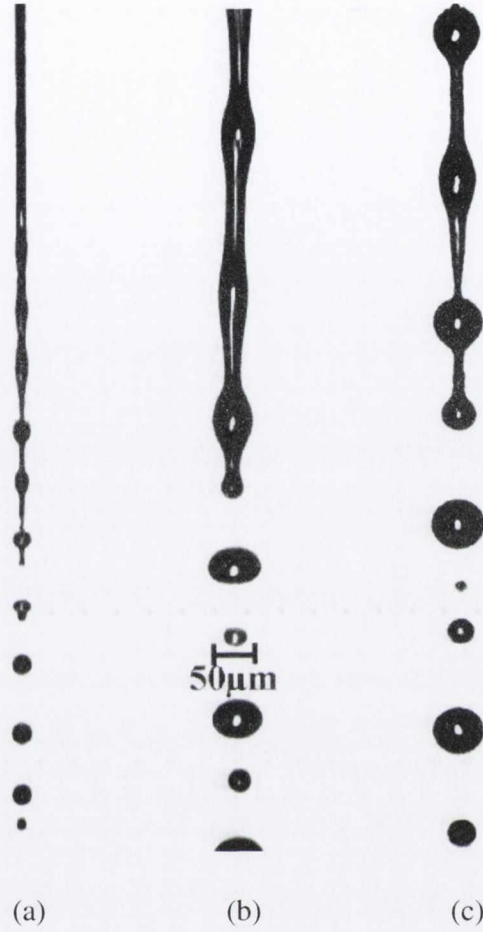


Figure 2.9: Cone-jet break-up in the varicose regime. n-butanol and iso-butanol, $Q = 50.4 - 294 \mu\text{L min}^{-1}$. Reprinted from Hartman et al. [90].

observation, a significant amount of research [4, 11, 27, 68, 76, 80, 83, 83, 89, 90, 93] has shown the mean droplet diameter D_d in the cone-jet regime of spraying depends on the working fluid flow rate, electrical conductivity, surface tension, viscosity, relative permittivity ϵ_r and the electric field strength, with a monotonic relationship observed between the droplet size and flow rate [11, 68, 83, 89]. If $D_j \ll D_c$ then the applied electric field will have no impact on the primary droplet size [13, 74]. For the monodisperse case, if the spray current is sufficient the jet will break up in a similar manner to that of an uncharged jet [74]. Primary droplets make up $\approx 97\%$ of the flow and $\approx 86\%$ of the total spray current [11].

Before breakup occurs, axisymmetric varicose waves propagate along the jet as a result of naturally occurring disturbances shown in Figure 2.9. This wave propagation leads to a changing pressure distribution along the jet. This pressure difference, due to surface tension

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stresses, drives the liquid away from the jet restrictions. This results in pinching of the jet into individual droplets [13, 90]. Satellite droplet production occurs during the detachment of the primary droplet from the jet [96]. It has been shown that the droplet diameter produced by varicose instabilities depends directly on the jet thickness. The following relationship was defined for low viscosity fluids such as ethanol [40, 74]:

Table 2.2: Electrospray parametric relationships.

Parameter (increasing)	Droplet size D_d	Jet thickness D_j	Jet length, L_j	Spray current I_{spray}	Emission frequency f_e
Flow rate Q	Increase [11, 27, 68, 83, 89]	Increase [11, 27, 68, 83, 89]	Increase [74, 87, 94]	Increase [27, 78]	Decrease [4, 27]
Electrical conductivity κ	Decrease [6]	Decrease [6]	Decrease [74, 78, 87]	Increase [6]	Increase [4, 6]
Surface tension γ	Increase [4]	Increase [4]	Increase [4, 87]	Increase [4]	Decrease [4]
Relative permittivity ϵ_r	Increase [27]	Increase [27]	Increases [27]	Decrease [27]	Decrease [27]
Viscosity μ	Increase [4, 87]	Increase [4, 87]	Increase [4, 74, 87, 95]	N/A	Decrease [4, 87]
Applied voltage V_n	Decrease ¹ [20] / No effect ² [13, 83]	Decrease ¹ [20] / No effect ² [13, 83]	Decrease ¹ [20] / No effect ² [13, 83]	Increase [76]	Increase ¹ [20] / No effect ² [13, 83]
Separation height H	No effect [11, 83]	No effect [11, 83]	No effect [11, 83]	No effect [11, 83]	No effect [11, 83]
Nozzle size D_i	No effect ² [13, 74]	No effect ² [13, 74]	No effect ² [13, 74]	No effect ² [13, 74]	No effect ² [13, 74]

¹ $D_j \sim D_c$

² $D_j \ll D_c$

$$D_d = 1.89D_j \quad (2.3)$$

Jet break up time has been shown to depend on the flow rate and primary droplet size [97]. A summary of the impact of key parameters from the literature is shown in Table 2.2.

2.2.3.3 Kink Instability

As the spray current is increased, the surface charge on the jet increases. This results in the break-up regime transitioning from the varicose instabilities axisymmetric break-up to lateral or azimuthal instabilities, also known as kink instabilities (see Figure 2.7 (b)) [90]. This type of break-up appears when the electrostatic pressure approaches the capillary pressure [3, 35, 74]. Droplets produced are normally polydisperse due to the whipping nature of the break-up [4, 90]. The charge on the larger droplets generally exceed that of the Rayleigh limit charge¹ $q_{e,max}$ [3, 88]. If the spray current is increased further lateral kink-type instabilities appear [3, 74]. This is denoted by the whipping motion becoming more pronounced with droplets produced from multiple places on the jet. The produced droplets are extremely polydisperse. This mode only occurs for certain working fluid conductivities and flow rates. The value of this minimum flow rate increases as the electrical conductivity decreases [3].

2.2.3.4 Scaling Laws

Several scaling laws have been developed to predict the droplet size and spray current for the cone-jet electrospray. They are derived by theoretical [27, 89] and numerical analysis [98]. Fernández De La Mora and Loscertales [27] developed a pair of scaling laws for the spray current and primary droplet size for polar working fluids with high electrical conductivities $\kappa > 10^{-4} \text{ S m}^{-1}$)². This case results in a nearly equipotential cone, except near tip where the convective time is too fast for the charge to be brought to the tip surface from the interior via conduction [13]. Their scaling law applied for a dimensionless flow rate parameter between the ranges of $0.51 < [\rho \kappa Q / (\gamma \epsilon_r \epsilon_0)] < 2.01$.

¹The Rayleigh limit charge is achieved when the pressure due to electrostatic forces on the jet is equal to ≈ 0.82 times the capillary pressure [74].

² $\kappa_{ethanol} = 1.35 \times 10^{-7} \text{ S m}^{-1}$.

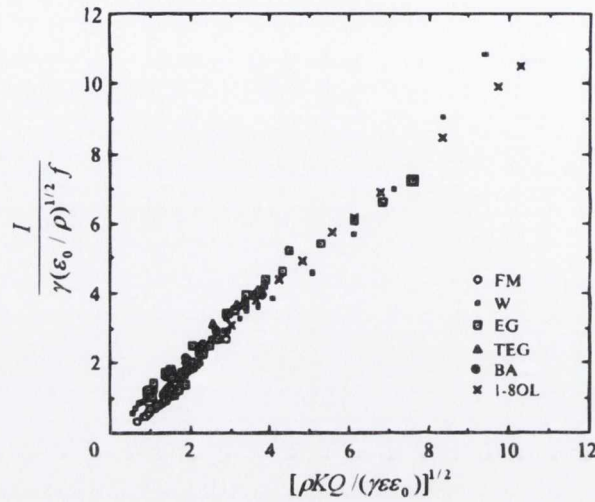


Figure 2.10: Spray current scaling law for highly conducting working fluids. Formamide (FM), water (W), benzyl alcohol (BA), ethylene glycol (EG), triathlon glycol (TEG) and 1-octanol (18OL). Reprinted from Fernandez de la Mora and Loscertales [27].

$$I_{spray} = f(\epsilon) \left(\frac{\gamma Q K}{\epsilon_r} \right)^{1/2} \quad (2.4)$$

$$D_d = G(\epsilon) (Q \epsilon_r \epsilon_0 / \kappa)^{1/3} \quad (2.5)$$

where $f(\epsilon)$ and $G(\epsilon)$ are dimensionless functions determined experimentally. Fernández De La Mora and Loscertales showed good agreement between their derived scaling law for spray current (Equation 2.4) and experimental results for various working fluids. These results are shown in Figure 2.10. No consensus on the functional form of $G(\epsilon)$ in Equation 2.5 has been shown [99].

In the case of lower electrical conductivities and viscosity ($[\epsilon_0^2 \gamma^3 / (\kappa^2 \mu^3 Q)]^{1/3} \gg 1$)³, Gañán-Calvo et al. [89] suggested a power law exponent of 0.25 and 0.5 for the spray current and droplet diameter scaling respectively. They derived their laws from two relationships. First, that the charge per unit volume scales with the Rayleigh limit (Equation 2.12) as shown by Cloupeau and Prunet-Foch [4] and Gomez and Tang [91].

$$I_{spray} / Q \sim D_d^{-3/2} \quad (2.6)$$

³Present research: $[\epsilon_0^2 \gamma^3 / (\kappa^2 \mu^3 Q)]^{1/3} = 51 - 102$

Second, that the advection current, which is of the order of the total current, is given by:

$$I_{\text{spray}}/Q \sim \pi D_j U_{js} \bar{q} \quad (2.7)$$

where u_{js} and $\bar{q} = (2\gamma\epsilon_0/D_j)^{1/2}$ are the velocity of the jet surface and charge per unit area respectively. The velocity of the working fluid on the surface of the jet is dependent on the the tangential electrical stress $\epsilon_0 E_n^0 E_s$ and the boundary layer thickness $\mu L_j / \rho U_{js}$ such that:

$$u_{js} \sim \left(\frac{\epsilon_0 \gamma}{\rho \mu} \right)^{1/3} \left(\frac{E_s^2 L_j}{D_j} \right)^{1/3} \quad (2.8)$$

The relationship between E_s^2 and D_j/L_j is unknown but Gañán-Calvo et al. observed that the best fit with their experimental data was achieved when $E_s^2 L_j / D_j$ is constant and independent of D_j or Q . In this case:

$$I_{\text{spray}} \sim D_j^{1/2} \quad (2.9)$$

If it is assumed that droplet diameter scales with the jet diameter (Equation 2.3) it is found that:

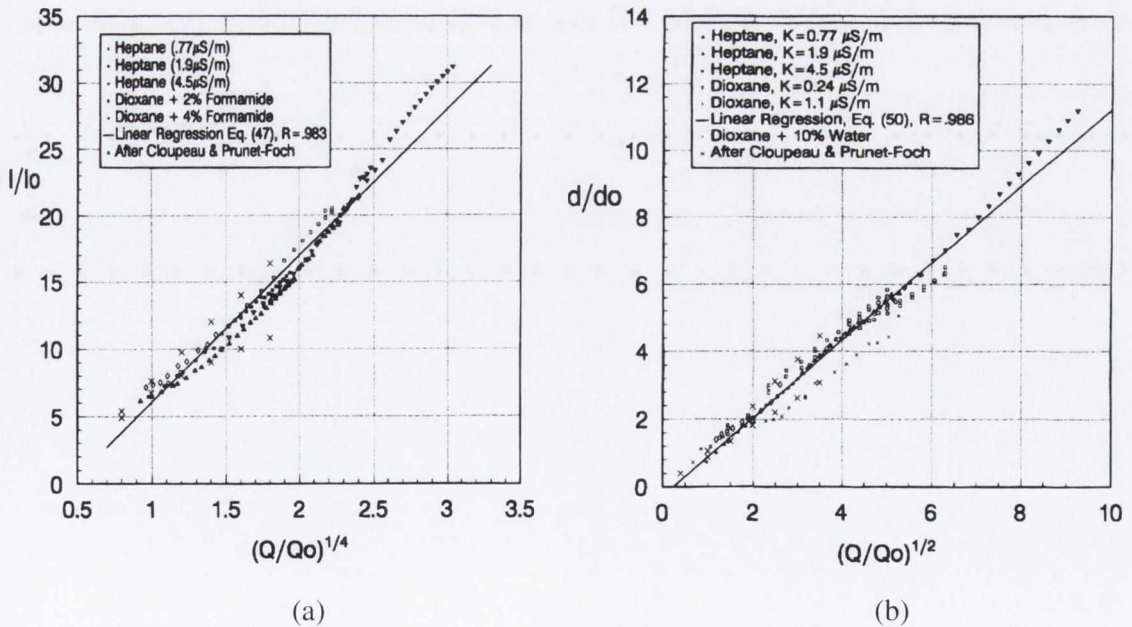


Figure 2.11: Scaling law agreement with experimental results. (a) Spray current; (b) Droplet size. Reprinted from Gañán-Calvo et al. [89].

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$$D_d \sim Q^{1/2}, \quad I \sim Q^{1/4} \quad (2.10)$$

They showed good agreement of their laws with experiments for doped hydrocarbons of varying electrical conductivities. The scaling law and experimental data correlation of Gañán-Calvo et al. [89] results are shown in Figure 2.11, where I_0 , Q_0 and D_0 are characteristic value for spray current, flow rate and droplet size respectively. Tang and Gomez [100] showed a power law exponent of 0.46 between droplet size and flow rate from their experimental data. They tested deionised water in a fixed electrode configuration over a wide range of working

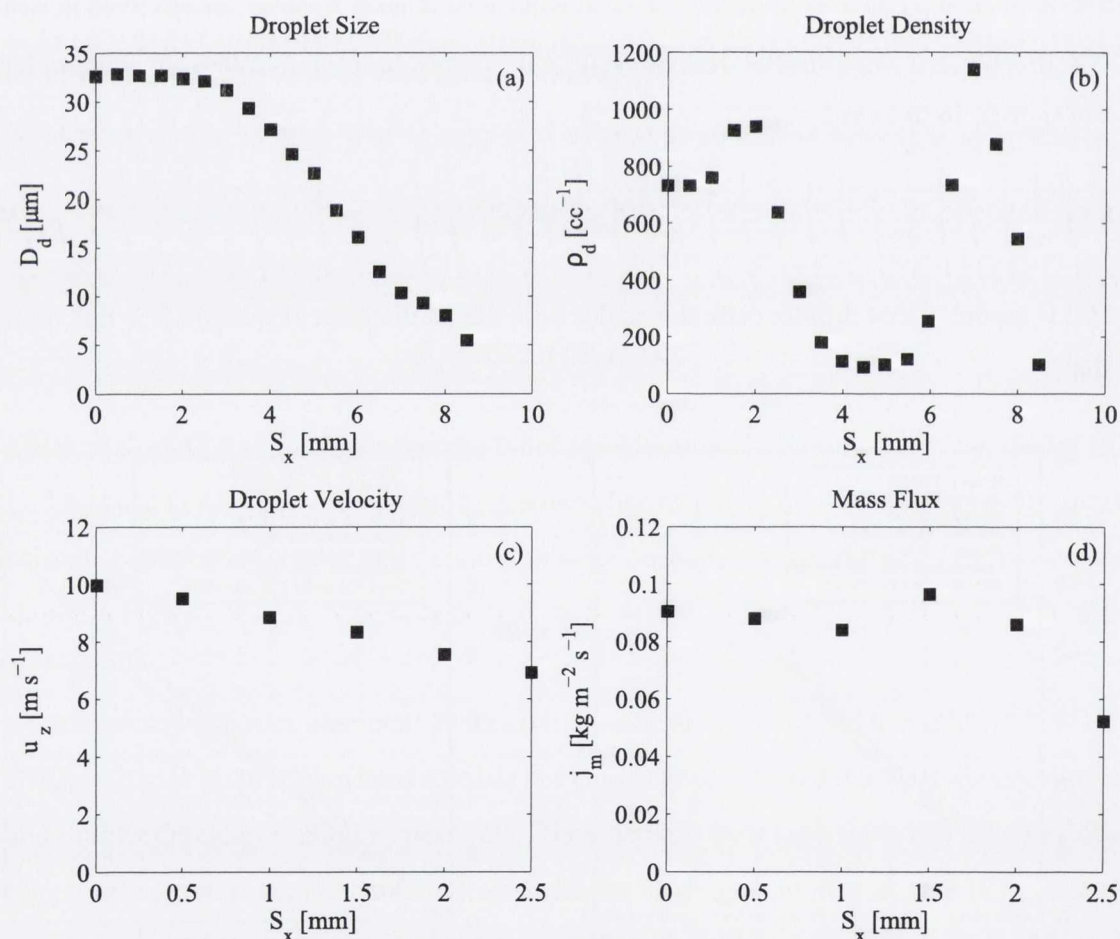


Figure 2.12: Spray plume dynamics. Heptane working fluid, $H = 30$ mm, $Q = 166.67 \mu\text{L min}^{-1}$, with measurements taken 12 mm from the source nozzle tip. (a) Droplet size, (b) dispersion density, (c) spray axis velocity and (d) mass flux vs. radial distance from nozzle centre. Adapted from Tang and Gomez [11].

fluid flow rates.

2.2.4 Droplet Transit

The generated charged droplets are propelled to the grounded target surface by the applied electric field and surrounding gravitational force [31]. While in transit, the similarly charged nature of the droplets enables plume dispersion and inhibits droplet coalescence [101]. Segregation of primary and satellite droplets in the spray plume has been noted [11, 91] and corroborated by numerous authors, with the spray core consisting principally of primary droplets with satellite droplets located around the periphery of the plume. These satellite droplets are segregated by inertial and electrostatic effects [97] and make up $\approx 3\%$ of the total flow rate and $\approx 14\%$ of the total charge [11]. These smaller droplets move at a lower velocity relative to the central spray core due to the radial decay of the electric field [87].

Figure 2.12 is an adaption of the experimental results from work carried out by Tang and Gomez [11] on the structure of a cone-jet electrospray plume. This test point corresponded to $H = 30$ mm, $Q = 166.67 \mu\text{L min}^{-1}$ with measurements taken 12 mm from the source nozzle tip using heptane as the working fluid. Figures 2.12a, b and c show that droplet size is uniform

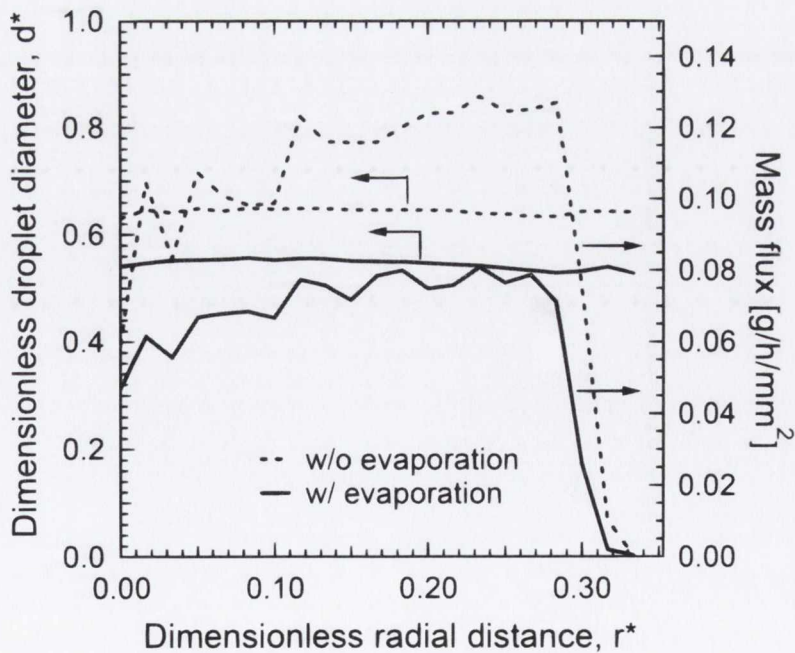


Figure 2.13: Numerical droplet size and electrospray mass flux. $H = 30$ mm, $Q = 45.5 \mu\text{L min}^{-1}$ with butanol as the working fluid. Reprinted from Wilhelm et al. [67].

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in the spray core ($S_x = 0 - 2$ mm), with increasing deposition density at the periphery of the spray core, and decreasing spray direction droplet velocity as S_x increases. The second peak in droplet density in Figure 2.12b at $S_x = 7$ mm corresponds to satellite droplet deposition. From Figure 2.12d it can be seen that the droplet mass flux is nearly uniform in the spray core with a standard deviation of 5.4%. A similar mass flux profile is shown in Figure 2.13 from the numerical work of Wilhelm et al. [67]. Their results were calculated for $H = 30$ mm, $Q = 45.5 \mu\text{L min}^{-1}$ with butanol as the working fluid. The graph highlights two cases: one of an unheated substrate; and one with the substrate heated to 300 K.

The path a droplet takes en-route to the target surface can be described quantitatively using Lagrangian tracking of an individual droplet in a dilute two phase flow [67]. The force balance for the transport of a single droplet, i , in a low density spray assuming no collisions is given by [87, 102]:

$$\frac{\pi}{6} D_{d,i}^3 \rho_l \frac{d\vec{u}_i}{dt} = C_D \frac{\pi}{8} \rho_g D_{d,i}^2 (u_i - u_g)^2 \vec{e}_i + q_{e,i} \vec{E}_e + \frac{1}{4\pi\epsilon_0} \sum_{ij, i \neq j} \frac{q_{e,i} q_{e,j}}{r_{ij}^3} \vec{r}_{ij} + F_{image} \quad (2.11)$$

The left side of the equation is the force due to the mass times the acceleration of the i th droplet, where $D_{d,i}$ is the diameter of the i th droplet ρ is the density with sub scripts l and g denoting the working fluid and surrounding gas respectively. The first term on the RHS is the drag force by the surrounding medium, where C_d is the drag coefficient on the falling droplet, u_i and u_g is the velocity of the droplet and surrounding medium respectfully and e_i is the velocity unit vector. The second term accounts for the force on the droplets by the external electric field, where $q_{e,i}$ is the droplet charge and E_e is the external electric field. The third term on the RHS is the electric force between adjacent charged droplets, with r_{ij} describing the displacement vector of the i th and the j th droplets. The final term is the image force which was shown by Deng and Gomez [70] to scale as $\sim A_d \gamma / 2D_{d,i}$, where A_d is the surface area of the droplet. Wilhelm et al. [67] solved Equation 2.11 to show the deposition pattern of a cone-jet electrospray on a heated substrate. This was done to investigate the deposited film quality. Yang et al. [102] implemented Equation 2.11 to investigate the deposition pattern of a multi-source electrospray array. This was done as a means of developing a design tool for future array design. Their model showed good agreement with experimental results.

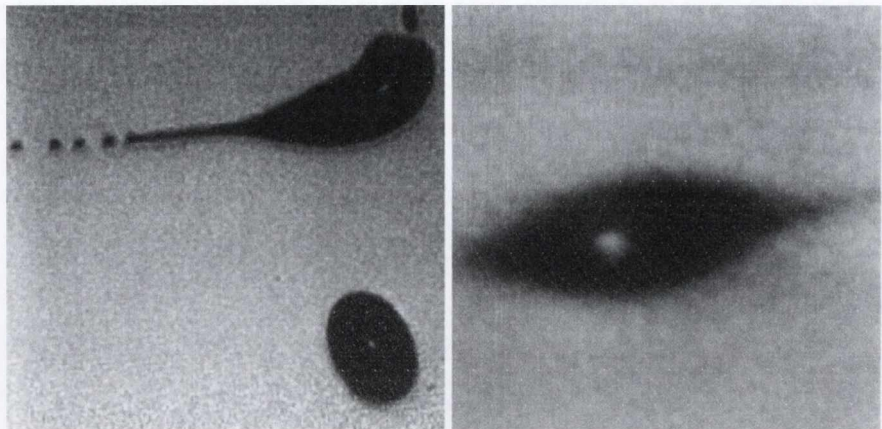


Figure 2.14: Coulomb fission of a charged droplet. Reprinted from Gomez and Tang [91].

2.2.4.1 Droplet Charge and Coulomb Fission

In 1882, Lord Rayleigh showed that a charged droplet becomes unstable once the repulsive force due to the trapped charge on the droplet surface is greater than the surface tension forces [103]. The maximum droplet charge is given by:

$$q_{e,max} = (8\pi^2 \epsilon_0 \gamma D_d^3)^{1/2} \tag{2.12}$$

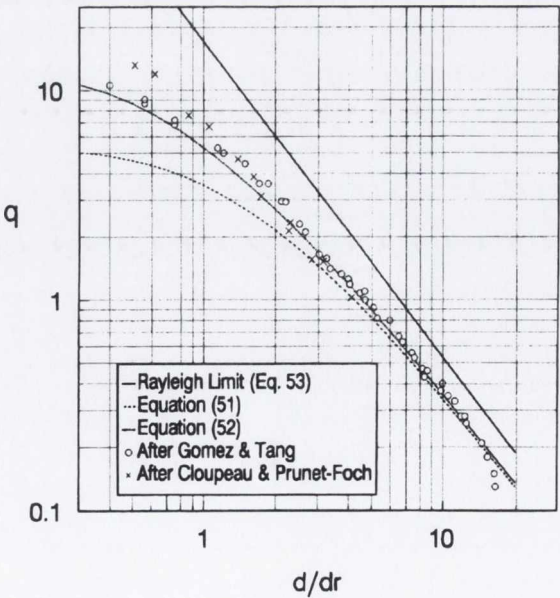


Figure 2.15: Droplet charge comparison of experimental data with Rayleigh prediction. Reprinted from Gañán-Calvo [89].

2.2. CONE-JET ELECTROSPRAY COOLING

De Juan and Fernandez de la Mora [106] showed that if the charge relaxation time $\tau = \epsilon_r \epsilon_0 / \kappa$ is an order of magnitude greater than the breakup time of the electrospray jet $t_b = \pi D_d^3 / 6Q$, the charge remains constant during break-up. This results in a uniform volumetric charge density for each droplet. If a droplet becomes charged above this Rayleigh limit Coulomb fission will occur. Coulomb fission is characterised by the formation of a permanent jet on the droplet, similar to that of the cone-jet at the source nozzle. From this jet a stream of smaller droplets are produced. An example of Coulomb fission is shown in Figure 2.14. Gomez and Tang [91] studied the fission of heptane droplets at charge levels below that of the Rayleigh limit generated by electrospraying. They showed that smaller droplets ($D_d < 50 \mu\text{m}$) underwent fission well below the Rayleigh limit (70 – 80%), but as the droplet diameter increased the rupture charge approached the Rayleigh limit. They suggested that aerodynamic effects played a significant factor, highlighting the results of Cloupeau and

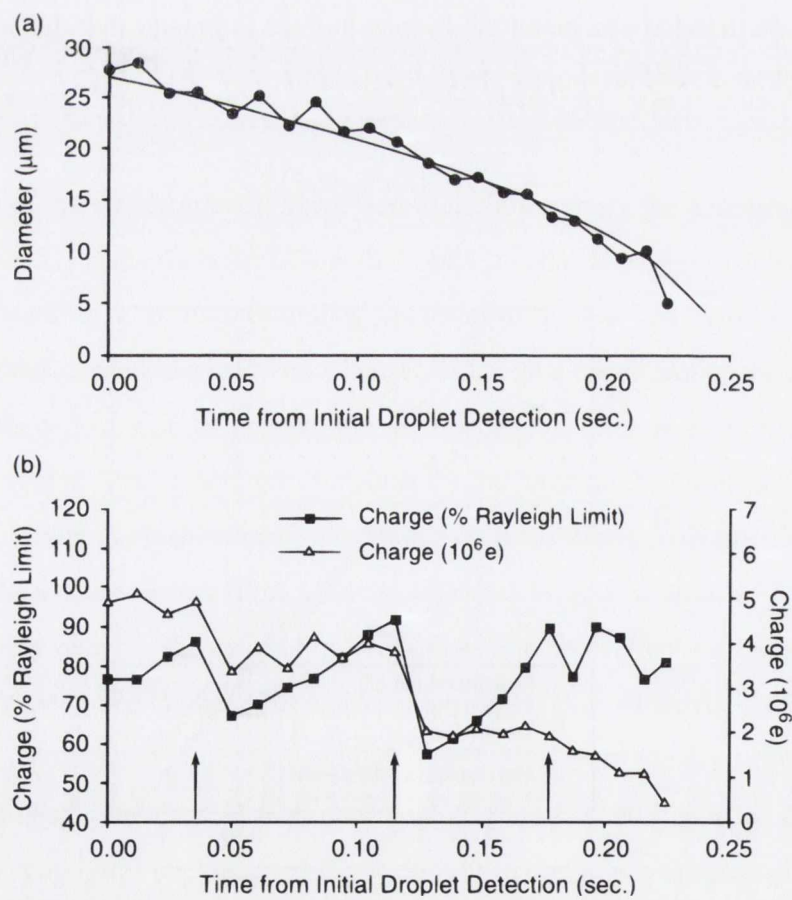


Figure 2.16: Droplet evaporation and fission process. The vertical arrows from the x-axis denote a fission event. Reprinted from Smith et al. [104].

Prunet-Foch [4], who showed that droplet fission could occur at any various fractions of the Rayleigh limit when the flow rate and droplet size are varied. Figure 2.15 shows a comparison of the experimental results of Cloupeau and Prunet-Foch [4] and Gomez and Tang [91] with that of the Rayleigh limit. As the droplet size increases the droplet charge asymptotically approach the Rayleigh limit.

Under certain spraying conditions, the produced charged droplets can undergo continuous evaporation due to the thermal energy supplied by the surrounding medium. This results in a decreasing droplet diameter and thus, a decreasing maximum charge value predicted by the Rayleigh stability criterion. As the droplet charge is conserved during the evaporation process, the droplet will reach a point where the electrostatic forces will overcome the surface tension forces resulting in Coulomb fission [104]. This process is shown in Figure 2.16 from the experimental work of Smith et al. [104] using neutral water droplets. They found each discharge event corresponded to a charge loss of 20 – 40% of the total charge. These satellite droplets will evaporate at a significantly higher rate than the parent droplet as droplet evaporation time varies inversely with the droplet diameter in diffusion controlled evaporation [91].

Gibson et al. [105] demonstrated droplet temperature fluctuation along the spray axis en-route to the target surface using radiometric fluorescence. Their experiments were conducted

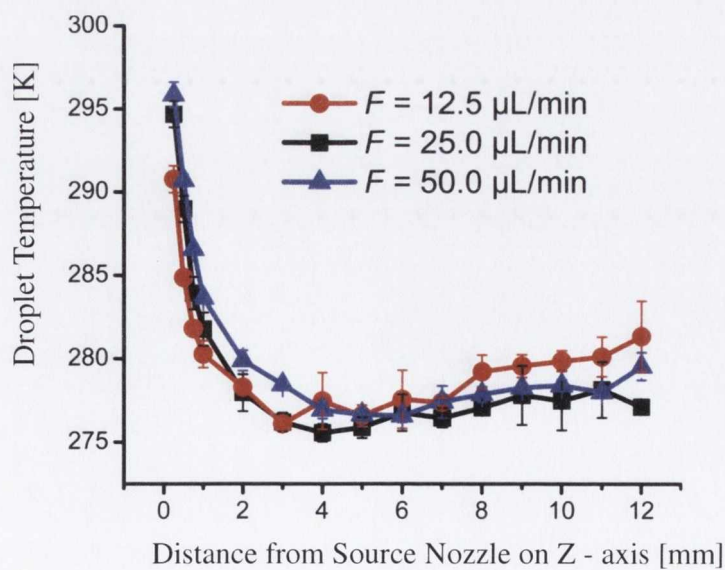


Figure 2.17: Droplet temperature during transit along centre spray axis. Reprinted from Gibson et al. [105].

with Methanol mixed with RhB and Rh110 reagents in the cone-jet regime. Figure 2.17 from their results shows initial significant droplet cooling due to endothermic solvent evaporation followed by marginal reheating attributed to collisional warming, the heated nebulizing gas used in their experiments and the ambient surroundings.

2.2.5 Droplet Impact

The charged nature of impacting droplets negates droplet rebound, lowers droplet contact angle and increases droplet spreading due to the applied image force between the charged droplet and grounded target surface [70, 71]. Droplet impingement and cooling can be split

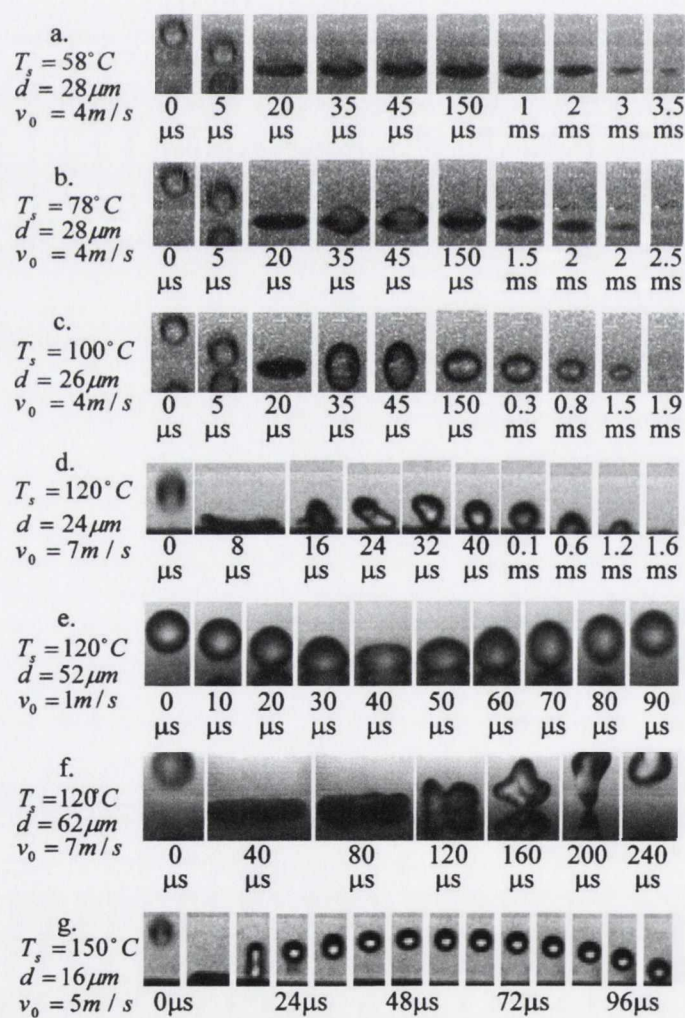


Figure 2.18: Charged microdroplet impact on temperature varied heated polished stainless steel surface. Reprinted from Deng and Gomez [70].

into three phases: impact, recoil and quasi-steady evaporation [70]. Impacting droplets can bounce, spread or splash, and their behaviour is governed by the properties of the working fluid, the impacted substrate, impact velocity, surrounding medium of the droplet, geometric aspects and the droplet shape [107]. Cone-jet electrospray droplet impact is relatively gentle in comparison with other atomisation techniques, with low Weber $We \sim O(10)$ and Ohnesorge $Oh \sim O(0.1)$ numbers [70]. These are shown in Equation 2.13 and 2.14. They relate inertial with surface tension forces and viscous to inertial and surface tension forces respectfully.

$$We = \rho D_d u_z^2 / \gamma \quad (2.13)$$

$$Oh = \frac{\mu}{(\rho \gamma D_d)^{1/2}} \quad (2.14)$$

The charged nature of impacting droplets has been shown to negate droplet rebound [70]. Deng and Gomez [70] investigated charged ethanol droplets impacting onto a grounded heated conductor for ambient conditions. As can be seen in Figure 2.18 they explored the effect that varying substrate temperature ($T_s = 58 - 150^\circ\text{C}$) had on charged droplet impact and the role that image forces played in avoidance of rebound. Droplet impact at the lowest temperature was similar to the three droplet impact phases outlined previously (impact, recoil and quasi steady evaporation). Larger droplet contact angles θ , higher recoil magnitude and shorter evaporation times were noted as the wall temperature was gradually increased up to 100°C . Typically droplet impact above the Leidenfrost point would lead to a contact angle of $\sim 180^\circ$, however only for some cases was the droplet rebound observed at this point due to the imposed image force. Droplet rebound is shown to be dependent on the droplet size and impact velocity. At the highest temperature of 150°C the droplet initially rebounds for all cases. However due to the charge and subsequent Coulombic attraction the droplet was quickly forced back to the grounded target surface. During impact the droplet deforms from a liquid sphere into a disk achieving its maximum solid-liquid interface. Ryu and Lee [71] demonstrated an increase in the maximum spreading radius of an impacting charged droplet on a dielectric surface in comparison with the uncharged case. In the recoil phase the liquid disk is retracted back to a hemispherical cap by surface tension forces, oscillation between the

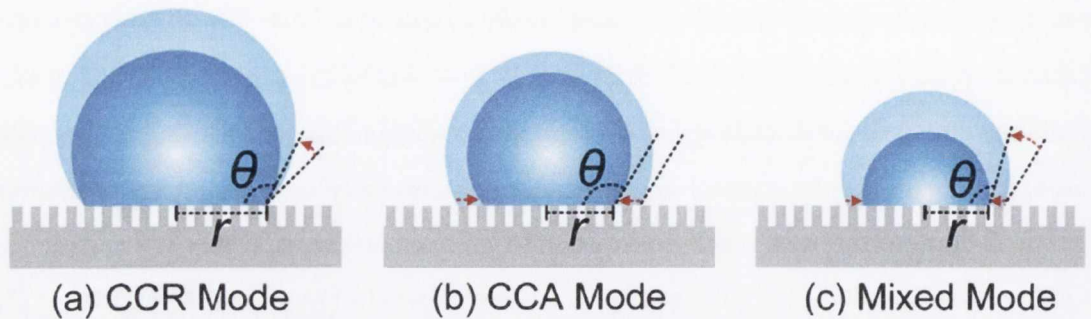


Figure 2.19: Illustration of sessile droplet evaporation modes: (a) constant contact radius (CCR), (b) constant contact angle (CCA), and (c) mixed mode. Adapted from Xu et al. [109].

liquid disk and hemispherical cap may occur multiple times after initial recoil [70, 107, 108]. The impacted stationary droplet then enters quasi-steady evaporation.

2.2.6 Droplet Evaporation

Droplet evaporation can occur in four primary modes: constant contact angle (CCA), constant contact radius (CCR), a mixed mode, and a stick slip mode. Figure 2.19 illustrates the CCA, CCR, and mixed mode evaporation regimes. Both CCA and CCR were first introduced by Picknett and Bexon [111]. In CCA, the droplet’s contact angle remains fixed with a decreasing base radius R_b . Whereas for CCR, the droplet’s radius is fixed (pinned) and droplet evaporation occurs with a decreasing contact angle. In mixed mode evaporation both the contact angle and radius decreases as the evaporation proceeds [109]. Finally in

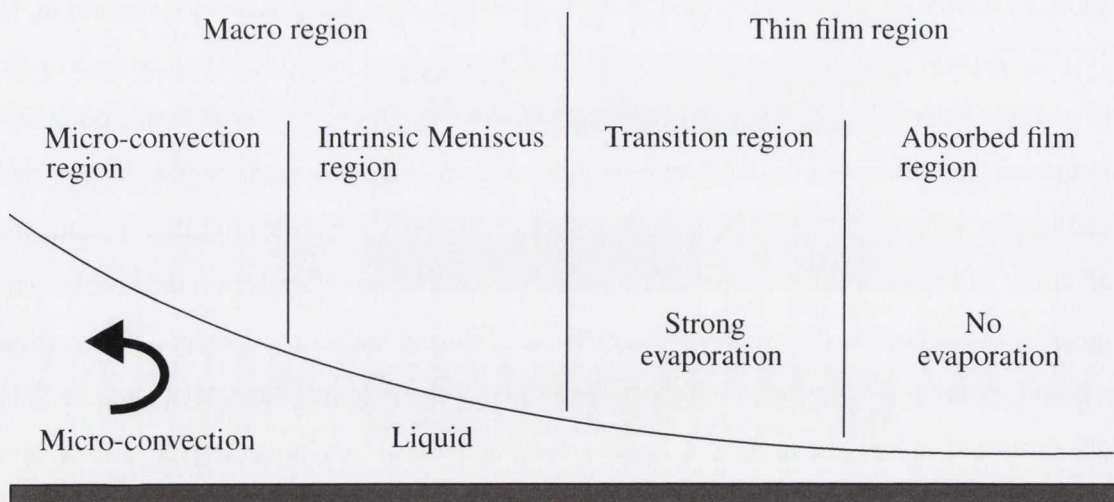


Figure 2.20: Droplet evaporation at the contact line. Adapted from Raghupathi and Kandlikar [110].

the stick-slip mode the contact line rapidly oscillates between pinning and unpinning [112]. Which mode occurs is dependent on the droplet and surface properties. For example, pinned droplets on a lyophilic surface, such as a highly wetting liquid on metal, normally evaporate in the CCR regime. Whereas weakly pinned droplets on a lyophobic surface, such as water/oil on a polymer, will evaporate in the stick-slip mode [113].

Droplet evaporation is dependent on the working fluid, surrounding gas, and heated substrate properties [115]. The highest local heat transfer for an evaporating droplet occurs at the contact line [116]. The contact line or triple line is defined as the region where the gas, liquid, and solid phases intersect. It can be broken-up into four distinct regions: micro-convection region, intrinsic meniscus region, transition region, and absorbed film region [110]. These are shown in Figure 2.20. The absorbed film region is characterised by long range intermolecular forces. This disjoining pressure results in a flat liquid-vapour interface of a few nanometers thick (10 - 20 nm) [117] and prevents evaporation occurring in this region [118]. The transition region is defined by growing film thickness which results in a reduction in the long range intermolecular forces. This region experiences the highest heat fluxes across the droplet as a result of the low thermal resistance from the small film thickness (1 - 3 μm) [117]. The total length of the absorbed film and transition region varies

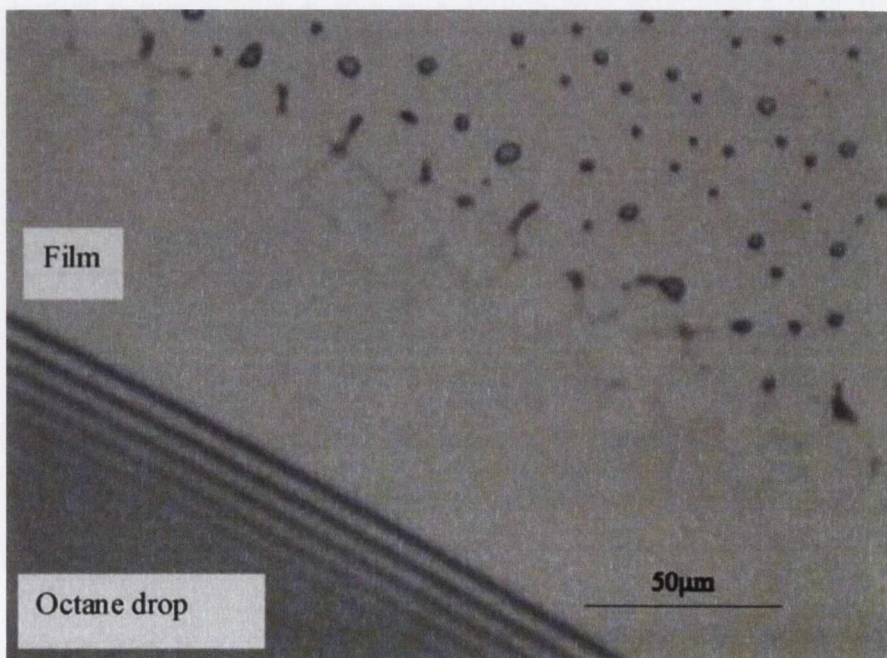


Figure 2.21: Receding contact line of an octane droplet. Reprinted from Guena et al. [114].

from $0.5\ \mu\text{m}$ to $10 - 20\ \mu\text{m}$ [117]. Figure 2.21 from Guena et al. [114] shows the receding contact line of an octane drop. They stipulated that the apparent contact line recedes on a tens on nanometer thick mesoscopic film. As the film thickness increases from the transition region into the intrinsic meniscus and micro-convection regions, so too does the thermal resistance resulting in a drop in the local heat flux. Both the intrinsic meniscus and micro-convection regions are characterised by surface tension and inertial forces [110]. Heat is transferred at the liquid-solid boundary by diffusion and advection [119]. The net thermal boundary resistance R_{net} is the sum of the solid-liquid R_{SL} and the liquid-gas R_{LG} thermal resistance, with R_{LG} typically 10 – 100 times greater in magnitude than R_{SL} [113]. If the evaporating vapour is lighter than the surround medium (air) convection may occur [119]. Vapour concentration gradients at the liquid-vapour interface establishes natural convection currents in the surrounding gas. These currents influence the advection and diffusion of the vapour [120, 121]. If an air draft over the droplet is present during evaporation, then advection will dominate at a given distance from the droplet surface [119]. It has been shown extensively in the literature both theoretically and experimentally that CCR evaporation is much faster than CCA evaporation [122–125]. The droplet evaporation rate in CCR is linear, while in CCA evaporation the droplet radius squared decreases linearly through time. While CCR evaporation is preferable in thermal management applications, CCA excels in the deposition of suspended particles. Particles deposited by CCR evaporation results in an inhomogeneous “coffee-ring” pattern owing to the pinned contact line, nonuniform evaporation heat flux, and capillary induced flow at the contact line due to changing curvature [110, 126, 127]. CCA evaporation enables homogeneous deposition of suspended particles due to its receding contact line during evaporation [126].

Picknett and Bexon [111] proposed a theoretical model based on the assumptions that the atmosphere just above the free surface of the droplet is saturated with vapour and that diffusion of liquid vapour away from the droplet is the rate-limiting evaporative process. Hu and Larson [128] derived a numerical solution to calculate the vapour concentration of diffusion based droplet evaporation. A peak evaporative heat flux across the liquid - gas boundary was calculated near the contact line, their result showed good agreement with previous analytical models. They found that the evaporative flux was near independent of contact angle for $\theta < 40^\circ$ and dependent for $40^\circ < \theta < 90^\circ$. This model was in agreement with

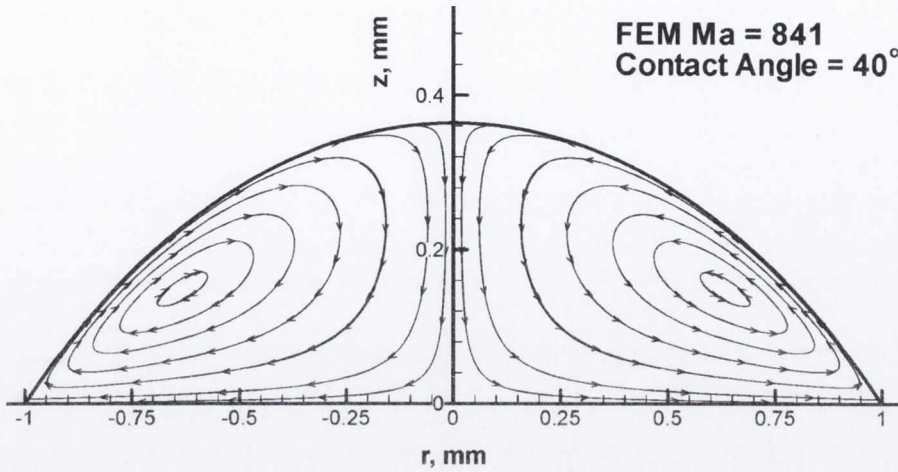


Figure 2.22: Streamline plots of the flow field from the finite element model for the Marangoni-stress boundary condition at contact angle of 40° from an FEM solution. Reprinted from Hu and Lawson [130].

their earlier experimental results [129]. In their later work Hu and Larson [130] improved their solution by incorporating thermocapillary stresses/Marangoni flow. Popov [131] developed an analytical solution for droplets of arbitrary contact angles using the analytical solution to the electric potential around a charged lens-shaped conductor [132].

Substrate thickness δ_f and thermal conductivity k_f play important roles in droplet evaporation mechanics. High values of δ_f and k_f , such as copper ($390 \text{ W m}^{-1} \text{ K}^{-1}$) and aluminium ($237 \text{ W m}^{-1} \text{ K}^{-1}$), result in the an isothermal substrate. Conversely, if δ_f and k_f are low, such as glass ($1 \text{ W m}^{-1} \text{ K}^{-1}$), and the evaporation rate is high a temperature gradient may be present across the heated substrate. If the thermal conductivity of the working fluid, k_{wf} , is low, Marangoni flow due surface tension gradients may occur. This Marangoni flow serves to reduce thermal gradients through mixing [119, 128]. Ristenpart et al. [133] showed using simple analytical arguments that the ratio between the solid and liquid thermal conductivity influences the flow direction of the Marangoni flow. This result was in keeping with that predicted by the numerical solution of Hu and Lawson [130]. Figure 2.22 shows a streamline plot of the droplet flow field due to the Marangoni-stress boundary condition from the numerical model developed by Hu and Lawson [130]. Savino et al. [134] and Nguyen and Stebe [135] reported that contaminants that collect on the free surface can almost entirely suppress Marangoni flow in an evaporating water droplet and that they are almost unavoidable when working with water. Sobac and Brutin [136] experimentally confirmed the theoretical

model by Hu and Lawson [130] and showed that for pinned natural evaporation of water that the thermal power to the droplet decreased monotonically with contact angle between $40^\circ < \theta < 90^\circ$, with the thermal power independent of contact angle for $\theta < 40^\circ$. Their receding contact line experiments (initial contact angle $\theta_0 < 40^\circ$) showed a monotonically decreasing thermal power as the droplet evaporated.

Droplet evaporation has been primarily investigated both experimentally and numerically from a diffusion based standpoint (natural evaporation) [111, 114, 125, 128–130, 136, 137]. Where the evaporation flux is controlled by the stationary diffusion of the liquid molecules. With an increase in substrate temperature the role of the linked mechanisms becomes increasingly important [138]. Sobac and Brutin [138] compared their experimental results for heated substrate droplet evaporation with the quasi-steady, diffusion-driven evaporation model employed widely in the literature [111, 128, 130, 137]. They showed that due to neglecting the buoyant convection contribution a 30% deviation between experimental and the diffusion based models results at a wall temperature of 75°C . This error was observed to increase with increasing substrate temperatures. Saada et al [139] showed similar results in their own comparison between the model and their experimental results. A 27.3% deviation in results was noted for 70°C wall temperature case. Both studies highlight the importance of buoyant convective vapour transport.

Briones et al. [140] noted that Marangoni flow decreases with increasing substrate temperature. Sobac and Brutin [138] observed that at elevated substrate temperatures that buoyancy driven recirculation flow and the ratio of thermal conductivity between the solid - liquid phases are prominent in the magnitude and direction of internal flow. From a thermal management viewpoint internal convection within the droplet has been shown to be almost inconsequential [141, 142]. Due to the no slip boundary condition and the fact that nearly all resistance to evaporative heat transfer exists at the liquid-vapour interface. Contact line motion is only important if one can augment the intrinsic meniscus region of the evaporating droplet [121, 141, 143, 144].

Vapour transport in the gas phase through diffusion and convection are crucial transport mechanisms governing droplet evaporation. For heated substrate evaporation, vapour gradients in the gas phase adjacent to the liquid - gas interface induce convective flow [120, 145–147]. This convective flow influences the vapour transport from the solid-liquid interface and

thus the overall evaporation rate [148]. Crafton and Black [149] investigated the evaporation of water and n-heptane droplets on isothermal aluminium and coppers substrates for a range of substrate temperatures between 80 – 95°C. The initial contact angle for water droplets ranged between 102° – 113°. Water droplets remained primarily pinned with unpinning occurring near the end of the evaporation period. While the n-heptane droplets evaporated with a receding triple line for a near constant contact angle. They noted an increasing evaporation rate with increasing wetted area and an increasing average heat flux across the solid - liquid interface for decreasing wetted area. Mollaret et al. [150] compared the evaporation of pinned water droplets on aluminum and PTFE substrates. They noted a steady linear decrease in both droplet volume and contact angle, with a longer evaporation time noted on the PTFE substrate. They also observed a substrate temperature dependency on contact line depinning. Gleason and Putnam [151] experimentally and numerically investigate water droplet evaporation for a forced pinned contact line on a heated substrate. They noted a linear decrease in both droplet volume and contact angle over the evaporation period. Numerical comparison of widely used evaporation model from the literature [127, 131, 137] showed poor correlation with experimental data. The model error is attributed to the implementation of a constant vapour concentration across the liquid - gas interface. Karchevsky et al. [117] investigated the thin foil conjugate heat transfer problem with evaporating sessile water droplets. Their experimental study highlighted the peak local conjugate heat transfer occurred at the evaporating droplet triple line.

Deng and Gomez [70] showed that during evaporation the image force between the charged droplet and grounded surface can reduce the droplet contact angle. A comparison with an uncharged droplet is shown in Figure 2.23. The electrostatic force in a charged

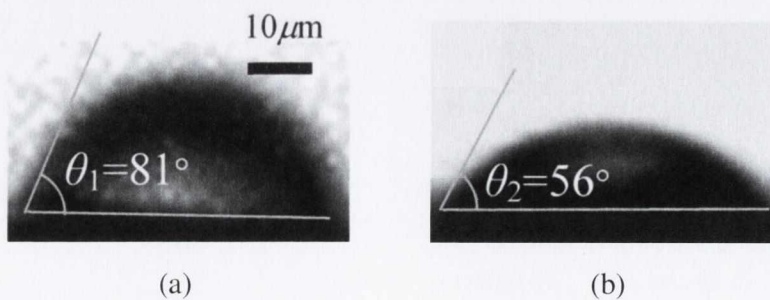


Figure 2.23: Droplet charge impact on contact angle. (a) Uncharged, (b) Charged. Reprinted from Deng and Gomez [70].

droplet opposes that of the surface tension force. This electrostatic force acts to pull the liquid towards the grounded target resulting in a reduced contact angle. They attempted to quantify the image force by assuming that the droplet was charged to 50 – 80% of the Rayleigh limit, that half of this charge was lost on impact, the droplet shape was that of spherical cap, and the charge was frozen during the recoil phase, which was short. They estimated that the image force is thousand fold greater than that due to droplet mass alone. Due to the reduced contact angle, induced by the image force, Deng and Gomez noticed a decrease in the evaporation time of the charge droplet.

2.3 Spray Cooling Heat Transfer

Several different heat and mass transfer mechanisms for spray cooling at the solid - liquid interface have been proposed in the literature: thin film evaporation, secondary nucleation and contact line heat transfer.

2.3.1 Thin Film Evaporation

In this system a continuous thin film forms on the heated substrate, where the top of the film is at the saturated temperature. This enables the liquid molecules on the top of the film to efficiently escape into the gas phase. Pais et al. [152] and Tilton [66] calculated that a 1.4 μm thick film of water is required to remove 1000 W cm^{-2} of thermal energy at a superheat of 20°C .

2.3.2 Secondary Nucleation

Yang et al. [65] achieved a heat flux removal of 820 W cm^{-2} for an air-atomised nozzle with water operating at a flow rate of 2 L min^{-1} . They proposed that the high heat transfer was due to the increased generation of nucleation sites within the film as a result of the impacting droplets on the liquid film. They attributed this to two factors. Impacting droplets possess a layer of absorbed gas on their surface that is released when the droplet has penetrated the film. The droplet can also puncture growing bubbles, increasing the bubble growth frequency. Rini et al. [153] visualised the solid - liquid interface using a high speed camera and a $500 \mu\text{m}$

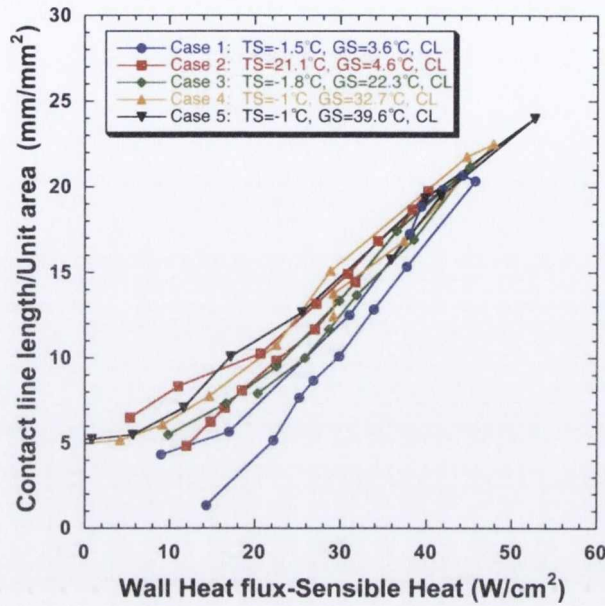


Figure 2.24: Average contact line length density as a function of the wall heat flux corrected for sensible heating. Reprinted from Horacek et al. [73].

semi transparent synthetic diamond substrate. They compared the heat transfer of their spray with that of pool boiling and noted ~ 3.9 fold increase ($900 - 3500 \text{ cm}^{-2}$) in nucleation sites for the spray case. Similar to Yang et al. [65] they noted that an increase in droplet number flux increased nucleation sites density. They observed an increase in heat flux with droplet number flux, they attributed this to the induced mixing as a result of droplet impact on the thin film.

2.3.3 Contact line heat transfer

Horacek et al. [73] measured the spatial and temporal resolved heat transfer distributions of an ISR spray nozzle using an array of individually controlled microheaters and FC-72 as working fluid. They visualised the solid - liquid interface using a Total Internal Reflectance (TIR) technique. This enabled measurement of the three phase contact line length density (CLL). Horacek et al. found no dependency between wetted area and heat transfer but observed a strong correlation between CLL and heat transfer with the maximum CLL occurring at the critical heat flux (CHF). This result is shown in Figure 2.24.

Sodtke and Stephan [72] investigated spray cooling on a smooth heater surface with water.

2.3. SPRAY COOLING HEAT TRANSFER

Using a thermal imaging camera to focus on the top of the heated substrate they were able to characterise the CLL during spraying. Similar to Horacek et al. [73] they observed an increase in CLL with heat flux (see Figure 2.25). As a result they designed micro-structured surfaces to exploit this dependency on CLL. The surface structure increased CLL by protruding

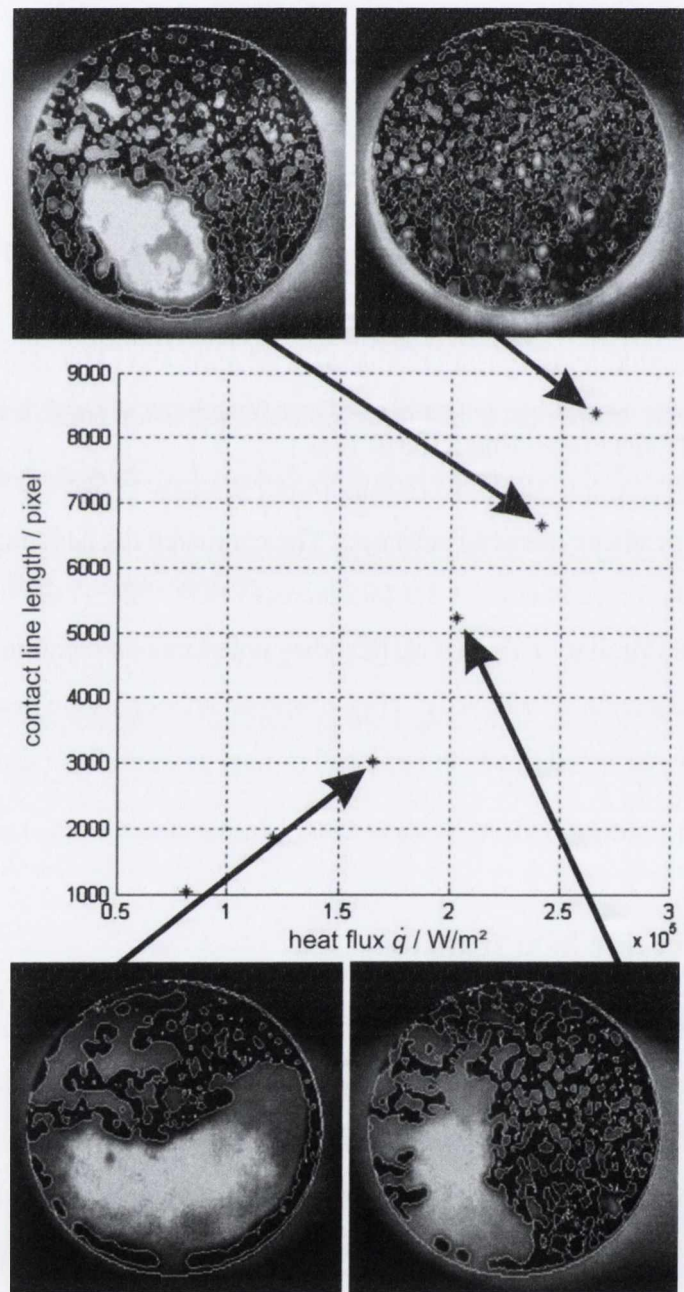


Figure 2.25: Contact line length density versus heat flux with corresponding IR-images on a smooth copper surface. Dark spots are the copper substrate and water elsewhere. Reprinted from Sodtke and Stephan [72].

through the deposited film. Thus acting to create a multitude pinned contact lines on the heater surface. Significant heat transfer enhancement over the smooth substrate (10%) was observed for the structure surface. The authors attributed this to the increased CLL.

Contact line heat transfer has clear application and impact on spray cooling heat transfer [72, 73]. The impact and subsequent spreading of droplets on the surface results in formation of multiple contact lines. Strong correlation between the heat transfer intensity and the contact line length per unit area has been observed [72, 73]. Studies of evaporating droplets on heated substrate are essential for further understanding this result [154], yet the field is still largely underdeveloped due to the challenges in modelling the contact line phenomena due to the complex coupling of several physical mechanisms such as evaporation, viscous flow, surface tension, and heat conduction in a highly localized region. These challenges lead to difficulties in modelling and experimental design. Thin foil thermography is a promising route in analysing the heat and mass transfer at the contact line region of an evaporating liquid. Further understanding of this mechanism will enable development of smarter thermal management solutions not only for spray cooling but for a host of thermal management applications.

2.4 Electrode Configuration

As previous noted electrospray electrode design can be divided into two configurations: two electrodes (source - target), and three electrodes (source - extractor - target), these are shown in Figure 1.1. Both designs achieve cone-jet electrospraying in the manner previously described. The first design is quite forgiving in geometrical terms. Once a sufficient electric field is established between the source and the target electrode spraying will occur. The second configuration, Figure 1.1b, incorporates an additional intermediate electrode, the extractor electrode, between the source and target electrode. The purpose of this electrode is to localise the electric field at the source electrode and shield the cone-jet from the highly charged generated spray cloud [5]. The extractor design enables a more stable spray process and a degree of plume dispersion control [102]. Two electric fields exist in this design, the jet forming electric field E_j between the source and extractor and the driving electric field E_d between the extractor and target electrodes. Once a suitable potential drop is established

between the source and extractor it is possible to vary the driving electric field to alter plume dynamics. Yang et al. [102] showed that a higher E_d resulted in a more concentrated plume. This is due to the increased droplet velocity in the spray direction arising from the increasing electric field. This results in a shorter droplet residence time allowing less time for plume dispersion. Deng and Gomez [97] defined a minimum E_d that is required to prevent “satellite trapping”. This is flow reversal of satellite droplets back to the extractor electrode. This can result in flooding between the source and extractor electrode which is detrimental to spraying. They showed that varying E_d has no impact on the produced droplet size, as is shown in Figure 2.26. The greatest advantage of the three electrode configuration is in the realisation of high density electrospray arrays operating the cone-jet regime of spraying.

2.5 Electrospray Array

Electrospray Ionization Mass Spectrometry [17] is the only widely applied practical application of electrospray technology. One of the primary impediments is the inherent low flow rate ($0.6 \mu\text{L min}^{-1} - 600 \mu\text{L min}^{-1}$) of cone-jet electrospraying [3, 60]. A solution to this is multiple source electrospraying. As noted previously, the droplet size and flow rate have a

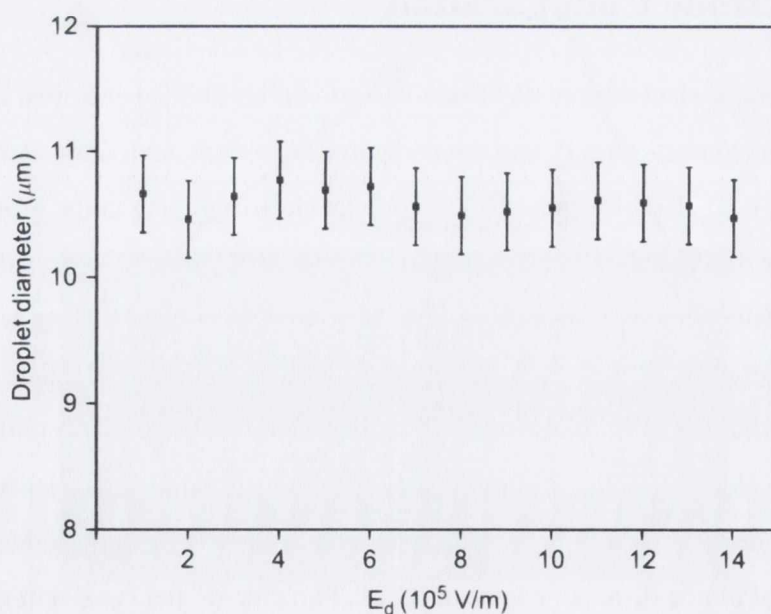


Figure 2.26: Driving electric field strength impact on primary droplet size. Reprinted from Deng and Gomez [97].

monotonic relationship. In the case of heat transfer, a 3 source array in a triangular pattern spraying at $1 \mu\text{L min}^{-1}$ will produce a thinner film, due to smaller produced droplets, than a single source operating at $3 \mu\text{L min}^{-1}$. This thinner film enables higher heat fluxes and thus greater cooling potential. Deng and Gomez [6] showed increased average heat fluxes through doping of their working fluid to achieve higher electrical conductivities and thus smaller droplet sizes.

Numerous different design route have been proposed over the past 25 years for electro-spray array design. They can be classified into four primary categories: capillary integration, where multiple stainless steel or fused silica capillaries are integrated into a single device [19, 28, 45, 45, 47, 155, 156]; silicon etching [5, 7, 157]; soft lithography [158]; and, conventional machining [8, 9, 159, 160]. Each method has its own advantages and limitations. For example silicon etching has the greatest fabrication accuracy with feature size accuracy on the submicron scale [60]. However it is limited to the size of a silicon wafer (0.775 mm thickness and 100 mm diameter) and are extremely fragile due to their fabrication process (deep reactive ion etching) [8, 60].

It is clear from the literature that in order to successfully implement a high density 3D array with uniform source to source performance the three electrode configuration of source - extractor - target is required [5]. If the intermediate extractor electrode is not implemented electrical interference between adjacent nozzles and the space charge from the generated plume will prevent cone-jet formation at the centre of the array [5]. Arrays operated without an extractor are said to be by “brute force” and struggle to perform for ring values greater than 1, thus limiting their coverage area greatly. As such only relevant non-heat transfer extractor designs are reviewed; a list of all reviewed array research is shown in Table 2.3.

Bocanegra et al. [9] demonstrated that an electrospray array could be achieved from a 37 source array of holes in a hexagonal arrangement, with a compactness of $115 N_n \text{ cm}^{-2}$, where N_n is the number of sources in an array. They reported improved cone-jet stability when the source holes are drilled in a dielectric material with a hydrophobic coating. Their results demonstrated good agreement of spray current, primary droplet size and fraction of satellite droplets to their capillary source counterparts. Figure 2.27 shows their array in operation. They also defined the compactness of a hexagonal patterned electrospray array as:

2.5. ELECTROSPRAY ARRAY

Table 2.3: Electro spray literature array design summary.

Author	Array design	Source density ρ_{N_n}	No. of sources N_n	Fabrication method
Rulison and Flagan [28]	Two electrode, linear	N/A	8	Capillary integration
Almekinders et al. [159]	Two electrode, linear, $D_i = 0.5$ mm, $D_o = 1.5$ mm,	$5 N_n \text{ cm}^{-1}$	8	Conventional machining
Kim and Knapp [158]	Two electrode, linear, 30 x 50 (width $\times$ depth), $L_n = 1$ mm	$9 N_n \text{ cm}^{-1}$	16	PDMS soft lithograph
Regel et al. [155]	Two electrode, triangular, $D_i = 0.8$ mm, $D_o = 1.27$ mm, $L_n = 10$ cm	$0.2 - 71.59 N_n \text{ cm}^{-2}$	4	Capillary integration
Kyritis et al. [156]	Two electrode, hexagonal, $D_i = 127 \mu\text{m}$, $D_o = 1.59$ mm, $L_n = 10$ cm	$4 N_n \text{ cm}^{-2}$	19	Capillary integration
Deng et al. [5]	Three electrode, hexagonal, $D_i = 120 \mu\text{m}$, $D_o = 210 \mu\text{m}$, $L_n = 450 \mu\text{m}$	$250 N_n \text{ cm}^{-2}$	37	Silicon etching
Kim et al. [157]	Two electrode, linear, 2 x 8 μm (width $\times$ depth)	$1000 N_n \text{ cm}^{-1}$	10	Silicon etching
Si et al. [19]	Two electrode, linear, $D_i = 0.5$ mm, $D_o = 1.5$ mm	$0.47 - 6.67 N_n \text{ cm}^{-1}$	3	Capillary integration
Bocanegra et al. [9]	Three electrode, hexagonal, $D_i = 500 \mu\text{m}$	$29 - 115 N_n \text{ cm}^{-2}$	37	Conventional machining
Deng et al. [7]	Three electrode, hexagonal, $D_i = 15 \mu\text{m}$, $D_o = 30 \mu\text{m}$, $L_n = 80 \mu\text{m}$	$10,000 N_n \text{ cm}^{-2}$	37	Silicon etching
Lojewski et al. [8]	Three electrode, linear, $D_i = 50 \mu\text{m}$, $D_o = 120 \mu\text{m}$, $L_n = 300 \mu\text{m}$	$20 N_n \text{ cm}^{-1}$	17	Conventional machining

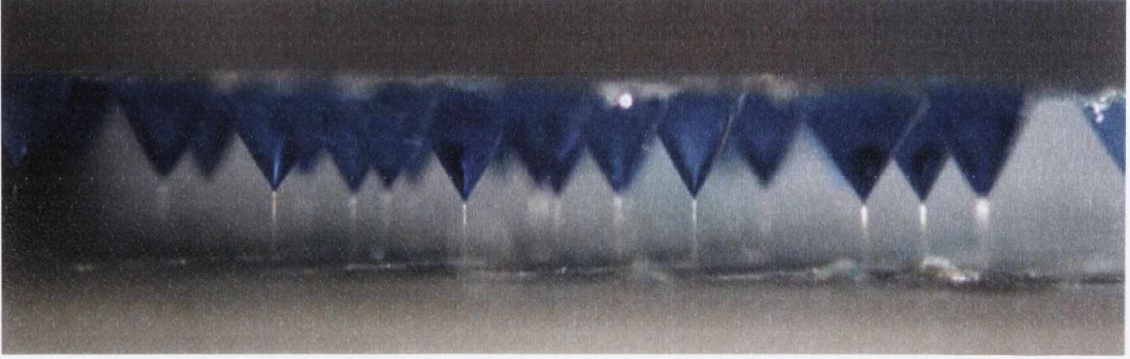


Figure 2.27: 37 source electrospay array of holes, $s = 2 \text{ mm}$, $Q = 2.25 \text{ } \mu\text{L min}^{-1}$, with colored ethylene glycol as the working fluid. Reprinted from Bocanegra et al. [9].

$$\rho_{N_n} = \frac{1}{s^2 \sin(\pi/3)} \quad (2.15)$$

where s is the nozzle spacing.

Deng et al. [5] developed several 37 source extractor electrospay arrays. The source nozzles were manufactured in silicon wafers using deep reactive ion etching (DRIE) and were arranged in a staggered manner. 3 nozzle spacings were investigated: 2.52, 1.26 and 0.675 mm. The nozzle length L_n and outer diameter D_o varied from 150 – 450 μm and 180 – 240 μm respectively, with a constant nozzle inner diameter D_i of 120 μm . They defined a number of length scale equalities in order to better define the electrospay array design process. They noted that as long as the source – extractor separation distance H_1 was of the same order as the nozzle length and both were much smaller than the source spacing, such that $H_1 < L_n < s$, ensures that the local electric field is uninfluenced from adjacent source nozzles.

Deng and Gomez [97] later investigated if the space charge from the generated plume would penetrate the orifice of the extractor electrode. If this were to happen the space charge would weaken the field near the cone and eventually destabilize spraying. They investigated the electric field strength variation near the source nozzle tip with and without space charge. They placed a liquid drop at the source nozzle tip, that had the same potential as the source nozzle. The driving field intensity was set to that of the electrical breakdown of air ($3 \times 10^6 \text{ V/m}$) for the first case and no applied electric field in the second. The local electric field at the top of the droplet was then compared for both cases. Deng and Gomez investigated this

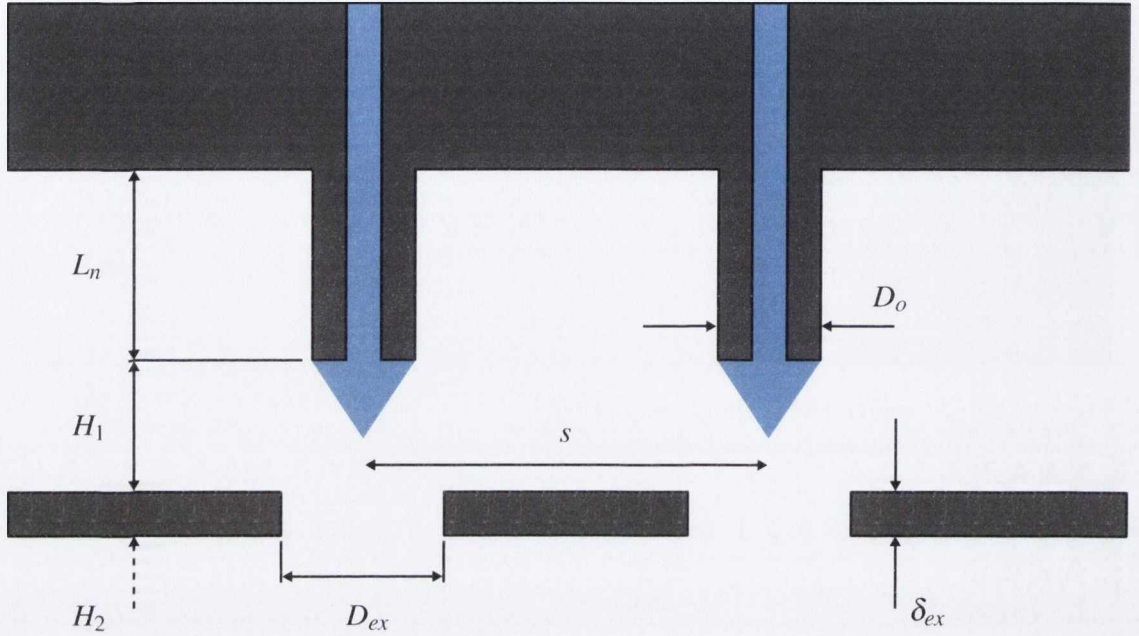


Figure 2.28: Electro spray array length scales.

for varied extractor orifice diameters, D_{ex} , and source – extractor separation heights. The imposed space-charge was shown to impact the local electric field less than 1.4% for $D_{ex}/H_1 = 1.14$ and 0.1% when $D_{ex}/H_1 = 0.57$. They suggested that $D_{ex}/H_1 < 1$ to minimise the space charge feedback. Figure 2.28 shows an illustration of the key length variable in 3D electro spray array design. From their results Deng and Gomez [97] derived an expression to determine the required driving field for a finite number of sources to prevent satellite trapping:

$$E_{d,array} > E_{z0} + 0.827 \left(\frac{H_2 I_{spray}}{u_z \epsilon_0 A_n} \right) [n/a - (n+1)n/2a^2] \quad (2.16)$$

where E_{z0} is the minimum required driving electric field to remove satellite trapping for a single source, A_n is the area covered by an individual spray plume, $A_n = \sqrt{3}/2s^2$, assuming hexagonal source pattern, $a = H_2/s$, where H_2 is the extractor – target separation height and n is the number of rings of nozzles around the central source. Deng et al. [5] computed and compared the electric field of selected numbered nozzles in a 19 source 3 electrode array, their results are shown in Figure 2.29. Their model was computed in FEMLAB with the following parameters: $D_i = 120 \mu\text{m}$, $D_o = 210 \mu\text{m}$, $H_1 = 400 \mu\text{m}$, $L_n = 450 \mu\text{m}$, $s = 0.675 \text{ mm}$ and a 1 kV potential drop between the source and extractor electrodes. Their computed

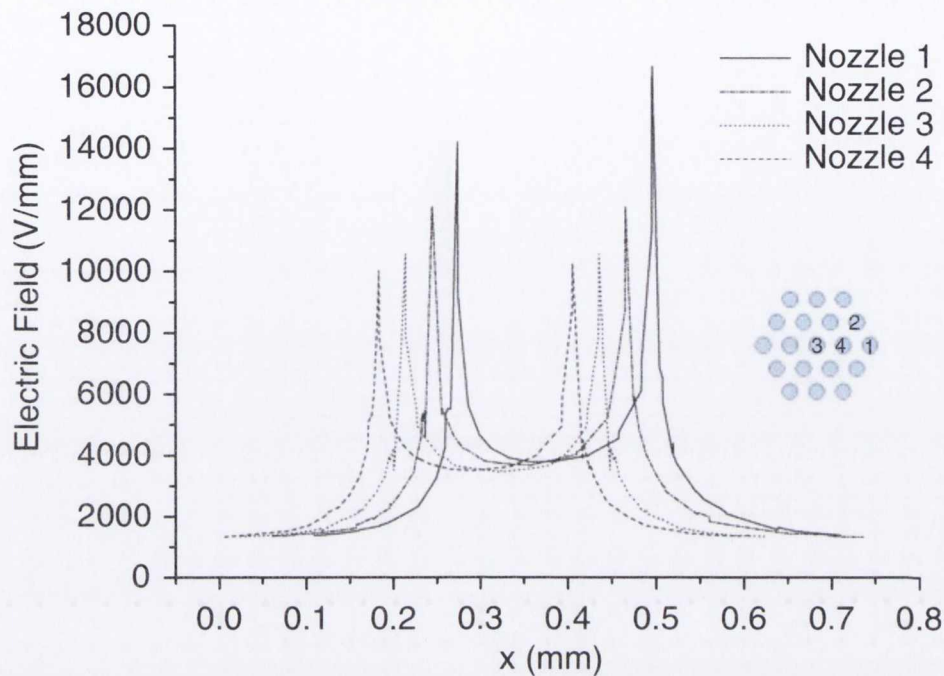


Figure 2.29: Details of the electric field on a array for individual numbered sourced nozzles. computed on each of the nozzles in a 19-nozzle hexagonal pattern. Ethanol working fluid, $D_i = 120\ \mu\text{m}$, $D_o = 210\ \mu\text{m}$, $H_1 = 300\ \mu\text{m}$, $L_n = 450\ \mu\text{m}$, $s = 0.675\ \text{mm}$ and source - extractor voltage drop = 1 kV. Reprinted from Deng et al. [5].

results demonstrate a higher electric field at the edge electrode in comparison to the centre of the array.

2.5.1 Edge Effects

A successful array design should ensure that all sources behave in a consistent manner. Nozzles at the edge of the array experience a more intense electric field which results in a greater electrostatic “pull” on the working fluid, which is on the order of the capillary force, γ/D_o [8, 13]. This causes a greater liquid flow rate for sources on the periphery of the array [5]. This increased flow rate produces larger droplets relative to central sources. This result is shown in Figure 2.30. Deng et al. suggested two different methodologies to circumnavigate this problem. The first was to implement an outside ring of “dummy” nozzles. Similar to that suggested by Rulison and Flagan [28] in their linear array work. These outside nozzles were plugged, allowing no fluid flow but do impact the electric field reducing edge effects on operational nozzles. Through this approach Deng et al. achieved a relative standard

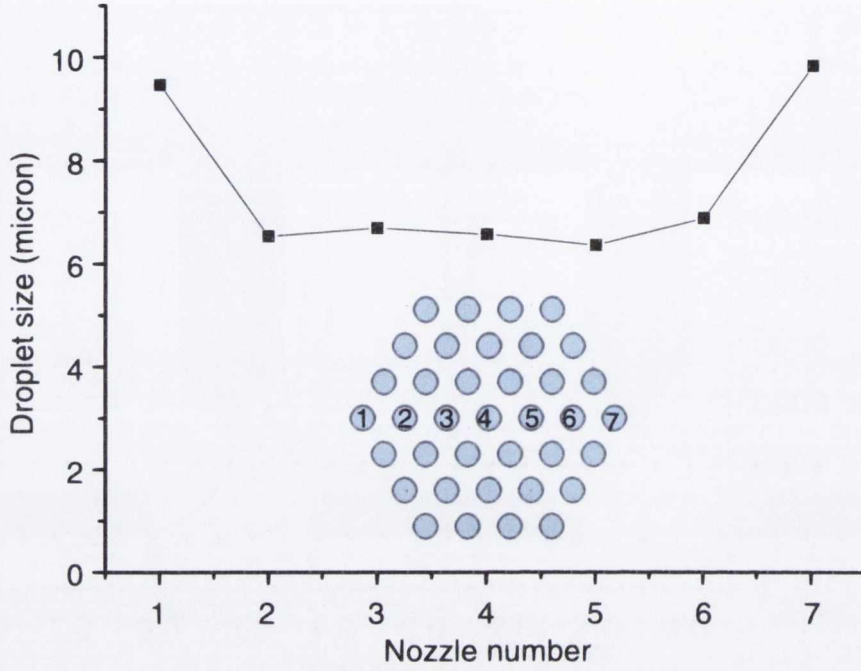


Figure 2.30: Electro spray array edge effect on droplet size distribution. Ethanol working fluid and $Q_{total} = 366.67 \mu\text{L min}^{-1}$, $D_i = 120 \mu\text{m}$, $D_o = 210 \mu\text{m}$, $H_1 = 300 \mu\text{m}$, $L_n = 450 \mu\text{m}$, $s = 0.675 \text{ mm}$ and source - extractor voltage drop = 1 kV. Reprinted from Deng et al. [5].

deviation (RSD) of 0.053 across all nozzles. The second solution was to ensure that the sum of the pressure drop across the reservoir and source nozzles is much greater than that of the electrostatic pulling force. They latter achieved this for a 91 source array by decreasing the inner nozzle diameter from $120 \mu\text{m}$ to $60 \mu\text{m}$ resulting in a 16 fold increase in pressure drop across the nozzles. This design achieved a RSD of 0.09 [60].

2.5.2 Spray Current

Bocanegra et al. [9] showed that if the flow rate is constant across all sources, the spray current was constant across all source nozzles, such that:

$$I_{total} = N_n I_{spray} \quad (2.17)$$

where I_{total} is the total spray current of the array. They also showed that:

$$I_{total} \propto N_n^{.5} Q^{.5} \quad (2.18)$$

This is in agreement with the spray current scaling law defined by Gañán-Calvo et al. [89] and is in agreement with the single source experimental results observed by Tang and Gomez [100].

2.6 Heat Transfer

Prior research in the field of electrospray cooling is very limited. To this authors knowledge on three authors have published work pertaining to electrospray enhanced cooling. Their research, especially that of Deng and Gomez [6, 70], has shown that EC is a viable option as a thermal management solution for small form factor electronics.

Feng and Bryan [45] were the first to investigate two phase EC. They studies the average heat transfer of different nozzle arrays in an enclosed chamber under various spraying regimes with Ethanol. The compared their results with that of no electric field case ($V_n = 0$ kV) and defined a heat transfer enhancement due to the applied electric field. They investigated a range of spraying regimes, operating primarily outside the cone-jet mode. This methodology resulted in the creation of largely polydisperse droplets. Feng and Bryan operated at relatively

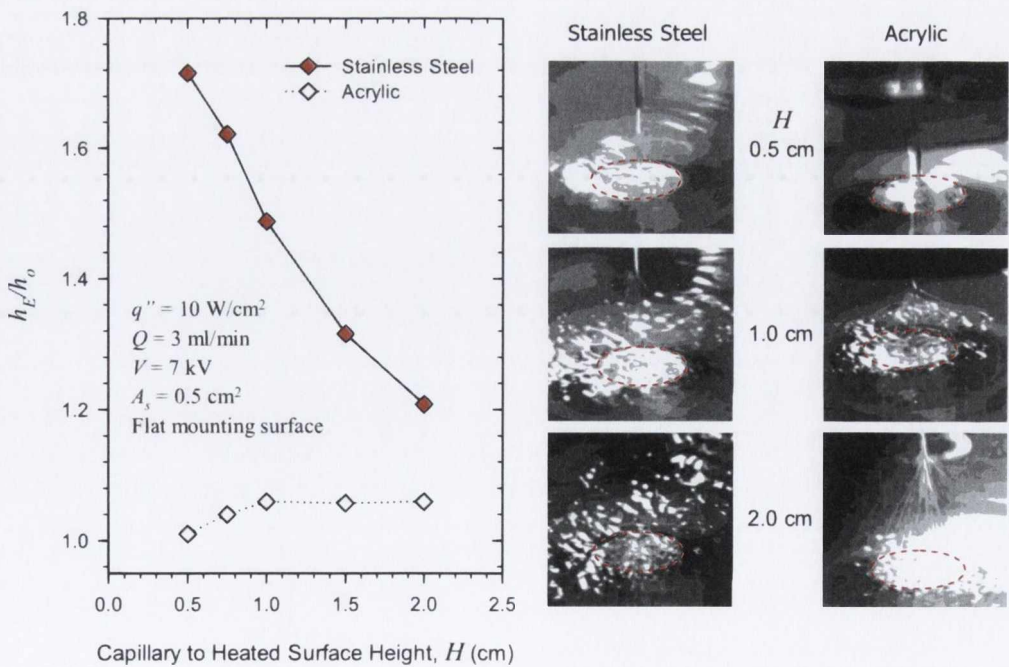


Figure 2.31: Heat transfer enhancement over no applied electric field for varied separation height. Reprinted from Feng and Bryan [45].

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high flow rates compared with the other EC studies [6, 161], $Q = 750 - 8600 \mu\text{L min}^{-1} N_n^{-1}$. They achieved a peak average heat transfer enhancement, ER_{field} , of 1.7 times greater than the no electric field case. This condition corresponded to a single nozzle, $V_n = 7 \text{ kV}$, $D_i = 0.25 \text{ mm}$, $H = 5 \text{ mm}$, and $Q = 3000 \mu\text{L min}^{-1}$. This fluid was deposited onto 50 mm^2 surface with a wall heat flux of 10 W cm^{-2} . A fixed liquid layer appears to have been maintained on the heated substrate at this point. The spraying regime was unspecified. This result is shown in Figure 2.31. Feng and Bryan observed that as H is increased the enhancement over natural convection decreases. It is important to note that for a fixed applied voltage, as in this data set, as H is increased the applied electric field decreases. This would suggest that several functioning modes were observed across these results. No specification of what spraying regime corresponds to a specific experimental point. Greater heat transfer enhancement over natural convection was noted for sloped target surface in comparison to a flat one at lower wall heat fluxes ($10 - 20 \text{ W cm}^{-2}$). The is shown in Figure 2.32a. As the heat flux was increased ($30 - 50 \text{ W cm}^{-2}$) the difference in sloped and flat became negligible. Finally as the wall heat flux was increased further ($60 - 70 \text{ W cm}^{-2}$) the flat mounted surface demonstrates

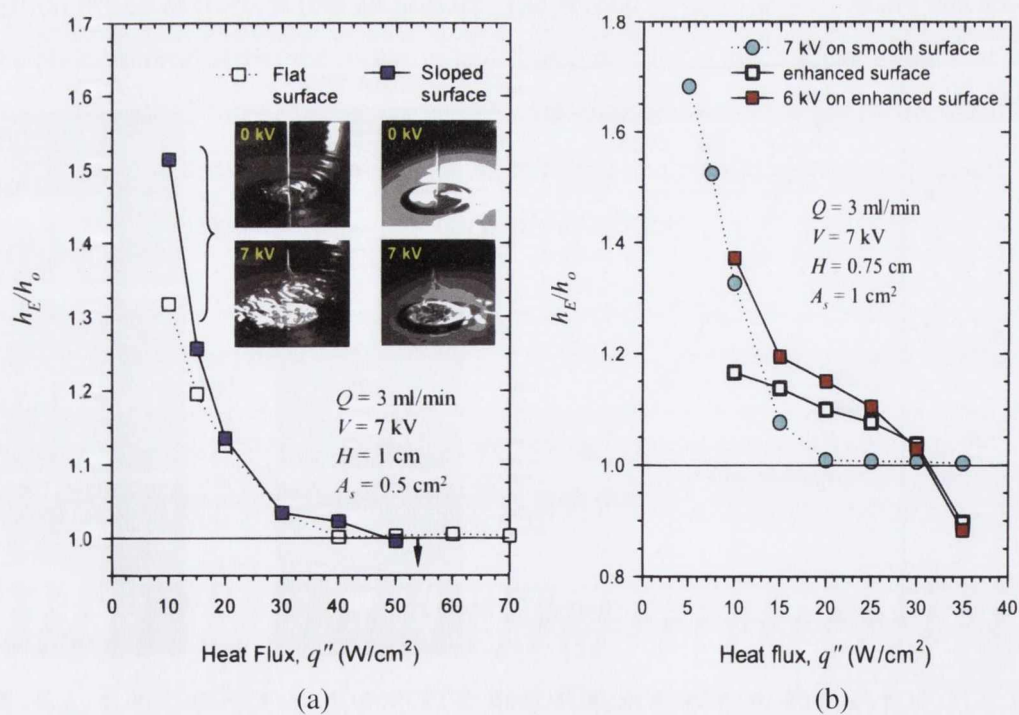


Figure 2.32: Heat transfer enhancement over no applied electric field for varied wall heat flux and substrate condition. Reprinted from Feng and Bryan [45].

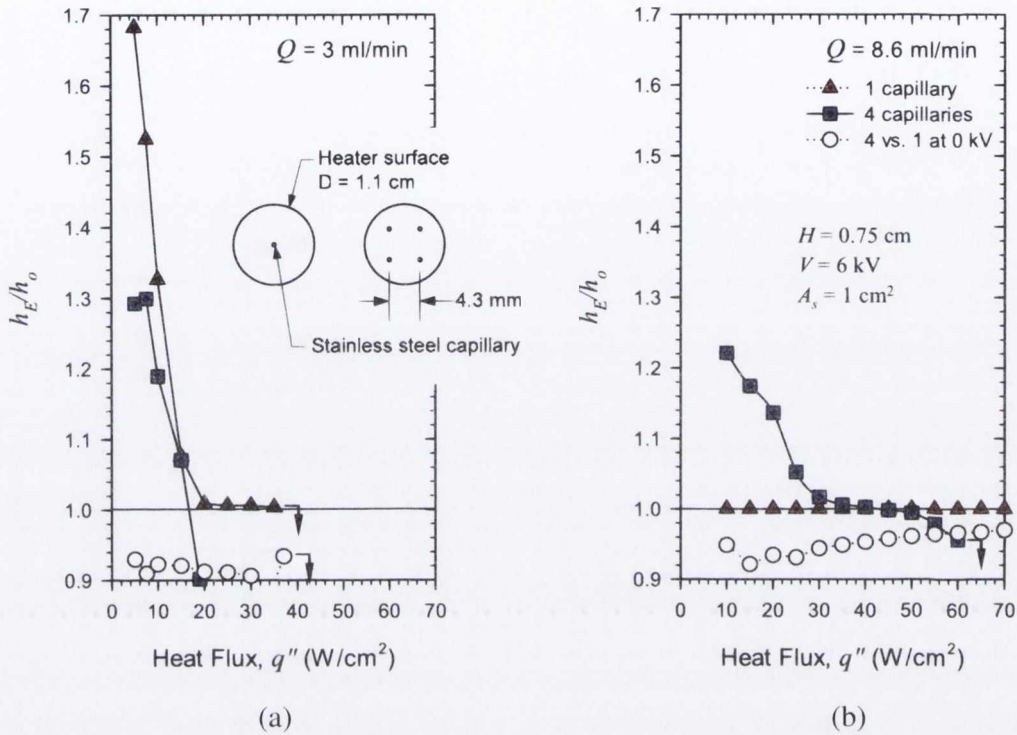


Figure 2.33: Heat transfer enhancement over natural convection cooling for varied wall heat flux, working fluid flow rate and number of spray sources. Reprinted from Feng and Bryan [45].

an enhancement of ≈ 1 while the sloped < 0.95 . Figure 2.32b demonstrate the impact of target surface enhancement on enhancement ratio. The enhanced surface achieved a higher enhancement ratio for all test point except the highest wall heat flux.

Feng and Bryan also investigated the impact of multiple sources, shown in Figure 2.33. However as they did not implement an extractor in their design, scalability to a larger number of sources was not feasible [97]. They generally observed a greater enhancement ratio from the single source across all wall heat fluxes for the 3,000 $\mu\text{L min}^{-1}$ case (see Figure 2.33a). Improved performance of the array over the single source is noted as the flow rate is increase to 8,600 $\mu\text{L min}^{-1}$ at lower wall heat fluxes (10 – 30 W cm^{-2}), Figure 2.33b. Both configuration perform with an enhancement ratio of ≈ 1 in the wall heat flux range of 30 – 55 W cm^{-2} . As the wall heat flux is increased further, the array performance decreases below an enhancement ratio of 1. As no consistent spraying regime is employed it is difficult to quantify the experimental results presented by Feng and Bryan. From the optical results of the heater surface presented and the relatively high flow rates quoted it appears that the electrospray was largely used as a fluid delivery device to a fixed pool.

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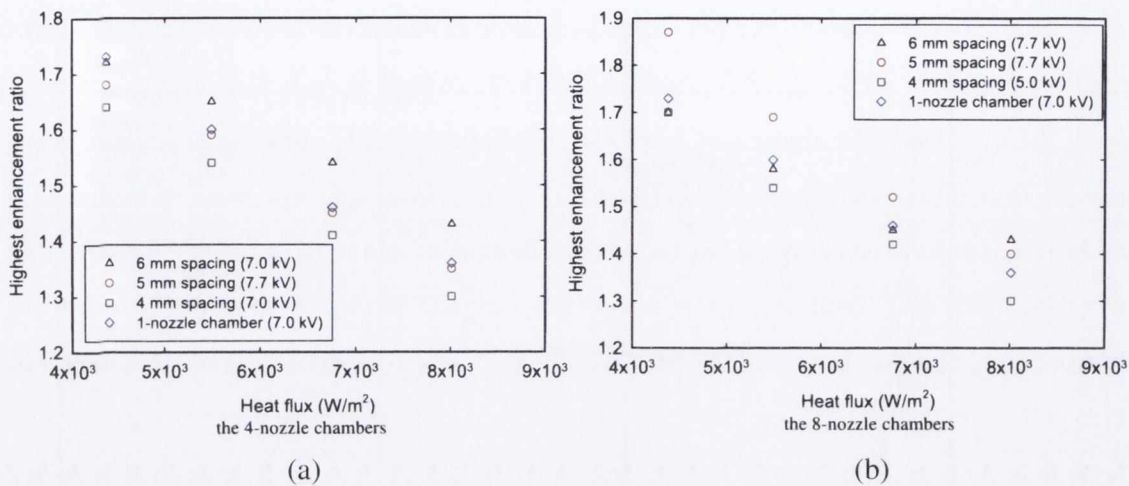


Figure 234: Heat transfer enhancement over natural convection cooling for increasing wall heat flux. (a) 4 nozzle array (b) 8 nozzle array. Reprinted from Wang and Mamishev [161].

Wang and Mamishev [46] investigated the average heat transfer of enclosed electrospraying chambers for three different geometry types in the multi-jet regime. They achieved a maximum average heat transfer enhancement of 1.87 fold over natural convection for their 8 source array at the lowest wall heat flux of $4,384 \text{ W m}^{-2}$ operating in the multi-jet regime. This case corresponded to $s = 5 \text{ mm}$, $H = 7.5 \text{ mm}$, $D_i = 0.210 \text{ mm}$, and $Q = 134 \text{ } \mu\text{L min}^{-1}$ and is shown in Figure 2.34. They noted an increasing enhancement for decreasing wall heat fluxes for all source designs. Like Feng and Bryan, they operated their array design without an extractor and as such no scale up of their device is feasible. They stipulated that

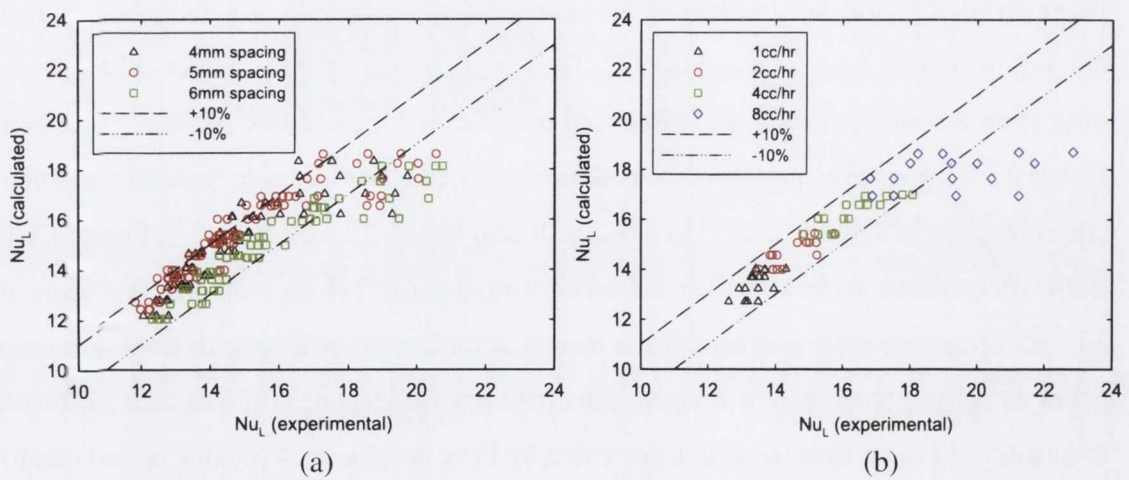


Figure 235: Nusselt number correlation comparison between experimental and predicted. (a) 4 nozzle array (b) 8 nozzle array, 5 mm spacing. Reprinted from Wang and Mamishev [47].

increasing the number of sources and their spacing does not improve the steady state heat transfer. However it does increase the transient average heat transfer performance.

Wang and Mamishev [47] later developed a set of correlations for the average heat transfer enhancement ratio and Nusselt number for an electrospray operated in the multi-jet regime. They based their model on impinging liquid jet correlations, modifying them to account for the electric field of the electrospray. They suggested that the enhancement ratio and Nusselt number were dependent on the Weber number, electric Weber number, Reynold number, the number of electrospray sources and a characteristic length normalised by the source nozzles inner diameter. Their model for the enhancement ratio and Nusselt number covered 83% of their experimental data within a $\pm 10\%$ deviation. This correlation agreement is shown in Figure 2.35. From their results it can be shown that their correlation perform well at lower flow rates with erroneous results appearing at higher flow rates. Wang and Mamishev stipulated that this was due to excessive coolant deposition on the target surface.

Deng and Gomez [6] constructed and tested 19 sourced multiplexed electrospray arrays with a source density of $253 N_n \text{ cm}^{-2}$. They operated in the cone-jet regime utilising an extractor design. They found that the micron sized quasi-monodisperse droplets resulted in the onset of two phase cooling at $\sim 20 \text{ K}$ below the boiling point of the working fluid, while typically this occurs at $3 - 5 \text{ K}$ above the boiling point. This is shown in Figure 2.36a. Figure 2.36b highlights their electrospray cooling curve for varied flow rates. A peak critical

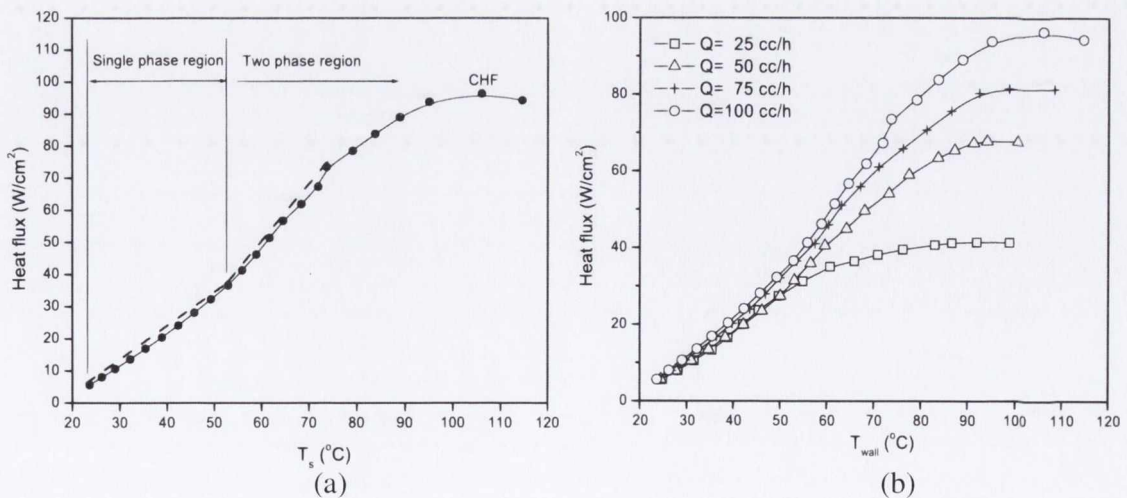


Figure 2.36: Electro spray cooling wall heat flux and temperature. (a) $Q = 88 \mu\text{L min}^{-1} N_n^{-1}$ (b) varied flow rate. Reprinted from Deng and Gomez [6].

2.6. HEAT TRANSFER

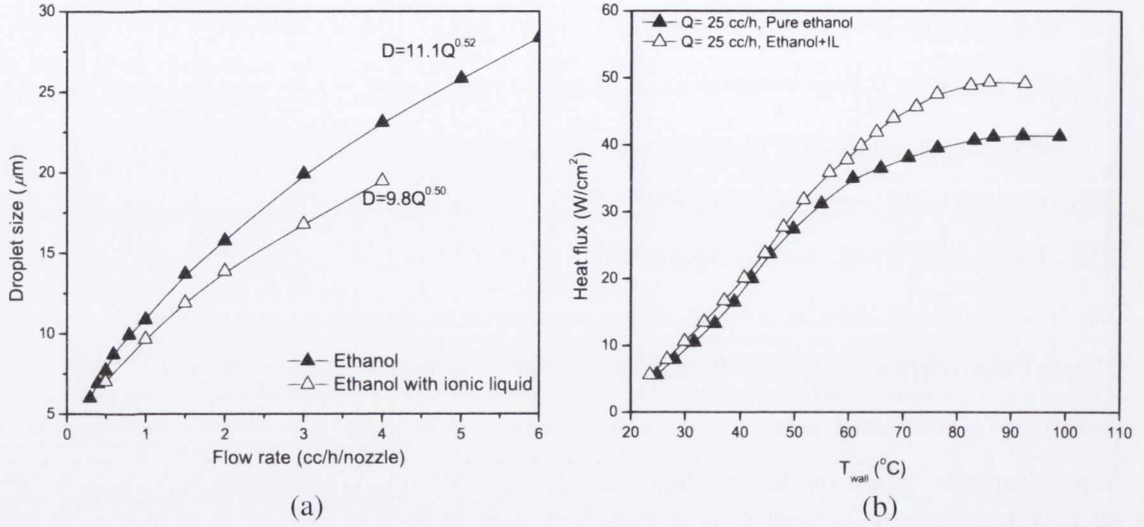


Figure 2.37: Influence of working fluid electrical conductivity. (a) droplet size for increasing flow rate (b) electrospray cooling average heat flux for increase wall heat flux. Reprinted from Deng and Gomez [6].

heat flux is observed at higher flow rates. These results correspond to a separation height of 5 mm. Deng and Gomez showed that by decreasing the droplet size, while maintaining all other parameters, an increase in the critical heat flux by $\approx 20 - 25\%$ is observed (see Figure 2.37b). They determined that this was a result of the shorter evaporation time of smaller droplets produced from the doped working fluid (see Figure 2.37a). Using the spray cooling efficiency, η , developed by Kim [64], they compared the removed heat flux by electrospray cooling compared to the maximum potential cooling capacity. This is given by:

$$\eta = \frac{q''_{\text{conv}}}{\dot{m}_l C_{p,l}(T_b - T_{\text{spray}}) + \dot{m}_l h_{\text{evap}} + \dot{m}_{\text{air}} C_{p,\text{air}}(T_s - T_{\text{spray}}) + \dot{m}_{\text{air}} C_{p,v}(T_s - T_b)} \quad (2.19)$$

where q''_{conv} is the heat flux removed by electrospray cooling, $\dot{m}_l$ is the liquid mass flow rate, $C_{p,l}$ is the specific heat of the coolant, T_b is the working fluid boiling point, T_{spray} is the working fluid and co-flow gas temperature. h_{evap} is the latent heat of the working fluid, $\dot{m}_{\text{air}}$ is the mass flow rate co-flow gas, $C_{p,g}$ is the specific heat of the surrounding medium at constant pressure, and $C_{p,v}$ is specific heat of the working fluid vapour at constant pressure. The co-flow gas mass flow was supplied by the array designed by Deng and Gomez. An optimal average heat flux removal of 96 W cm^{-2} and a cooling efficiency of 60% was noted for the

highest flow rate ($88 \mu\text{L min}^{-1} N_n^{-1}$), while the lowest flow rate ($22 \mu\text{L min}^{-1} N_n^{-1}$) produced a cooling efficiency of 82% for a heat flux removal of 40 W cm^{-2} . This drop in efficiency was attributed to the increasing droplet size with increasing flow rate. By increasing the electrical conductivity of the working fluid they achieved a peak spray efficiency of 97% for an average heat flux removal of 50 W cm^{-2} for the lowest flow rate. This result is shown in Figure 2.37b and corresponds to an increase in average heat flux removal of 25% compared to the undoped ethanol case. From these results Deng and Gomez [6] suggested that producing the smallest droplets at the highest rate would optimise both spray efficiency and average heat flux removal. This can be achieved by optimising the packing of source nozzles to achieve small droplets for larger total flow rates.

2.7 Summary

An extensive review of the literature surrounding electrospray cooling and heat and mass transfer to an evaporating fluid has been provided. It is clear that work on the average heat transfer to an impinging electrospray and evaporating droplets exists. However, to the best knowledge of the author no previous work has been done on the local heat transfer to an electrospray and there is a demand for further investigation into the heat and mass transfer mechanisms of the deposited working fluid, especially that at the triple line. The gaps in literature identified here help define the research objectives.

2.8 Research Objectives

Unlike previous electrospray cooling research, a summary of which is included in Table 2.4, all of which focused on surface averaged heat transfer. And building on the observations of Horacek et al. [73] and Sodtke and Stephan [72] who noted the intrinsic link between spray cooling and contact line evaporation. The objectives of this research are as follows:

- Experimentally quantify the influence of separation height, nozzle size and flow rate on single source electrospray cooling, and examine their impact on the heat and mass transfer from the heated substrate to the deposited liquid.

2.8. RESEARCH OBJECTIVES

- Explore the local heat transfer performance of an electrospray array. Analysing the interaction between adjacent spray plumes and the “driving” electric field and their role on the local heat and mass transfer to the impinging fluid.
- Investigate triple line evaporation. Quantify the influence of surface wettability and an applied electric field on the local heat and mass transfer to an evaporating liquid on a heated substrate.

This work aims to provide a foundation for future spray cooling optimisation of convective heat transfer enhancement. A brief summary of the experimental parameters of all four studies are included in Table 2.5.

Table 2.4: Electrospray cooling literature summary.

Author	Design	N_n	Working Fluid	Spraying mode	Wall flux [W m ⁻²]	Flow per sources [μL min ⁻¹]	Peak cooling	Heat flux measurement
Feng and Bryan [45]	Two electrode, $D_i = 250\text{ }\mu\text{m}$, $D_o = 470\text{ }\mu\text{m}$, $H = 5 - 20\text{ mm}$, $s = 4.3\text{ mm}$, $\rho_{N_n} = 5.4\text{ [}N_n\text{ cm}^{-2}\text{]}$	1, 4 square	Ethanol	Multiple regimes	$4 \times 10^4 - 80 \times 10^4$	750 – 8600	$1.7 \times q''_{0kv}$	Average
Wang et al. [161]	Two electrode, $D_i = 210\text{ }\mu\text{m}$, $D_o = 410\text{ }\mu\text{m}$, $H = 7.5\text{ mm}$, $L_n = 6.98\text{ mm}$, $s = 4, 5, 6\text{ mm}$, $\rho_{N_n} = 2.8 - 12.5\text{ [}N_n\text{ cm}^{-2}\text{]}$	1, 4 square, 8 ring	Ethanol	Multi-jet	$4364 - 8013$	2 – 133	$1.87 \times q''_{nc}$	Average
Deng and Gomez [6]	Three electrode, $D_i = 60\text{ }\mu\text{m}$, $D_o = 210\text{ }\mu\text{m}$, $H_2 = 5\text{ mm}$, $L_n = 450\text{ }\mu\text{m}$, $s = 0.675\text{ mm}$, $\rho_{N_n} = 253\text{ [}N_n\text{ cm}^{-2}\text{]}$	19 hexagonal	Ethanol	Cone-jet	$10 \times 10^4 - 96 \times 10^4$	22 – 88	$96 \times 10^4\text{ [W m}^{-2}\text{]}$	Average

Table 2.5: Present work experimental parameters.

Present work	Design	Working Fluid	Cooling mechanism	Wall flux [W m ⁻²]	Heat flux measurement
Study 1: Single source electrospray cooling	Two electrode, $D_i = 0.330 - 1.844$ mm, $D_o = 0.621 - 1.371$ mm, $L_n = 12.5$ mm, $H = 2.5 - 17.5$ mm, $N_n = 1$, $Q = 2 - 16$ $\mu\text{L min}^{-1}$	Ethanol	Cone-jet EC	1,395	Local
Study 2: Array electrospray cooling	Three electrode, $D_i = 200$ μm , $D_o = 400$ μm , $L_n = 570$ μm , $H_1 = 500$ μm , $H_2 = 7.5$ mm, $s = 1$ mm, $\rho_{N_n} = 115$ [N_n cm ⁻²], $N_n = 7$, hexagonal, $Q = 2 - 4$ $\mu\text{L min}^{-1} N_n^{-1}$	Ethanol	Cone-jet EC	1,395	Local
Study 3: Droplet evaporation	$V_0 = 80$ μL $\theta_A = 90 - 160^\circ$, $\theta_R = 53 - 159^\circ$ $Ra_f = 59.52 - 176.24$ nm	Water	Sessile droplet	912	Local
Study 4: Droplet evaporation in an electric field	Two electrode, $E_{field} = 0 - 11$ kV cm ⁻¹ $V_0 = 80$ μL $\theta_A = 90 - 160^\circ$, $\theta_R = 53 - 159^\circ$ $Ra_f = 59.52 - 176.24$ nm	Water	Sessile droplet	912	Local

Chapter 3

Experimental Test Facility and Analysis

This chapter describes the experimental apparatus, procedures and the data analysis applied during this research. Three experimental configurations are utilised: single source electro-spray cooling, multiple source electro-spray cooling and droplet evaporation for varied surface wettability with and without an applied electric field.

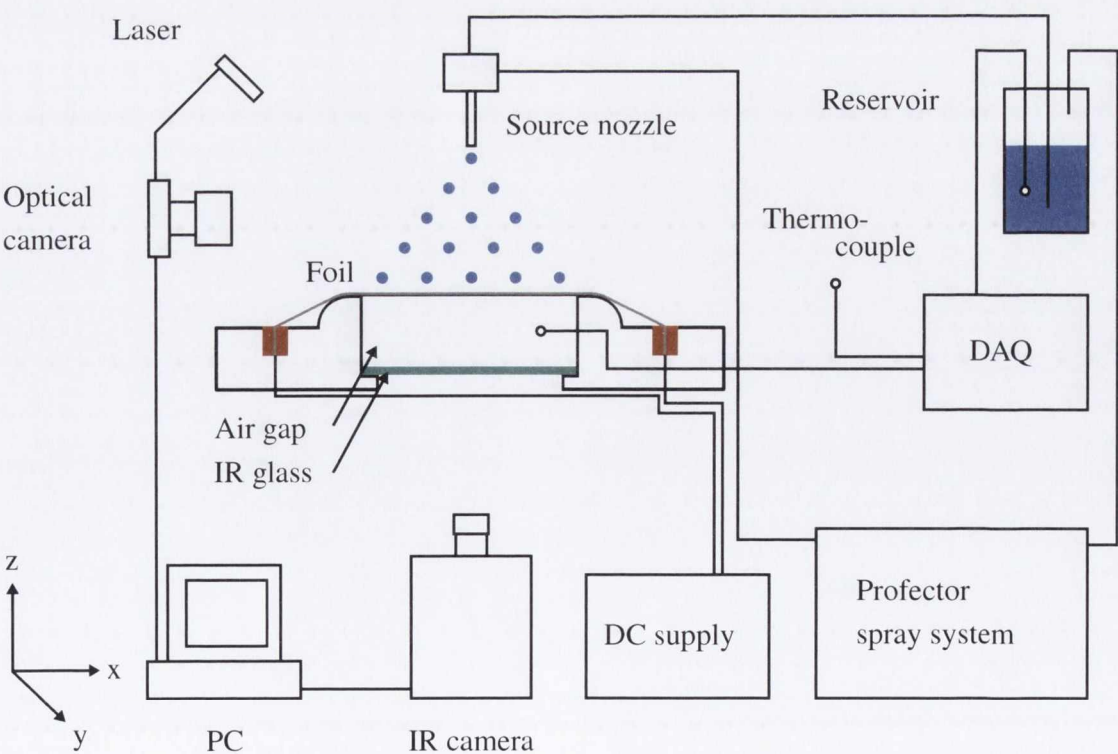


Figure 3.1: Experimental set-up schematic of single source electro-spray cooling.

3.1 Test Facility

The experimental apparatus consists of two primary sections; the thermal base and a cooling component. The thermal base was unchanged across experimental studies, while the means of thermal dissipation were interchanged for each study. A schematic of the test facility for the first experimental set-up is illustrated in Figure 3.1. In this case a single source electrospray was implemented as the source of cooling.

3.1.1 Thermal Base

The thermal base consists of a number of key components: thermal exchange section, infrared camera, optical camera, and data acquisition system.

3.1.1.1 Thermal Exchange Section

The thermal exchange surface is shown in Figure 3.2. It consisted of a $25\text{ }\mu\text{m}$ thick, $140 \times 80\text{ mm}^2$ Stainless Steel AISI 316 foil (Fe/Cr18/Ni10/Mo3, P/N: 505-400-04) (Figure 3.2d). The foil was clamped between two copper bus bars (Figure 3.2c). Both bus bars were

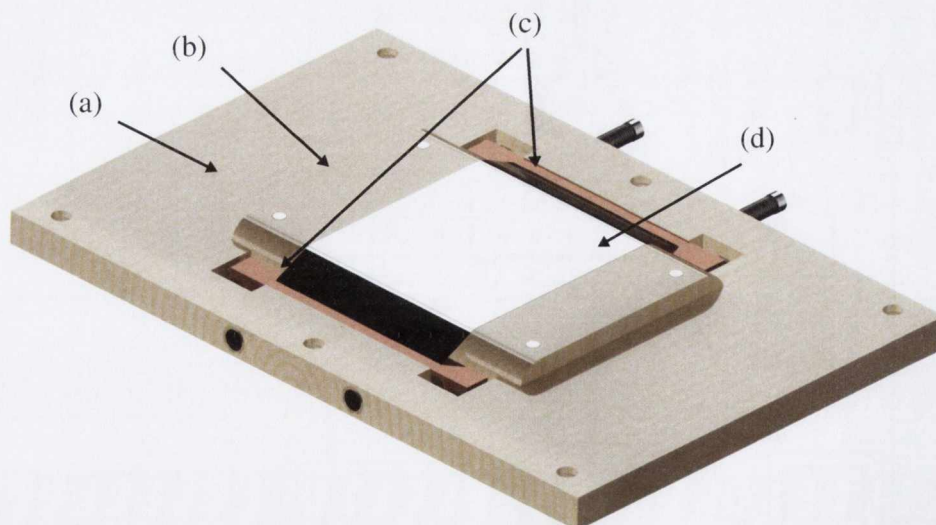


Figure 3.2: Thermal exchange section. (a) PEEK housing, (b) Central PEEK piece, (c) Copper bus bars, and (d) Stainless steel foil.

connected at each end to 6 mm² copper cables, through which a direct current (DC) was supplied from a Lambda GENESYS DC power supply (0 – 6 V, 0 – 200 A, and P/N: GEN 6-200 ABCDE). Constant current heating enables accurate control of the thermal exchange surface temperature. The copper bus bars were mounted on a polyether ether ketone (PEEK, Ketron 1000 PEEK) housing, measuring 275 × 180 × 11 mm³, shown in Figure 3.2a.

A tensioning system was employed in order to counteract foil warping at higher foil temperature gradients. The 25 µm foil was stretched across a central 140 × 100 × 13 mm³ PEEK piece (Figure 3.2b) that secures into a supporting PEEK housing. One set of the copper bus bars was rigidly fixed to this PEEK support structure while the other was spring loaded. The springs can be adjusted to vary the tensile force on the foil. The central PEEK component also serves to house a 70 × 70 × 2 mm³ infrared (IR) transparent Calcium Fluoride (CaF₂) piece on a supportive ledge. The top of this IR glass was positioned 7 mm below the underside of the foil. This configuration establishes a 71 × 71 × 7 mm³ air cavity. During experimentation this trapped air acts as a thermal barrier, ensuring that the heat lost from the underside of the foil was minimised. A 1.6 mm exposed T-type thermocouple was used to measure the air temperature at the mid point of the cavity. Direct temperature measurement of the underside of the foil was taken by an infrared camera. This surface was coated in thin

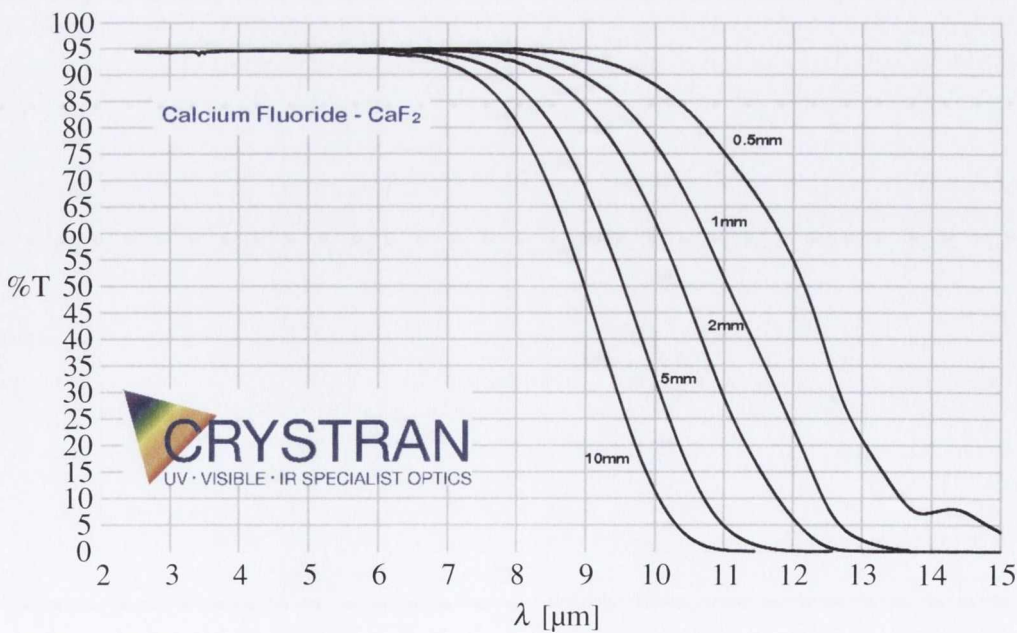


Figure 3.3: Calcium Fluoride IR transmissibility chart. Adapted from Crystran Ltd [162].

3.1. TEST FACILITY

layer of matt black paint to minimise reflections and provide a surface of known emissivity ($\epsilon_p \sim 0.95$). Calcium Fluoride was chosen due to its high percentage transmissibility within the operating spectrum of the infrared camera (3 – 5 μm). A transmissibility graph to wavelength, λ , for the calcium fluoride piece is shown in Figure 3.3.

Table 3.1: Thermal exchange substrate foil and paint layer properties.

Foil thickness, δ_f	25	μm
Density, ρ_f	7960	kg m^{-3}
Thermal conductivity, k_f	16.3	$\text{W m}^{-1} \text{K}^{-1}$
Specific heat, $C_{p,f}$	502	$\text{J kg}^{-1} \text{K}^{-1}$
Surface Roughness, $Ra_{f,\text{uncoated}}$	59.52	nm
Surface Roughness, $Ra_{f,\text{coated}}$	176.24	nm
CaF ₂ Emissivity, ϵ_{CaF_2}	0.17 - 0.25	-
Paint layer thickness, δ_p	10.52	μm
Density, ρ_p	1261	kg m^{-3}
Thermal conductivity, k_p	0.095	$\text{W m}^{-1} \text{K}^{-1}$
Specific heat, $C_{p,p}$	2835	$\text{J kg}^{-1} \text{K}^{-1}$
Paint Emissivity, ϵ_p	0.95	-

In analysing the thermal exchange surface it is important to define the relevant material properties, listed in Table 3.1. The high emissivity matt black coating of the thermal exchange section was applied using an airbrush. The paint thickness was determined using a Dektak 6m programmable surface profiler measuring system. This was a non-destructive test of the test surface. The Dektak had a resolution of ± 0.5 nm. Measurement were taken at 10 mm intervals along the side of the foil starting at 5 mm from the edge. The surface profiler operated by “scratching” a 2 mm section, consisting of both an uncoated and coated portion. The uncoated part was used as a reference height for the paint thickness. The results of this are shown in Figure 3.4. The acquired data was averaged and a paint thickness of 10.52 μm was found with a standard uncertainty of 2.7 μm . The foil properties were defined using the supplied Goodfellow data sheet. While the paint layer properties were determined from data tabulated by Donoghue et al. [163] and Raghu and Philip [164].

3.1.1.2 Material Properties

The surface roughness of the steel foil was determined using a MicroXAM White light Interferometric Surface Profiler. Two surface conditions were characterised; uncoated and

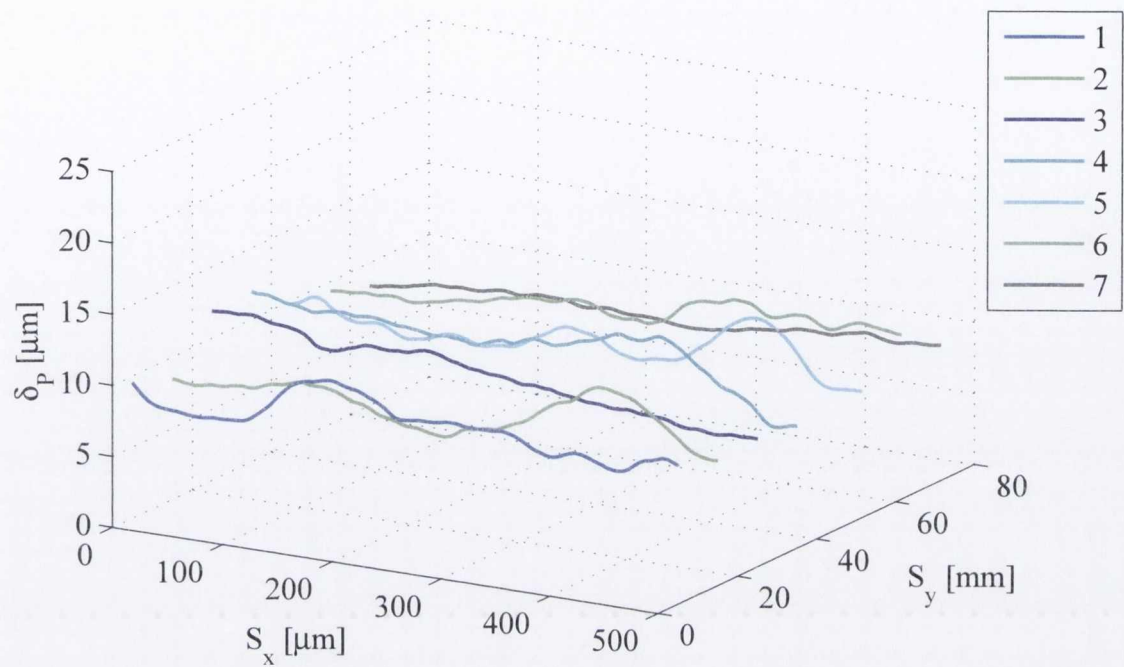


Figure 3.4: Paint layer thickness at different locations on the foil.

coated. The first corresponded to the normal raw foil, a surface roughness of 59.52 nm was found with a standard uncertainty of 1.23 nm for this case. For the coated case the foil with coated with Glaco surface coating [165], a surface roughness of 176.24 nm was found with a

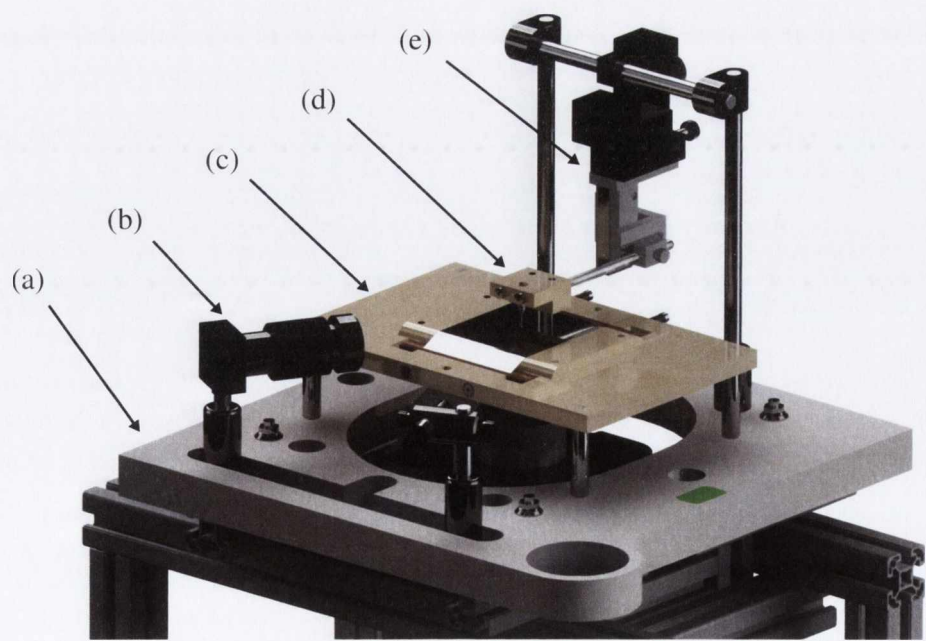


Figure 3.5: Thermal baseplate configuration. (a) Ertacetal baseplate, (b) Optical camera. (c) Thermal exchange section, (d) Heat flux removal device, and (e) XYZ micropositioner.

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standard uncertainty of 4.83 nm for this case. The surface roughness was calculated using the average of the absolute values of the profile heights in the measured area.

The thermal exchange section was mounted on a acetal (Ertacetal C, Quadrant engineering plastic products) baseplate (Figure 3.5a), $400 \times 320 \times 30 \text{ mm}^3$, and was supported on $40 \times 40 \text{ mm}^2$ Bosch Rexroth aluminium profile frame (P/N: 3842993120/3000). A XYX micropositioner (Figure 3.5e) was used to control placement of the heat flux removal device. The separation height can be set accurate to $\pm 0.5 \text{ }\mu\text{m}$.

3.1.1.3 Thin Foil Thermography

The thin foil thermography technique was implemented across all experimental studies. This technique is a physical approximation of the ideal constant wall heat flux (CWF) boundary condition. A thermal imaging camera is used to focused on a surface of known emissivity to non-intrusively acquire temporal and spatial temperature data. In a standard set-up the thermal imaging camera is focused on the opposite side of a thin foil to the cooling phenomenon (see Figure 3.1). The technique relies on the assumption that the temperature on the top and bottom sides of the foil at a given point are equal. This was verified by calculating the Biot number (see Section 3.3.1.2). By acquiring the temporal and spatial temperature data of the substrate in tandem with other easily measured values (ambient temperature, foil current and voltage drop), it is possible using an energy balance approach to calculate the local thermal energy dissipated by the implemented cooling (see Section 3.3.1.2). This local heat transfer data can be coupled with the fluid mechanics on the top surface to develop a greater understanding of the heat and mass transfer mechanisms taking place. This allows the development of smarter thermal solutions than would be achievable through bulk/average experimental heat transfer techniques.

Figure 3.6 contrasts the constant wall heat flux, (left column), constant wall temperature

Table 3.2: Numerical model properties.

Experimental case	Substrate thickness [μm]	Thermal conductivity [$\text{Wm}^{-1} \text{K}$]
CWF	2.5	16.3×10^{-12}
CWT	250	16.3×10^3
Thin foil	25	16.3

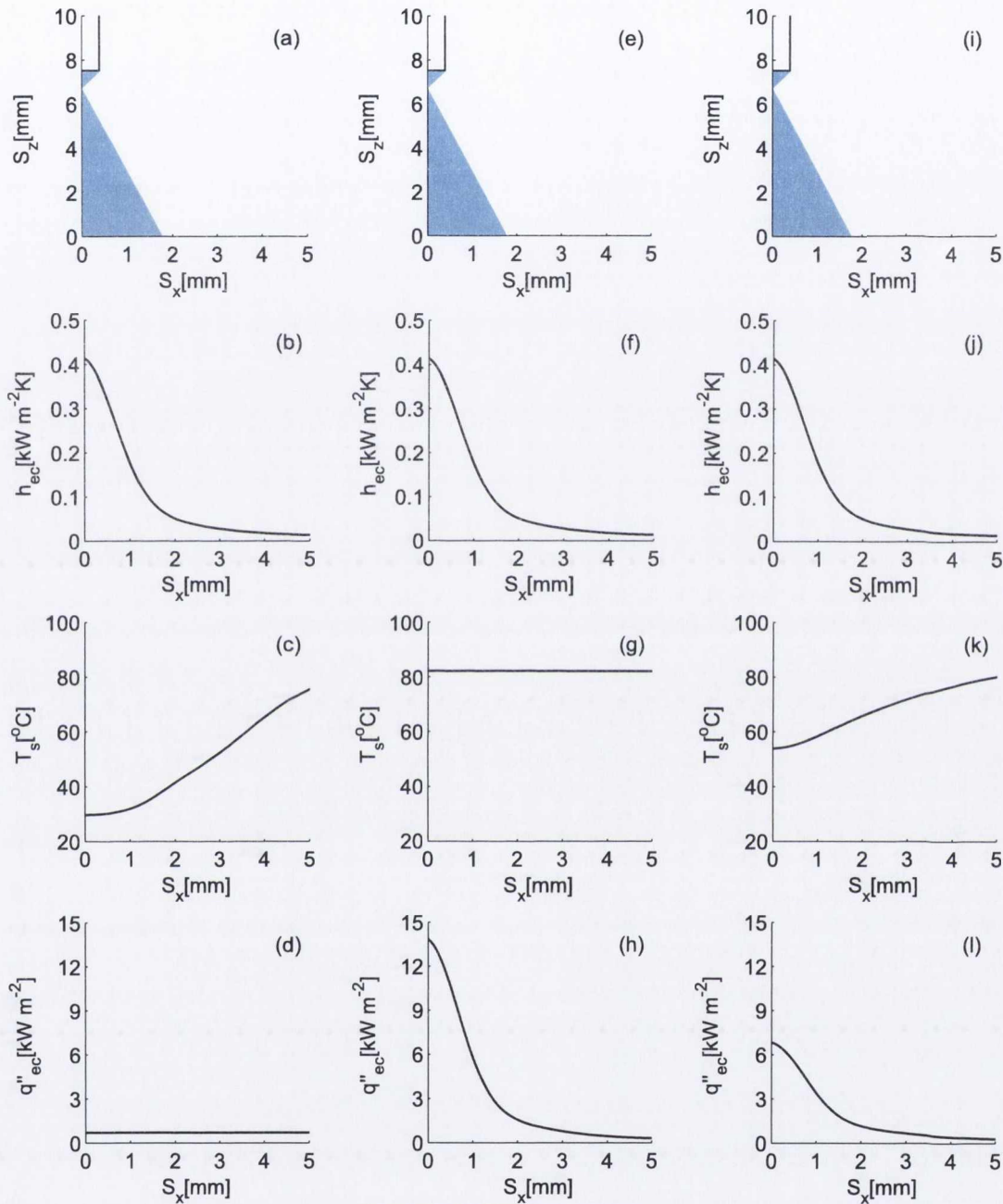


Figure 3.6: Thin foil comparison. Left: CWT, centre: CWF, and right: thin foil

(CWT) (centre column) with that of the experimentally implemented thin foil technique. The results were computed using COMSOL Multiphysics. The model implemented is largely the same as outlined in Appendix A. The same experimentally calculated heat transfer coefficient is applied to the top surface of the heated substrate for all three cases. For the thin foil case (right column) this results in the experimentally observed radial temperature and heat flux

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profile for this test point (see Figure 4.2, $V_n = 3$ kV, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 7.5$ mm, $Q = 4$ $\mu\text{L min}^{-1}$ and $q''_{gen} = 1,395$ W m^{-2}). The substrate thickness and thermal conductivity are varied to establish the CWF or CWT cases. These values are highlighted in Table 3.2

For the constant wall heat flux case (left column) the thermal conductivity and thickness of the substrate are reduced to limit energy movement within the foil. This case results in a constant radial heat flux profile (Figure 3.6d) and a radial temperature profile (Figure 3.6c) that mirrors the inverse of the applied radial heat transfer coefficient. Realisation of the ideal CWF condition is not physical practical for thin foil thermography, given the thermal, electrical and structural requirement. Foil thickness below 10 μm can be difficult to work with, resulting in a non-infinitely thin substrate. Coating an IR transparent substrate with a thin emissive black layer such as chromium nitride is an option [166]. Thermal conductivity of the IR material then plays an important factor as conjugate heat transfer within this surface can break down the CWF implementation. This selection of material is also limited to those compatible with the IR spectrum of the IR camera.

For the isothermal case the thermal conductivity and foil thickness are increased as shown in Table 3.2. This lowers the thermal resistance within the substrate. As a result of the unrestricted conjugate heat transfer, a uniform radial temperature profile is noted (Figure 3.6g). This is despite the localised cooling at $S_x = 0$ mm. This free movement of thermal energy within the substrate enables the maximum heat flux from the heated substrate for the applied heat transfer coefficient (Figure 3.6h). The shape of this radial heat flux profile matches that of the heat transfer coefficient. Analysis of the local heat and mass transfer in this case is not possible due to the constant wall temperature, but this set-up lends itself well to average/bulk heat transfer analysis [6, 45, 136, 149, 150, 167, 168]. This experimental condition can be approximated using a highly conductive and thick substrate material such as copper or aluminium.

The final column of results is that of thin foil implemented in this research. By comparing the radial temperature (Figure 3.6c, g and k) and heat flux plots (Figure 3.6d, h and l) it can be seen that this experimental condition lies between that of the constant wall heat flux and temperature cases. The internal thermal resistance of the foil is not high enough to achieve the ideal CWF case but is sufficient to prevent the CWT scenario. As outlined in

Section 3.3.1.2, the acquired temperature gradient can be used to account for the conjugate heat transfer in the foil. This enables investigation into the local heat and mass transfer of electrospray plumes and evaporating liquids which is not possible using bulk/average heat transfer techniques [6, 45, 136, 149, 150, 167]. This enable greater understanding of the physical mechanisms occurring during heat transfer.

3.1.1.4 Data Acquisition

The data acquisition was controlled by LabVIEW. A LabVIEW virtual instrument (VI) was developed to simultaneously trigger all data acquisition devices during testing. The relevant systems were; IR image recording, optical imaging, thermocouple measurement, working fluid flow rate (Chapters 4 and 5 only), and the current and voltage drop across the thermal exchange section foil and paint layers.

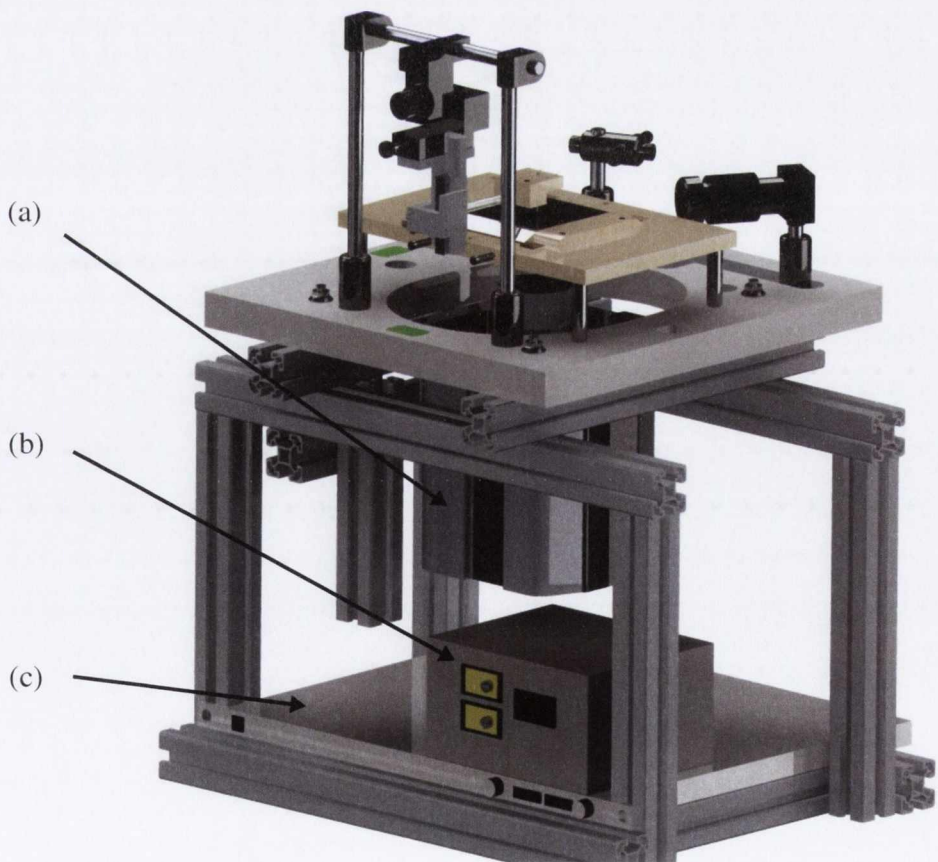


Figure 3.7: Thermal base. (a) IR camera, (b) Spraybase DC voltage and pressure supply, and (c) substrate DC power supply.

Infrared Thermography

A FLIR SC6000 infrared camera and ExaminIR (Version 1.30.0, build 72, PN: 24796-200) integrated software was used to acquire the IR images. The camera was mounted to the aluminium profile frame, directly below the thermal exchange surface as shown in Figure 3.7a. To reduce external reflections, the path between the camera lens and the test surface was surrounded by a black card enclosure. The camera has an indium antimonide (InSb) 640×512 pixel sensor. A Stirling motor cooled the sensor to approximately 78 K, thereby reducing the effects of noise in the captured images. The sensor is cooled to avoid flooding of infrared radiation from its own surroundings. The focal plane array (FPA) is fitted with an infrared filter to restrict the sensitivity of the sensor to the $3 - 5 \mu\text{m}$ range, which is known as the Mid-Wavelength InfraRed (MWIR) range. For the present study a 25 mm focal length lens was utilized. Data from the camera was transferred to the high speed data recorder (HSDR) via a CameraLink connection, which bypasses the dedicated computer's motherboard. A 1 Gb s^{-1} ethernet connection provides instructions and live image to and from the camera. Thermal image recording was controlled by a NI-9401 data acquisition module (DAQ). The DAQ sends a 5 V TTL signal at 200 Hz to the thermal imaging camera. The signal was generated through a coaxial cable to a female BNC connector, then passed directly to "sync. in" connection on the rear of the FLIR SC6000 camera. The camera was set to "listen" for the external trigger in the FLIR ExaminIR software. The camera records one frame on the rising edge of the triggering signal. Record length and timing was set using the LabVIEW VI. The thermal camera was set to record an image that was 400×400 pixels at a frequency of 200 Hz. As the image is no longer full frame, an offset must be selected. In order to reduce reflections, the centre of the area of interest was offset from the centre of the focal array. This was necessary because of a phenomenon known as the Narcissus effect, in which the camera sees its own sensor due to internal lens reflections. The camera was tilted at a 5° angle to the thermal exchange surface to reduce this Narcissus effect. The IR camera has a 14 bit sensor meaning that each pixel can detect between 0 and 16383 individual levels, or counts. An integration (exposure) time of 0.22 ms at 200 Hz was chosen so that the highest temperature recorded ($< 100^\circ\text{C}$) corresponded to a count value of approximately 15000. For every operation of the camera, seven files are saved, with file extensions of .sfmov, .pod, .scg, .sbp, .sco, .cal and .inc. The .sfmov file contains the raw infrared measurement data, in

counts, from the camera. The .pod file contains data on the exact time of each frame obtained from the Inter-range instrumentation group (IRIG) clock in the HSDR computer. The .scg, .sco and .sbp are the gain, offset and bad pixel image correction files. The .cal file relates to the counts to temperature calibration of the IR camera and the .inc has information on the camera set-up during data acquisition.

Optical

A Chameleon CMLN-1352M optical camera was focused on the heated substrate during experimentation (see Figure 3.5b). The optical camera operates in conjunction with Point Grey FlyCap2 imaging software (Version: 2.04.3-010). The camera is fitted with a Sony ICX445 mono CCD sensor (4.86×3.62 mm) which has a maximum resolution of 1296×964 pixels and can record at 18 Hz. It has a 12 bit ADC chip enabling a dynamic range of colours with an optimum exposure time of 10 μ s. The camera was fitted with a long distance microscopic lens (Infinity Proximity InfiniMite Gamma lens with a DL Tube/2 $\times$, P/N: 57-724 and 39-686). At the optimum working distance of 122 mm this results in a primary magnification factor of 1 $\times$. This gives a pixel size of 3.5 μ m and a field of view of 4.86×3.62 mm. During testing the optical camera can be used in conjunction with an Acculase diode laser (650 nm, 5 mW, P/N: ACCULASE-PWM-650-5-S) fitted with a 9 $^\circ$ line lens (Global Laser Interchangeable Cross 8d, P/N: 697-3579) to illuminate and capture the cooling phenomenon.

Temperature

Four temperature measurements are taken during experimentation: air gap, working fluid reservoir, and 2 $\times$ ambient surroundings. The air gap and ambient readings are taken with exposed T-type copper-constantan inconel sheath thermocouples and the working fluid with an ungrounded T-type copper-constantan inconel sheath thermocouple. A NI-9214 DAQ records the 4 thermocouple signals at 2 Hz each. 2 Hz was the maximum record rate of the DAQ while maintaining a "High" resolution setting.

Current and Voltage

The voltage drop and current through the heated substrate was measured using two Hewlett-Packard 34401A digital multimeters (DMM). The voltage reading is taken across the steel

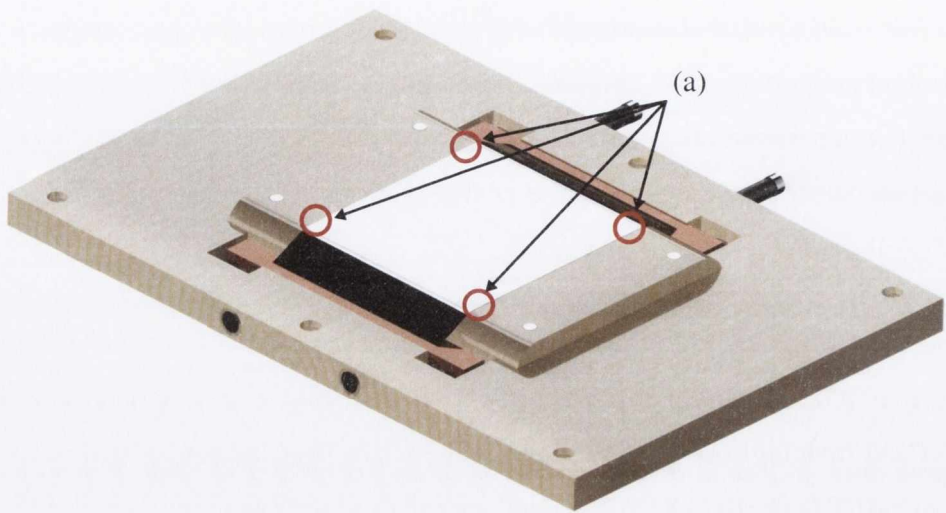


Figure 3.8: Foil voltage measurement. (a) Voltage reading placement.

foil between the copper busbars. These measuring points are highlighted in Figure 3.8. The DMMs are connected in series via a GPIB serial connection and National instrument (NI) USB – GPIB cable to the dedicated computer. The current was scaled into an easily measurable voltage drop using a Murata 50 A 50 mV $\pm 0.25\%$ shunt (P/N: 3020-01096-0). These multimeters record at a frequency 50 Hz for a time period defined in the core VI.

Flow Rate

A Sensirion liquid flow sensor (IPA full scale flow rate: $500 \mu\text{L min}^{-1}$, Accuracy: 20% of

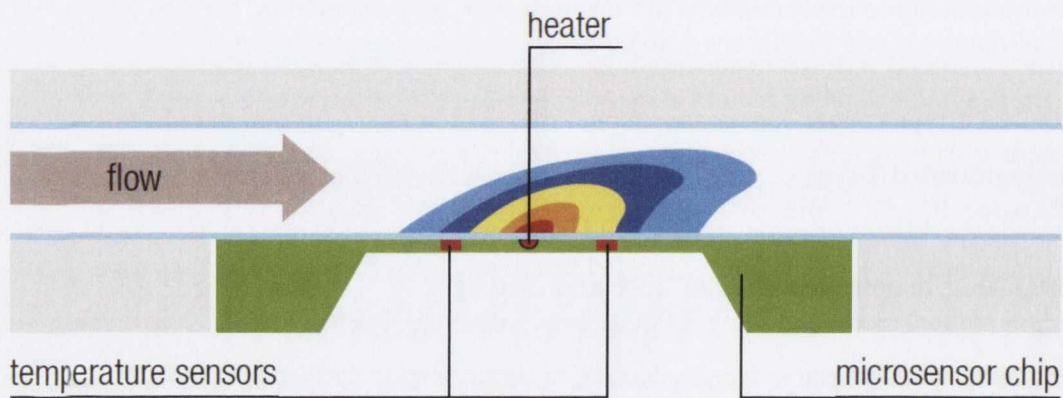


Figure 3.9: Sensirion liquid flow sensor principle of flow measurement. Reprinted from Sensirion AG [169].

measure value / 1% of full scale, Repeatability: 1% of measure value / 0.05% of full scale, and P/N: SLI-0430) was used to record the working fluid flow rate. The sensors primary novelty was the ability to non-intrusively measure ultra low working fluid flow rates on the scale of $0.01 \mu\text{L min}^{-1}$. The flow rate was determined by the integrated digital CMOS microchip (see Figure 3.9) bonded to the outside of the sensor's $430 \mu\text{m}$ flow channel. The microchip added a minimal amount of heat to the working fluid for the thermal flow measurement. Temperature readings were taken by two temperature sensors positioned before and after the heat source. This thermal data was directly related to the flow rate through the working fluid properties [169]. The Sensirion flow sensor recorded at 2 Hz.

3.1.2 Thermal Dissipation

Three different cooling configurations are implemented throughout this research: single source electrospray cooling, electrospray array cooling and sessile droplet evaporation.

3.1.2.1 Study 1: Single Source Electrospray

The spray source was an stainless steel nozzle (Adhesive Dispensing, P/N TS23SS-1/2) (Figure 3.10d) with a blunt end finish. The nozzle consisted of a chrome nickel plated Luer lock hub to grade 304 stainless steel burr-free capillary. The capillary is electro-polished for unobstructed and consistent fluid flow. The nozzle was attached to the fluid path by a female Luer lock to 1/4 inch 28 UNF adapter (Figure 3.10c). The nozzle was held in place by a

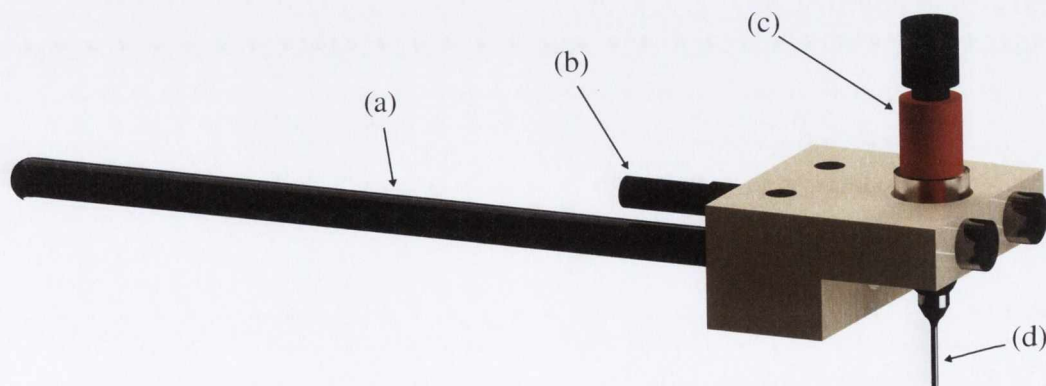


Figure 3.10: Single source electrospray. (a) optical wand, (b) source electrode, (c) fluidic connector, and (d) source nozzle.

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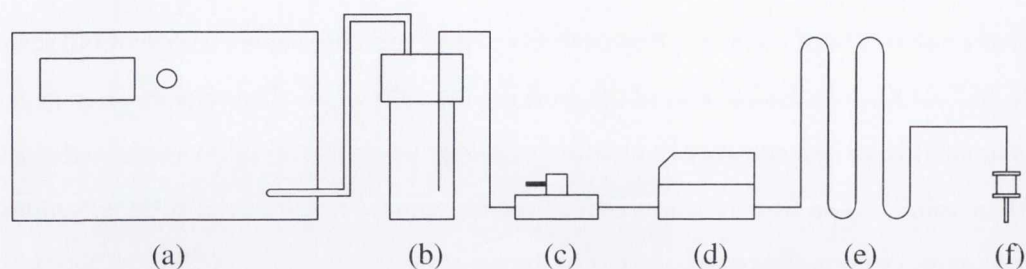


Figure 3.11: Electro spray flow path. (a) Spraybase controller, (b) working fluid reservoir, (c) 2-way shut-off valve, (d) Sensirion flow sensor, (e) flow resistance, and (f) source nozzle.

PEEK holder and electrically connected using a gold spring loaded pin electrode inserted into the PEEK holder (Figure 3.10b). The optical wand (Figure 3.10a) was clamped into the XYZ micropositioner (Figure 3.5d) which sets the source nozzle separation height. A flow path schematic is shown in Figure 3.11. The working fluid was HPLC grade ethanol from Fisher Scientific ($\text{C}_2\text{H}_5\text{OH}$, density 790 kg m^{-3} , boiling point: 78.37°C , Assay (GC): $>99.8\%$, Water $<0.3\%$, and P/N: E/0665DF/17). A variable pressure head was supplied to the working fluid reservoir (Figure 3.11b) from the Spraybase controller (Avectas, DC power: $0 - 30 \text{ kV}$, pressure head: $0 - 5 \text{ bar}$, and P/N: CAT000040) (Figure 3.11a). The controller takes in 10 bar from an outside source and regulates it to an output pressure between $0 - 5 \text{ bar}$. This output can be set to a precision of 1 mbar . The applied pressure drives the ethanol flow through the fluidic circuit (Figure 3.11). The ethanol flows from the reservoir through the 2 way shut-off valve (Figure 3.11c) to the Sensirion flow sensor (Figure 3.11d). Additional flow resistance was implemented (Figure 3.11e) in order to ensure a steady and consistent flow. This flow resistance consisted of FEP and PEEK tubing of predefined lengths (see Table 3.3). The FXX resistors (Dolomite microfluidics, P/N 32002YY) consisted of fused silica (wetted material), polyimide (inner sheath) and PEEK (outer sheath). The FXX number corresponded

Table 3.3: Single source electro spray flow path.

Flow section	Channel diameter [μm]	Channel length [mm]
Reservoir – shut-off valve	500	580
Shut-off valve – flow sensor	500	62
Flow sensor	100	100
F10 flow resistor	100	1470
F30 flow resistor	100	490
Flow resistor – source nozzle	100	200

to the flow rate in μL for a 1 bar pressure drop across the resistor. They are applied in series to increase the applied flow resistance. The amount of flow resistance applied was defined by the maximum required flow rate such that this was achieved for a ~ 4.8 bar applied pressure. The applied flow resistance is shown in Table 3.3. Using a pressurised system to drive the working fluid flow is more advantageous than other methods as it removes flow pulsation that can exist in systems where flow is driven by syringe, piezoelectric and peristaltic pumps.

All tests were conducted in the cone-jet regime of spraying. The applied nozzle voltage and flow rate ranged between 2 – 5 kV and 2 – 16 $\mu\text{L min}^{-1}$ respectively during experimentation. The nozzle to target separation height was varied from 2.5 – 17.5 mm in increments of 2.5 mm. Four inner diameter nozzle sizes were characterised; 0.330, 0.712, 1.066, and 1.371 mm. Experiments were conducted at atmospheric pressure and room temperature after steady state conditions were reached. The heated substrate temperature was assumed to be uniform across its thickness at a constant heat flux generation of $1,395 \text{ W m}^{-2}$ for all test points. This corresponded to a surface temperature of $\sim 90^\circ\text{C}$ in the absence of EC.

3.1.2.2 Study 2: Electrospray Array

An electrospray array was developed (see Figure 3.12) and the relevant design length scales are shown in Table 3.4. An exploded view of the array assembly is shown in Figure 3.13 and a

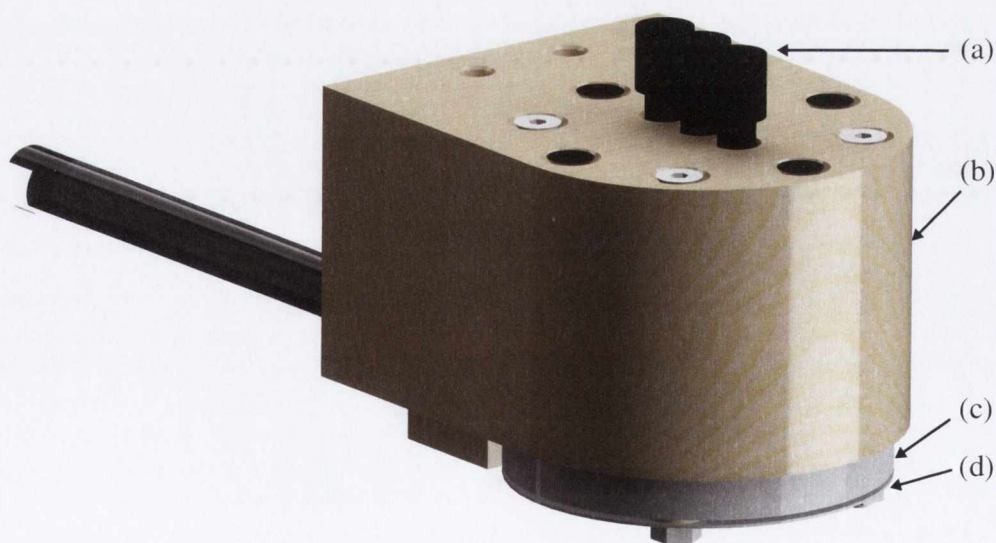


Figure 3.12: Electrospray array. (a) Fluid input, (b) PEEK reservoir, (c) Source electrode, and (d) Extractor electrode.

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Table 3.4: Electrospray array design summary.

Configuration	Triangular	
Number of sources, N_n	7	
Nozzle spacing, s	1	mm
Nozzle density, ρ_{N_n}	115	$N_n\text{cm}^{-2}$
Nozzle inner diameter, D_i	200	μm
Nozzle outer diameter, D_o	400	μm
Nozzle height, L_n	570	μm
Nozzle channel length, L_{ch}	34	mm
Source – Extractor separation, H_1	500	μm
Extractor hole diameter, D_{ex}	600	μm
Extractor thickness, δ_{ex}	200	μm

section view is presented in Figure 3.14. The array consisted of three key components: PEEK reservoir, source electrode, and extractor electrode. The source electrode was constructed by integrating 19 stainless steel AISI 304 capillaries (Figure 3.13d) into a 3 mm thick stainless steel plate (Figure 3.13e). The capillaries were fixed in place first using electrically conductive silver epoxy and later using Loctite 9497 epoxy. The source electrode was mounted and sealed against the PEEK reservoir during testing (Figure 3.14h).

The PEEK reservoir (Figure 3.14d) was filled with 250 – 500 μm diameter glass beads (Retsch, P/N: 22.222.0002) to minimise dead volume and increase flow resistance within the reservoir. During testing the reservoir is completely filled with ethanol to remove all air in the system using the input channel (Figure 3.14b) and bleed valve (Figure 3.14a). Edge effect were eliminated by plugging the outside ring of 12 source nozzles and by applying an appropriate source flow resistance. This was done to ensure a uniform flow rate source - source across the array. A flow resistance comparison with an array designed by Deng et al. [7, 60] is shown in Table 3.5. Deng et al. reported consistent droplet size across all sources during testing. The array used in the present study has a flow resistance $4.89\times$ greater than that employed by Deng et al. in their study. Flow resistance was calculated assuming laminar

Table 3.5: Source nozzle flow resistance – edge effect comparison.

Author	Number of sources	Channel diameter [μm]	Channel length [mm]	Flow resistance [m^{-3}]
Deng et al. [7]	91	60	.775	2.68×10^{13}
Present Study	7	200	36	1.31×10^{14}

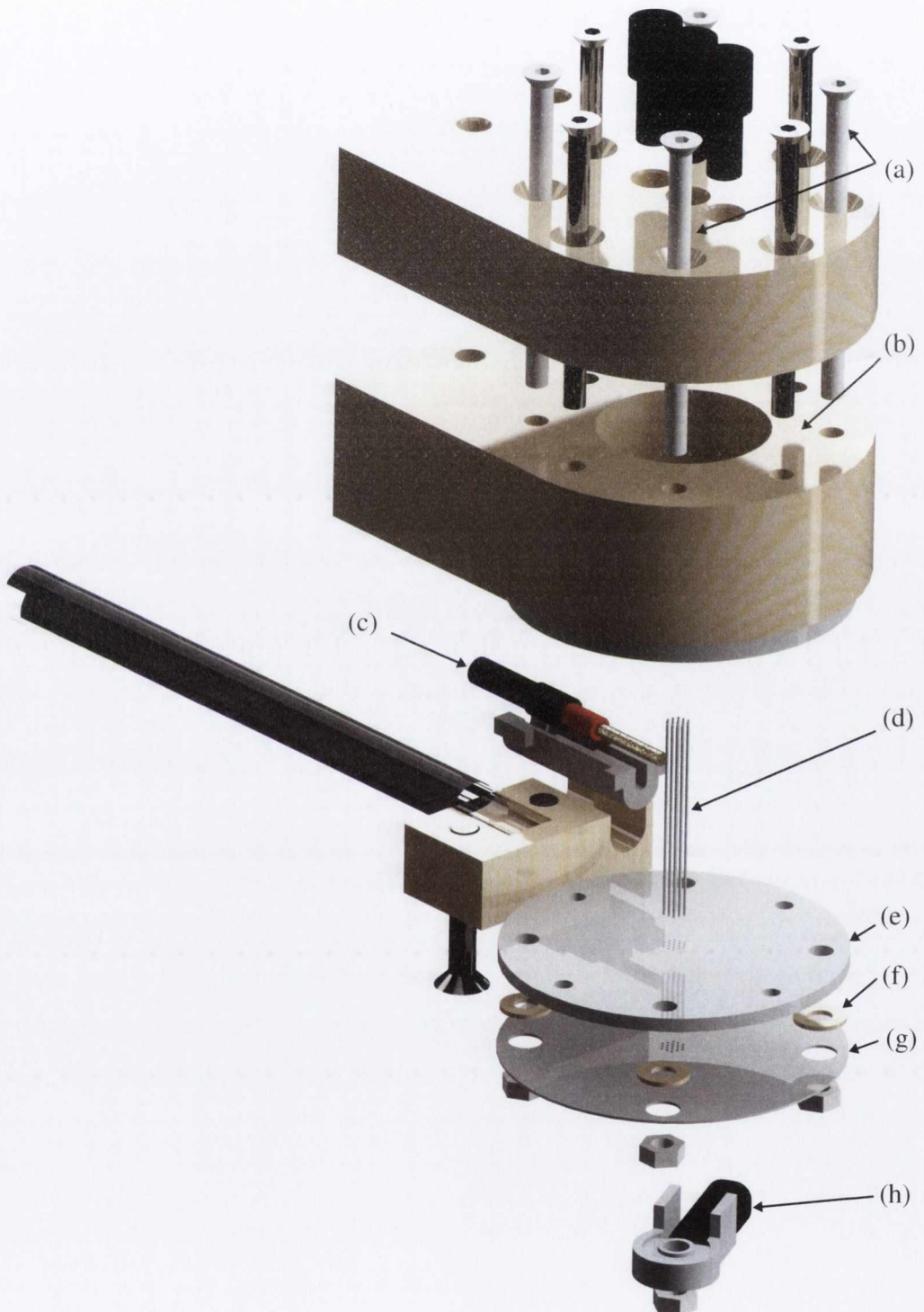


Figure 3.13: Electro spray array section view. (a) nylon screws, (b) PEEK reservoir, (c) source electrode connection, (c) 19 $\times$ source nozzles, (d) source electrode, (e) PEEK spacers, (f) extractor electrode, and (g) extractor electrode connection.

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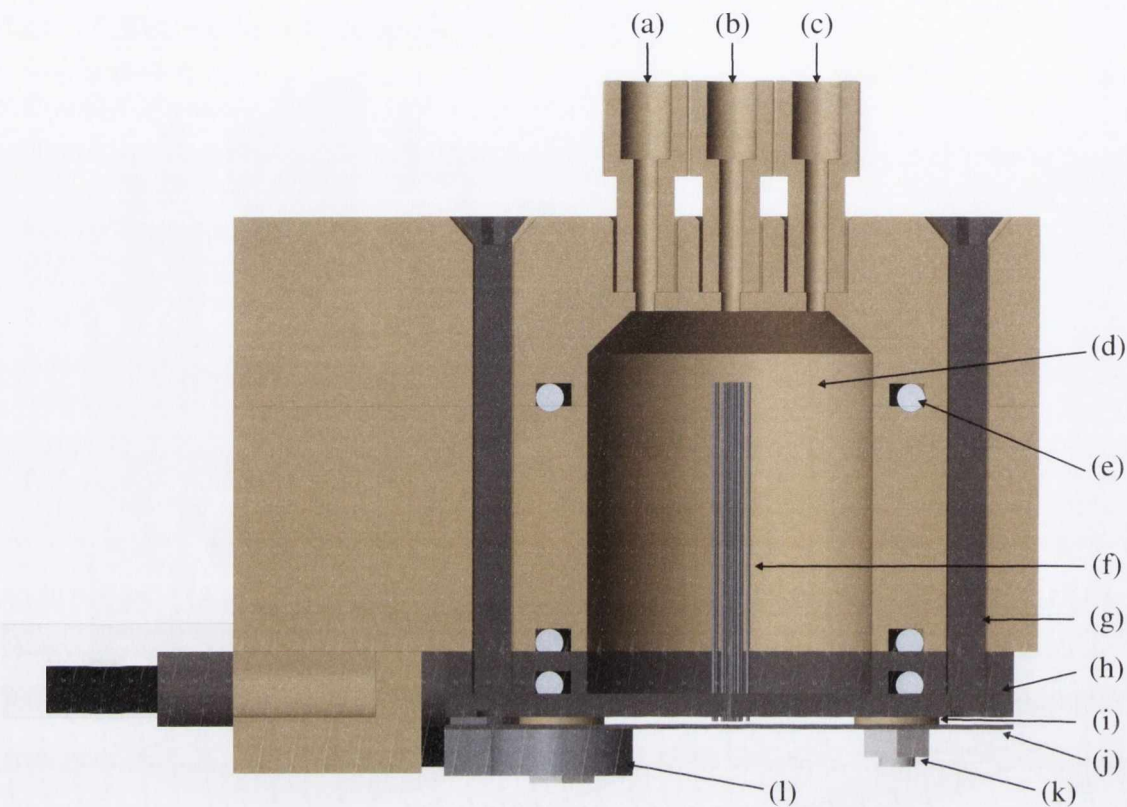


Figure 3.14: Electrospray array exploded view. (a) bleed valve, (b) working fluid input, (c) thermocouple measurement access, (d) working fluid reservoir packed with glass beads, (e) o-ring seal, (f) 19 × source nozzles, (g) stainless steel screw, (h) source electrode, (i) PEEK spacers, (j) extractor electrode, (k) nylon screw and nut, and (l) extractor electrode connection

flow using the major head loss equation in pipe flow, $h_{l,major} = \frac{2f_nL_{ch}Q^2}{D_i^5}$, where f_n is the friction factor [170]. By rearranging this equation, removing fluid properties, pressure head, and solving for the reciprocal of the flow rate, the flow resistance in a laminar pipe flow is given by: $R_Q = \frac{128L_{ch}}{\pi D_i^4}$.

A similar fluid delivery system to study 1 was used (see Figure 3.11). The primary difference being lower flow resistance as a result of the higher total flow rate required by

Table 3.6: Electrospray array flow path.

Flow section	Channel diameter [μm]	Channel length [mm]
Reservoir – shut-off valve	500	560
Shut-off valve – flow sensor	500	60
Flow Sensor – flow resistance	500	1000
Flow Resistance – Peek Reservoir	100	192
Array Nozzle	200	36

the array. This flow path is shown in Table 3.6. The flow was again driven by a 0 – 5 bar pressure controller. The source electrode potential was supplied by 0 – 30 kV DC power supply used in study 1 and the extractor electrode by a 0 – 20 kV DC power supply (Advectus, P/N: CAT000005). Flow rate was monitored using the Sensirion flow sensor. The source and extractor electrodes were electrically insulated from each other using nylon screws (Figure 3.13a) and M4 PEEK spacers (Figure 3.13f). All tests were conducted in the cone-jet regime of spraying. The applied voltages and flow rate ranged between 2 – 10 kV and 2 – 4 $\mu\text{L min}^{-1} N_n^{-1}$ respectively during experimentation. The extractor electrode – heated substrate separation height was set at 7.5 mm. Experiments were conducted at atmospheric pressure and room temperature after steady state conditions were reached. The heated substrate temperature was assumed to be uniform across its thickness at a constant heat flux generation of $1,395 \text{ W m}^{-2}$ for all test points. This corresponded to a surface temperature of $\sim 90^\circ\text{C}$ in the absence of EC.

3.1.2.3 Study 3 and 4: Droplet Evaporation

Droplet evaporation was observed both with and without an applied electric field for two substrate wettability condition. A schematic of the experimental set-up is illustrated in

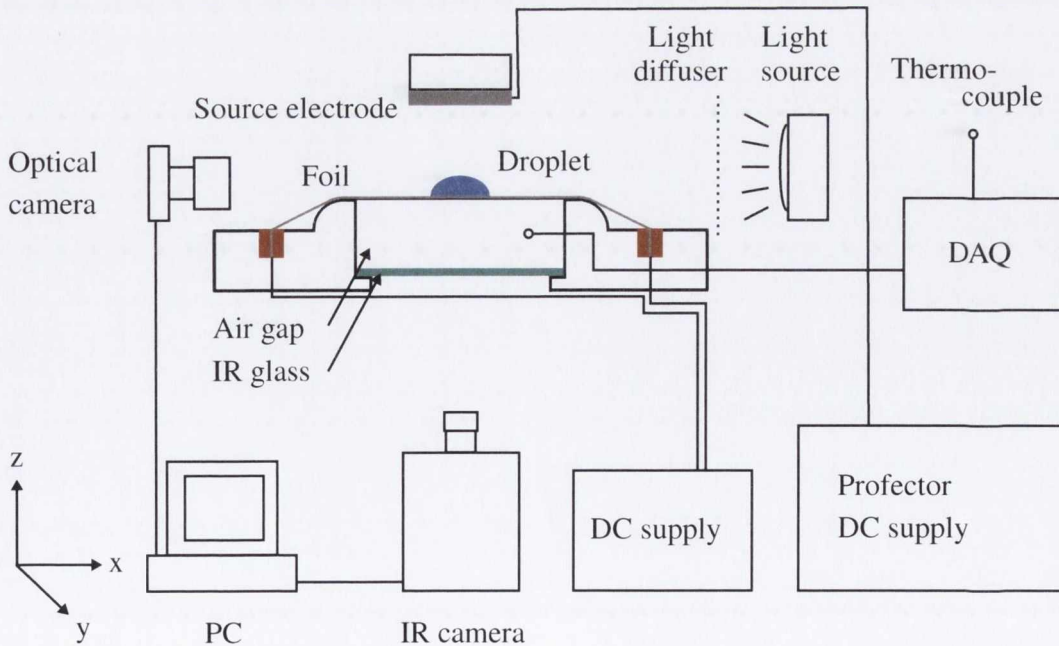


Figure 3.15: Experimental set-up schematic for droplet evaporation.

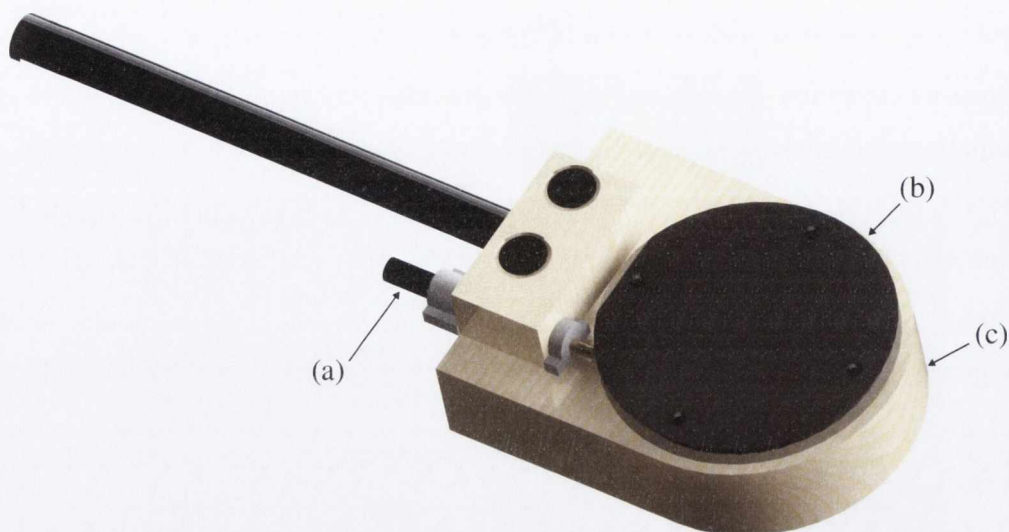


Figure 3.16: Droplet evaporation electrode. (a) Electrode connection, (b) Disc electrode, and (c) PEEK reservoir.

Figure 3.15. A flat disc electrode was positioned over the droplet during experimentation. This electrode is shown in Figure 3.16. The disc electrode was 5 mm thick and 55 mm in diameter. It was positioned over the heated substrate using the XYZ micropositioner (Figure 3.5d). The electrode was maintained at a 10 mm separation height for each test. The source electrode potential was supplied by the 0 – 20 kV DC power supply used in study 2. Two substrate conditions were explored during testing; uncoated and coated. The coated condition was achieved using Glaco Mirror Coat Zero. It was applied in a manner similar to that of Malavasi et al. [165]. The surface was cleaned using acetone and then ethanol. It was allowed to dry for 3 - 5 hours before the Glaco coating was applied. A 24 hour drying period was implemented before use. The coating was removed and reapplied after each test. The working fluid used was deionised water from Sigma-Aldrich (boiling point: 100°C, P/N: 99053).

The initial droplet volume was set at 80 μL for all test points. The volume was achieved using a KDS 200 dual syringe infusion pump (linear force: 18 kg, advance per microstep: 0.165 μm , max step rate: 1600 s^{-1} , min step rate: 100 s^{-1} , accuracy $\pm < 1\%$, reproducibility $\pm 0.1\%$, and P/N: 789200) and a Hamilton 1750 gastight syringe (capacity: 500 μL , inner bore: 3.26 mm, and P/N: 81262). The syringe pump was set to the desired volume and deposited on the surface using 100 μm inner diameter FEP tubing (DuPont, P/N: 1474-20).

The average applied electric field was $0 - 11 \text{ kV cm}^{-1}$ for the uncoated substrate tests and $0 - 6 \text{ kV cm}^{-1}$ for the coated tests. The maximum electric field in both cases was the highest electric field that could be set during experimentation without the occurrence of coronal discharge between the droplet and the disc electrode. Thermal and optical images were captured using the FLIR SC6000 and Chameleon CMLN-1352M cameras respectively. A GS Vitec MultiLED light source (24 LED, Power: 7,700 Lumen, P/N: GS 01372) was used to illuminate the droplet. Tracing paper was placed between the light and the droplet to diffuse the light source. The heated substrate temperature was assumed to be uniform across its thickness at a constant heat flux generation of 912 W m^{-2} for all test points. This corresponded to a surface temperature of $\sim 70^\circ\text{C}$ in the absence of droplet evaporation at steady state.

3.1.3 System Calibration

3.1.3.1 Thermocouples

Six T-type thermocouples were used during experimentation. The first three were; T_{s1} , T_{s2} , and T_{s3} , were glued to the heated substrate using Loctite 9497 thermal adhesive. These thermocouples were only used for calibration of the FLIR SC6000 thermal imaging camera. The second three thermocouples; T_∞ , T_{wf} , and T_{ag} , were used in data acquisition during testing and have already been discussed previously. The six thermocouples were calibrated using a resistance temperature detector (RTD) probe. Calibration points were taken at both 2.5 and 5°C intervals. The lower interval points were for temperature ranges of greater required accuracy. Temperature data was acquired using a custom built VI over a 60 second period at 2 Hz ch^{-1} . The 120 samples per channel were averaged to give an adequately accurate result. The linear fit equation for each thermocouple is shown in Equations 3.1 – 3.6 and Figure 3.17.

$$T_{RTD} = 0.9998T_{s1} - 0.1334 \quad (3.1)$$

$$T_{RTD} = 1.0003T_{s2} - 0.1261 \quad (3.2)$$

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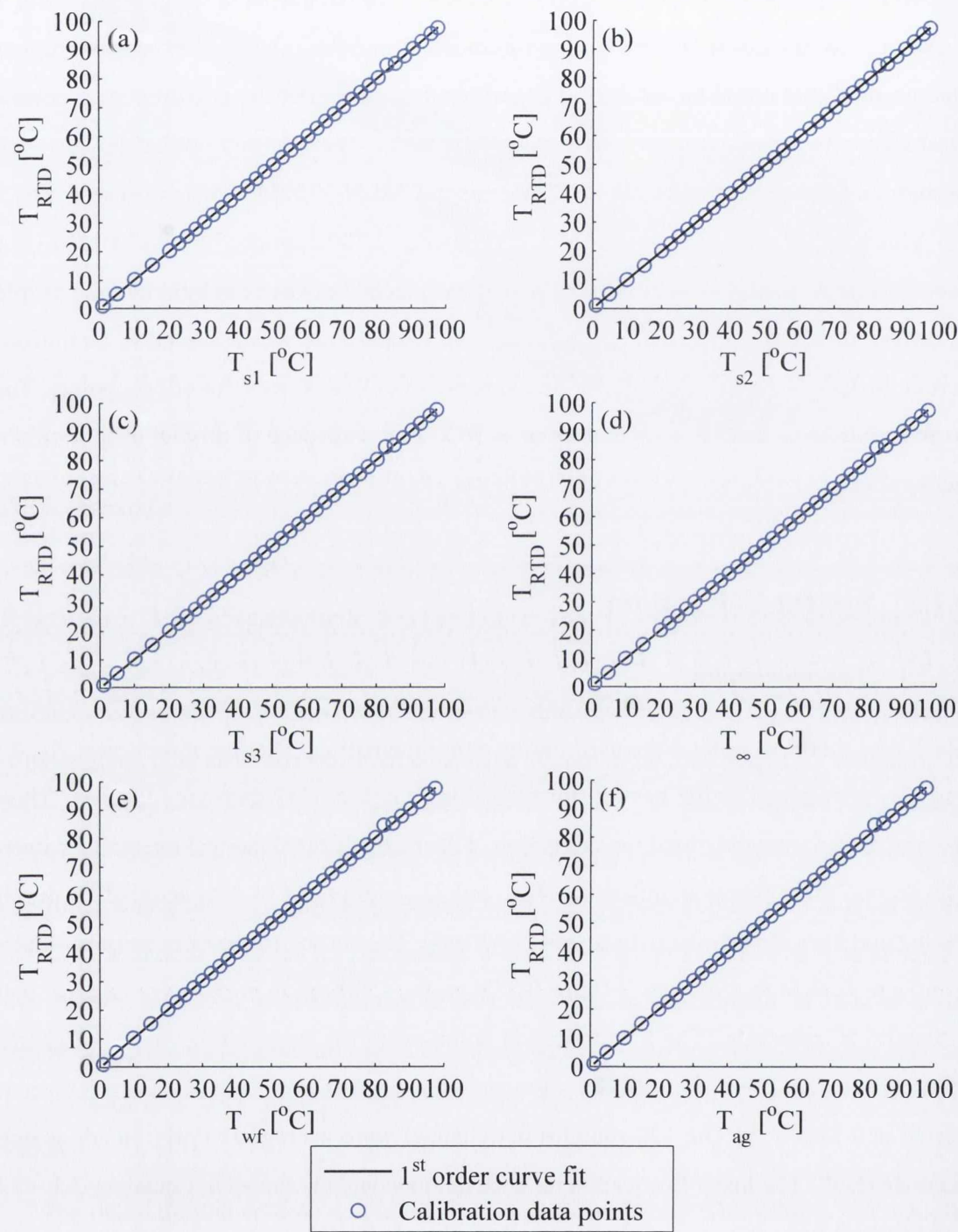


Figure 3.17: Thermocouple calibration.

$$T_{RTD} = 1.0001T_{s3} - 0.1054 \quad (3.3)$$

$$T_{RTD} = 1.10922T_{\infty} - 0.0793 \quad (3.4)$$

$$T_{RTD} = 1.0023T_{wf} - 0.0629 \quad (3.5)$$

$$T_{RTD} = 1.0020T_{ag} - 0.1302 \quad (3.6)$$

3.1.3.2 Thermal Imaging Camera

To correctly calibrate the FLIR SC6000 thermal imaging camera both a temperature calibration and a non-uniformity correction (NUC) for the IR sensor was required. This is due to the infrared focal plane arrays showing different responses of individual pixels, as well as accounting for measurement discrepancies introduced by the IR camera lens and CaF₂ window. During calibration the IR camera is mounted in its experimental testing position (Figure 3.1), looking through the IR window at the underside of the heated substrate. The NUC was applied by placing a flatfield black-body plate at two temperature directly in front of the camera lens ($\sim 70\%$ of the maximum temperature observed by the camera, and the other at room temperature) fully occupying its entire field of view. The ExaminIR software then calculates the gain, bad pixel, and offset correction for the sensor (.scg, .sbp and .sco), for the defined integration time and window size. When the correction is applied, each pixel should read the same value when a uniform temperature source is placed in front of the camera. Prior to calibration, the software asks for a values of atmospheric attenuation in the 3 – 5 μm range, which is important for correctly calculating surface radiance. During calibration the foil is uniformly heated to different temperature set points by adjusting the current on the DC power supply. A thermally insulative block is placed over the entirety of the foil during calibration to ensure a uniform foil temperature. This calibration method is known as an *in-situ* calibration, as presented by Schulz [24] and Ochs et al. [171], and ensures that the effect of the IR window and paint emissivity are accounted for. At each calibration point the surface temperature of the foil was allowed to reach steady state and then 20 temperature samples at 2 Hz were recorded per channel using LabVIEW and a NI-9214 DAQ connected to three T-type thermocouples glued to the heated substrate. In tandem the IR camera was

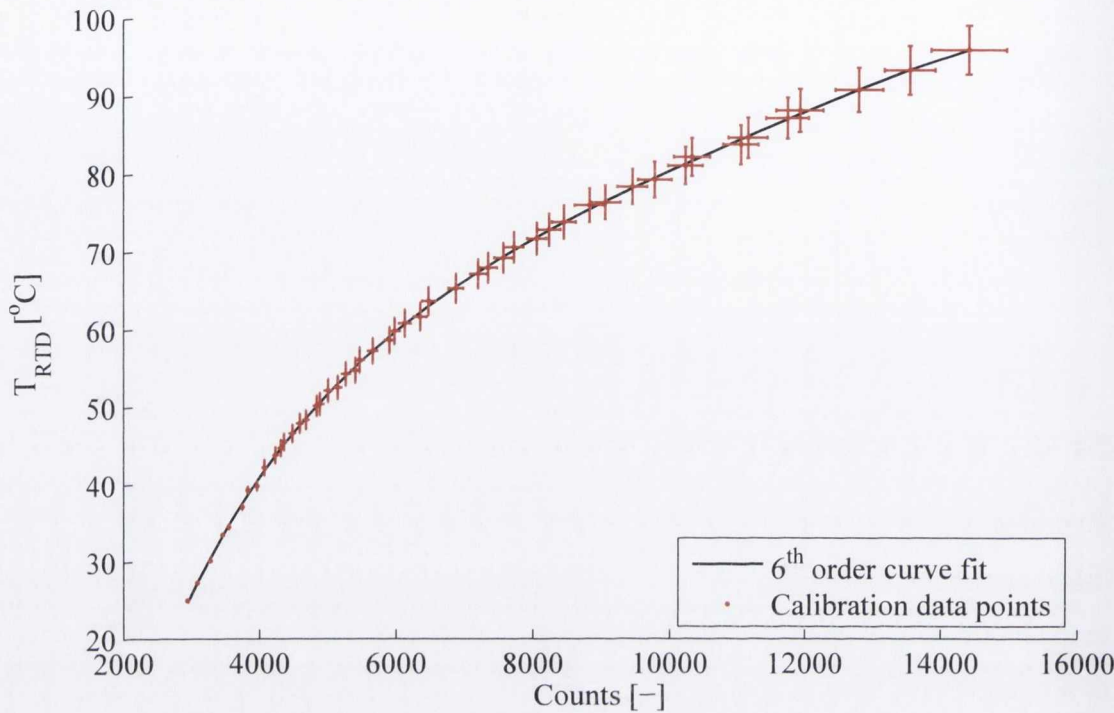


Figure 3.18: FLIR SC6000 calibration curve.

triggered recording 2000 images. The calibration ranged from 25 to 100 °C. Calibration points are taken at both 1 and 2.5°C intervals. The lower interval was implemented for temperature ranges of greater interest. It was then possible to directly convert the recorded count values to temperature values using the averaged values for temperature and counts in MATLAB. The 6th order fit is shown in Equation 3.7 and Figure 3.18.

$$\begin{aligned} T_s = & -1.01 \times 10^{-22} c^6 + 5.75 \times 10^{-18} c^5 \\ & - 1.37 \times 10^{-13} c^4 + 1.77 \times 10^{-9} c^3 - 1.36 \times 10^{-5} c^2 \\ & + 0.0649c - 84.44 \quad (3.7) \end{aligned}$$

3.1.3.3 Flow Rate Sensor

The Sensirion flow rate sensor was calibrated for ethanol using the KDS 200 dual syringe infusion pump and the Hamilton 1750 gastight syringe. The syringe pump was set to the

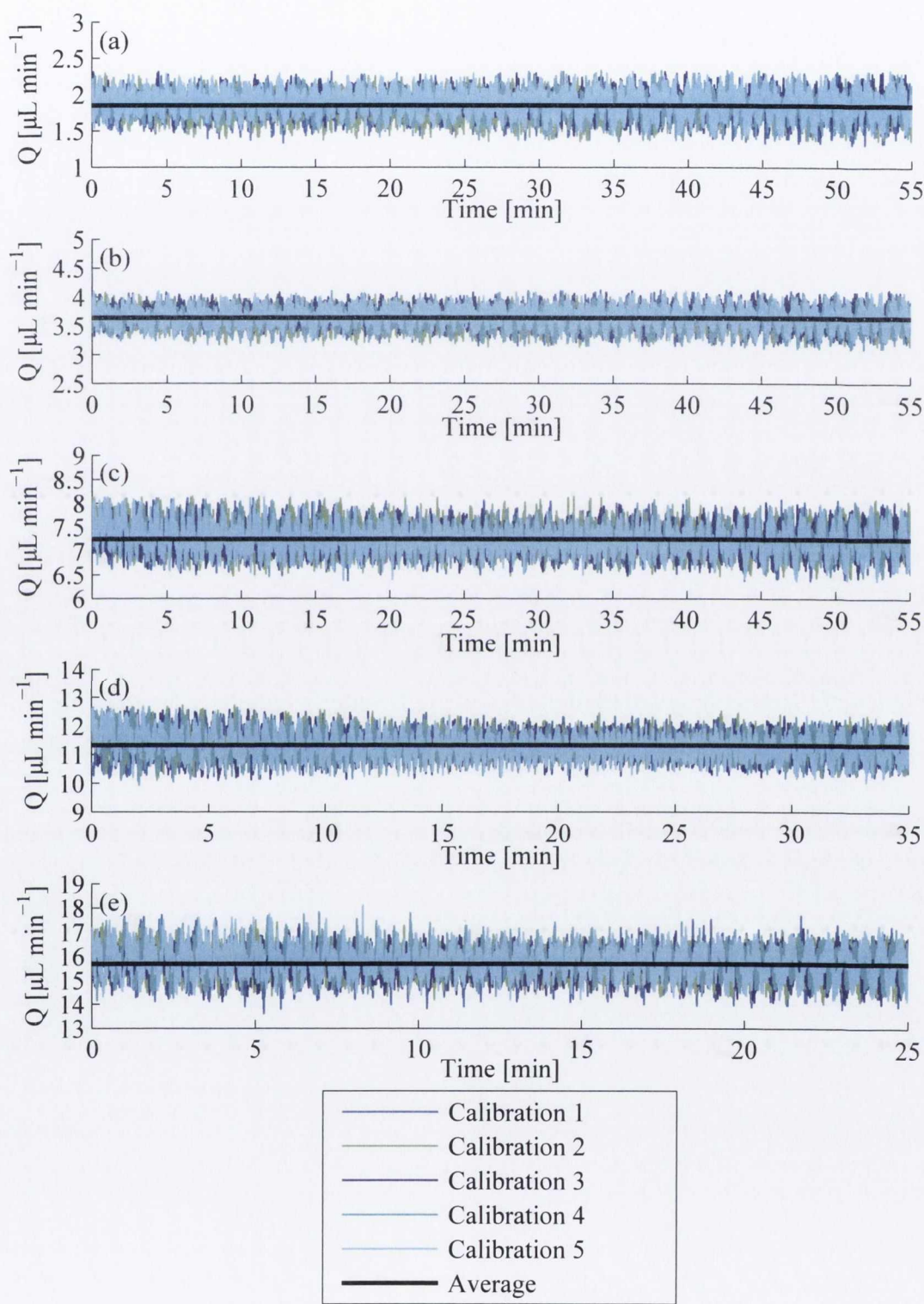


Figure 3.19: Flow sensor calibration - single source. (a) $2 \mu\text{L min}^{-1}$, (b) $4 \mu\text{L min}^{-1}$, (c) $8 \mu\text{L min}^{-1}$, (d) $12 \mu\text{L min}^{-1}$, (e) $16 \mu\text{L min}^{-1}$.

3.1. TEST FACILITY

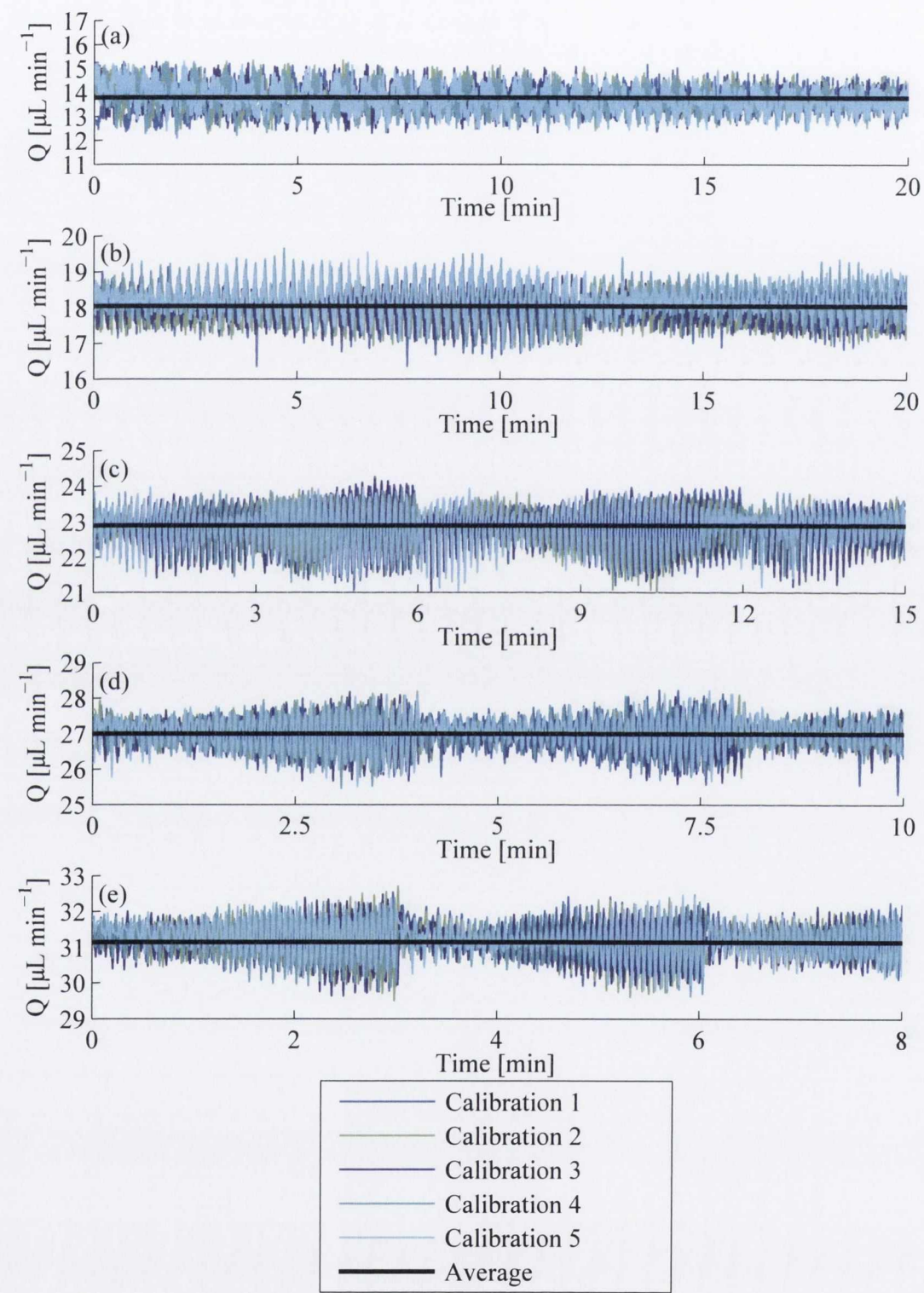


Figure 3.20: Flow sensor calibration - array. (a) $14 \mu\text{L min}^{-1}$, (b) $17.5 \mu\text{L min}^{-1}$, (c) $21 \mu\text{L min}^{-1}$, (d) $24.5 \mu\text{L min}^{-1}$, and (e) $28 \mu\text{L min}^{-1}$.

Table 3.7: Flow sensor calibration summary.

Flow rate [$\mu\text{L min}^{-1}$]	Sensor average reading [$\mu\text{L min}^{-1}$]	Expanded uncertainty, $U(y)$ [$\mu\text{L min}^{-1}$]
2	1.85	± 0.40
4	3.64	± 0.79
8	7.25	± 1.48
12	11.32	± 2.37
16	16.67	± 3.52
14	13.79	± 2.93
17.5	18.05	± 3.74
21	22.89	± 4.74
24.5	27.00	± 5.14
28	31.14	± 5.17

desired flow and a LabVIEW program was implemented to record the flow rate and ambient temperature conditions. The single source calibration is shown in Figure 3.19. Each flow rate was examined 5 times. Temperature and flow rate reading were recorded at 2 Hz. Fluid lines were primed prior to data acquisition. The sinusoidal nature of each plot is as a result of the screw based system employed by the syringe pump. Although the local flow rate varied over time it was assumed a constant volume was dispensed by the pump. Figure 3.20 shows the flow sensor calibration for the seven source array. Calibration was carried out in a similar manner to that of the single source. Each flow rate was examined 15 times. These 15 points were split into 5 groups of 3 points for presentation in Figure 3.20. Table 3.7 shows the the results of the calibration as well as the expanded uncertainty attached to each point.

3.2 Experimental Procedure

3.2.1 Study 1: Single Source Electrospray

Experimental test points were conducted in ambient surroundings and atmospheric pressure. Prior to testing the heated substrate temperature was set and allowed to reach steady-state. Fluid lines were primed with the working fluid ethanol. The experimental procedure for the single source electrospray was as follows:

1. Nozzle size, working fluid flow rate and source – target separation height were set.

3.2. EXPERIMENTAL PROCEDURE

Table 3.8: Single source electrospray sample rate and size.

Measurement	Sample rate [Hz]	Sample size [-]
IR camera	200	2000
Optical camera	15	150
Ambient temperature , T_{∞}	2	21
Working fluid temperature, T_{wf}	2	21
Air gap temperature, T_{ag}	2	21
Flow measurement, Q	2	20
Surface voltage drop, V_s	20	200
Surface current, I_s	20	200

2. Shut-off valve was opened and fluid was allowed flow from nozzle tip
3. Source voltage was adjusted to establish the cone-jet regime of spraying. Cone angle was as close to 49.3° without introducing asymmetric spraying. This cone formation zone was observed with the optical camera and the jet was illuminate by the Acculase laser.
4. System was allowed 15 minutes to settle and ensure steady state spraying conditions. Steady state was confirmed in the post processing through investigating the energy storage term in the foil and paint layers.
5. Data acquisition was initiated using the developed VI in labVIEW. The VI triggered recording of the IR camera, optical camera, temperature measurements, flow measurements, and surface voltage drop and current. A list of the sampling rates of each measurement are shown in Table 3.8. The sampling period lasted for ten seconds in total.

3.2.2 Study 2: Electrospray Array

Experimental test points were conducted in ambient surroundings and atmospheric pressure. Prior to testing the heated substrate temperature was set and allowed to reach steady-state. Fluid line, the array reservoir, and the source nozzles were primed with ethanol removing all air from the system. The experimental procedure for the electrospray array was as follows:

1. Working fluid flow rate and source – target separation height was set.

Table 3.9: Electrospray array sample rate and size.

Measurement	Sample rate [Hz]	Sample size [-]
IR camera	200	2000
Optical camera	15	150
Ambient temperature , T_{∞}	2	21
Working fluid temperature, T_{wf}	2	21
Air gap temperature, T_{ag}	2	21
Flow measurement, Q	2	20
Surface voltage drop, V_s	20	200
Surface current, I_s	20	200

2. Source and extractor voltages were set prior to opening the shut-off valve. This was done to avoid flooding of the jet forming region. A potential difference of $\Delta V = 1,050 - 1,100$ V was noted between the source and extractor for all test points.
3. Shut-off valve was opened and fluid was allowed flow from source nozzles.
4. The potential drop across source and extractor electrodes were adjusted to ensure the Taylor cone angle. Cone angle was as close to 49.3° without introducing asymmetric spraying. This cone formation zone was observed with the optical camera and the jet was illuminate by the diode laser.
5. System was allowed 15 minutes to settle and ensure steady state spraying conditions. Steady state was confirmed in the post processing through investigating the energy storage term in the foil and paint layers.
6. Data acquisition was initiated using the developed VI in labVIEW. The VI triggered recording of the IR camera, optical camera, temperature measurements, flow measurements, and surface voltage drop and current. A list of the sampling rates of each measurement are shown in Table 3.9. The sampling period lasts for ten seconds in total.

3.2.3 Study 3 and 4: Droplet Evaporation

Experimental test points were conducted in ambient surroundings and atmospheric pressure. The relative humidity during experimentation ranged between 35 – 45%.

3.2. EXPERIMENTAL PROCEDURE

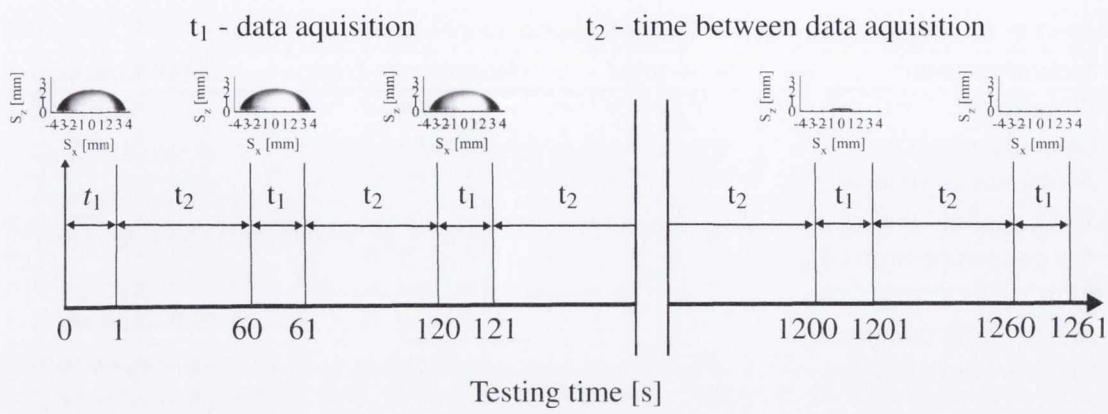


Figure 3.21: Droplet evaporation - data acquisition timeline

1. The 80 μL deionised water droplet was placed in the centre of the stainless steel substrate.
2. The electrode was positioned concentrically above the droplet at the 10 mm separation height.
3. A calibration image was captured to determine pixel size in post processing.
4. The start of testing was triggered using an in-house developed LabVIEW program. Once the VI was started it triggers the thermal imaging camera, optical camera and surface power supply, while simultaneously acquiring thermocouple, surface voltage drop and circuit current readings. Data was acquired for 1 second every 60 seconds for each sub-point. The total test time for all tests was 25 minutes for the uncoated case and 35 minutes for the coated case. This results in 26 and 36 sub-points per test, with

Table 3.10: Droplet evaporation sub-point sample rate and size.

Measurement	Sample rate [Hz]	Sample size [-]
IR camera	200	200
Optical camera ¹	2	31
Ambient temperature 1, $T_{\infty,1}$	2	3
Ambient temperature 2, $T_{\infty,2}$	2	3
Air gap temperature, T_{ag}	2	3
Surface voltage drop, V_s	20	21
Surface current, I_s	20	21

¹ Optical camera was continuously recording across test period.

each sub-point separated by 59 seconds. The acquisition timing sequence is outlined in Figure 3.21. All droplets were completely evaporated by the completion of each test. A list of the sampling rates and number of samples for an individual sub-point are shown in Table 3.10.

3.3 Data Analysis

3.3.1 Thermal Image Processing

The captured thermal images have a resolution of 400×400 pixels, an example of which is shown in Figure 3.22. These images are exported from FLIR ExaminIR as .sfmov video file. Each image pixel records a temperature intensity value in counts during data acquisition. As shown previously these count values have an associated temperature value. The processing

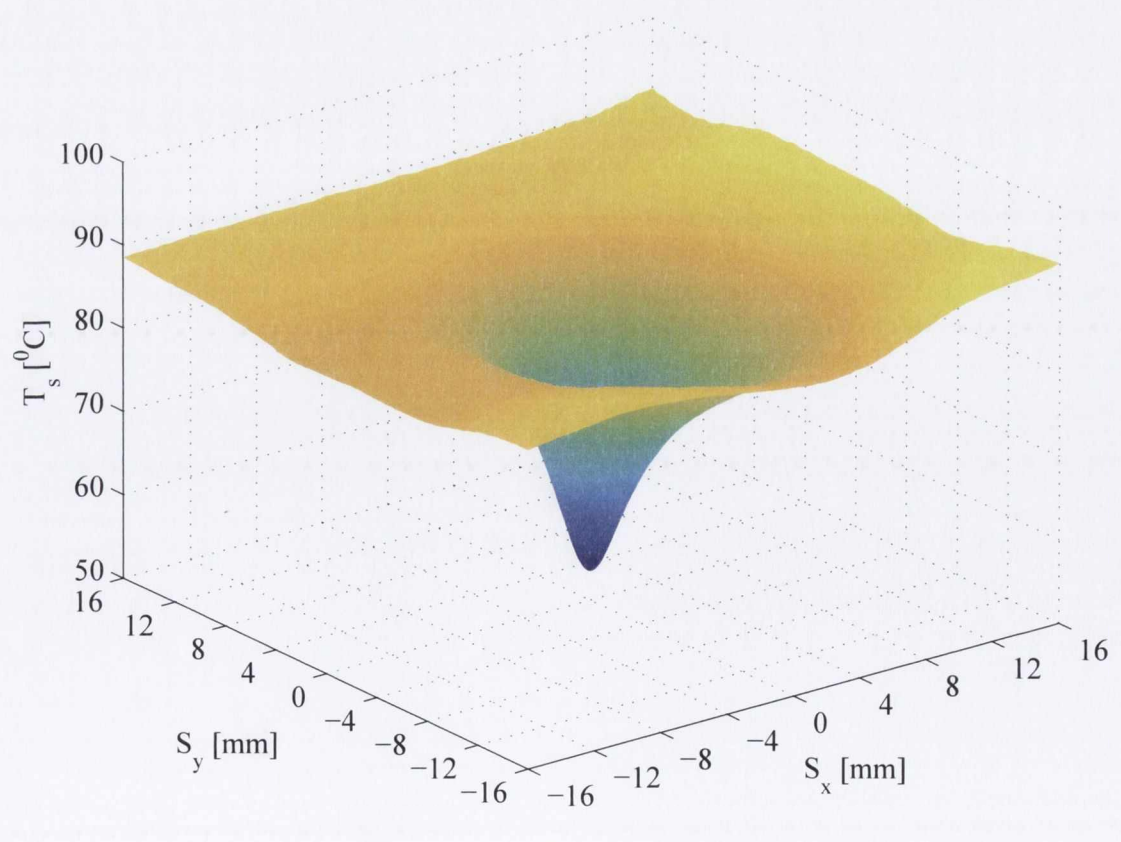


Figure 3.22: Heated substrate temperature distribution during single source electrospraying.

of these images and other recorded data was split into four phases; conversion from counts to temperature, heat transfer analysis, averaging and centering. All analysis was done using in-house developed MATLAB code.

3.3.1.1 Image Conversion

The .sfmov video file was imported into MATLAB and each image was extracted and the bad pixel, gain, offset and counts conversion are applied. This results in $2000\ 400 \times 400$ thermal images stored in a 3D array for the single source and array cases.

3.3.1.2 Heat Transfer Analysis

In order to compute the thermal energy convected by the heat flux dissipation method, various fluid, foil and paint properties (see Table 3.1) are input along with relevant calibration curves for the ambient air, air gap and reservoir temperatures. It was assumed that there was uniform heat generation within the foil and paint layers and that the temperature through the thickness of these layers was constant. In order to investigate the second assumption a simple lumped capacitance method was used by considering the Biot number, Bi . The Biot number is a ratio which compares the resistance to conduction within a solid, to the resistance to convection to the working fluid. If the Biot number (Equation 3.8) is very small ($Bi \ll 0.1$) then the temperature of the solid can be approximated as spatially uniform at any instant.

$$Bi = \frac{h_{max}L_c}{k} < 0.1 \quad (3.8)$$

where h_{max} is the maximum heat transfer coefficient, L_c is a characteristic length. For this case, of wall of thickness δ which is cooled on either sides, the characteristic length can be taken as $L_c = \delta/2$ [172]. From this and using the relevant material properties from Table 3.1, a Biot number of 5.08×10^{-4} and 3.66×10^{-2} was calculated for the foil and paint layers respectively.

Element-wise Energy Balance

In order to determine the thermal energy convected to the working fluid during testing an element-wise analysis was required. Incropera et al. [172] outlined an energy conservation

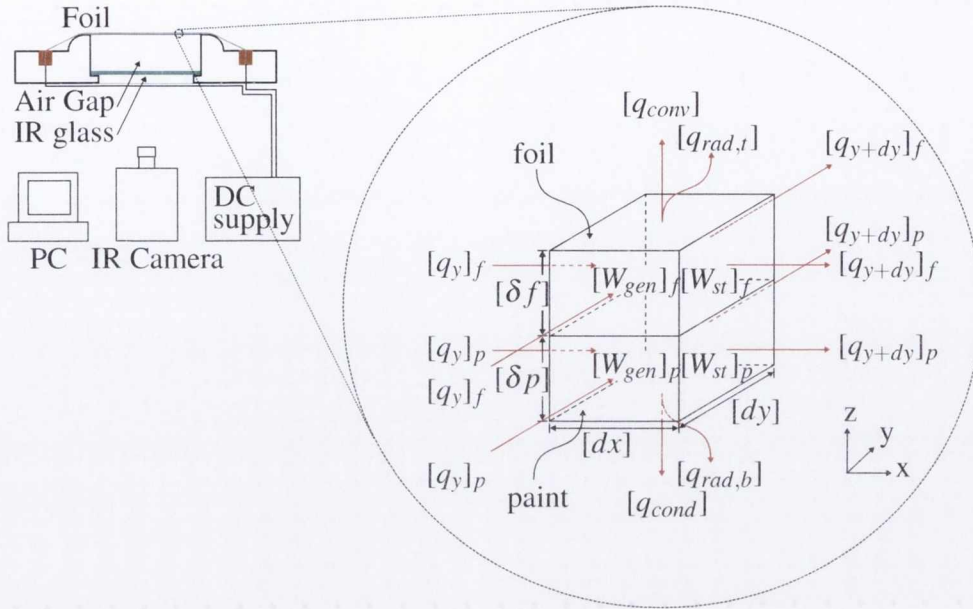


Figure 3.23: Thermal exchange surface heat transfer energy flow balance within a single $dx \times dy$ element of thickness δ_f and δ_p .

method shown in Equation 3.9

$$\dot{W}_{in} + \dot{W}_{gen} - \dot{W}_{out} = \Delta \dot{W}_{st} \quad (3.9)$$

where $\dot{W}_{in}$ and $\dot{W}_{out}$ refer into the energy transport in and out of the control section. $\dot{W}_{gen}$ and $\dot{W}_{st}$ are the generated and stored thermal energy in the control volume respectively. This energy balance for the differential control volume is depicted in Figure 3.23, where f and p denote the foil and paint respectively. For experiments performed under steady state conditions, the energy storage term, $\Delta \dot{W}_{st}$, is usually neglected. Combining similar terms the energy balance of the control volume can be expressed as:

$$\begin{aligned} q_{x,f} + q_{y,f} + q_{x,p} + q_{y,p} + \dot{q}_{gen} dx dy \delta_f - \\ q_{x+dx,f} + q_{y+dy,f} + q_{x+dx,p} + q_{y+dy,p} - \\ q_{conv} - q_{cond} - q_{rad} - q_{cap} = 0 \end{aligned} \quad (3.10)$$

where q_{conv} represents the convective heat transfer to the working fluid, q_{cond} is the conductive heat loss from the underside of the thermal exchange surface to the air gap and q_{rad} is the total

3.3. DATA ANALYSIS

radiative heat transfer from the upper and lower surfaces of the thermal exchange surface. The q_{cap} term describes the change in internal energy within the control volume in response to temporal fluctuations in material temperature. This unsteady term describes the heat stored in the foil and paint layers.

The conduction heat flux through the control volume edges at x and y , for both layers, can be calculated using Fourier's law:

$$q_x = -k\delta dy \frac{\partial T}{\partial x} \quad (3.11)$$

$$q_y = -k\delta dx \frac{\partial T}{\partial y} \quad (3.12)$$

Similarly, the conduction heat flux on the opposite side of the control volume can be determined as:

$$q_{x+dx} = q_x + \frac{\partial q_x}{\partial x} dx \quad (3.13)$$

$$q_{y+dy} = q_y + \frac{\partial q_y}{\partial y} dy \quad (3.14)$$

Substituting these equations into the energy balance equation 3.10 gives:

$$\begin{aligned} (k_f\delta_f + k_p\delta_p) \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) dxdy + \\ \dot{q}_{gen}\delta_f dxdy - q''_{conv}dxdy - q''_{nc}dxdy - \\ q''_{rad,total}dxdy - q''_{cap}dxdy = 0 \end{aligned} \quad (3.15)$$

Equation 3.15 assumes a constant thermal conductivity, along with a constant thickness, δ . Rearranging the terms one can solve for the heat flux dissipated by cooling, q''_{conv} , which has units of $W m^{-2}$:

$$q''_{conv} = q''_{gen} - q''_{cond} - q''_{rad} - q''_{cap} + (k_f\delta_f + k_p\delta_p) \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (3.16)$$

In this study the control volume element was square, such that dx was equal to dy . The heat generated within each element by Joule effect and was denoted by:

$$q''_{gen} = \frac{I_s V_s}{A_s} \quad (3.17)$$

where I_s is the current and V_s is the voltage drop across the heated substrate of surface area, A_s . To account for the conduction through the 7 mm air gap, Fourier's law was applied. Bergman et al. [172] noted that, in the absence of radiation, for heating from above, heat transfer from the heated substrate to the IR window was exclusively by conduction ($Nu_L = 1$), irrespective of the value of Ra_L . The heat transfer is given by:

$$q''_{cond} = -k_{\infty} \frac{dT}{dz} \quad (3.18)$$

where k_{∞} is the thermal conductivity of the bulk fluid (air) and dT/dz is the temperate gradient across the air gap to the IR window. The heat lost through radiation from the bottom surfaces of the heated substrate to the top surface of the IR window is given by:

$$q''_{rad,b} = \frac{\sigma (T_s^4 - T_{CaF_2}^4)}{1/\epsilon_p + 1/\epsilon_{CaF_2} - 1} \quad (3.19)$$

where ϵ is the surface emissivity, σ is the Stefan-Boltzmann constant. The heat transfer by lateral conduction in the foil and paint layers, q''_{lc} , comes from spatial derivative terms in the energy balance, given by:

$$q''_{lc} = (k_f \delta_f + k_p \delta_p) \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} \right) \quad (3.20)$$

The partial derivative is also known as the Laplacian, $\nabla^2 T$. This partial derivative was solved by means of a central-difference approach using the inbuilt `de12` MATLAB function. Finally, the time-fluctuating capacitive effects associated with the foil and paint layer, q''_{cap} , are determined by equation 3.21:

$$q''_{cap} = (\rho_f C_{p_f} \delta_f + \rho_p C_{p_p} \delta_p) \frac{\partial T_s}{\partial t} \quad (3.21)$$

3.3. DATA ANALYSIS

where ρ and C_p are the density and the specific heat capacity of an element, and $\partial T_s / \partial t$ is the change in temperature of the heated substrate with respect to time. This was solved using the MATLAB function `gradient`. To calculate the change in internal energy storage, three successive thermal images are required.

3.3.1.3 Centering and Averaging

A search algorithm finds the centre of the spray plume or droplet by analysing the sum of the recorded temperatures and the calculated heat flux distribution. Using the data found by the search, the 400×400 pixel thermal image is then translated into an empty 500×500 array with the centre of the cooling now located at the centre of the array. All images are then averaged. This is done so that radial averaging can be carried out at a later stage in the processing.

3.3.1.4 Thermal Set-up Validation

A base natural convection case was investigated in order to validate the thermal experimental set-up. The foil was heated and allowed to reach steady state conditions with only natural convection cooling applied to the top surface of the foil. 2000 images were captured using the SC6000 camera and the data was processed using Equations 3.9 – 3.21. Figure 3.24 shows the thermal losses from the foil and paint layers of the heated substrate. The radiation from the top surface of the heated substrate to surrounding, $q''_{rad,t}$, was calculated from [172]:

$$q''_{rad,t} = \sigma \epsilon_f (T_s^4 - T_\infty^4) \quad (3.22)$$

where ϵ_f is the emissivity of the foil top surface. While Figure 3.25 highlights the heat flux dissipated and heat transfer coefficient due to natural convection from the top surface of the heated substrate. A central $20 \times 20 \text{ mm}^2$ portion of the heated substrate is shown in both Figure 3.24 and 3.25 for all terms. The average value of each term and twice their associated expanded uncertainty is listed in Table 3.11. This value for the heat transfer coefficient h_{nc} was found using the formula [172]:

$$h_{nc} = \frac{q''_{nc}}{T_s - T_\infty} \quad (3.23)$$

This experimental value of h was compared with that calculated using Equations 3.24 – 3.26 [172]:

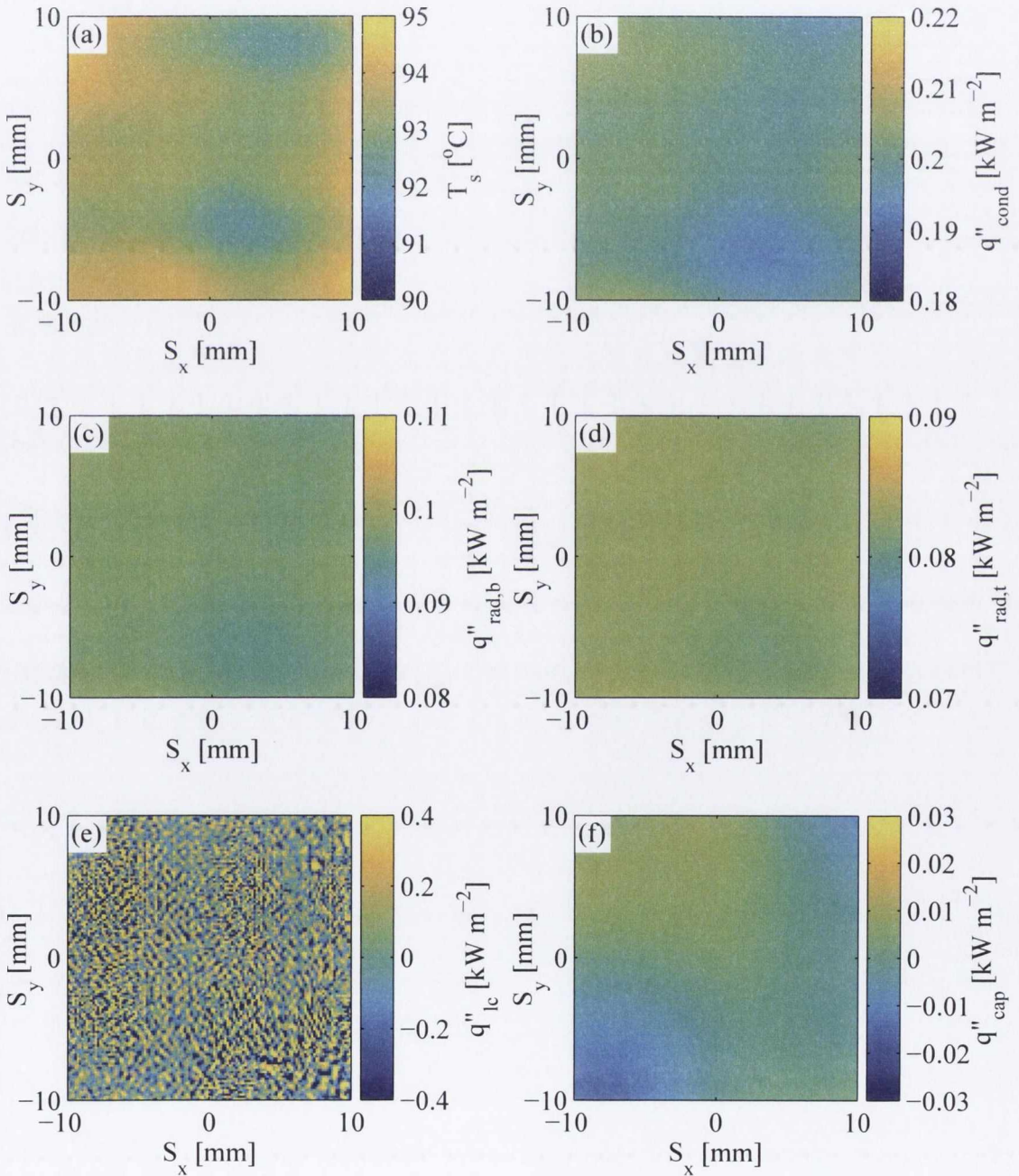


Figure 3.24: Thermal set-up validation. (a) T_s , (b) q''_{cond} , (c) $q''_{rad,b}$, (d) $q''_{rad,t}$, (e) q''_{ic} , and (f) q''_{cap} .

3.3. DATA ANALYSIS

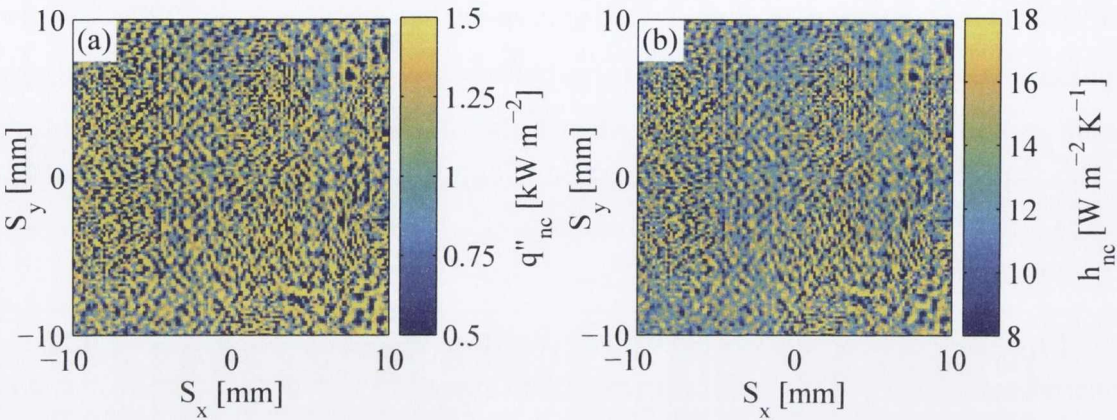


Figure 3.25: Thermal set-up validation. (a) q''_{nc} and (b) h_{nc} .

$$Ra_L = \frac{g\beta(T_s - T_\infty)L_f^3}{\nu\alpha_t} \tag{3.24}$$

$$\overline{Nu}_L = 0.54Ra_L^{0.25} \tag{3.25}$$

$$\overline{h}_L = \frac{k\overline{Nu}_L}{L_f} \tag{3.26}$$

where Ra_L is the Rayleigh number, β is the volumetric thermal expansion coefficient, α_t is the thermal diffusivity, L_f is the characteristic length of the heated substrate, $L_f = A_s/p_s$,

Table 3.11: Thermal set-up validation, individual terms and their 95% coverage.

Term	Average value	Unit	Expanded uncertainty
T_s	92.9	$^{\circ}\text{C}$	± 0.77
q''_{gen}	1,395.2	W m^{-2}	± 5.63
q''_{cond}	197.62	W m^{-2}	± 0.34
$q''_{rad,b}$	96.35	W m^{-2}	± 0.21
$q''_{rad,t}$	81.97	W m^{-2}	± 0.22
q''_{lc}	-5.87	W m^{-2}	± 173.80
q''_{cap}	2.39	W m^{-2}	± 0.57
q''_{nc}	1008.3	W m^{-2}	± 174.53
h_{nc}	12.64	$\text{W m}^{-2} \text{K}^{-1}$	± 2.14

$\overline{Nu}_L$ is the Nusselt number. The theoretical heat transfer coefficient $\bar{h}_L$ was calculated to be $11.7 \text{ W m}^{-2} \text{ K}^{-1}$. This calculation assumes that ambient air is quiescent, surface radiation effects are negligible, that the surround medium behaves like an ideal gas and all thermophysical properties are constant. The theoretical value is 9.5% smaller than the experimental value for h_{nc} but is within expanded uncertainty, $EU = \pm 2.14 \text{ W m}^{-2} \text{ K}^{-1}$, found by applying the formulae and methodology outlined in Section 3.3.3.

Finally these results were compared with that calculated for the COMSOL model outlined in Appendix A. A natural convection boundary condition applied to the top surface of the foil and all external boundaries. An internal heat source of $1,395 \text{ W m}^{-2}$ was applied to the foil domain. A resultant foil temperature of 95.15°C was found from the computed model. This value shows good agreement with that observed experimental data. In conclusion good agreement of experimental, theoretical and numerical data is observed. These results verify the validity of the thin foil technique employed in this work.

3.3.2 Optical Image Processing

The captured optical data has a resolution of 1280×960 pixels. All images were recorded in .tif format to preserve image quality. Images are processed using in-house developed

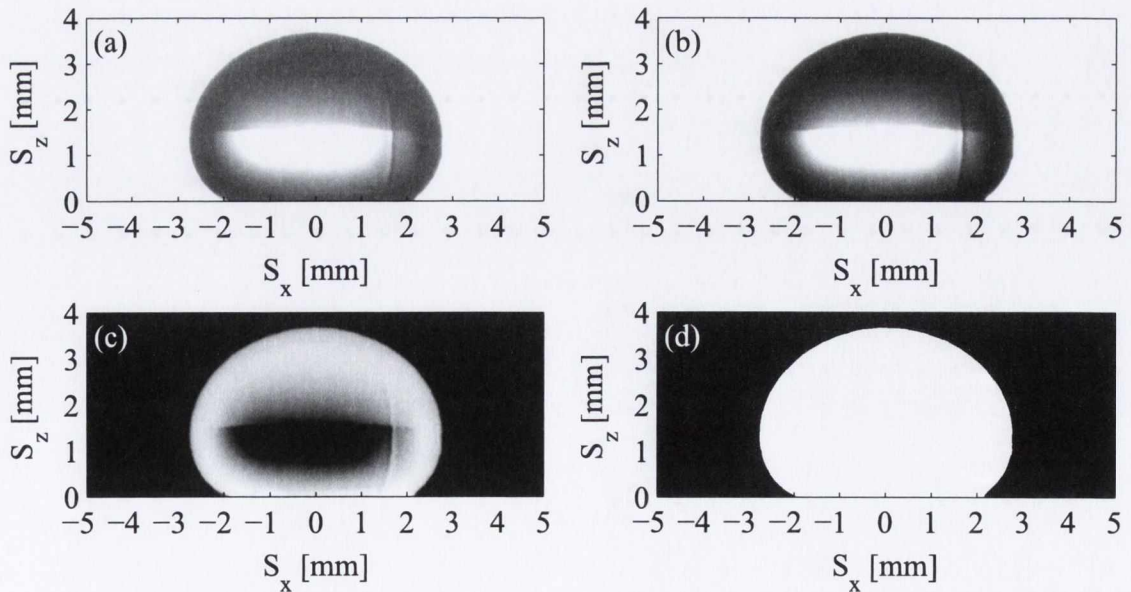


Figure 3.26: Droplet optical image processing. (a) raw image, (b) grayscale, (c) subtracted background, and (d) binarised, sensitised, and filled

MATLAB code. Image processing was broken down into three phases: boundary detection and filtering, image analysis, force analysis.

3.3.2.1 Boundary Detection and Filtering

An example of a droplet image is shown in Figure 3.26a. The captured image was converted to grayscale (Figure 3.26b) and the background was subtracted (Figure 3.26c). Finally the image was binarised, sensitised, and filled (Figure 3.26d). Once the droplet shape has been determined the boundary was traced and the pixel size was applied to correctly scale the image. The resultant boundary is shown in Figure 3.27 and was the average of the left and right sides of the droplet. The detected boundary was then broken up into 200 evenly spaced segments and filtered using a Savitzky - Golay filter to smooth the droplet boundary removing noise. Comparisons of the raw and filtered boundaries are shown in Figure 3.27.

3.3.2.2 Image Analysis

From the filtered boundary a number of key parameters such as; droplet height H_d , width D_b , volume V_d , centroid, contact angle, and radii of curvature can be calculated. These parameters

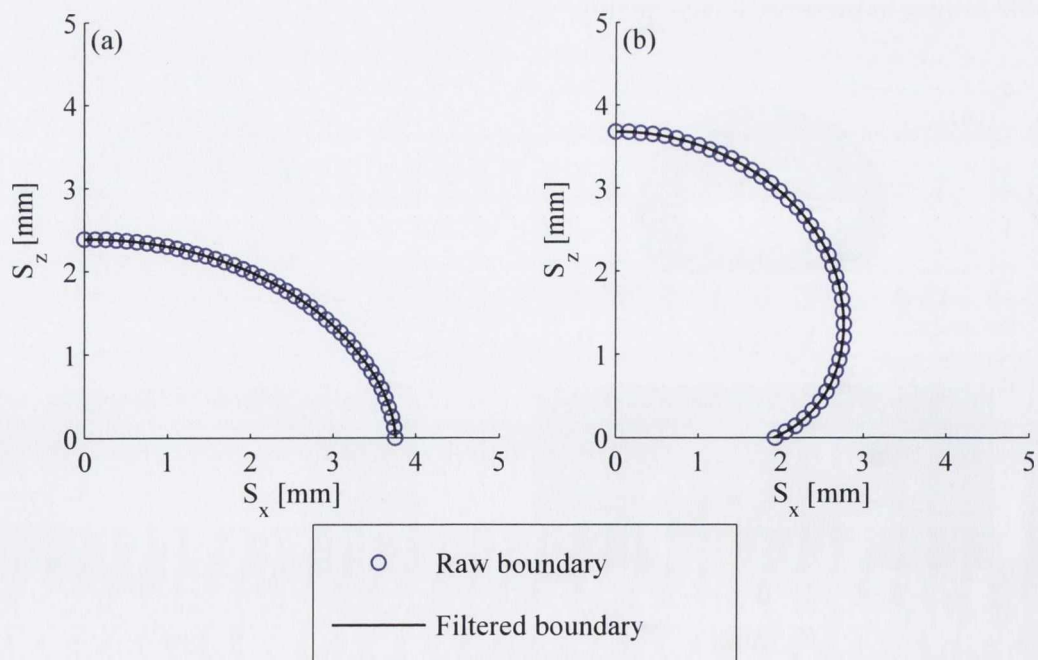


Figure 3.27: Droplet interface filtering. (a) partial wetting and (b) partial non-wetting.

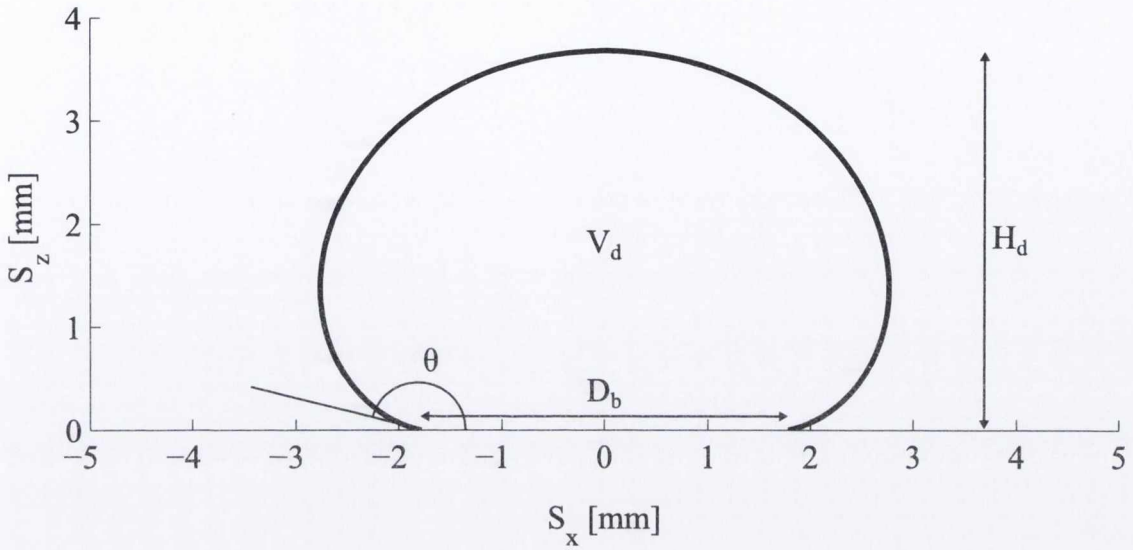


Figure 3.28: Schematic of droplet parameters.

are shown in Figure 3.28. Figure 3.29a, b and c illustrates the method in determining the droplet contact angle. The droplet contact angle was determined by using the three points on the droplet interface adjacent to the base of the droplet (Figure 3.29a, points 1, 2, and 3). The centre of a circle which passes through these three points (Figure 3.29c, point c) was found by bisecting points 1 – 2 (Figure 3.29b, point m_1) and 2 – 3 (Figure 3.29b, point m_2) and solving where the respective lines of bisection intersect (Figure 3.29c, point c). A line was then connected between the circle centre point and the point on the interface at the base of the droplet (Figure 3.29a, point 1). The slope of the line perpendicular to this was the droplet contact angle. The method of determining the local principal radii (R_1 and R_2) is shown in Figure 3.29d, e, and f. Using a central and adjacent points (Figure 3.29d, points i , $i+1$, and $i-1$) the first principal radius, R_1 , was calculated in a similar manner to that of the contact angle. First the bisecting points of $i-1 - i$ (Figure 3.29e, point m_1) and $i - i+1$ (Figure 3.29b, point m_2) are computed. The point of intersection between the perpendicular lines from these bisecting points gives the centre of a circle (Figure 3.29f, point c) whose arc passes through i , $i+1$, and $i-1$. The radius of this circle is the first principal radius (Figure 3.29f, R_1). As a result of the axisymmetric assumption the second principal radius, R_2 , must lie on the z - axis and is found where the plane perpendicular to that which R_1 lies on (Figure 3.29f, points i and c) intersects with the z - axis (Figure 3.29f, o). From these two principal radii one is able to determine the local curvature of the droplet, C , which is given by:

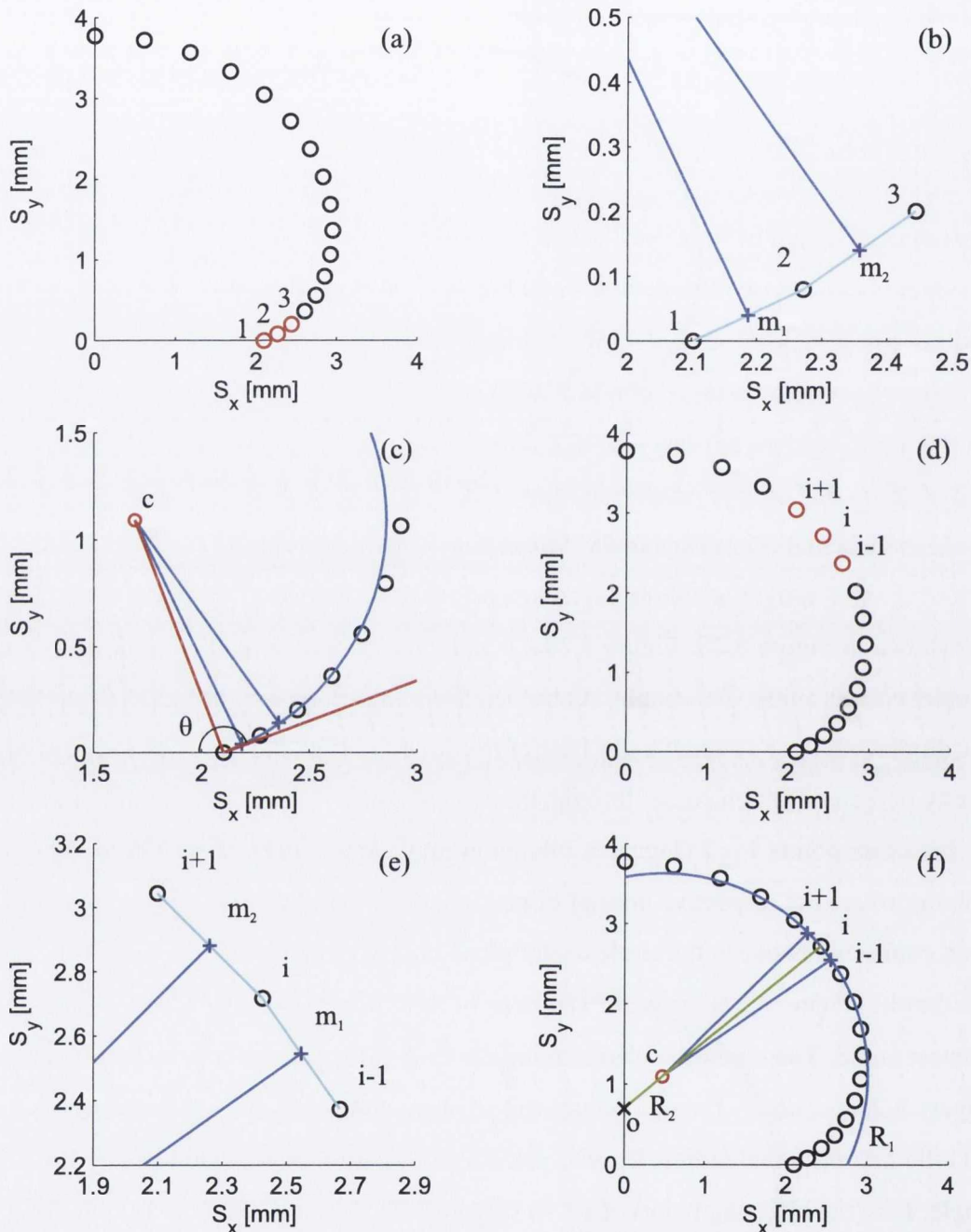


Figure 3.29: Droplet contact angle and curvature calculation.

$$2C = \frac{1}{R_1} + \frac{1}{R_2} \quad (3.27)$$

This curvature can be used to determine the pressure difference across a gas liquid interface using the Young - Laplace equation.

$$\Delta P = \gamma 2C \tag{3.28}$$

3.3.2.3 Force Analysis

The stresses acting on an evaporating droplet dictates its shape and motion. Analysis of the droplet force balance during evaporation enables one to quantify the impact of an applied electric, as is applied in Chapter 7, and can thus be linked to any change in heat and mass transfer from the solid substrate. Two techniques can be applied when analysing the forces acting on an evaporating droplet; the first is a bulk force analysis and the second is a local force analysis.

Bulk Analysis

A bulk force analysis is key in understanding the droplet dynamics and evaporation process. A number of assumptions are made when characterising the forces acting on the evaporating droplet.

- The evaporation process was axisymmetric.
- Forces due to momentum, surface tension gradients, liquid inertia and added mass are negligible.
- Fluid and air properties are evaluated using data acquired during thermal data acquisition. It was assumed that these properties remain constant between thermal test

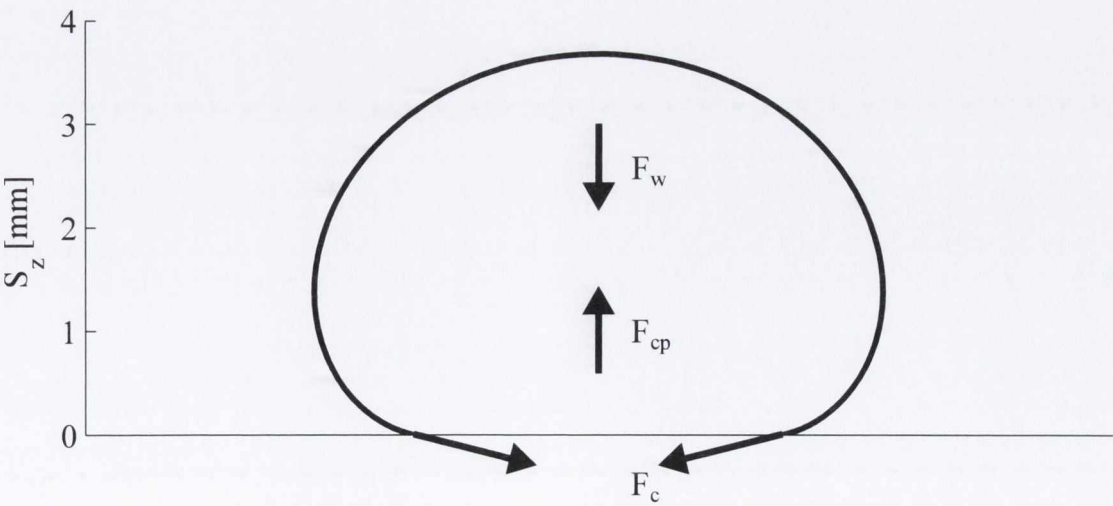


Figure 3.30: Droplet bulk force balance.

points.

The momentum conservation law states that for a given control volume the sum of the external forces were equal to the variation of momentum of that control volume. These bulk force are illustrated in Figure 3.30. The momentum variation provides information about the change of mass of a control volume or the motion of its centre of gravity, but does not account for change of droplet morphology if the centre of gravity remains fixed [173]. The integral form of the momentum equation considered in this study is given by:

$$\frac{d}{dt} \iiint_V \rho_l \vec{u}_l dV = \vec{F}_{cp} + \vec{F}_w + \vec{F}_c \quad (3.29)$$

with $\vec{F}_{cp}$, $\vec{F}_w$, $\vec{F}_c$ the resultant of the contact pressure, weight and surface tension and adhesion force. The contact pressure force considers the pressure difference across the liquid - solid interface and is given by [173]:

$$\vec{F}_{cp} = \left(\frac{2\gamma}{R_T} + (\rho_l - \rho_g)gH_d \right) S_b \vec{k} \quad (3.30)$$

where R_T is the radius of curvature of the droplet at its apex, g is gravity, and S_b is the contact area of the droplet base with the solid surface. The weight force can be considered as the resultant of the hydrostatic pressure force acting on the droplet surface and is given by [173]:

$$\vec{F}_w = -(\rho_l - \rho_v)V_d g \vec{k} \quad (3.31)$$

Droplet forces at the triple line are very complex and are governed by the surface tension between the liquid - gas phase and the wettability of the solid surface. This term can be expressed by [173]:

$$F_c = -D_b \pi \gamma \sin \theta \quad (3.32)$$

Thus the momentum integral equation can be written as:

$$\frac{d}{dt} \iiint_V \rho_l \vec{u}_l dV = \vec{F}_{cp} + \vec{F}_w + \vec{F}_c + \vec{\epsilon}_{res} \quad (3.33)$$

where $\vec{\epsilon}_{res}$ is residual error in evaluating the momentum integral.

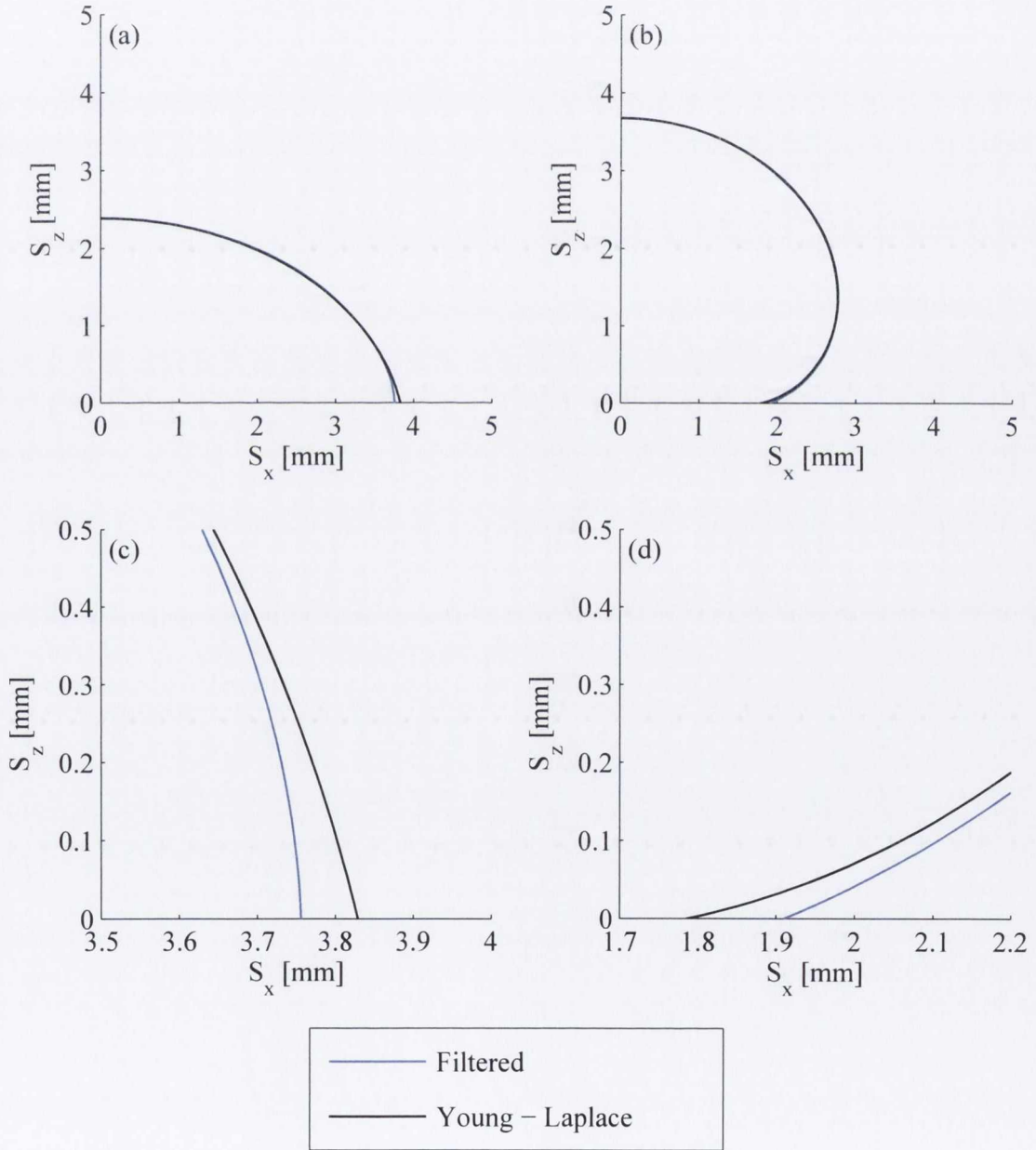


Figure 3.31: Filtered boundary compared with Young - Laplace solution. (a) and (c) partial wetting, (b) and (d) partial non-wetting droplet.

Local Analysis

The local mechanical equilibrium between each point of the droplet liquid - gas interface can be expressed by the Young - Laplace equation. Accounting for only hydrostatic effects this equation is given by:

$$2C = \frac{2\gamma}{R_T} - g(\rho_l - \rho_v)(z - z_0) \quad (3.34)$$

where z_0 is the reference height, usually taken as the droplet height and z is height of point on the interface being evaluated. This equation evaluates the balance between the pressure difference across the interface and the stress in the interface, expressed as the product of the surface tension and the curvature. Due to droplet reflections, shadowing and mirage effect arising from the evaporating droplet at the base, the droplet shape near the triple line can be distorted. For the hydrostatic only case it is possible, from experimental data, to calculate the droplet shape according to the Young-Laplace equation. Similar to Siedel et al. [173] an iterative method can be employed to calculate the droplet interface. The method requires the input of the apex curvature, droplet height and droplet volume. The equation was solved for the initial apex curvature and the volume of the produced Young - Laplace curve was compared with that of the experimental data. The tip curvature was then adjusted based on this volume ratio. This process continues until the Young - Laplace solution volume converges on that of the experimental data. A $\pm 0.01\mu\text{L}$ threshold was employed. As the experimental boundary may be distorted at the base only the volumes of the upper 95% of the Young-Laplace and experimental boundaries are compared. A comparison of experimental data and the boundary calculated from the Young-Laplace equation for both the partial wetting and partial non-wetting droplet cases are shown in Figure 3.31.

3.3.3 Uncertainty Analysis

The following section will outline the equations and methodology utilised in determining the uncertainty associated with all experimental measurements and their subsequent results. In the context of a measurement, an error is defined as the difference between the measured value and the true value. A measurement error can be split into two parts; random and

systematic error [174].

3.3.3.1 Systematic Errors

A systematic error is an error that remains fixed when the measurement is repeated under the same conditions, such as a constant offset in a measured quantity. Systematic errors cannot be diminished by large sampling and averaging. The amount by which the value differs from the true value may be a constant or may be proportional to the magnitude of the quantity being measured. A systematic error, also known as a Type B error, can be found by two different approaches. The first is a review of a previously obtained calibration report. The second is to change the experimental setup to measure the same quantity but by a different means or a known quantity by the same means.

Single Measurements

A single measurement accounting for systematic error can be written as;

$$M = x + Z \quad (3.35)$$

where M is the true value of the quantity being measured, x is the recorded value and Z is the best estimate of the correction which accounts for the systematic errors. This best estimate of the offset depends on several quantities such as calibration error and resolution error. The standard uncertainty in the offset, $u(Z)$, due to a resolution error is given by;

$$u(Z) = \frac{a}{\sqrt{3}} \quad (3.36)$$

The proportional uncertainty $pu(Z)$ is a ratio of the resolution error compared to the actual measurement, and is found by:

$$pu(Z) = \frac{a}{x} \times 100 \quad (3.37)$$

The degree of freedom ν associated with this single measurement can then be found from:

$$v = \frac{0.5}{\sqrt{pu(Z)}} \quad (3.38)$$

3.3.3.2 Random Errors

A random error, also known as a Type A error, is variation in a measurement despite consistent experimental conditions. Random errors occur due to our inability to completely control the measurand or its surroundings. Random errors can be reduced by taking numerous samples and averaging, if sufficient samples are recorded a zero net error can be assumed to a high degree of confidence.

Multiple Measurements

The standard deviation s of a range of measurements is given by;

$$s = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n - 1}} \quad (3.39)$$

where n is the number of measurements taken, x_i is a specific i th measurement and $\bar{x}$ is the mean of the n measurements. The standard uncertainty, $u(\bar{x})$, of the mean, assuming uncorrelated measurements, is given by;

$$u(\bar{x}) = \frac{s}{\sqrt{n}} \quad (3.40)$$

The degree of freedom of this measurement again assuming uncorrelated measurements is found from:

$$v = n - 1 \quad (3.41)$$

3.3.3.3 Correlation

Correlations describe the dependency that one data point x has on another data point y . Autocorrelation is often used for time varying points, and crosscorrelation for data points captured simultaneously. The linear correlation coefficient r_{cor} is defined as [174]:

$$r_{cor} = \frac{\text{covariance}(x,y)}{s(x)s(y)} \quad (3.42)$$

where the $\text{covariance}(x,y)$ is given by:

$$\text{covariance}(x,y) = \frac{\sum_{i=1}^n (x_i - \bar{x})(y_i - \bar{y})}{n - 1} \quad (3.43)$$

Uncorrelated data will have a r_{cor} close to zero and perfectly correlated $|r_{cor}| = 1$. During data processing autocorrelation of the thermal data through time was calculated in determining the uncertainty of the q''_{cap} term and a crosscorrelation of the thermal data temperature field was applied to determine the uncertainty attached to the q''_{lc} term.

3.3.3.4 Combined Uncertainty

The standard uncertainty in combined measurements $u(y)$ between inputs is given by:

$$u(y) = \left(\left(\frac{\partial y}{\partial x_1} \right)^2 u^2(x_1) + \left(\frac{\partial y}{\partial x_2} \right)^2 u^2(x_2) + \dots + \left(\frac{\partial y}{\partial x_n} \right)^2 u^2(x_n) \right. \\ \left. + 2r_{cor}(x_1, x_2) \frac{\partial y}{\partial x_1} \frac{\partial y}{\partial x_2} u(x_1)u(x_2) + 2r_{cor}(x_1, x_3) \frac{\partial y}{\partial x_1} \frac{\partial y}{\partial x_3} u(x_1)u(x_3) + \dots \right. \\ \left. 2r_{cor}(x_i, x_j) \frac{\partial y}{\partial x_i} \frac{\partial y}{\partial x_j} u(x_i)u(x_j) + \dots \right)^{\frac{1}{2}} \quad (3.44)$$

where the measurand y is the result of multiple individual measurements and there own attached uncertainty. The uncertainty in the combined measurement consists of the the partial differential $\left(\frac{\partial y}{\partial x_i} \right)$ of the combined measurement with respect to the individual measurement

3.3. DATA ANALYSIS

Table 3.12: Experimental uncertainty 95% coverage factor, c as a function of the number of degrees of freedom, ν . Adapted from Kirkup & Frenkel [174].

Degrees of Freedom, ν	Coverage factor, c	Degrees of Freedom, ν	Coverage factor, c
2	4.3	14	2.14
3	3.18	15	2.13
4	2.78	16	2.12
5	2.57	17	2.11
6	2.45	18	2.1
7	2.36	19	2.09
8	2.31	20	2.09
9	2.26	30	2.04
10	2.23	40	2.02
11	2.2	50	2.01
12	2.18	100	1.98
13	2.16	∞	1.96

being examined and the standard uncertainty in that measurement $u(x_i)$. This methodology can also be used in combining uncertainties from both Type A and Type B components of uncertainties. The degree of freedom of this measurand is called the effective degree of freedom ν_{eff} and is given by;

$$\nu_{eff} = \frac{u^4(y)}{\sum_{i=1}^n \frac{\left(\frac{\partial y}{\partial x_i}\right)^4 u^4(x_i)}{\nu_i}} \tag{3.45}$$

3.3.3.5 Coverage Factor

The coverage factor c is the interval within which the true value of the measurand is located with high probability, normally 95%. It is determined using the degrees of freedom of the measurement and Table 3.12. The results of Table 3.12 are derived from Student’s t-distribution [174].

3.3.3.6 Expanded Uncertainty

The expanded uncertainty $U(y)$ is given by the coverage factor multiplied by the standard uncertainty.

$$U(y) = cu(y) \tag{3.46}$$

This result indicates that the range of values that the experimental result should exist within a 95% level of confidence. For the case where y is the reported measurement value, this value and its associated expanded uncertainty can be written as;

$$y \pm U(y) \tag{3.47}$$

3.3.3.7 Study 1: Single Source Electrospray Cooling

The experimental uncertainty for all parameters was calculated using the outlined methodology using code developed with MATLAB. A list of the relevant parameter and their associated expanded uncertainty (EU) and percentage uncertainty (PU) is outlined in Table 3.13. All listed values are to a 95% confidence level. First the standard uncertainty (SU) of the acquired test data was determined. A combined uncertainty approach was then applied on a pixel by pixel basis to Equation 3.16 - 3.21. The uncertainty of the foil surface temperature is the experimental standard deviation of the mean. This approach was taken due to the high autocorrelation value attached to sequential T_s values. This uncertainty was determined from the calibration curve fit, accounting for the uncertainty attached to the polynomial fit between counts (IR camera data acquisition) and temperature. The expanded uncertainty for the radial profile of q''_{ec} , q''_{lc} , q''_{cap} , $q''_{rad,b}$, and q''_{cond} are shown graphically in Figure 4.2 and 4.3. These values are highlighted at three key points on the radial profile.

Table 3.13: Single source electrospray cooling experimental uncertainty.

Parameter		EU	Unit		PU [%]
T_s	$\pm$	0.77	K	$\pm$	0.25
T_∞	$\pm$	0.05	K	$\pm$	0.016
T_{ag}	$\pm$	0.06	K	$\pm$	0.016
Q	$\pm$	0.09	$\mu\text{L min}^{-1}$	$\pm$	0.99
q''_{gen}	$\pm$	5.56	W m^{-2}	$\pm$	0.38

3.3. DATA ANALYSIS

Table 3.14: Electrospray array cooling experimental uncertainty.

Parameter		EU	Unit		PU [%]
T_s	$\pm$	0.77	K	$\pm$	0.25
T_∞	$\pm$	0.05	K	$\pm$	0.016
T_{ag}	$\pm$	0.06	K	$\pm$	0.016
Q	$\pm$	0.15	$\mu\text{L min}^{-1}$	$\pm$	1.12
q''_{gen}	$\pm$	5.56	W m^{-2}	$\pm$	0.38

3.3.3.8 Study 2: Electrospray Array Cooling

A list of the relevant parameter and their associated expanded uncertainty (EU) and percentage uncertainty (PU) is outlined in Table 3.14. The uncertainty was calculated in an identical manner to study 1 and all values are to a 95% confidence level. The expanded uncertainty for the radial profile of q''_{ec} , q''_{lc} , q''_{cap} , $q''_{rad,b}$, and q''_{cond} are shown graphically in Figure 5.3. These values are highlighted at three key points on the radial profile.

3.3.3.9 Study 3 and 4: Droplet Evaporation

The uncertainty to a 95% confidence for study 3 and 4 is outlined in Table 3.15. All value were achieved by the same methodology as study 1 and 2. A much higher uncertainty was noted in comparison with study 1 and 2 due to the shorter capture time resulting in a reduced number of samples per subpoint in comparison with test points of study 1 and 2. The expanded uncertainty for the radial profile of q''_{de} , q''_{lc} , q''_{cap} , $q''_{rad,b}$, and q''_{cond} are shown graphically in Figure 6.3. These values are highlighted at three key points on the radial profile.

Table 3.15: Droplet evaporation experimental uncertainty.

Parameter		EU	Unit		PU [%]
T_s	$\pm$	0.77	K	$\pm$	0.25
T_∞	$\pm$	0.12	K	$\pm$	0.04
T_{ag}	$\pm$	0.13	K	$\pm$	0.04
q''_{gen}	$\pm$	17.58	W m^{-2}	$\pm$	1.97
V_d	$\pm$	2.06	μL	$\pm$	5.31
θ	$\pm$	20.05	mm	$\pm$	34.09
D_b	$\pm$	0.08	mm	$\pm$	2.69
H_d	$\pm$	0.05	mm	$\pm$	1.52
R_T	$\pm$	0.07	mm	$\pm$	2.54

3.4 Summary

A comprehensive and detailed description of the experimental apparatus and procedures used in Chapters 4 – 7 has been provided. Next the experimental analysis applied in processing both the thermal and optical acquired data was outlined. Finally the methodology practiced in defining the experimental uncertainty attached to the processed data was defined.

Chapter 4

Single Source Electrospray Cooling

Greater knowledge on the impact of various spraying parameters can aid and lead to the design of smarter more efficient spray cooling solutions. Electrospray cooling is an attractive solution for thermal dissipation given its; uniform mass flux, micron sized droplets, self spray dispersion, minimal energy and low total working fluid volume requirements. In this chapter single source electrospray cooling was used in tandem with thin foil thermography to characterises the local heat and mass transfer to an impinging single source electrospray plume. The influence of working fluid flow rate, source - target separation height, and nozzle were investigated. Generated droplets impact onto the heated substrate forming multiple

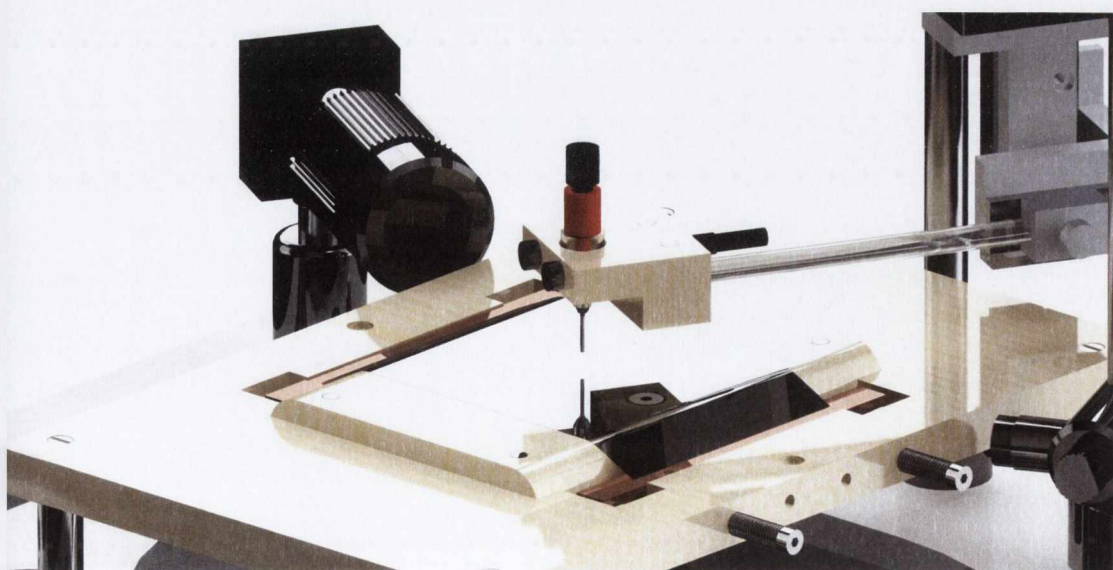


Figure 4.1: Single source electrospray cooling.

contact lines or coalescing with already evaporating fluid deposited onto the heated substrate. Characterisation of a single source is the first step in development of this smarter cooling solution.

4.1 Electro spray Cooling Modes

Two distinct electro spray cooling modes were observed during experimentation: evaporative electro spray cooling and pool electro spray cooling. The first corresponds to the case where droplets impinge on the heated substrate, forming multiple contact lines, and evaporate. Whereas, pool cooling was denoted by a coalescence of generated droplets on the heated substrate to form a quasi-steady liquid film. The first regime was noted for greater separation

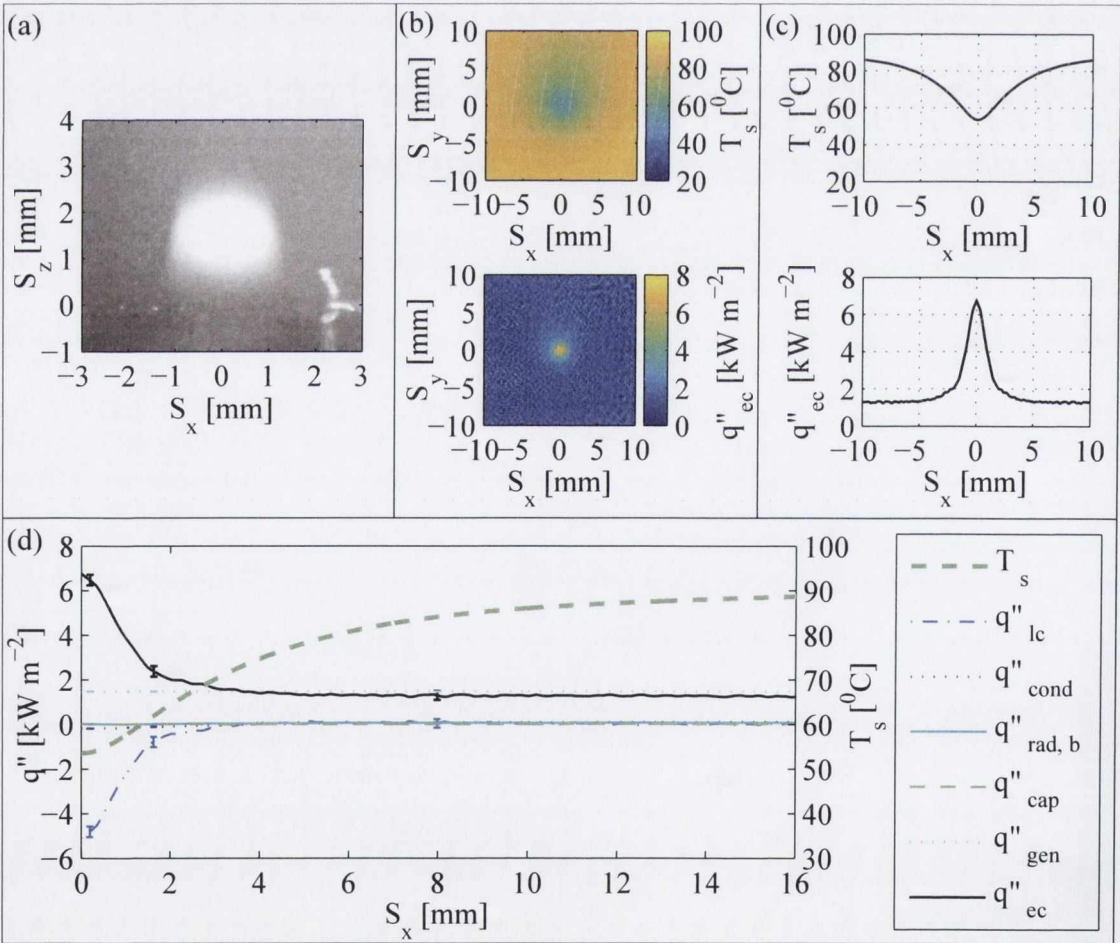


Figure 4.2: Radial evaporative energy balance. $V_n = 3 \text{ kV}$, $D_i = 0.330 \text{ mm}$, $D_o = 0.629 \text{ mm}$, $H = 7.5 \text{ mm}$, $Q = 4 \text{ }\mu\text{L min}^{-1}$ and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

heights and lower flow rates with the pooling regime noted for lower separation heights and larger flow rates.

An example of the evaporative cooling case is shown in Figure 4.2. This test corresponded to $V_n = 3$ kV, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 7.5$ mm, $Q = 4$ $\mu\text{L min}^{-1}$ and $q''_{gen} = 1,395$ W m^{-2} . Figure 4.2a is captured from the optical camera which is focused on the interface between the electrospray plume and the heated substrate. A section of the electrospray plume is illuminated using the low power laser. The temperature and convective heat flux maps of the foil are shown in Figure 4.2b with their corresponding radial profiles shown in Figure 4.2c. The final plot, Figure 4.2d, highlights the individual radial energy balance terms of the evaporative electrospray cooling. From this plot it can be seen that peak convective electrospray cooling is located directly below the nozzle, at $S_x = 0$ mm, and corresponds with a value of $6,711$ W m^{-2} , notably larger than q''_{gen} . This convective radial heat flux profile is due to the relatively uniform mass flux (j_m) in the region around $S_x = 0$ mm [11, 67]. The high mass flux in the central region of the spray allows higher evaporation rates and is thus consistent with observation that the convective heat flux is also higher [11, 67]. Both the energy generation term and storage term remain constant at $1,395$ W m^{-2} and 24 W m^{-2} respectively as the radial distance from the centre of the source nozzle increases. In the case of q''_{gen} this is due to the uniform heat generation through the foil and paint layers. At higher separation heights and lower flow rates the position of the spray core was observed to flutter slightly across the foil due to small air currents. The low value of q''_{cap} for all values of S_x demonstrates the quasi-steady nature of the EC process and that these minute fluctuations are not significant. Examining each term moving radially outward from the centre for $S_x = 0 - 4$ mm, it can be seen that the magnitude of q''_{ec} decreases steadily across this region. A peak lateral conduction value ($q''_{lc} = 5,071$ W m^{-2}) is noted at $S_x = 0$ mm due to the high thermal gradients in the central region ($\partial T / \partial S_x = 4.96^\circ\text{C mm}^{-1}$) resulting from the high convective heat flux caused by the evaporation of impacting droplets. From $S_x = 4 - 16$ mm the convective term q''_{ec} remains relatively constant at $\approx 1,142$ W m^{-2} and the lateral conduction is low due to the low thermal gradient ($\partial T / \partial S_x = 1.02^\circ\text{C mm}^{-1}$). The small difference between the q''_{ec} and q''_{gen} ($\Delta q'' = 253$ W m^{-2}) is due to the minor yet not negligible q''_{cond} and $q''_{rad,b}$ heat loss terms.

Figure 4.3 is an example of the pool cooling electro spraying regime. This test corre-

4.1. ELECTROSPRAY COOLING MODES

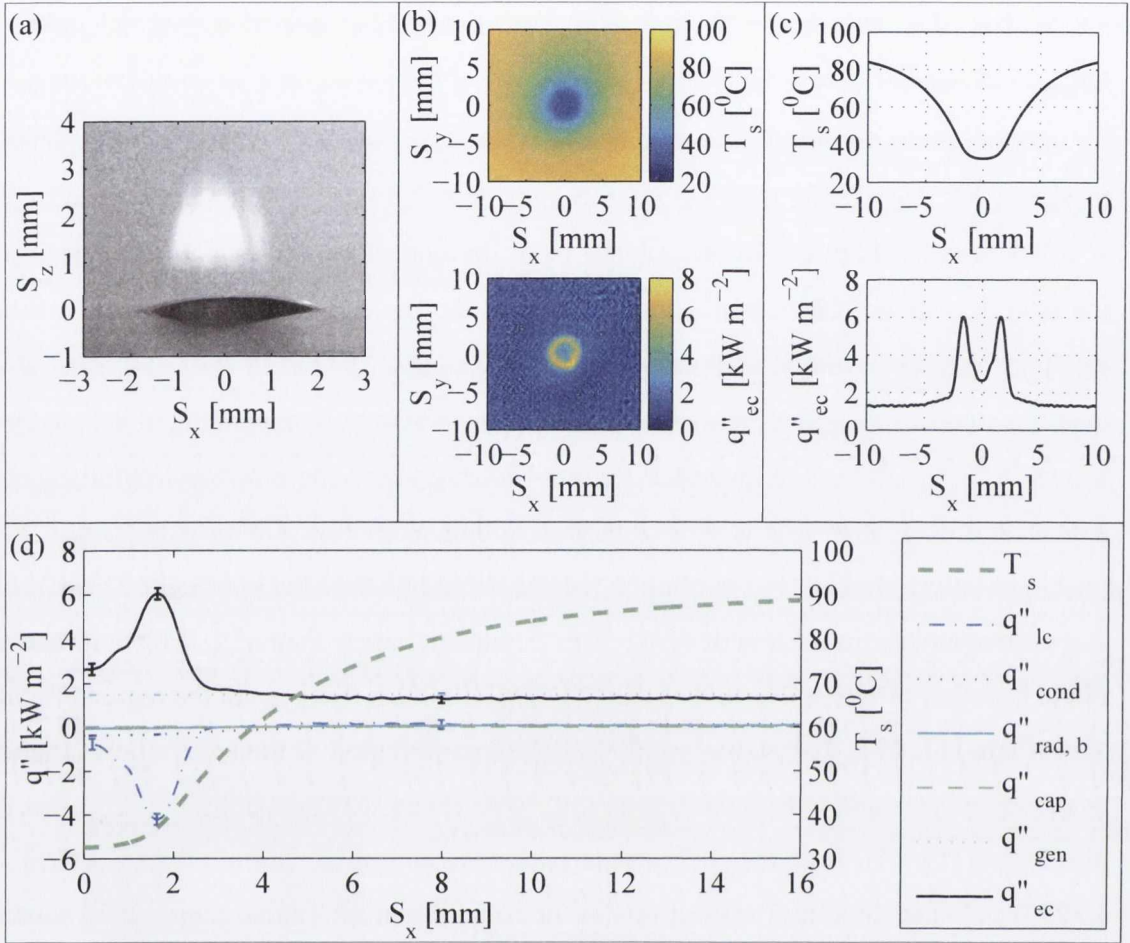


Figure 4.3: Radial pooling energy balance. $V_n = 3.2$ kV, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 7.5$ mm, $Q = 16 \mu\text{L min}^{-1}$ and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

sponded to $V_n = 3.2$ kV, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 7.5$ mm, $Q = 16 \mu\text{L min}^{-1}$ and $q''_{gen} = 1,395 \text{ W m}^{-2}$. The fixed liquid pool can be clearly seen in Figure 4.3a. Unlike the previous evaporative case, the convective heat flux distribution (Figure 4.3b) shows that greatest cooling occurs in the triple line region of the quasi-steady film ($q''_{ec, peak} = 6,043 \text{ W m}^{-2}$ and $S_{x, q''_{ec, peak}} = 1.64$ mm). This profile shape is due to the high evaporation rate at the triple line, and is similar to that observed by Marchuk et al. [116] and Karchevsky et al. [117] in their investigation of the lateral conduction problem for thin foil droplet evaporation. This region experience the highest heat fluxes across the droplet as a result of the low thermal resistance from the small film thickness (1 - 3 μm) [117].

The radial profile of the energy balance (Figure 4.3d) can be separated into three distinct regions. In the central region, $S_x = 0 - 1$ mm, it can be seen that the magnitude of $q''_{ec} \approx q''_{gen}$.

This is due to the low temperature gradient across this region ($\partial T / \partial S_x = 1.05^\circ\text{Cmm}^{-1}$) resulting in low lateral conduction. Similarly, the low temperature value of the foil ($\approx 33.12^\circ\text{C}$) results in small heat losses by conduction and radiation to the IR glass from the bottom of the foil. In the second region ($S_x = 1 - 4 \text{ mm}$) the convective heat flux increases to a peak at the triple contact line, as this is the region where significant evaporation is occurring. Similar to that discussed for the centre region of the evaporative mode, the high convective heat flux causes significant lateral conduction within the foil due to the large induced temperature gradients. In the far-field region, $S_x = 4 - 16 \text{ mm}$, the situation is similar to the evaporative regime as there is no longer heat transfer to a liquid phase.

4.1.1 Cooling Regime Transition

The cooling transition point (*CTP*) between evaporative and pool cooling is dependent on six factors; heated substrate properties, applied generated power, the liquid mass flux impinging on the surface, the ambient surroundings, and the latent heat of evaporation of the cooling fluid h_{fg} , such that:

$$CTP = f(k_f, \delta_f, q''_{gen}, j_m, T_\infty, h_{fg}) \quad (4.1)$$

In the current experimental set-up, the energy available to evaporate an impinging droplet is supplied from that generated, q''_{gen} , and that which flows within the foil, q''_{lc} . Once the mass flux is large enough that the required evaporative heat flux exceeds that which can be supplied by q''_{gen} and q''_{lc} , pooling of the working fluid on the heated substrate occurs. This is similar to that noted by Horacek et al. [73] and Sodtke and Stephan [72]. They observed a maximum heat flux for the greatest cumulative CLL. This can be interpreted as the point when numerous droplets, all with their own individual contact lines, are evaporating on the heated substrate. Once the mass flux is increased, or in Horacek et al. [73] and Sodtke and Stephan [72] case the wall heat flux is decreased, an increased coalescence of impinging droplets is noted resulting in a decrease in CLL and subsequent heat flux. This highlights the dependency of spray cooling on triple line heat and mass transfer.

4.2 Influence of Flow Rate

Figure 4.4 and 4.5 demonstrates the influence of varying the coolant flow rate on the convective heat transfer. Figure 4.4 shows the electrospray plume - heated substrate interface (left), the convective heat flux map (middle) and the radial convective profile (right) for $V_n = 1.95 - 2.15$ kV, $Q = 2 - 16 \mu\text{L min}^{-1}$, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 2.5$ mm, and $q''_{gen} = 1,395 \text{ W m}^{-2}$. As the flow rate is initially increased from $2 \mu\text{L min}^{-1}$ the peak heat flux,

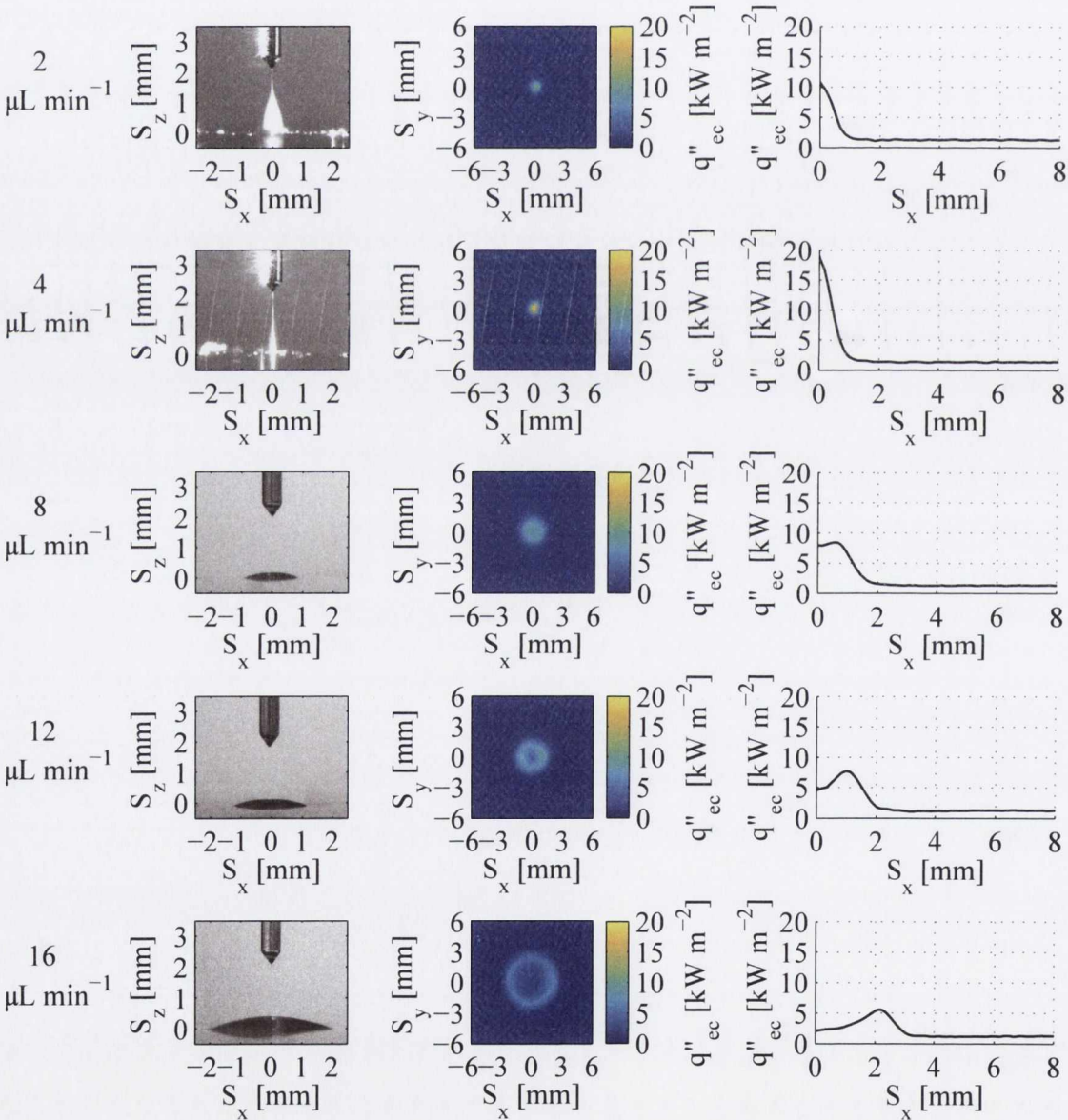


Figure 4.4: Impact of working fluid flow rate. $V_n = 1.95 - 2.15$ kV, $Q = 2 - 16 \mu\text{L min}^{-1}$, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 2.5$ mm, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

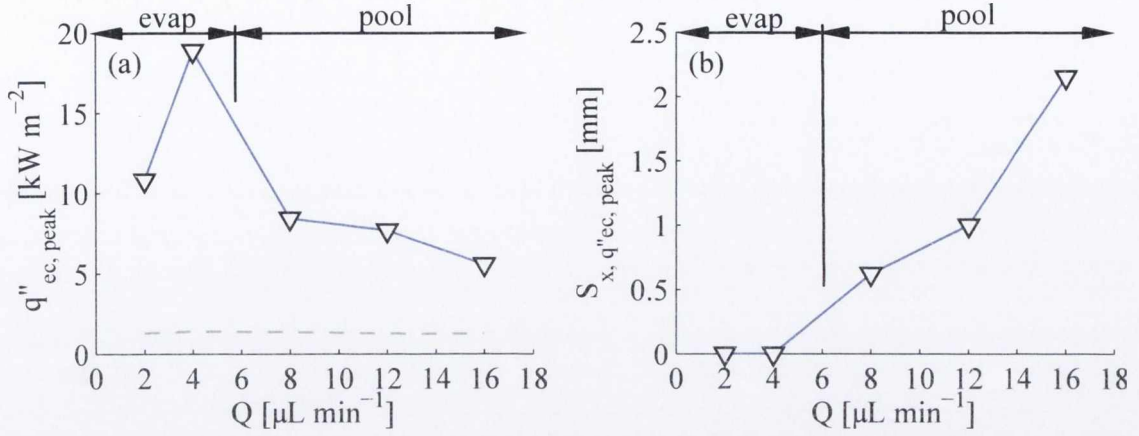


Figure 4.5: $q''_{ec,peak}$ magnitude and radial location for varying flow rate. $V_n = 1.95 - 2.15$ kV, $Q = 2 - 16$ $\mu\text{L min}^{-1}$, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $H = 2.5$ mm, and $q''_{gen} = 1,395$ W m^{-2} .

which is initially centrally located, increases (Figures 4.4 and 4.5). This is due to increasing droplet mass flux resulting in an increased CLL due to individual impinging droplets. A peak heat flux of 18.87 kW m^{-2} was observed at this point. This corresponded to a 18.71 times enhancement over natural convection. As the flow rate is increased further, cooling transitions from evaporative ($2 - 4$ $\mu\text{L min}^{-1}$) to pool electrospay cooling ($8 - 16$ $\mu\text{L min}^{-1}$). As the mass flux is increased from $4 - 8$ $\mu\text{L min}^{-1}$ the required evaporative heat flux exceeds that which can be supplied by the heated substrate, impinging droplet coalesce resulting in pooling of the working fluid on the heated substrate. This results in a drop $q''_{ec,peak}$ due to the reduced CLL on the heated substrate. The location of this reduced $q''_{ec,peak}$ ($S_{x,q''_{ec,peak}}$) is found at the triple line of the quasi-steady formed liquid droplet. This can be seen from the optical images by the appearance of a visible liquid film on the heated foil surface for the 8 $\mu\text{L min}^{-1}$ case. After transition to the pool cooling regime, the radial profile shows a decreasing $q''_{ec,peak}$ and increasing $S_{x,q''_{ec,peak}}$ with increasing liquid flow rate. This increasing $S_{x,q''_{ec,peak}}$ (Figure 4.5b) is due to the increasing base diameter of the quasi-steady pool from the increased flow rate. The drop in $q''_{ec,peak}$ can be attributed to the increasing thermal resistance at the triple line as the pool grows in thickness. From Figure 4.5a one would expect $q''_{ec,peak}$ for this set-up to occur at or just after 4 $\mu\text{L min}^{-1}$. This result demonstrates that increased fluid deposition rate on the surface does not necessarily result in improved heat transfer, which may be counter intuitive. This is due to the transition to the pool cooling regime, where the electrospay plume becomes more of a fluid delivery system to the fixed

4.3. INFLUENCE OF SEPARATION HEIGHT

pool of liquid on the substrate surface, with heat transfer defined by the singular contact line of the fixed droplet. The higher convective heat fluxes achieved by direct impact of the charged micron sized droplets on the heated substrate are thus diminished in this scenario. For reference, the dashed line in Figure 4.5a denotes the cooling in the far field of each test corresponding to that of natural convection alone, thus highlighting the notable enhancement that EC has over natural convection.

4.3 Influence of Separation Height

Figure 4.6 and 4.7 show the observed influence of varying the source - target separation height on the electrospray cooling profile, magnitude and position of $q''_{ec,peak}$ for $V_n = 2.07 -$

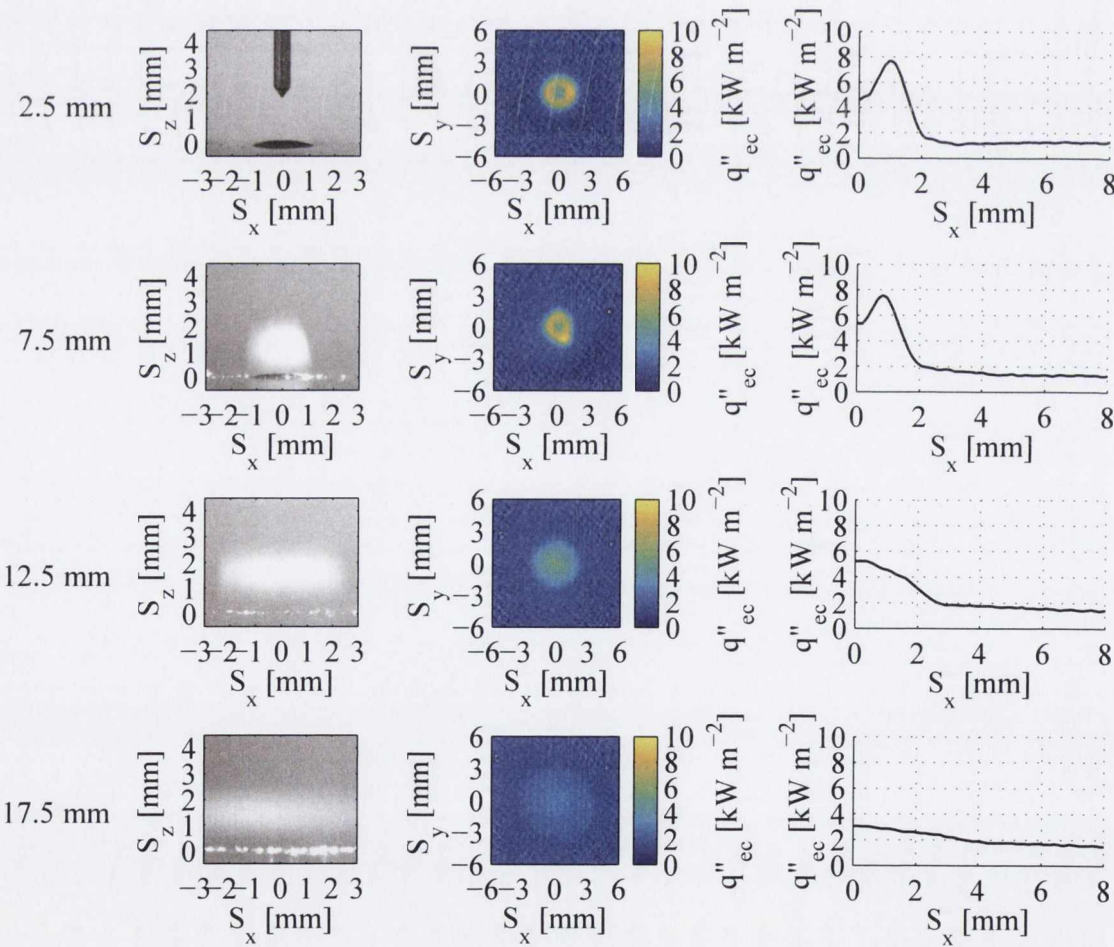


Figure 4.6: Impact of varied separation height. $V_n = 2.07 - 4.3$ kV, $H = 2.5 - 17.5$ mm, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $Q = 12 \mu\text{L min}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

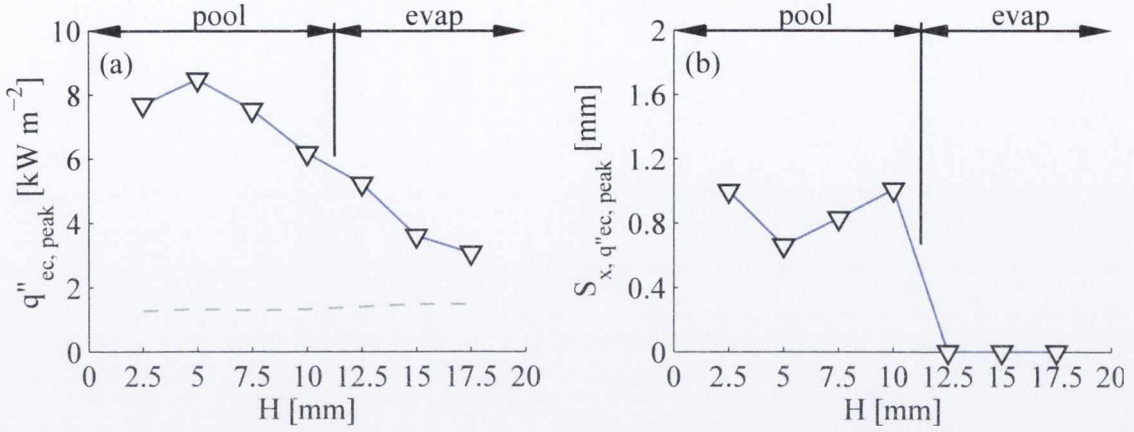


Figure 4.7: $q''_{ec,peak}$ magnitude and radial location for varying separation height. $V_n = 2.07 - 4.3$ kV, $H = 2.5 - 17.5$ mm, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $Q = 12 \mu\text{L min}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

4.3 kV, $H = 2.5 - 17.5$ mm, $D_i = 0.330$ mm, $D_o = 0.629$ mm, $Q = 12 \mu\text{L min}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$. The electro spray pool cooling mode is noted at lower separation heights, with transition to the evaporative regime ($H = 10 - 12.5$ mm) as H is increased. This is due to decreasing local droplet mass flux as H increases, as a result of greater plume dispersion. As H increases the distance over which the droplets have to spread increases. As the separation height is increased initially ($H = 2.5 - 5$ mm) a drop in $S_{x,q''_{ec,peak}}$ is noted with a modest increase in $q''_{ec,peak}$ (Figure 4.7a and b). This is due to the coincidental droplet triple line and primary droplet spray core resulting in a lower thermal resistance in this region. For $H = 5 - 10$ mm an increase in $S_{x,q''_{ec,peak}}$ is coupled with a monotonic decrease in $q''_{ec,peak}$. For $H = 10 - 17.5$ mm the spray transitions to the evaporative mode, such that $S_{x,q''_{ec,peak}} = 0$ mm and $q''_{ec,peak}$ continues to decrease as H increases due to the reduced mass flux of the droplets striking the surface due to the greater plume dispersion. This is confirmed by the radial plots of Figure 4.6 for the $H = 12.5 - 17.5$ mm cases, where it can be seen that as the separation height is increased a greater area of the substrate is cooled due to droplet dispersion, albeit at a lower q''_{ec} magnitude due to the lower local mass flux.

4.4 Parametric Study

Figures 4.8 and 4.9 are an attempt to synthesise the trends related to the working fluid flow rate, separation height, and nozzle size. Figure 4.8 shows the effect of separation height for

4.4. PARAMETRIC STUDY

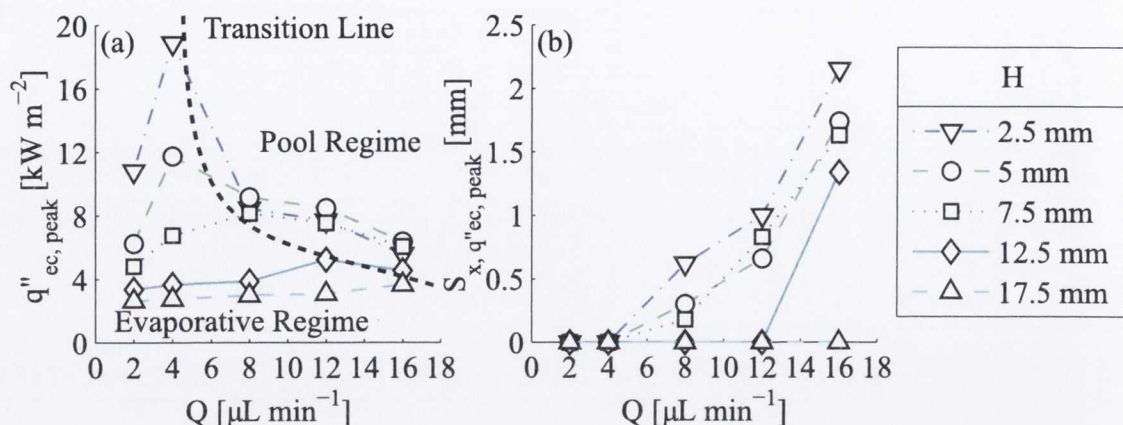


Figure 4.8: $q''_{ec,peak}$ magnitude and radial location for varying flow rate and separation height. $V_n = 1.95 - 4.67$ kV, $Q = 2 - 16$ μL min⁻¹, $H = 2.5 - 17.5$ mm, $D_i = 0.330$ mm, $D_o = 0.629$ mm, and $q''_{gen} = 1,395$ W m⁻².

$V_n = 1.95 - 4.67$ kV, $Q = 2 - 16$ μL min⁻¹, $H = 2.5 - 17.5$ mm, $D_i = 0.330$ mm, $D_o = 0.629$ mm, and $q''_{gen} = 1,395$ W m⁻².

For a fixed H , $q''_{ec,peak}$ increases with increasing Q in the evaporative regime due to the associated increase in droplet mass flux and subsequent CLL on the heated substrate. However, transition to pool cooling occurs when the combination of the internal heat generation and lateral conduction is insufficient to supply the energy required to evaporate the impinging droplets. The highest values of $q''_{ec,peak}$ for a given separation height are observed close to the transition from evaporative to pool cooling (CTP), with higher values of $q''_{ec,peak}$ occurring for smaller separation heights and lower flow rates. This can be attributed to a thinner liquid films due to smaller produced droplet from lower liquid flow rates. The relationship between H and Q with CTP is shown in Figure 4.8a where a “transition line” highlights CTP. Transition from evaporative to pool cooling occurs at lower flow rates for reduced separation heights. Plume dispersion increases as H increases resulting in a drop in the liquid mass flux for a fixed flow rate. Thus an increased Q is required to induce regime transition for increasing H . For evaporative cooling the magnitude of $q''_{ec,peak}$ decreases as H increases. This is due to the reduced spreading of the electrospray plume at lower separation heights and the radial nature of the system. As H is increased the area over which the plume is acting increases parabolically, which decreases the mass flux dramatically for modest increase in H .

Once transition to pool cooling occurs, $q''_{ec,peak}$ is largely insensitive to H as cooling is governed by triple line evaporation. For a fixed Q in the pool regime, $S_{x,q''_{ec,peak}}$ increases as

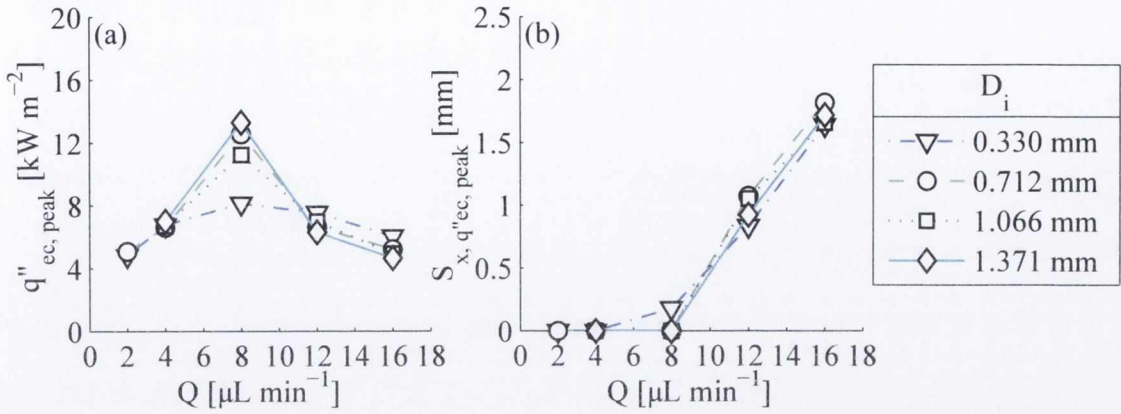


Figure 4.9: $q''_{ec,peak}$ magnitude and radial location for varying flow rate and nozzle size. $V_n = 2.95 - 4.08$ kV, $Q = 2 - 16$ $\mu\text{L min}^{-1}$, $D_i = 0.330 - 1.371$ mm, $D_o = 0.629 - 1.844$ mm, $H = 7.5$ mm and $q''_{gen} = 1,395$ W m^{-2} .

H decreases. This can be attributed to increased droplet evaporation due to increasing transit times and decreasing j_m as H increases. In the pool regime for a fixed H , as Q increases $q''_{ec,peak}$ decreases and $S_{x,q''_{ec,peak}}$ increases. This is due to the increasing mass flux causing an increased droplet thickness and spreading.

Figure 4.9 shows the influence of nozzle size for $V_n = 2.95 - 4.08$ kV, $Q = 2 - 16$ $\mu\text{L min}^{-1}$, $D_i = 0.330 - 1.371$ mm, $D_o = 0.629 - 1.844$ mm, $H = 7.5$ mm and $q''_{gen} = 1,395$ W m^{-2} . $q''_{ec,peak}$ and $S_{x,q''_{ec,peak}}$ are observed to be largely insensitive to varied nozzle diameter. If D_i is greater than the acceleration zone base then the jet break up is similar regardless of nozzle size [3]. For $Q = 8$ $\mu\text{L min}^{-1}$ and $D_i = 0.330$ mm, a drop in $q''_{ec,peak}$ is observed in comparison with other nozzle sizes as it has already transitioned to pool cooling, whereas the other nozzle sizes have not. This early transition can be accounted for as CTP occurs at this point. Applying Equation 2.1 for the electric field between a hyperboloid and plate, the applied average electric field for the smallest nozzle ($D_i = 0.330$ mm and $D_o = 0.629$ mm) is shown to be 1.26 times greater than the next smallest nozzle size of $D_o = 1.095$ mm. Yang et al. [102] showed that an increasing applied average electric field results in a reduction of the radial spread of the the primary droplet core resulting in an increase in the mass flux around $S_{x,q''_{ec,peak}} = 0$. These factors combine to account for the early onset of pool cooling for the smallest nozzle size.

In the evaporative regime all nozzle sizes show an increase in $q''_{ec,peak}$ as Q increases,

4.5. SUMMARY

which is consistent with the observations in Figure 4.9a. Apart from the smallest nozzle size, the CTP is consistent for all nozzle sizes, occurring between $Q = 8 - 12 \mu\text{L min}^{-1}$. Post CTP, $q''_{ec,peak}$ and $S_{x,q''_{ec,peak}}$ are observed to be insensitive to all nozzle size. This is again due to the similar means of jet break up due to the acceleration zone base being smaller than D_i and in this mode the electrospray is essentially acting as a fluid delivery system to a quasi-steady evaporating droplet film.

4.5 Summary

A parametric study of a single source electrospray was carried out using thin foil thermography to investigate, for the first time, the local convective heat transfer to a single source electrospray. Two distinct cooling regimes were observed; evaporative and pool electrospray cooling. The evaporative regime was shown to exist at greater separation heights and lower flow rates, with the converse true for pool cooling. Neither were shown to have a significant dependence on nozzle size.

The local convective heat flux profile and its dependency on flow rate, separation height and nozzle size was highlighted. In the evaporative regime an increase in working fluid flow rate caused an increase in the peak convective heat flux but no change was noted in its position. In the pool regime an increase in working fluid flow rate was shown to decrease the peak convective heat flux while increasing the radial location of this peak from the centre of the source nozzle. In the evaporative regime an increase in the separation height causes a decrease in the peak convective heat flux but an increase in the area of enhanced electrospray cooling. For the pooling case, an increase in the separation height resulted in an initial increase in $q''_{ec,peak}$ at higher flow rates before displaying a monotonic decrease thereafter. The location of this peak was observed to initially decrease at higher flow rates followed by a temporary increase before decreasing towards the evaporative regime of cooling for an increasing separation height. For nozzle size only a small dependency was observed in the evaporative regime for low flow rates and the smallest nozzle size. Nozzle size was shown to be largely independent for both cooling regimes otherwise. The largest values of $q''_{ec,peak}$ were noted for the evaporative regime prior to CTP at the smallest separation heights. These results are similar to that observed by Horacek et al. [73] and Sodtke and Stephan [72]. Both

authors noted that the peak heat flux corresponds to the point where the fluid deposited on the heated substrate achieve the greatest CLL (evaporative cooling, impact and subsequent spreading of individual droplets on the surface results in formation of multiple contact lines) and that the heat flux decreased with increased wetted area (pool cooling). A peak heat flux of 18.87 kW m^{-2} was observed in the evaporative regime at $H = 2.5 \text{ mm}$, $Q = 4 \text{ } \mu\text{L min}^{-1}$, $D_i = 0.330 \text{ mm}$, $D_o = 0.629 \text{ mm}$, and $V_n = 2.03 \text{ kV}$. This corresponded to a 18.71 times enhancement over natural convection.

The results of this work highlight the importance of the local droplet mass flux on the convective heat flux as well as heat transfer regime. In general, any parameter change that tends to increase the droplet mass flux also increases the convective heat flux in the evaporative cooling regime. However, due to the limited thermal energy that can be supplied due to the internal generation and lateral conduction, transition to pool cooling occurs when insufficient energy can be supplied to accommodate full evaporation of the impinging droplets. Both regimes are observed to be dependent on triple line heat transfer; evaporative due to the multiple contact lines induced by impacting and spreading of droplets, while pool cooling is governed by the much larger singular contact line established by the quasi-steady droplet formed on the heated substrate. Further investigation into the fundamentals of triple line evaporation is required to fully understand the heat and mass transfer from the heated substrate by spray cooling.

Chapter 5

Electrospray Array Cooling

Multiple source electrospraying enables higher droplet mass flux and greater cooling coverage area in comparison with single source electrospraying. For similar total working fluid flow rates the multiple source produces smaller droplets enabling greater CLL for higher heat transfer. The designed electrospray array and thermal exchange surface are shown in Figure 5.1. Figure 5.2 shows the designed seven source array in operation. A central nozzle at $S_x = 0$ mm is surrounded by 6 adjacent nozzle in a hexagonal pattern. An additional ring of 12 plugged (“dummy”) encircle these nozzle to reduce and eliminate edge effects stemming from the higher electric field near the edge of the array.

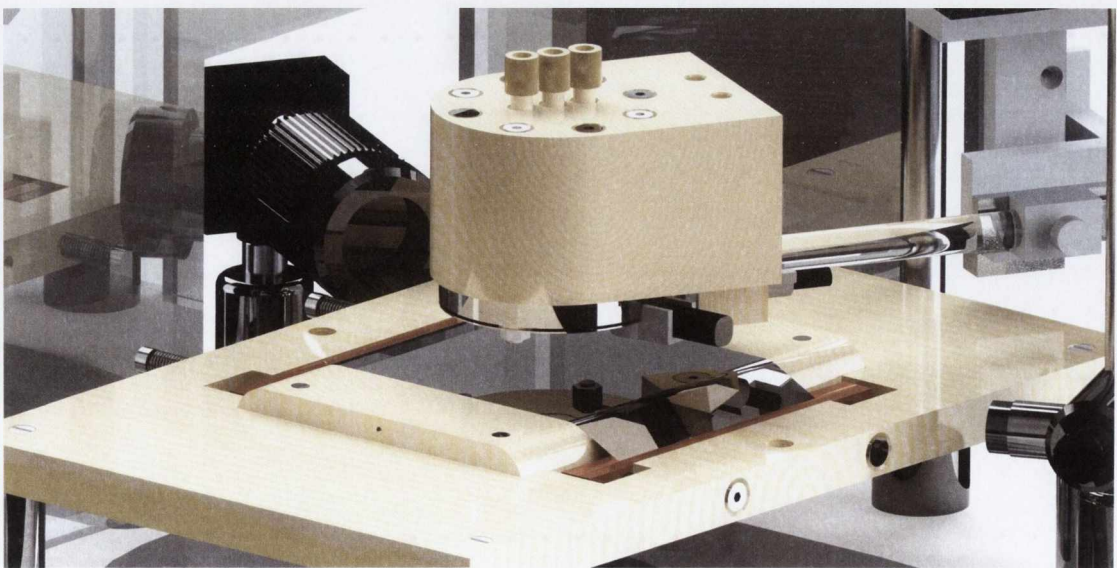


Figure 5.1: Array electrospray cooling.

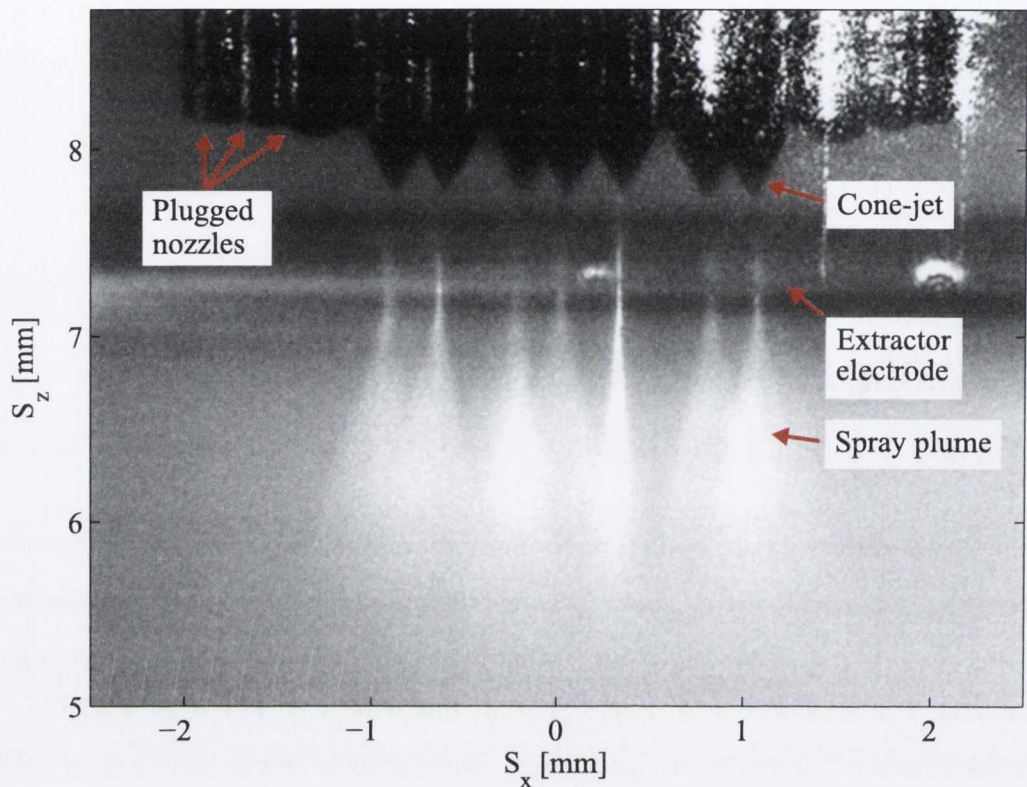


Figure 5.2: Electro spray array, cone-jet and spray plume. $D_i = 200 \mu\text{m}$, $D_o = 400 \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $Q = 3 \mu\text{L min}^{-1} N_n^{-1}$, $V_n = 7.1 \text{ kV}$, $V_{ex} = 6 \text{ kV}$, $E_d = 8 \text{ kV cm}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

As shown in Chapter 4 the greatest electro spray cooling is achieved in the evaporative regime of spraying, where the impact and subsequent spreading of droplets on the surface results in formation of multiple contact lines resulting in peak CLL and thus heat transfer. Following this, the electro spray array was exclusively operated in the evaporative regime for this investigation. An evaporative array cooling case is illustrated in Figure 5.3. This test corresponded to $D_i = 200 \mu\text{m}$, $D_o = 400 \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $Q = 3 \mu\text{L min}^{-1} N_n^{-1}$, $V_n = 7.1 \text{ kV}$, $V_{ex} = 6 \text{ kV}$, $E_d = 8 \text{ kV cm}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$. Figure 5.3a is captured from the optical camera which is focused on the jet forming region and generated spray plume. The plume is illuminated using the low power laser.

The temperature and convective heat flux distributions of the heated foil are shown in Figure 5.3b with their corresponding radial profiles shown in Figure 5.3c. The radial profiles are found by taking lines radially outward from the array centre at 60° , 0° , 300° , 240° , 180°

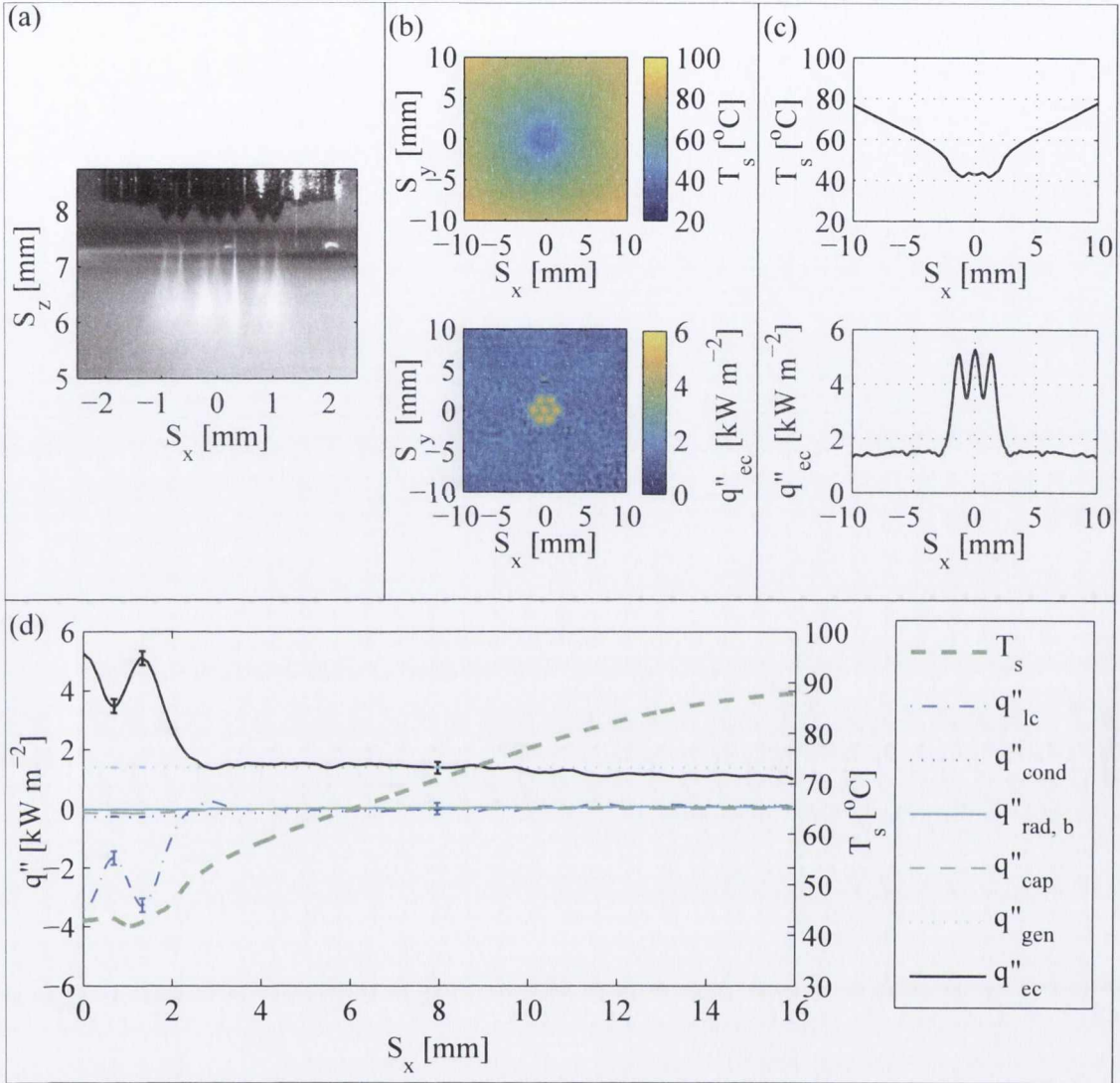


Figure 5.3: Electro spray array radial evaporative energy component balance. $D_i = 200 \mu\text{m}$, $D_o = 400 \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $Q = 3 \mu\text{L min}^{-1} N_n^{-1}$, $V_n = 7.1 \text{ kV}$, $V_{ex} = 6 \text{ kV}$, $E_d = 8 \text{ kV cm}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

and 120° and averaging.

The final plot, Figure 5.3d, shows the individual radial energy balance terms of the array electro spray cooling. A peak convective heat flux of $5,275 \text{ W m}^{-2}$ is noted directly below the central array nozzle, at $S_x = 0 \text{ mm}$, and is notably larger than q''_{gen} . This result is in agreement with that observed for all single source evaporative results. A second peak of $5,111 \text{ W m}^{-2}$ is observed at $S_x = 1.3 \text{ mm}$, and this peak corresponds to the centre of the adjacent nozzles spray plume. The peak is displaced by 0.3 mm due to repulsion between adjacent spray plumes en-route to the target surface. Both the energy generation term and

5. ELECTROSPRAY ARRAY COOLING

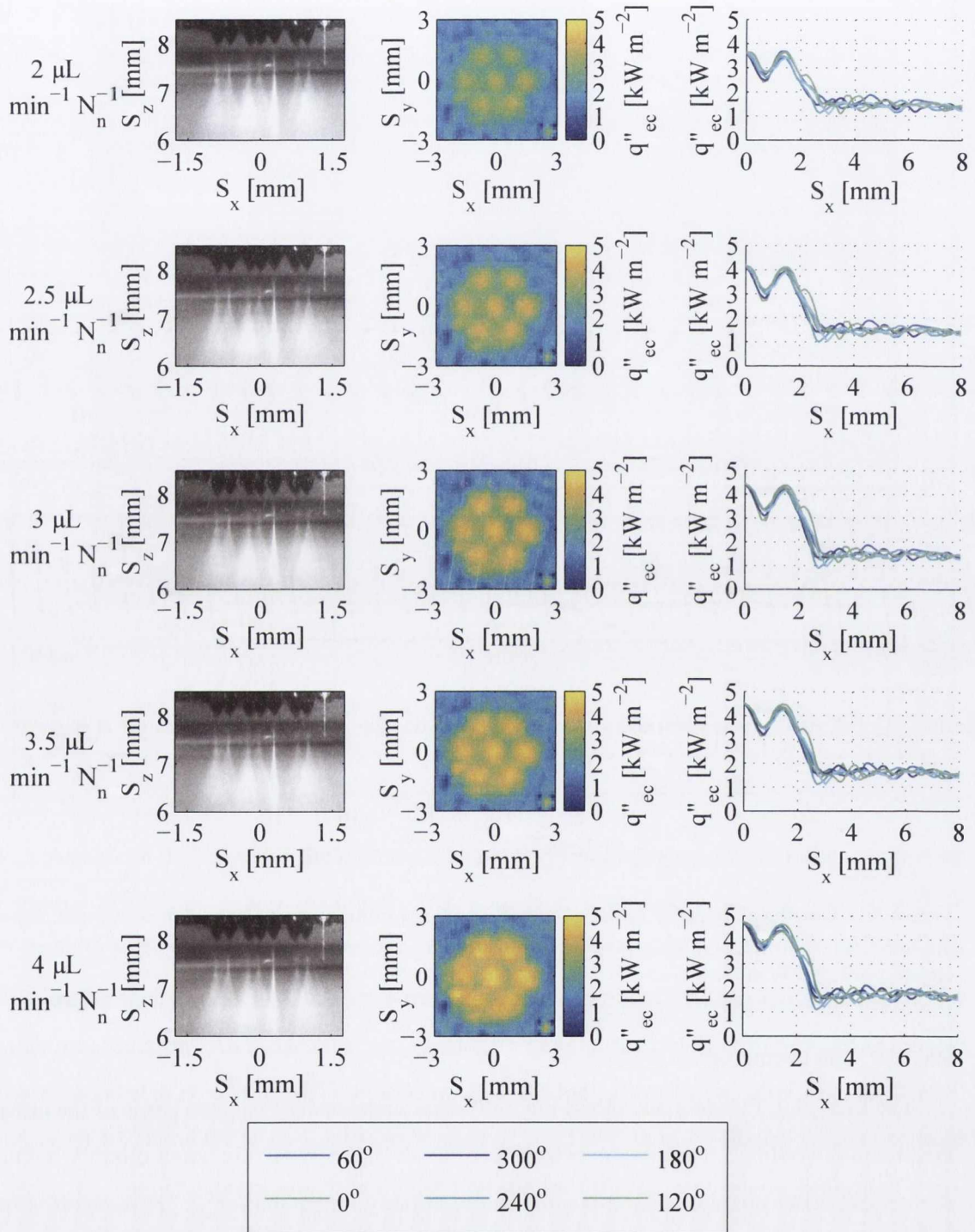


Figure 5.4: Influence of varied working fluid on plume, heat flux map and radial heat flux profile. $Q = 2 - 4 \mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200 \mu\text{m}$, $D_o = 400 \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $V_n = 5.55 \text{ kV}$, $V_{ex} = 4.5 \text{ kV}$, $E_d = 6 \text{ kV cm}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

storage term remain constant at $1,395 \text{ W m}^{-2}$ and 8 W m^{-2} respectively as the radial distance from the centre of the array increases. In the case of q''_{gen} this is due to the uniform heat generation through the foil and paint layers. Local peak lateral conduction values of $q''_{lc} = 3,448 \text{ W m}^{-2}$ and $3,280 \text{ W m}^{-2}$ are observed at $S_x = 0 \text{ mm}$ and 1.3 mm respectively. These locations correspond to the centre of the central and adjacent spray plumes. They are caused by high thermal gradients in the central region resulting from the high convective heat flux caused by the evaporation of impacting droplets. From $S_x = 3 - 16 \text{ mm}$ the convective term q''_{ec} decreases steadily from $\approx 1,500 - 1,010 \text{ W m}^{-2}$ as the foil temperature increases. This increasing T_s results in increasing values of q''_{cond} and $q''_{rad,b}$. The lateral conduction is low across this region due to the low thermal gradient. The small difference between the q''_{ec} and q''_{gen} for $S_x = 10 - 16 \text{ mm}$ ($\Delta q'' = 280 \text{ W m}^{-2}$) is due to the minor yet not negligible q''_{cond} and $q''_{rad,b}$ heat loss terms.

5.1 Influence of Flow Rate

Figure 5.4 and 5.5 demonstrates the influence of the working fluid flow rate on the convective heat transfer. Figure 5.4 shows the electrospray jet forming region and generated spray plume (left), the convective heat flux distribution (middle) and the radial convective profile (right) for varied flow rates at a fixed nozzle size, separation height, driving electric field and input

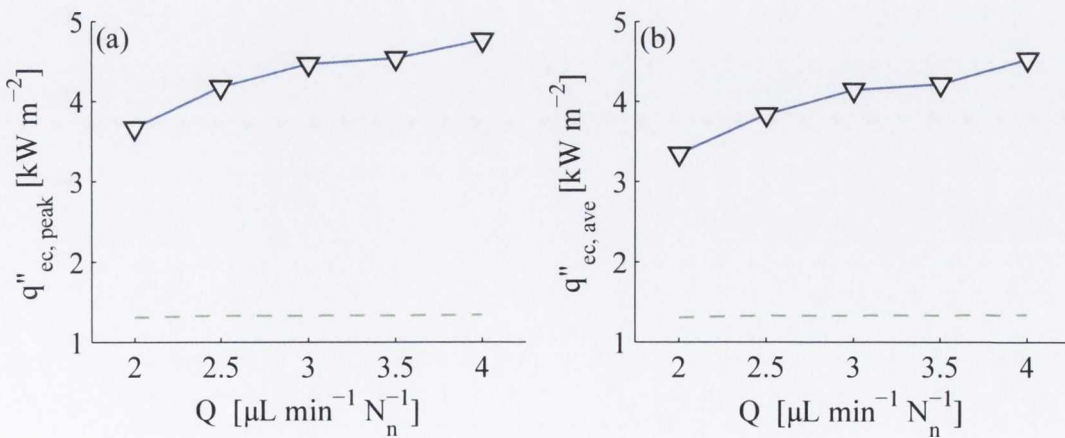


Figure 5.5: $q''_{ec,peak}$ and $q''_{ec,ave}$ for varying flow rate. $Q = 2 - 4 \mu\text{L min}^{-1} \text{N}_n^{-1}$, $D_i = 200 \mu\text{m}$, $D_o = 400 \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $V_n = 5.55 \text{ kV}$, $V_{ex} = 4.5 \text{ kV}$, $E_d = 6 \text{ kV cm}^{-1}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

5.1. INFLUENCE OF FLOW RATE

heat flux. The radial convective profile consists of lines taken from the centre source at 60°, 0°, 300°, 240°, 180° and 120°. Figure 5.4 and 5.5 show that as the working fluid flow rate is increased so to does the peak, radial profile, and average convective heat flux. This is due to increasing droplet mass flux with increasing flow rate. The consistent radial heat flux profiles in shape and magnitude for all flow rates demonstrates the uniform performance across all source nozzles. The average heat flux corresponds to a circular area of 1 mm diameter around the centre nozzle. For reference, the dashed line in Figure 5.5a and b denotes the cooling in the far field of each test corresponding to that of natural convection alone, thus highlighting the notable enhancement that EC has over natural convection.

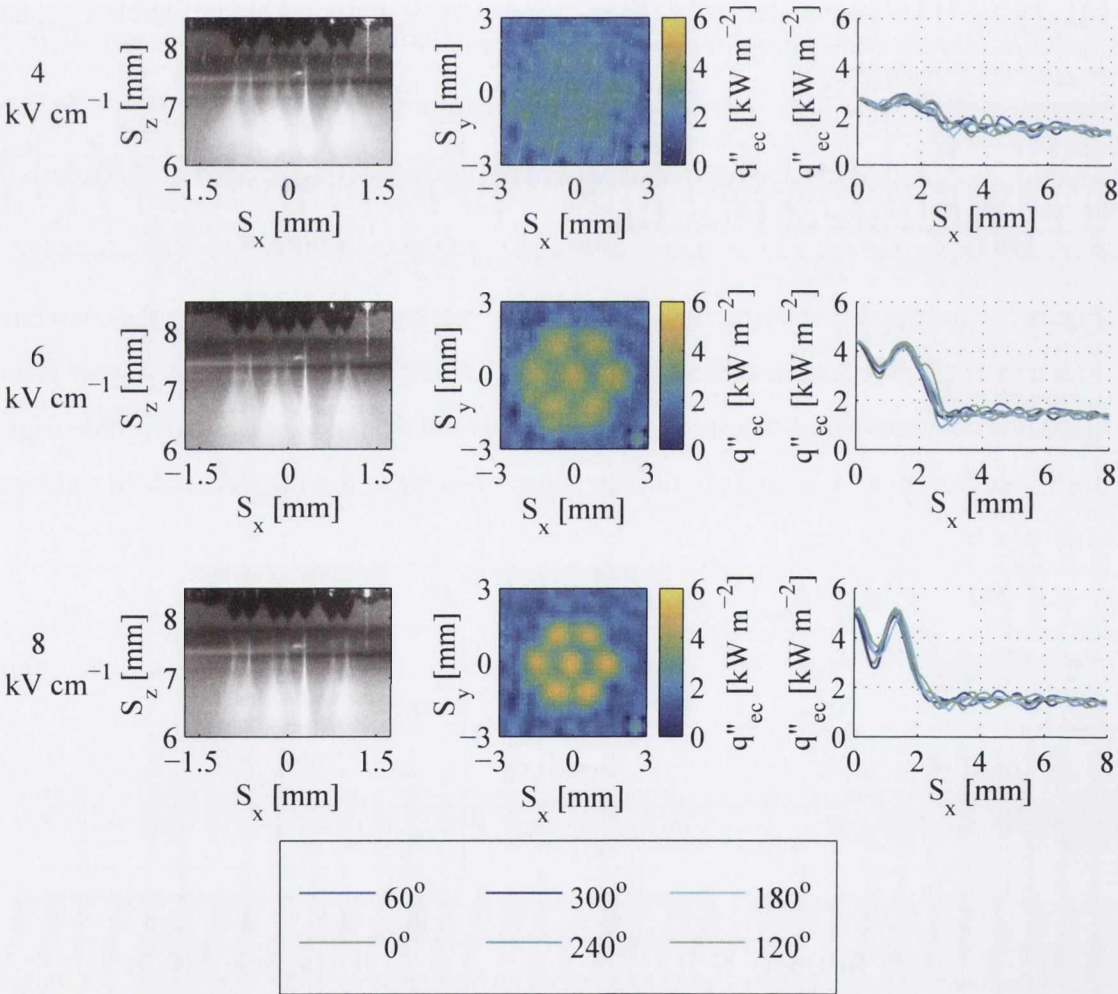


Figure 5.6: Influence of driving electric field on plume, heat flux map and radial heat flux profile. $E_d = 4 - 8 \text{ kV cm}^{-1}$, $Q = 3 \text{ } \mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200 \text{ } \mu\text{m}$, $D_o = 400 \text{ } \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $V_n = 4.1 - 7.1 \text{ kV}$, $V_{ex} = 3 - 6 \text{ kV}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

5.2 Influence of Driving Electric Field

Figure 5.6 and 5.7 demonstrates the influence of varied extractor – target electric field strength on the convective heat transfer. From the first column in Figure 5.6 it can be observed that an increasing driving electric field narrows the individual spray sources. This is due to the increasing downward acceleration of the charged droplets as the electric field increases. This reduces the droplet residence time between source and target. As a result the charged droplets have less time over which to repel each other, resulting in a more concentrated spray plume per source. Figures 5.6 and 5.7 show that as the driving electric field is increased a subsequent increase in the peak, radial profile and average heat flux is measured. The more defined radial heat flux profile at higher driving electric fields (Figure 5.6 column three) highlights the narrowing spray plume and subsequent increased mass flux around the centre line of each plume. The reduced droplet residence time as a result of the increasing driving electric field also serves to reduce the impact of droplet evaporation while in transit to the target substrate. This increases the mass flux impacting on the target surface, which subsequently increases the convective heat flux. The dashed line in Figure 5.7a and b denotes the cooling in the far field of each test corresponding to that of natural convection alone showing significant enhancement.

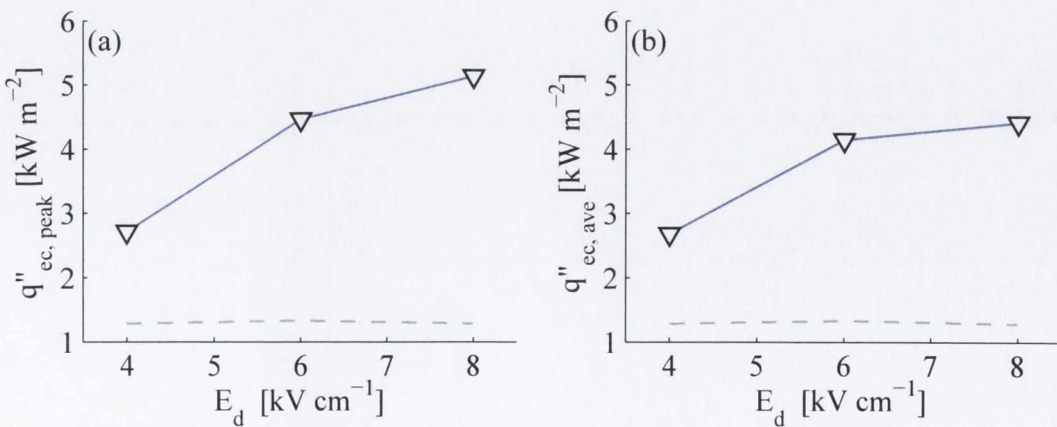


Figure 5.7: $q''_{ec,peak}$ and $q''_{ec,ave}$ for varying driving electric field. $E_d = 4 - 8 \text{ kV cm}^{-1}$, $Q = 3 \text{ } \mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200 \text{ } \mu\text{m}$, $D_o = 400 \text{ } \mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $V_n = 4.1 - 7.1 \text{ kV}$, $V_{ex} = 3 - 6 \text{ kV}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$.

5.3. SINGLE SOURCE - ARRAY COMPARISON

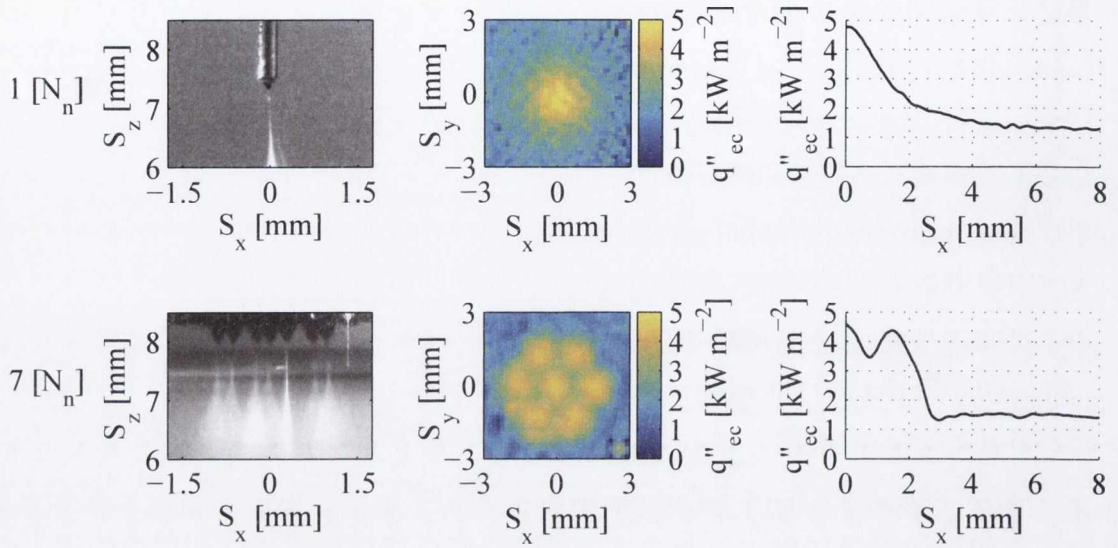


Figure 5.8: Electro spray comparison. Single source: $V_n = 2.46$ kV, $H = 7.5$ mm, $D_i = 180$ μm , $D_o = 321$ μm , $Q = 4$ $\mu\text{L min}^{-1}$, and $q''_{gen} = 1,395$ W m^{-2} . Array: $E_d = 6$ kV cm^{-1} , $Q = 4$ $\mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200$ μm , $D_o = 400$ μm , $H_1 = 0.5$ mm, $H_2 = 7.5$ mm, $V_n = 5.6$ kV, $V_{ex} = 4.5$ kV, and $q''_{gen} = 1,395$ W m^{-2} .

5.3 Single Source - Array Comparison

Figure 5.8 compares an electro spray array cooling with that of a single source under similar operation conditions. Single source: $V_n = 2.46$ kV, $H = 7.5$ mm, $D_i = 180$ μm , $D_o = 321$ μm , $Q = 4$ $\mu\text{L min}^{-1}$, and $q''_{gen} = 1,395$ W m^{-2} . Array: $E_d = 6$ kV cm^{-1} , $Q = 4$ $\mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200$ μm , $D_o = 400$ μm , $H_1 = 0.5$ mm, $H_2 = 7.5$ mm, $V_n = 5.6$ kV, $V_{ex} = 4.5$ kV, and $q''_{gen} = 1,395$ W m^{-2} . Both the single source and the array have comparable peak heat fluxes. Comparing both systems over a 1.5 mm radial area from $S_x = 0$ mm, the array achieves a 1.14 times greater average heat flux over this region. This is sensible given the larger total working fluid flow rate of the array. An average wall temperature of 37.69°C with a standard deviation of 0.3°C is noted for the array. Whereas, for the single source case these values were 54.71°C with a standard deviation of 0.41°C over the same cooling coverage area.

5.4 Summary

The convective heat transfer to an electro spray array operating in the evaporative regime using thin foil thermography was parametrically investigated. The influence of the working

fluid flow rate and driving electric field in this cooling regime was studied. Increasing peak, radial profile, and average convective heat flux was observed for increasing flow rate due to the increasing droplet mass flux as the flow rate is increased. An increase in the peak, radial profile, and average heat flux was also noted for increasing driving electric fields. An 89.2% and 64.86% increase in the peak and average heat flux were noted between the lowest and highest applied electric fields. This was due to increasing droplet mass flux, resulting in increasing CLL, from reduced droplet evaporation en-route to the target and narrowing of the spray plume. This narrowing spray plume, at higher driving electric fields, also resulted in a more defined radial heat flux profile. Comparing the arrays performance with that of a single source under similar experimental conditions showed a 1.14 times greater average heat flux and a 31% lower average wall temperature. This lower wall temperature was noted for a 28.82% decrease in the standard deviation of the wall temperature over the compared areas. The results of this work highlight the importance of electrospray arrays in achieving large coverage area uniform cooling and demonstrates the role that the working fluid flow rate and driving electric field play in optimising thermal performance.

Chapter 6

Droplet Evaporation

Evaporation from the contact line is an important factor in spray cooling heat transfer [64, 72, 73, 154]. The impact and subsequent spreading of droplets on the heated substrate results in formation of multiple contact lines. Strong correlation between the heat transfer intensity and the contact line length per unit area has been observed [72, 73]. Studies of evaporating droplets on heated substrate are essential for understanding this result [154].

Heat and mass transfer to the triple contact line region of an evaporating film is among the most important unsolved physical problems. Further investigation and understanding of this phenomena has applications in several other processes as well as spray cooling:

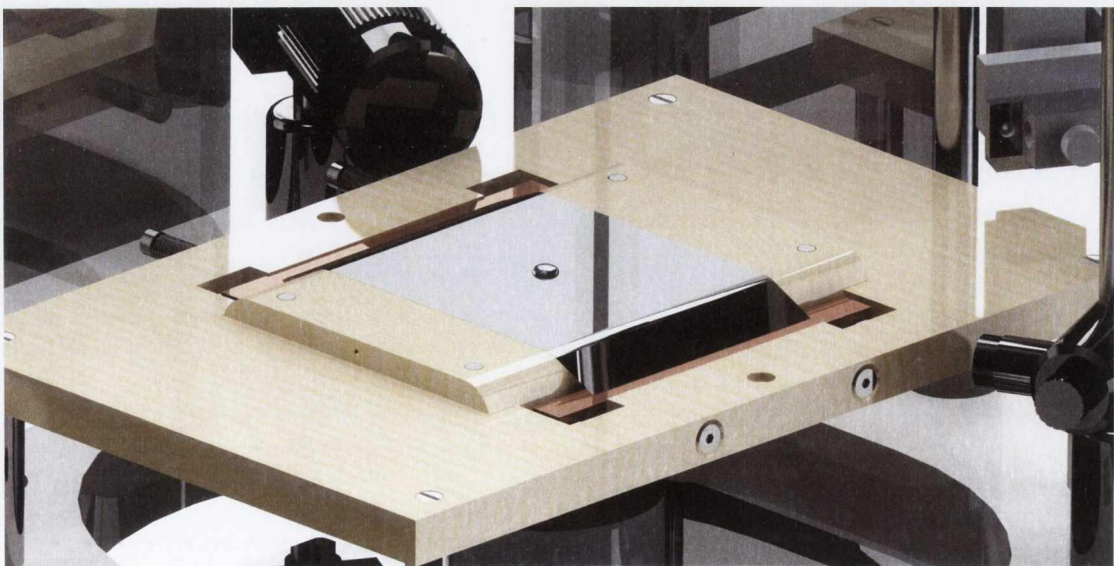


Figure 6.1: Droplet evaporation test facility.

droplet evaporation, droplet spreading, boiling, condensation, cavitation, rivulet flows and destruction of liquid films [175–178]. Greater knowledge on the mechanisms of the heat and mass transfer to an evaporating droplet is vital for both evaporative and pool electrospray cooling. In the evaporative case individual microscale droplet impact, forming multiple contact lines, and evaporate on the heated substrate. Whereas, for the pool cooling case impinging droplets coalesce and the heat transfer process proceeds as a quasi-steady fixed droplet evaporating on the heated surface in an applied electric field. Both cases are governed by heat and mass transfer at the triple contact line [64, 72, 73, 154]. Significant curvature of the liquid - gas interface at the triple line, non-equilibrium effects due to the temperature jump at the interface [179], the disjoining pressure due to Van der Waals forces, and substrate structure mean that the local heat transfer at the contact line is still an underdeveloped topic [154]. The following studies in Chapters 6 and 7 investigate heat and mass transfer at triple line of an evaporating droplet and define the impact of contact line dynamics, surface wettability and the application of an externally applied electric field on subsequent heat transfer.

Evaporation of an 80 μL droplet was investigated for two heated surface conditions; partial wetting ($0^\circ < \theta < 90^\circ$) and partial non-wetting ($90^\circ < \theta < 180^\circ$). The 80 μL droplet volume was chosen from the literature [117, 180, 181]. This volume allowed a longer evaporation period on the heated substrate and enabled the quasi steady and thermal equilibrium approximation during data acquisition. This volume resulted in an initial droplet base radius of 3.9 mm and 2.1 mm for the partial wetting and partial non-wetting cases respectively. The capillary length for water is 2.7 mm [119, 138]. This results in flattening of the partial wetting droplet due to gravitational forces [119]. Previous research much of which was for an unheated substrate has focused on lower initial droplet volumes (0.45-15 μL) which result in a base radius much smaller than that of the capillary length [114, 125, 129, 136, 137, 149–151]. This methodology is primarily done so that the solid - liquid interface can be approximated by a spherical cap. This enables the comparison with the natural evaporation model derived by Picknett and Bexon [111] and the subsequent improved models by Deegan [137], Hu and Lawson [128, 130], and Popov [131]. Briones et al. [140] noted that decreasing the droplet curvature weakened the Marangoni flow within the droplet reducing evaporation. However, as reported by Savino et al. [134] and Nguyen and Stebe [135] contaminants that collect on

Table 6.1: Contact angle hysteresis.

	Uncoated foil	Coated foil
$\theta_A[^\circ]$	79	160
$\theta_R[^\circ]$	50	159

the free surface can almost entirely suppress Marangoni flow in an evaporating water droplet and that they are almost unavoidable when working with water. Briones et al. [140] and Sobac and Brutin [138] observed that Marangoni-driven recirculating flow weakens with increasing substrate temperature. As already discussed, droplet evaporation is governed by the micro region which is the same size regardless of macro region length scales [154]. Furthermore, good correlation of droplet evaporation parameters (evaporation rate and contact angle) during evaporation have been shown between macro and micro evaporating droplets on heated substrates in the literature [149, 150, 181]. This highlights the relevance of the evaporation studies in Chapters 6 and 7 to both the evaporative and pool cooling regimes of electrospray cooling observed in Chapter 4 and 5.

The partial wetting substrate wettability condition corresponded to the uncoated stainless steel foil, while the partial non-wetting was achieved by coating the foil with Glaco surface coating [165]. Both case are thermally identical; only the physicochemical and surface roughness properties were modified. As covered in Section 3.1.1.2, the surface roughness of both cases was investigated by white light Interferometry and a surface roughness value of 59.52 nm and 176.24 nm was found respectively. To define the surface wettability the advancing and receding contact angles, θ_A and θ_R , were measured by expanding and contracting sessile water droplets on both substrate conditions. The results were processed using the steps outlined in Section 3.3.2.1 and are shown in Table 6.1.

6.1 Partial Wetting Droplet Evaporation

The baseline case of droplet evaporation is that of the uncoated stainless steel surface. Figure 6.2 shows a three-dimensional (3D) representation of the partial non-wetting droplet evaporation process at three time points. Figure 6.2 demonstrates that the droplet evaporates with a fixed droplet base radius and thus a pinned triple contact line over this period. The x-y plane of Figure 6.2 shows the local heat flux into the evaporating droplet q''_{de} . The highest

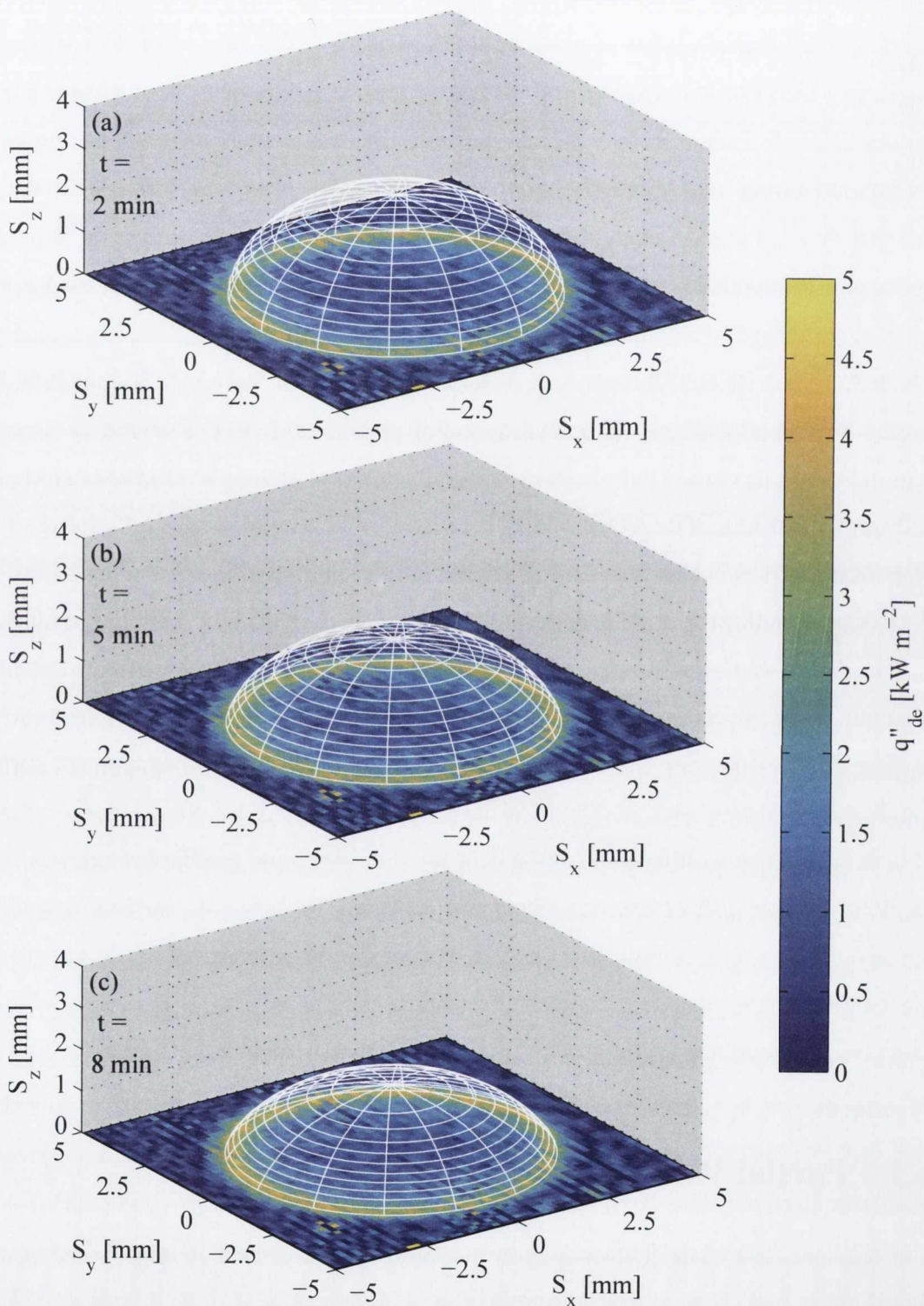


Figure 6.2: Partial wetting droplet evaporation. (a) $t = 2$ min, (b) $t = 5$ min, and (c) $t = 8$ min. Total evaporation time ≈ 21 min. $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

heat flux is observed around the triple contact line region for all time points. This result is further highlighted in Figure 6.3.

Figure 6.3 compares the droplet boundary captured by the optical camera (Figure 6.3a) with that of the radial heat flux energy balance calculated from temperature data acquired by the IR camera. Figure 6.3a incorporates the droplet boundary defined from the processing

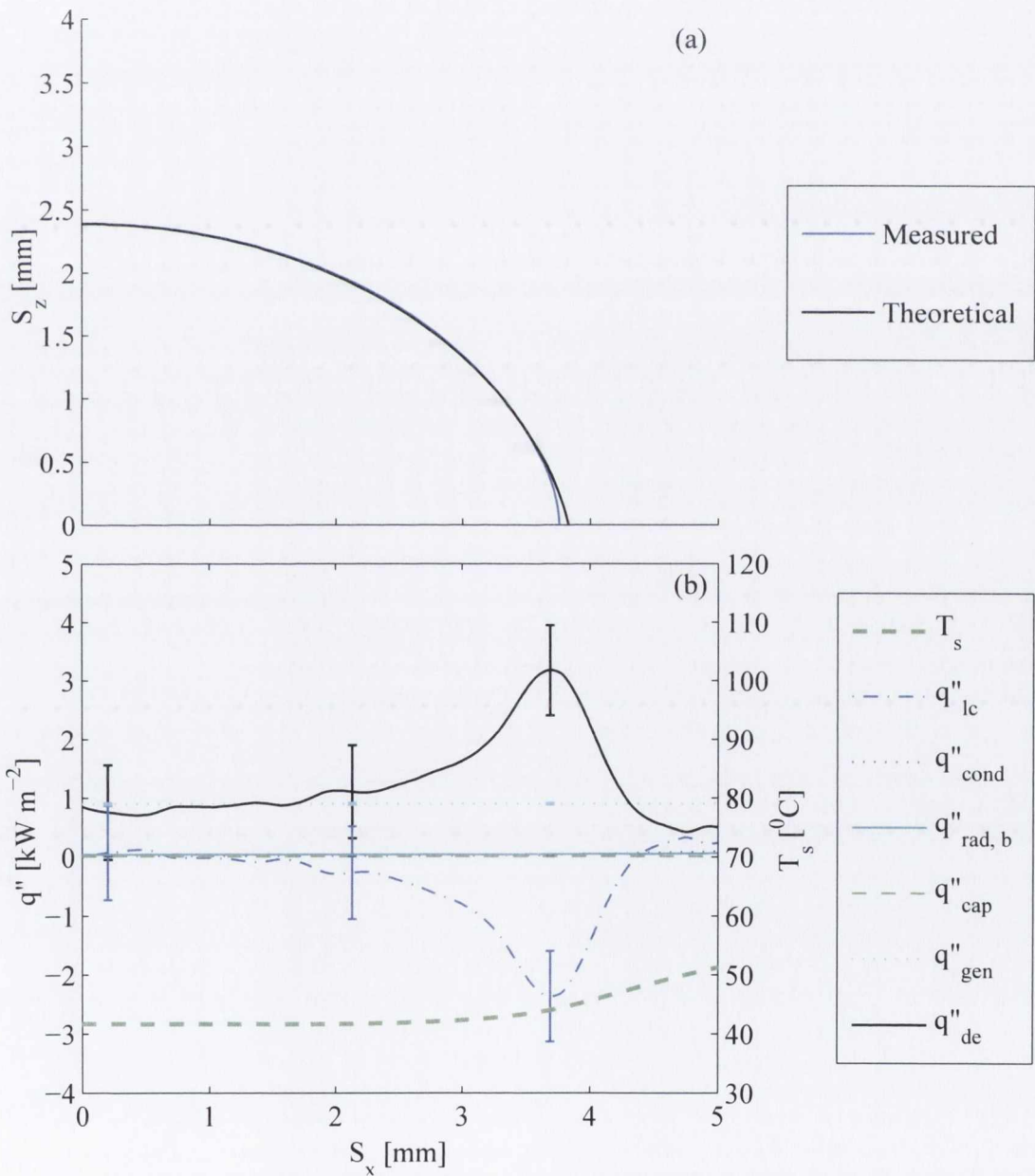


Figure 6.3: Partial wetting droplet (a) liquid-gas interface and (b) local heat flux energy balance. $t = 2$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^{\circ}$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

6.1. PARTIAL WETTING DROPLET EVAPORATION

Table 6.2: Lateral conduction literature comparison [116, 117].

	Present study	Karchevsky et al. [116, 117]
Material	Stainless steel AISI 316	Constantan
Foil thickness, δ_f [μm]	25	25
Droplet volume, V_d [μL]	8 – 80	5.1 – 39.3
Contact angle, θ [$^\circ$]	52.5 – 160	50 – 75
Base radius, R_b [mm]	0.7 – 3.8	1.84 – 3.59
q''_{gen} [W m^{-2}]	912	170 – 1,700
q''_{lc} [W m^{-2}]	2,500 – 5,300	940 – 3,357

steps outlined in Section 3.3.2.1 as well as that calculated from the solution of the Young-Laplace equation (Equation 3.34). Good agreement between the measured and theoretical boundaries was observed. Some discrepancy is noted in the region near the triple line where reflections, shadowing and mirage effects can distort the measured droplet boundary. These theoretical and measured results are discussed further in Figure 6.6.

Figure 6.3b shows the individual energy balance terms and their associated uncertainty in calculating the local convective heat flux to the evaporating droplet. A peak local heat flux value of 3.29 kW m^{-2} is calculated at $S_x = 3.86 \text{ mm}$. This corresponds with the region around the triple line of the evaporating droplet. This result stems from the lower thermal resistance from the reduced film thickness in the transition region as discussed in Section 2.2.6. Total droplet evaporation was completed after approximately 21 minutes.

The lateral conduction profile shape and magnitude in figure 6.3b shows good agreement with the results presented by Karchevsky et al. [116, 117]. A comparison of the experimental parameters between Karchevsky et al. and the present study are shown in Table 6.2. Karchevsky et al. only presented data on instantaneous conjugate heat transfer and did not indicate the transient nature of the triple line. Direct comparison of experimental data is difficult as this is the first thin foil study of the droplet evaporation evolution. The majority of previous droplet evaporation experiments on heated substrate have been conducted on isothermal substrates for smaller contact line length and initial volumes than the present study [149–151, 181]. As highlighted in Section 3.1.1.3 the CWT boundary condition enables much higher heat fluxes than the thin-foil technique under similar experimental conditions. Also as shown by Horacek et al. [73] and Sodtke and Stephan [72] triple line length, which

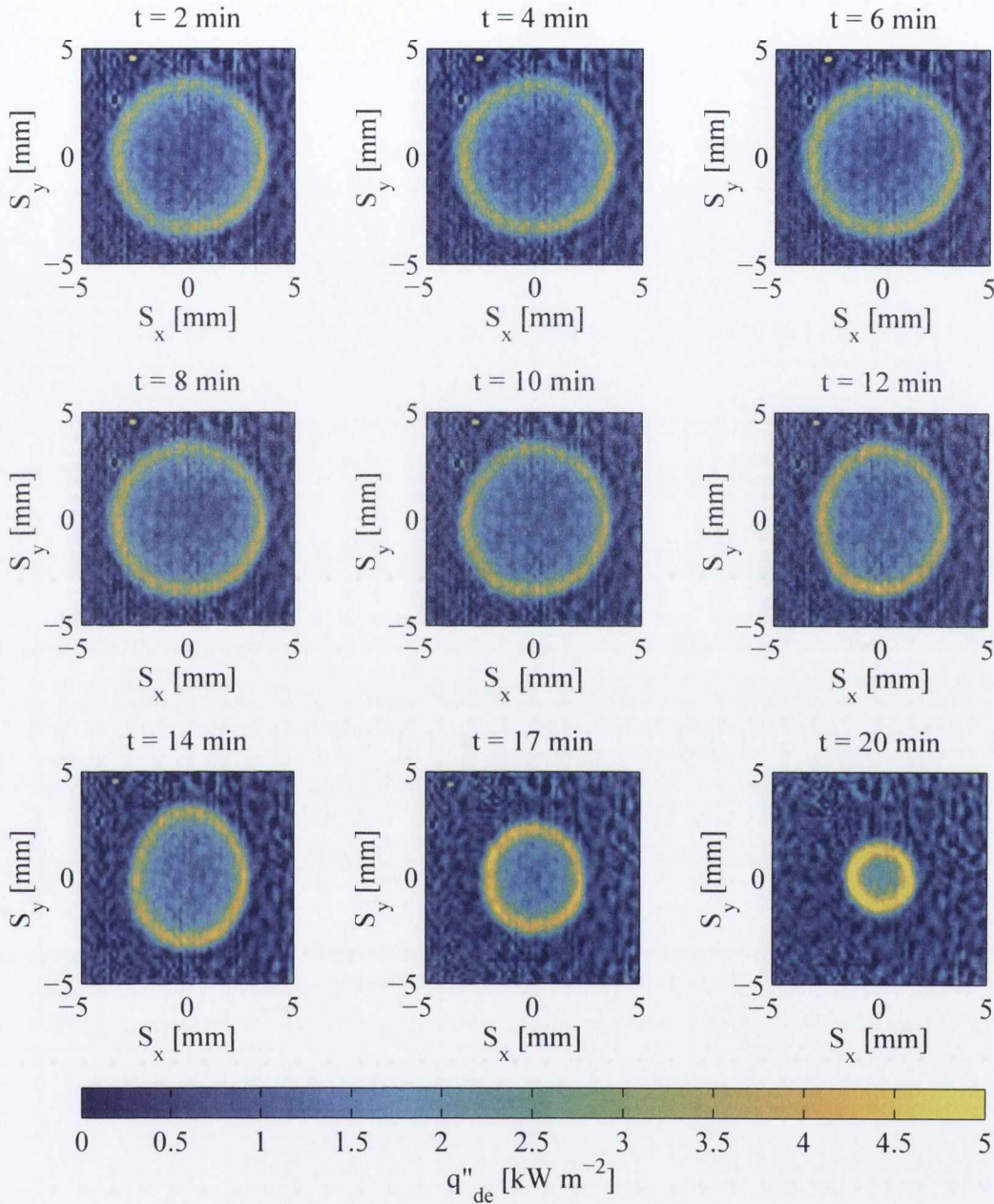


Figure 6.4: Partial wetting droplet evaporation heat flux distribution over time. $t = 2 - 20$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

scales with base droplet radius, is a key parameter for heat and mass transfer to an evaporating droplet. As such, only droplet evaporation trends and not direct magnitudes have been compared.

The local heat flux distribution of the evaporating droplet over time is shown in Figure 6.4. From $t = 2 - 8$ min the droplet evaporates in the constant contact radius (CCR) regime. At

6.1. PARTIAL WETTING DROPLET EVAPORATION

about $t = 10$ min, the droplet contact angle decreases to the receding contact angle of this droplet - substrate case. As a result from $t = 10 - 14$ min the contact line becomes unpinned, occurring in a asymmetric manner resulting in an oblate thermal footprint, evaporating in a stick-slip mode. This mode is characterised by rapid oscillation of the triple contact line between pinned and unpinned evaporation. From $t = 17 - 20$ min the droplet evaporates with a receding contact line with a near circular footprint once again. This evaporation process is similar in proceedings to that previously observed in the literature [125, 136, 137, 149, 150, 181]. Droplet evaporation for $t < 2$ min is not considered here as the system behaves in a transient manner before this point as a result of the step input of heat into the foil and subsequent rapid transients in temperature at $t = 0$ min. Of particular interest is the evaporation behaviour of the CCR regime and thus fixed CLL. Investigation of heat

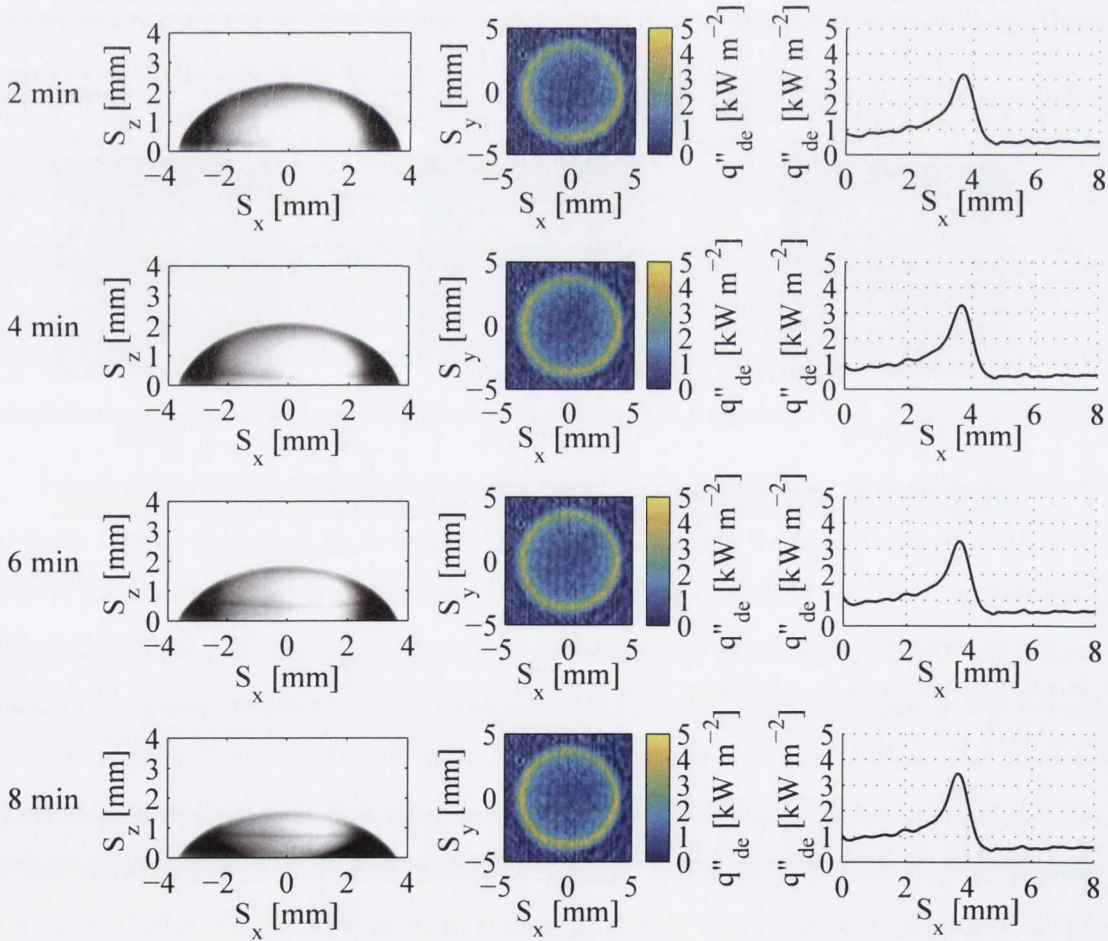


Figure 6.5: Partial wetting droplet evaporation, shape, heat flux distribution and radial heat flux profile over time. $t = 2 - 8$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

transfer over this time scale demonstrates the influence if any on the droplet contact angle. As such only the initial pinned evaporation ($t = 2 - 8$ min) will be considered for further analysis. Moving triple line evaporation behaviour will be detailed for the coated substrate in Section 6.2.

Figure 6.5 demonstrates the droplet shapes, heat flux distributions and local radial heat flux profiles of the evaporating droplet over the pinned evaporation regime. The left column displays the droplet boundary profile, the centre column shows the local convective heat flux distribution and the last column shows the radial convective heat flux profile. This data indicates the droplet is evaporating in the CCR mode. The radial heat flux profile highlights that the peak convective heat transfer is at the triple line region for all time points. Clear change in the droplet morphology over this period yields no significant change in the shape or magnitude of the local heat flux profile due to the pinned triple line. From this result one can conclude that despite a clear change in the apparent droplet contact angle no change occurs in the thin film transition region due to the near constant heat flux distribution.

Figure 6.6 shows how multiple droplet properties vary across the CCR evaporation phase. Data from both the measured interface and the Young - Laplace solution is shown for comparison. As the Young - Laplace solution is dependent on the apex radius, droplet volume, and height, good agreement is observed for these parameters since they are measured accurately. Therefore the most useful aspect of the Young-Laplace solution is in determining a better approximation of the droplet contact angle, as shadowing, reflections and mirage effects can distort the droplet boundary at the triple line [173]. Droplet volume, contact angle, height, and the surface area of the liquid-gas interface are seen to decrease linearly over the pinned evaporation regime (Figure 6.6a, b, d, and f). These results correlate well with that previously observed in the literature [149–151, 181] for pinned droplet evaporation on heated substrates. The radius remains constant, which is indicative of the CCR evaporation regime, while the apex radius increases. Decreasing contact angle, droplet height and surface area is sensible given the pinned CCR evaporation regime, as does the increasing apex radius for a fixed triple line with decreasing droplet volume. The linear nature of the decreasing droplet volume suggests a constant thermal energy input across the liquid-solid interface. This is in agreement with the local radial heat flux profiles of Figure 6.5 and is further confirmed by the data presented in Figure 6.7.

6.1. PARTIAL WETTING DROPLET EVAPORATION

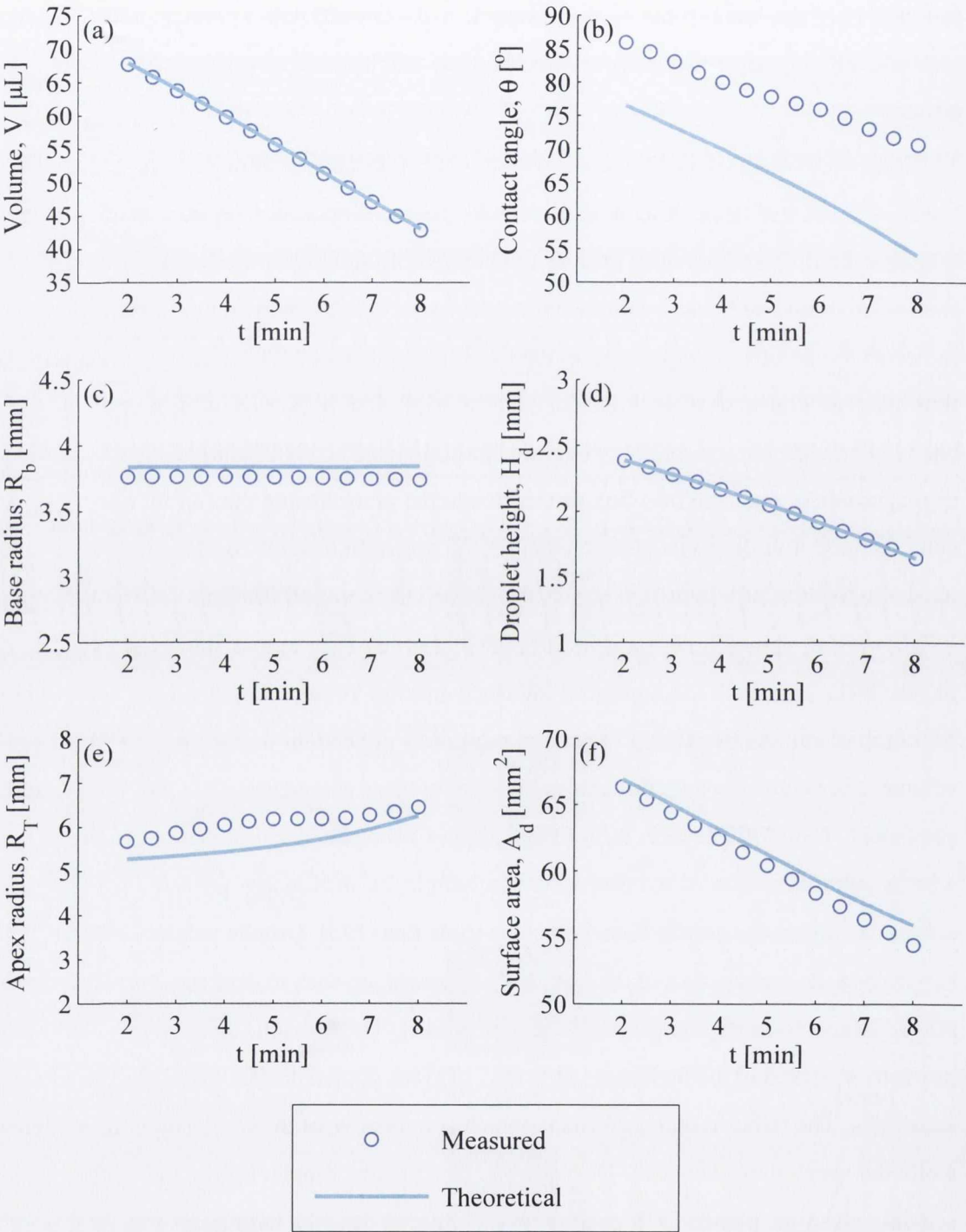


Figure 6.6: Partial wetting droplet geometric properties through time. (a) volume, (b) contact angle, (c) base radius, (d) height, (e) apex principal radii, and (f) liquid-gas surface area. $t = 2 - 8$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

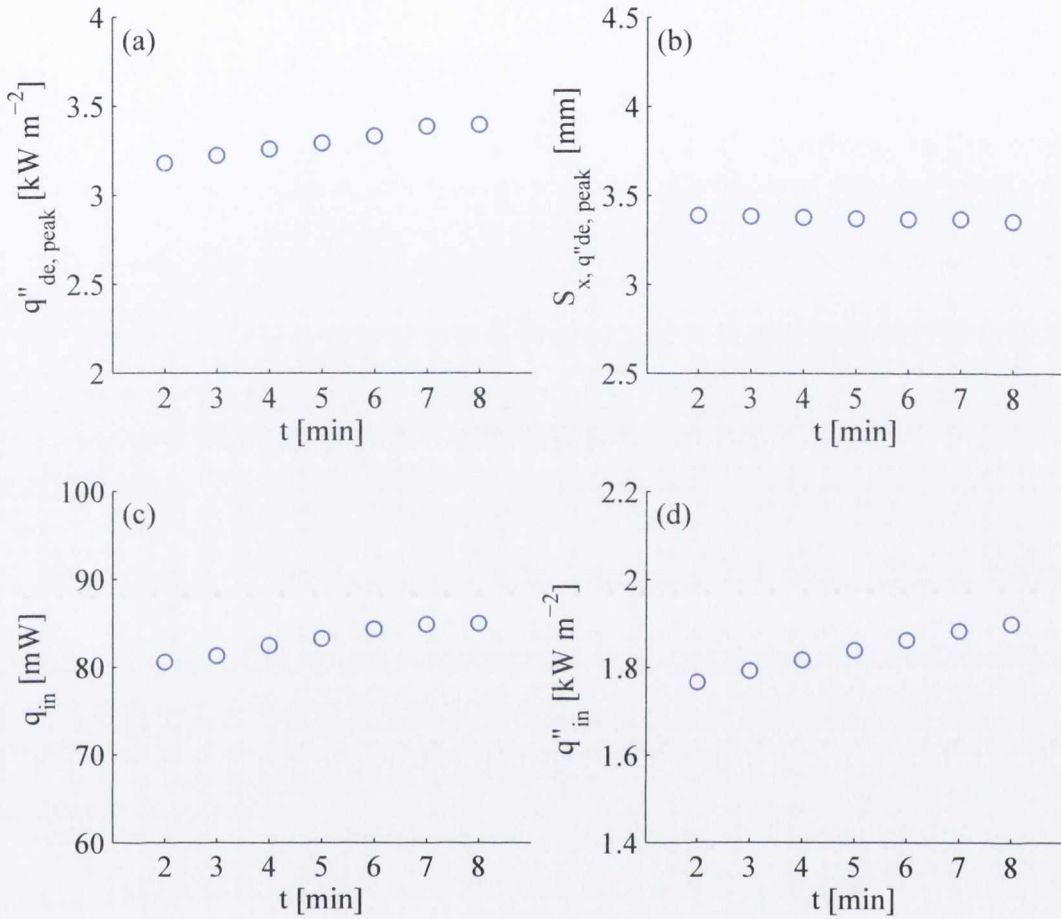


Figure 6.7: Partial wetting droplet evaporation heat transfer. (a) Peak heat flux magnitude, (b) location, (c) total power into droplet, and (d) average heat flux. $t = 2 - 8$ min, $V_{d,0} = 80 \mu L$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 W m^{-2}$.

Figure 6.7a and b show the magnitude and location of the peak local heat flux of the evaporating droplet. Figure 6.7c and d demonstrates the total power and the average heat flux into the droplet across the solid-liquid interface. A marginal increase of 5.5% is observed in q''_{in} and q_{in} across this range. As the droplet evaporates in the CCR regime the interface area between the solid substrate and liquid droplet and thus CLL remains constant, while the liquid-gas interface decreases. As the evaporation proceeds the thickness of the droplet decreases. This marginally reduces the thermal resistance across the height of the droplet resulting in an increase in temperature of the liquid at the liquid-gas boundary. This increasing temperature at the liquid-gas boundary results in greater convective heat transfer from the boundary to surroundings. Thus, the decreasing liquid-gas surface area is offset by an

6.1. PARTIAL WETTING DROPLET EVAPORATION

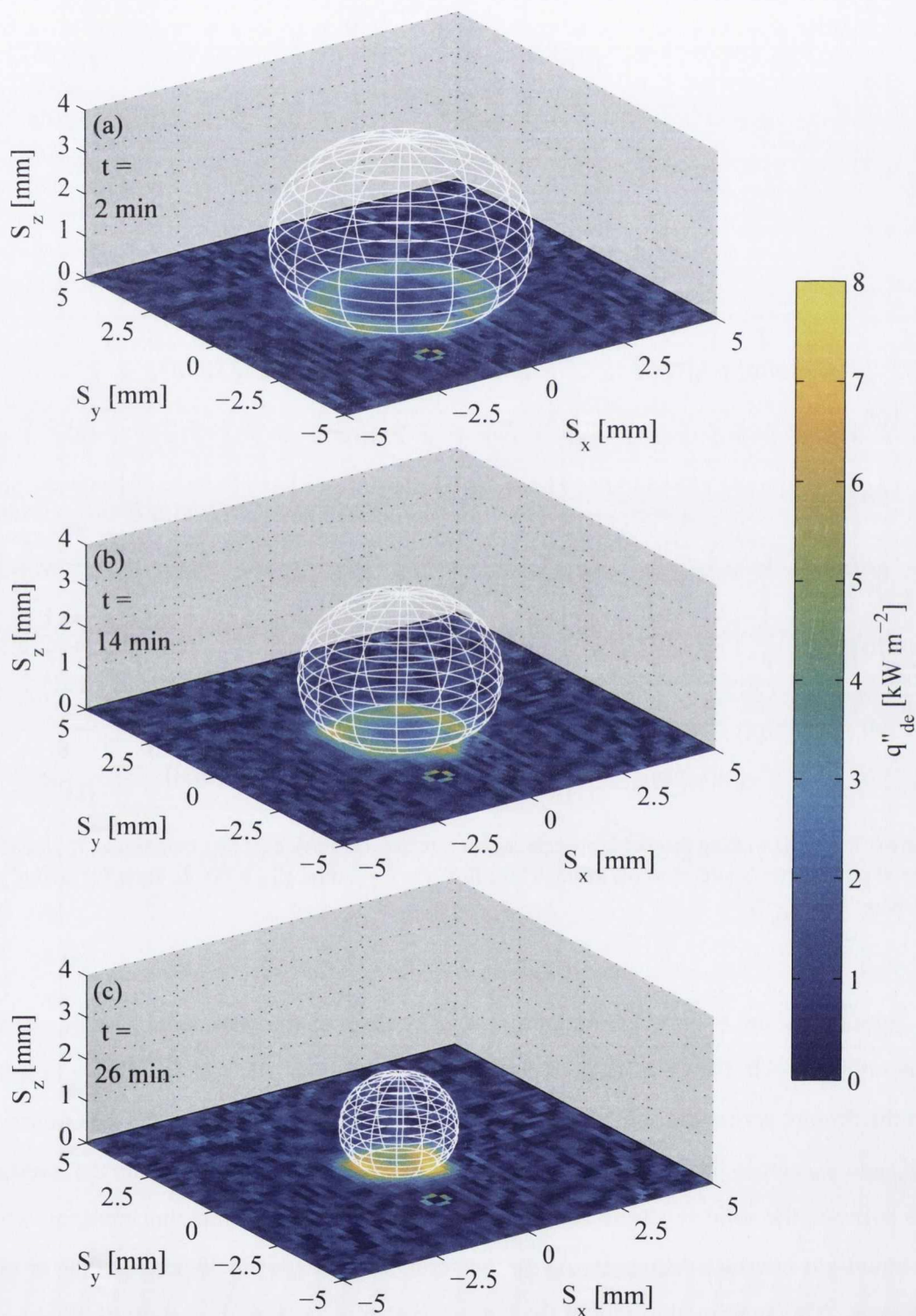


Figure 6.8: Partial non-wetting droplet evaporation. (a) $t = 2$ min, (b) $t = 14$ min, and (c) $t = 26$ min. Total evaporation time ≈ 31 min. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

increasing liquid-gas interface temperature [181]. This is highlighted by the near constant thermal energy across the solid-liquid boundary (Figure 6.7a, c and d). This is despite a 16% decrease in the surface area of the liquid-gas interface across this CCR evaporative regime. This result shows good agreement with that observed in the literature [149, 150, 181], where the evaporation rate is limited by buoyant convective vapour transport at the liquid-gas interface.

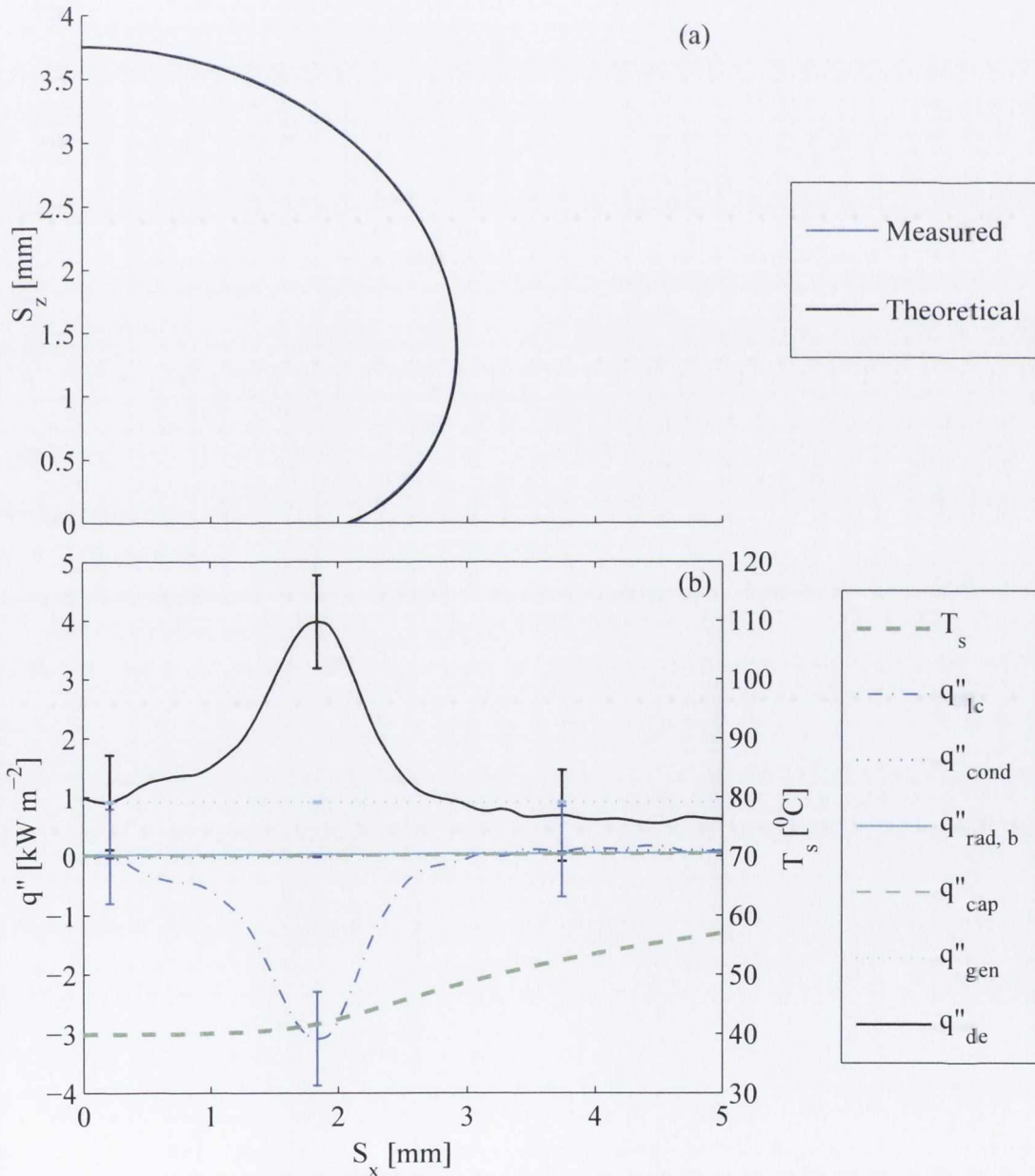


Figure 6.9: Partial non-wetting droplet. (a) liquid-gas interface and (b) local heat flux energy balance. $t = 2$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^{\circ}$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

6.2 Partial Non-Wetting Droplet Evaporation

In contrast to the pinned evaporation observed for the partial wetting substrate, droplet evaporation on the coated stainless steel surface was also investigated. The coating resulted in a pretesting droplet contact angle of $\theta_A = 160^\circ$ and $\theta_R = 159^\circ$, which corresponds to a partially non-wetting droplet. Figure 6.8 shows the 3D droplet evaporation process at three different times. The droplet evaporated in a stick-slip regime over the presented time scale, and thus increasing CLL. As with the partial wetting case, discussed in the previous section, the highest heat flux is observed around the triple contact line region. This point is further highlighted in Figure 6.9.

Figure 6.9 compares the droplet boundary captured by the optical camera (Figure 6.9a) with that of the radial heat flux energy balance calculated from data acquired by the IR camera at $t = 2$ min. A peak local heat flux value of 3.95 kW m^{-2} occurs at $S_x = 1.68 \text{ mm}$. This peak heat flux is again located in the thin film transition region near the droplet triple line. Total droplet evaporation was completed after approximately 31 minutes.

Figure 6.10 highlights the local heat flux distribution of the partial non-wetting evaporating droplet over time. The droplet footprint remains near axisymmetric over the evaporation period. The droplet is observed to evaporate with a stick-slip contact line. This is due to the droplet contact angle achieving the receding contact angle of this droplet - substrate condition for $t < 2$ min. The droplet heat flux footprint decreases through time, while the magnitude of the heat flux increases. This stick - slip mode continues until $t \approx 27$ min at which stage the rate at which the triple line sticks and slips begins to slow with the contact line unpinning in a stochastic manner to final complete droplet evaporation ($t = 28 - 31$ min). The stick-slip evaporation behaviour is of particular interest and is in contrast to the pinned evaporation observed in the partial wetting case. Only data on the stick - slip mode of evaporation ($t = 2 - 26$ min) will be discussed here.

Figure 6.11 shows the droplet morphology, convective heat flux distribution and local convective radial heat flux profiles of the partial non-wetting evaporating droplet. The stick-slip mode of evaporation is qualitatively observed from the decreasing droplet base, height and contact angle. As the droplet evaporates, the radial heat flux profile indicates that the location of the peak heat flux decreases while its magnitude increases over time. As will

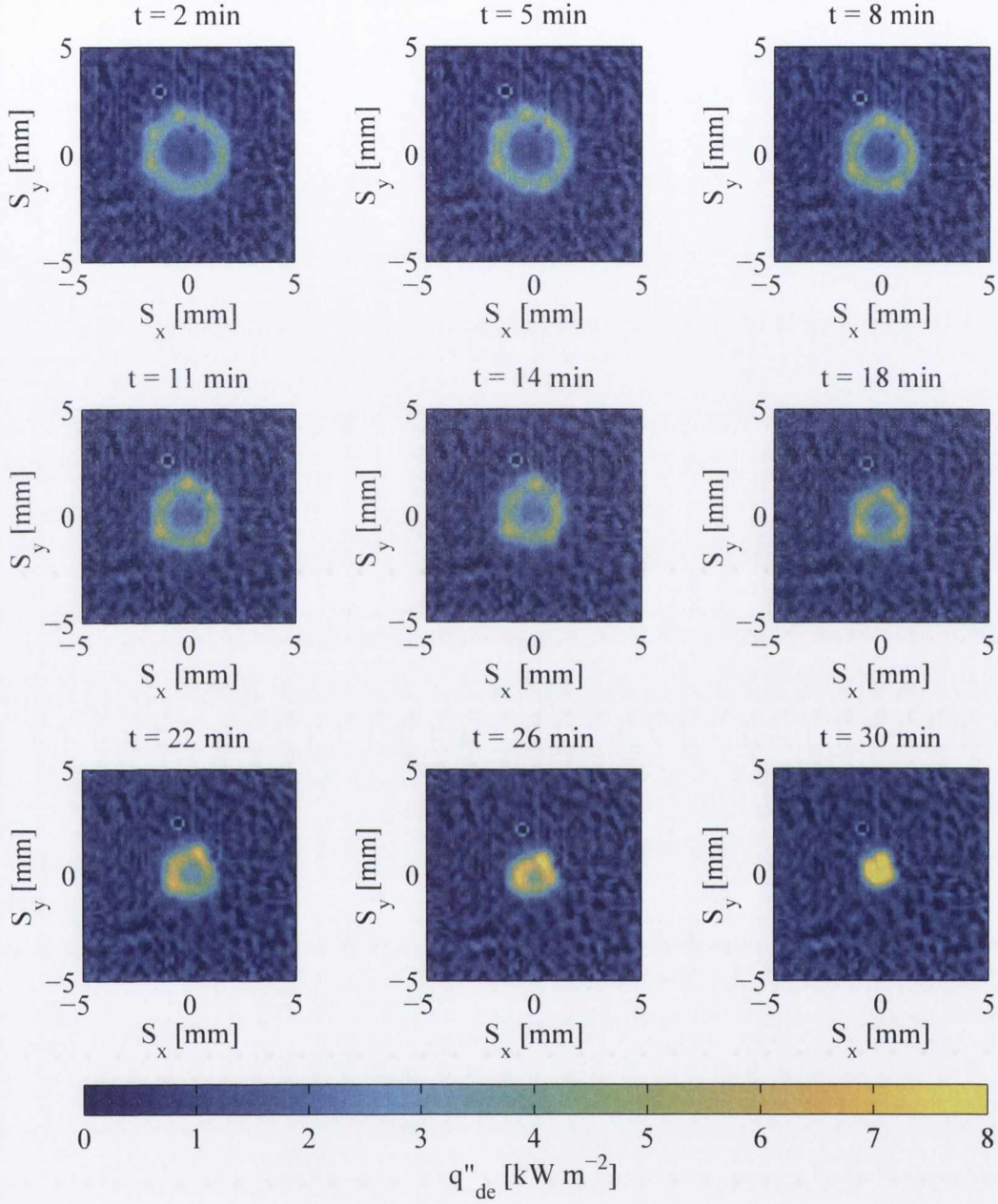


Figure 6.10: Partial non-wetting droplet evaporation heat flux distribution over time. $t = 2 - 30$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

be discussed this is an interesting result since the decreasing liquid - solid interface area is partially offset by the increase in the triple line convective heat flux.

Figure 6.12 shows how the geometric properties of the droplet vary across the partial non-wetting evaporation. Results from both the measured and the Young - Laplace solution are shown for comparison. As with the partial wetting droplet evaporation discussed in

6.2. PARTIAL NON-WETTING DROPLET EVAPORATION

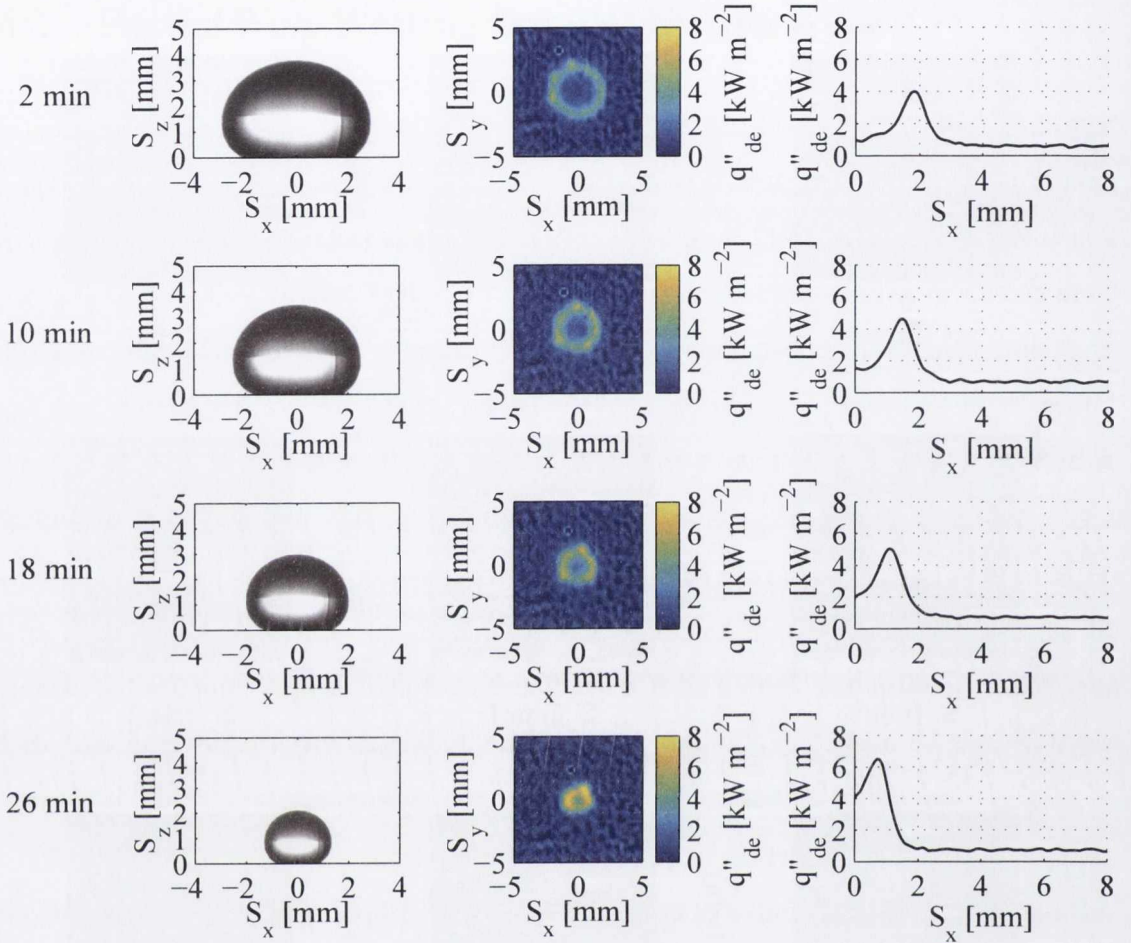


Figure 6.11: Partial non-wetting droplet evaporation, shape, heat flux distribution and radial heat flux profile over time. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

Section 6.1 the largest discrepancy between the measured interface and the Young - Laplace solution is in the contact angle results. As discussed this stems from difficulties measuring the triple line region of the evaporating droplet. All parameters are generally observed to decrease over the course of the droplet evaporation. The decreasing contact angle, droplet radius and height give credence to the stick-slip evaporation regime. The trends and shape for contact angle, droplet height and base radius are in good agreement with those observed by Bourges-Monnier and Shanahan [125] for their unpinned evaporation of n-heptane droplets on PTFE. They attributed the stick-slip nature of the evaporation to the surface roughness of their substrate.

Figure 6.13a and b shows the magnitude and location of the peak local heat flux of the evaporating droplet. A generally increasing $q''_{de,peak}$ and decreasing $S_{x,q''_{de,peak}}$ is observed.

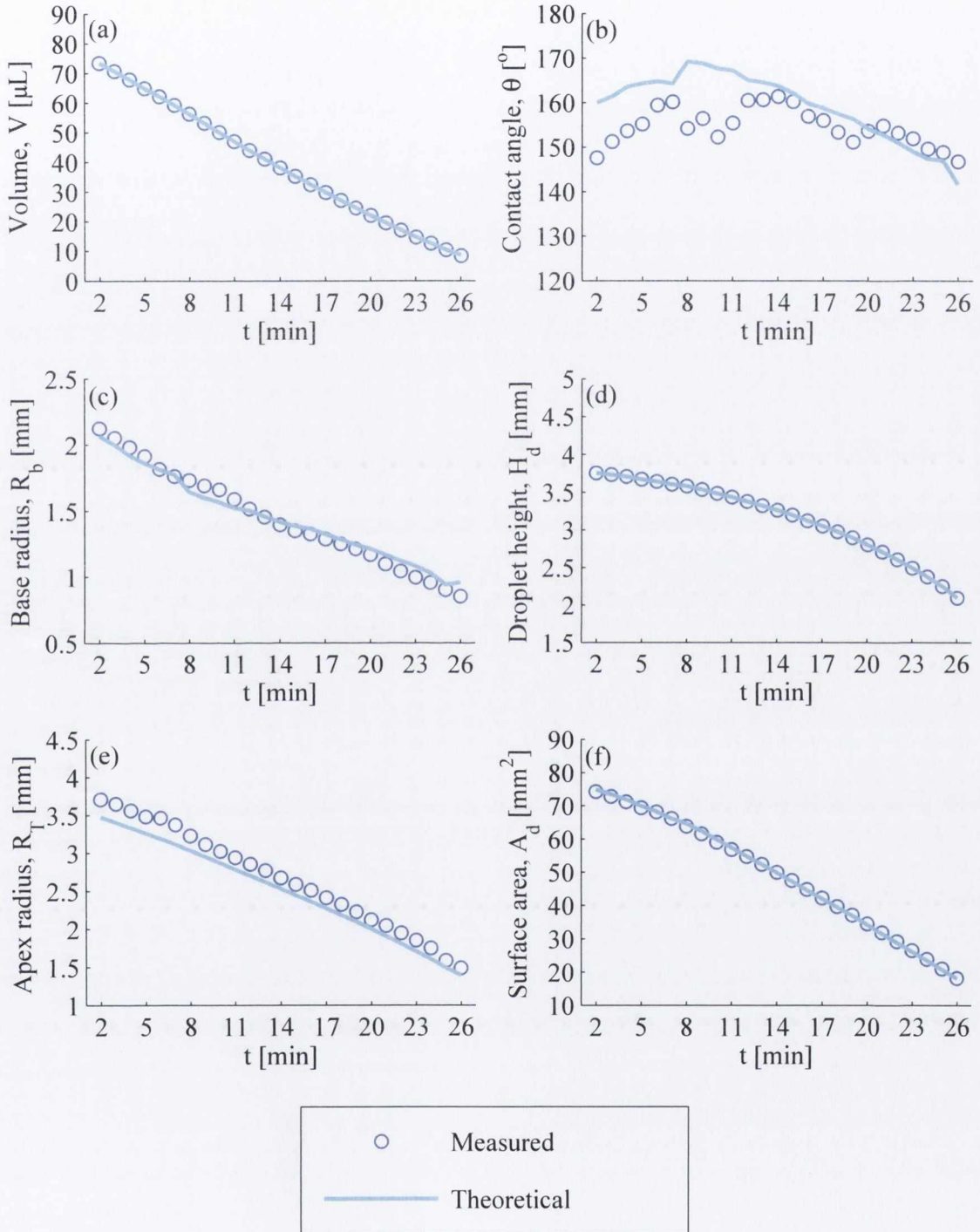


Figure 6.12: Partial non-wetting droplet geometric properties through time. (a) Volume, (b) contact angle, (c) base radius, (d) height, (e) apex principal radii, and (f) liquid-gas surface area. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

6.2. PARTIAL NON-WETTING DROPLET EVAPORATION

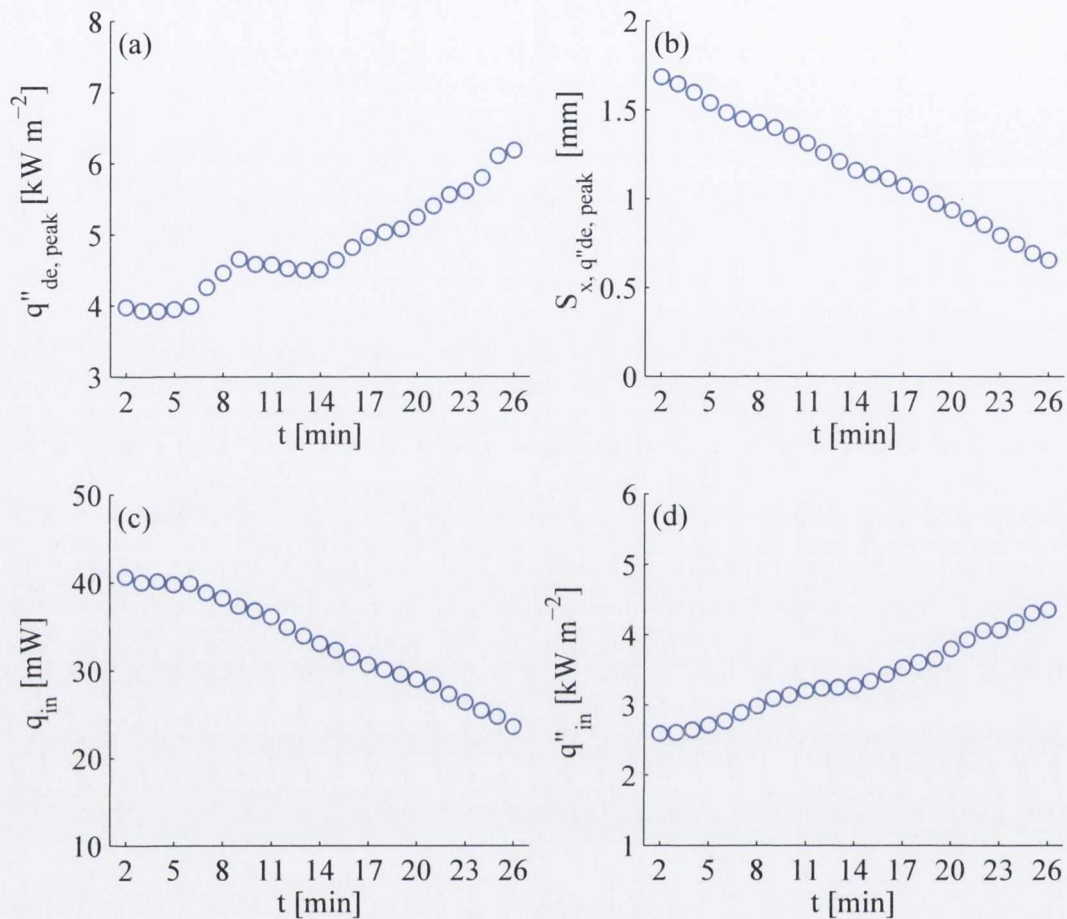


Figure 6.13: Partial non-wetting droplet evaporation heat transfer. (a) Peak heat flux magnitude, (b) location, (c) total power into droplet, and (d) average heat flux. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

Figure 6.13c and d plots the total power and the average heat flux into the droplet across the solid-liquid interface. Decreasing values of q_{in} and increasing values of q''_{in} are observed for increasing CLL. As the droplet evaporates in the stick-slip regime the interface area between the solid substrate and liquid droplet decreases. As this occurs the thickness of the droplet also decreases. This 79% and 75% reduction in the respective solid-liquid and liquid-gas interface results in a reduction in the total thermal energy q_{in} into the droplet. However an increase in the heat flux q''_{in} into the droplet is observed due to the reduced thermal resistance through the droplet as the droplet evaporates, resulting in an increase in the liquid-gas boundary temperature. Thus, the rate at which the heat transfer drops due to the decreasing area of heat transfer is moderated by the increase in the convective heat flux.

The trend for q_{in} over time in Figure 6.13c correlates well with that observed by Sobac and Brutin [136] for their unpinned natural evaporation of ethanol and toluene on PTFE coated isothermal substrate. The result of increasing q''_{in} with increasing CLL is in good agreement with that observed by Crafton and Black [149] and Gatapova et al. [181] for their unpinned heated substrate evaporation of n-heptane and water droplets respectively. Good correlation of the trend of $q''_{de,peak}$ is observed for receding contact line to that shown in Chapter 4 for the pool cooling case. As the flow rate is decreased in the pool cooling regime the wetted area decreases and CLL increases resulting in a similar increase in $q''_{ec,peak}$ (Figure 4.8) as shown in Figure 6.13a.

6.3 Evaporation Regime Comparison

Figures 6.14 and 6.15 contrast the partial wetting and partial non-wetting evaporation modes in terms of interface profile and radial heat flux distribution over the time take to evaporate $\sim 50\%$ of their initial volume. Figures 6.14 shows the pinned mode of evaporation and

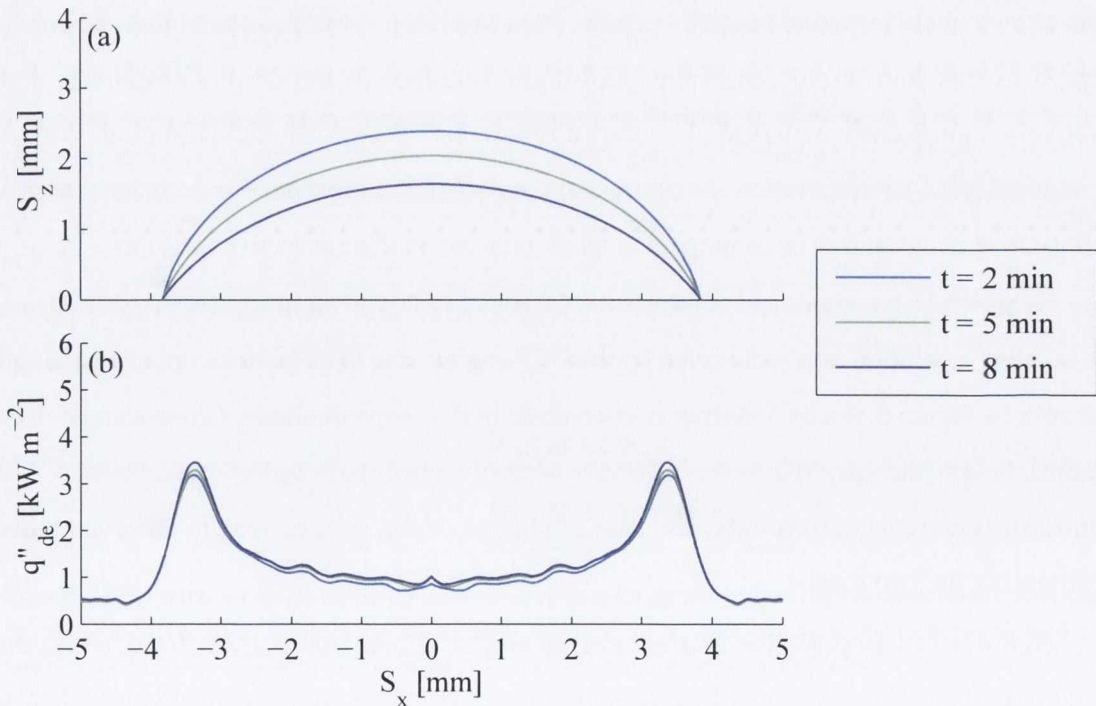


Figure 6.14: Partial wetting droplet evaporation. (a) liquid-gas interface, (b) convective heat flux profile. $t = 2 - 8$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

6.3. EVAPORATION REGIME COMPARISON

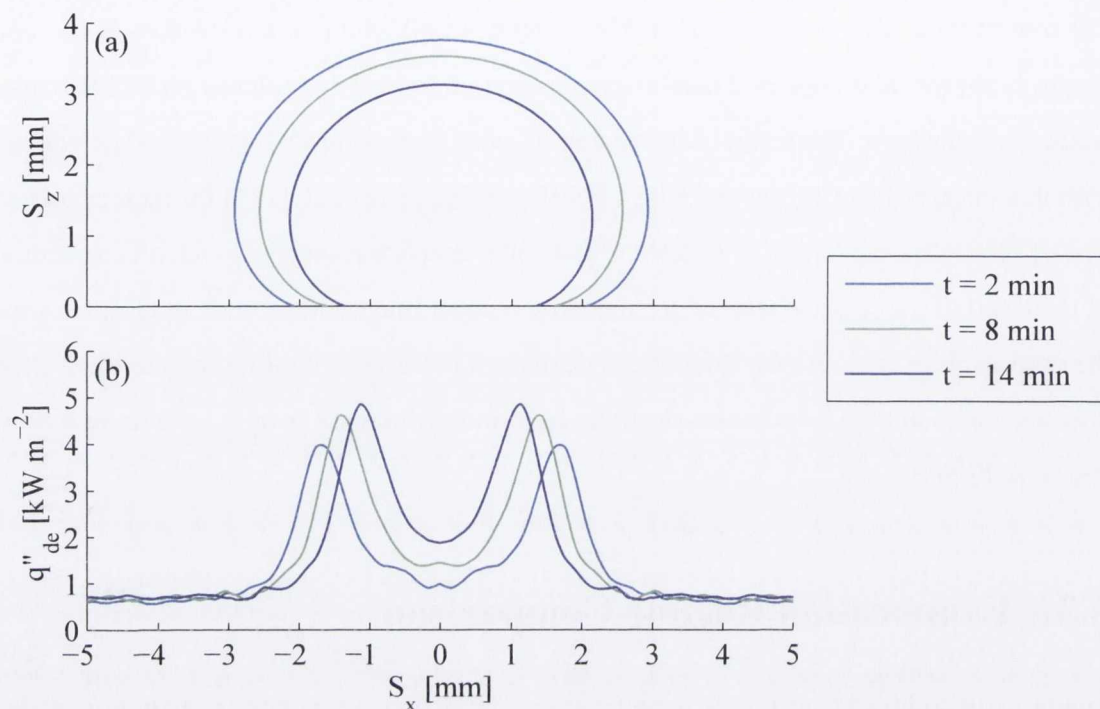


Figure 6.15: Partial non-wetting droplet evaporation. (a) liquid-gas interface, (b) convective heat flux profile. $t = 2 - 14$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

the almost invariant radial heat flux profile. As a result the subsequent total thermal energy and heat flux transferred across the solid-liquid interface as the droplet evaporates does not vary significantly with time. In contrast Figure 6.15 shows the stick-slip mode with its moving triple line evaporation. In contrast to the partial wetting case, the decreasing area of convective heat transfer is accompanied by an increase in the convective heat flux.

Figures 6.16 compares and contrasts the CCR and stick-slip regime geometric parameters. A higher evaporation rate is observed by the CCR regime due to its greater contact line length as seen in Figure 6.16a by a steeper rate decrease in the droplet volume. Contact angle, base radius, height and apex radius are indicative of their respective evaporation regimes. A near uniform liquid-gas surface area and rate of change of the surface area is observed when comparing the two modes.

Figure 6.17 compares the thermal energy and its spread that is transferred across the solid-liquid interface. A greater local peak heat flux is noted for the stick-slip regime, whereas a greater location of this peak is found for the CCR mode of evaporation. Despite their similar liquid-gas surface areas the CCR regime dissipates over 2 times as much thermal

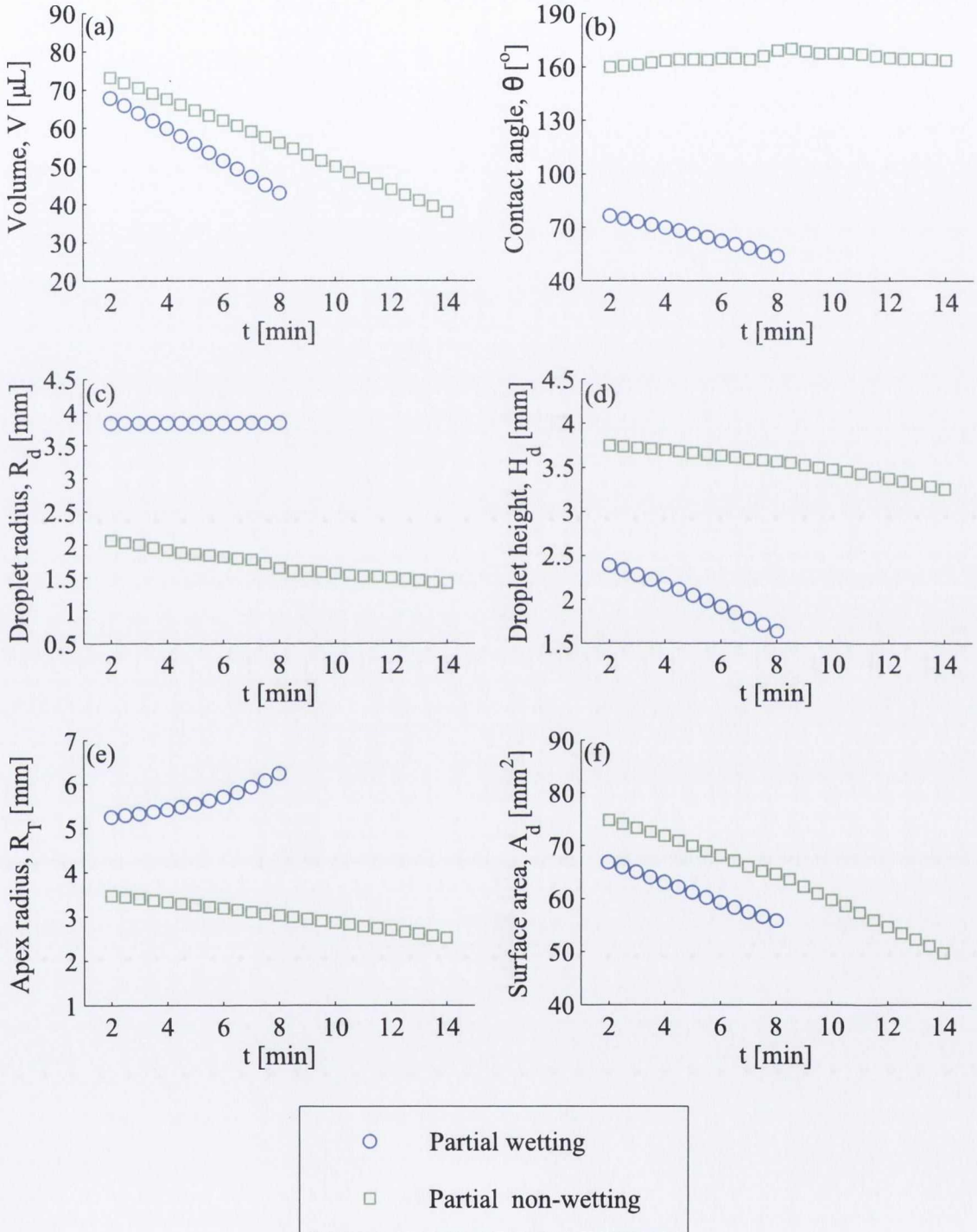


Figure 6.16: Partial wetting and partial non-wetting droplet geometric properties comparison over time. (a) volume, (b) contact angle, (c) base radius, (d) height, (e) apex principal radii, and (f) liquid-gas surface area. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79 - 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

energy. This is in keeping with the observations of Bourges-Monnier and Shanahan [125] and Picknet and Bexon [111] that for a constant volume the droplet with the greater contact

6.3. EVAPORATION REGIME COMPARISON

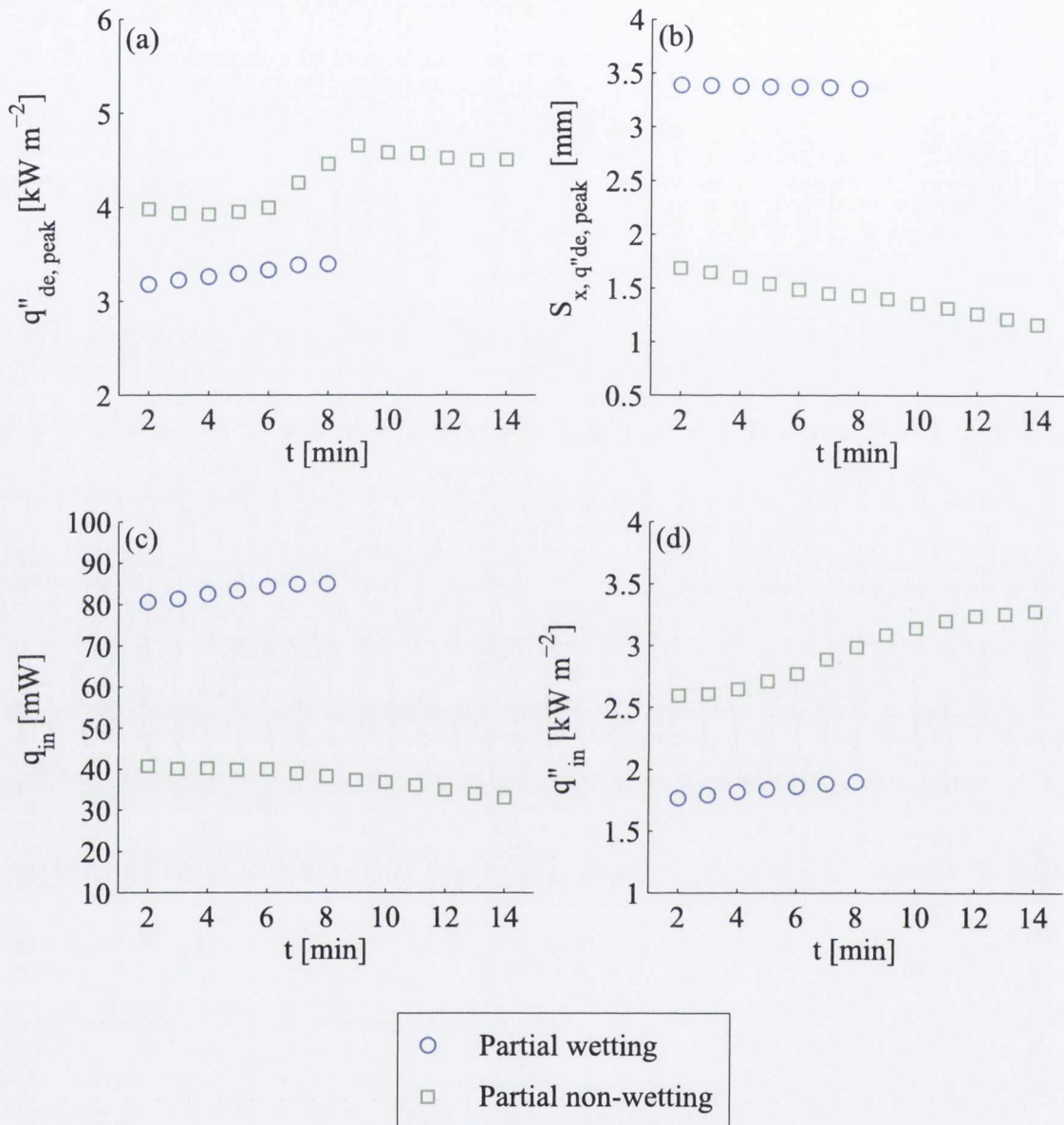


Figure 6.17: Partial wetting and partial non-wetting droplet evaporation heat transfer over time. (a) Peak heat flux magnitude, (b) location, (c) total power into droplet, and (d) average heat flux. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79 - 160^\circ$, and $q''_{gen} = 912 \text{ W m}^{-2}$.

area and thus triple line length with evaporate faster. A greater average heat flux is noted for the partial non-wetting case due to its reduced solid-liquid interface area but increased CLL for a similar liquid-gas surface area in comparison with the CCR regime of evaporation. If the objective is to evaporate a droplet as fast a possible the CCR mode with a partial wetting substrate and maximum CLL is optimal. As it supports the highest thermal power into the droplet due to its larger triple line transition region area, and thus lower thermal

resistance [K W^{-1}]. However, if the aim is to remove thermal energy as efficiently possible, the partial non-wetting substrate in the stick-slip mode is desired. As it achieves a larger average convective heat flux across its solid-liquid interface, due to its larger CLL and thus lower specific thermal resistance [cm K W^{-1}].

Comparing the partial non-wetting droplet at $t = 2$ min and $t = 26$ min over a similar wetted area highlights the importance of CLL in cooling. A 4.66 times greater wetted area (solid-liquid interfacial area) is noted for the $t = 2$ min case. Linear scaling of the partial non-wetting droplet properties is assumed, such that 4.66 partial non-wetting droplets at $t = 26$ min are evaporating on a heated substrate, to compare the cases over a similar wetted areas. The relevant parameters of the two cases and the calculated scaled partial non-wetting droplet results are shown in Table 6.3. The fundamental comparison is the thermal energy dissipated from the substrate. 2.7 times more thermal energy ($40.68 - 109.91$ mW) is dissipated by having multiple smaller droplets in comparison to a singular large droplet. This increase in q_{in} is achieved despite a 40% decrease in working fluid volume for the multiple droplet case. The total liquid-gas interfaces of both cases is similar ($74.91 - 84.01$ mm²) with a 12% increase for the multiple droplet case. The droplet height decreased from $3.75 - 2.10$ mm, resulting in a reduced thermal resistance for the multiple droplet case. This would result in a greater liquid-gas interface temperature and thus convective heat flux to surroundings. The CLL increases by 2.15 times from $0.97 - 2.09$ mm⁻¹. These result demonstrate the importance CLL on thermal energy dissipation. Given the similar wetted area and liquid-gas interface, the 2.7 times increase of q_{in} can be strongly attributed to the increasing CLL. This

Table 6.3: Wetted area partial non-wetting CLL comparison.

	Partial non-wetting	Partial non-wetting	Scaled partial non-wetting	
Time	2	26	26	[min]
Volume	73.17	8.71	40.55	[μL]
Contact angle	160	141.6	141.6	[$^{\circ}$]
Height	3.75	2.10	2.10	[mm]
Liquid-gas interface	74.91	18.04	84.01	[mm ²]
Wetted area	13.40	2.88	13.40	[mm ²]
CLL	0.97	2.09	2.09	[mm ⁻¹]
q_{in}	40.68	23.6	109.91	[mW]
q''_{in}	2.60	4.35	4.35	[kW m ⁻²]

6.3. EVAPORATION REGIME COMPARISON

result builds on the work of Horacek et al. [73] and Sodtke and Stephan [72] and further explains the higher heat flux noted for the evaporative electrospray cooling (multiple micro droplets all with their own contact lines) in comparison to the pool cooling case (singular droplet and contact line).

Applying a similar methodology but this time comparing the two droplet wetting conditions over a similar wetted area yields further understanding of the influence of substrate wettability on cooling. Taking the partial wetting droplet at $t = 2$ min and comparing it with the partial non-wetting droplet at $t = 26$ min. The partial wetting droplet has a 16.61 times greater wetted area (solid-liquid interfacial area) than the partial non-wetting droplet at the defined values of t for each case. Again, we assume linear scaling of the partial non-wetting droplet properties, such that 16.61 partial non-wetting droplets are evaporating on a heated substrate, to compare the two cases over a similar wetted areas. The relevant parameters of the two cases and the scaled partial non-wetting droplet results are shown in Table 6.4. 4.87 times more thermal energy (80.55 – 392.02 mW) is dissipated by having multiple smaller partial non-wetting droplets in comparison to a singular large partial wetting droplet. This increase in q_{in} is achieved for a 2.13 times increase in working fluid volume for the multiple partial non-wetting droplet case. A 4.48 times increase in the liquid-gas surface area is noted for the multiple droplet case (66.91 – 299.66 mm²). The droplet height is similar for both cases 2.38 – 2.10 mm, resulting in a comparable thermal resistance through the droplet in both cases. The CLL increases by 4.08 times from 0.51 to 2.09 mm⁻¹. These results highlight the influence of surface wettability. Due to the larger droplet contact angle of the

Table 6.4: Wetted area surface wettability comparison.

	Partial wetting	Partial non-wetting	Scaled partial non-wetting	
Time	2	26	26	[min]
Volume	67.82	8.71	144.65	[μL]
Contact angle	76.82	141.6	141.6	[°]
Height	2.38	2.10	2.10	[mm]
Liquid-gas interface	66.91	18.04	299.66	[mm ²]
Wetted area	47.78	2.88	47.78	[mm ²]
CLL	0.51	2.09	2.09	[mm ⁻¹]
q_{in}	80.55	23.6	392.02	[mW]
q''_{in}	1.77	4.35	4.35	[kW m ⁻²]

partial non-wetting case a greater increase in CLL and liquid-gas surface area for a similar droplet height (thermal resistance) is noted. This results in a greater increase (2.7 – 4.87 times) in thermal energy dissipated compared to the previous case. These results demonstrate that the optimum heat transfer for fixed wetted area is achieved for the evaporation that gives the greatest CLL and liquid-gas surface area, with greater dependency on CLL.

For the context of spray cooling of a fixed heated area, the ideal case would be a perfectly wetting substrate with droplet sizes on the same scale as the thickness of the transition region. In this system a continuous thin film, of the same thickness as the transition region (1 - 3 μm), forms on the heated substrate, where the top of the film is at the saturated temperature. This enables the liquid molecules on the top of the film to efficiently escape into the gas phase [66, 152]. For real world application this wetting condition and droplet size scale may not be possible. In this case, the optimum cooling corresponds to the evaporative regime with the smallest achievable droplet size to maximise CLL. In general for a constant wetted area and CLL, a droplet with a lower height, liquid-gas surface area, and contact angle results in greater cooling. As seen in the partial wetting droplet evaporation (Section 6.1), for a fixed wetted area from $t = 2 - 8$ min a 16% and 31% decrease in the liquid-gas surface area and droplet height respectively resulted in a 5.5% increase in q_{in} (Figures 6.6f and 6.7c). This is due to the increasing liquid-gas boundary temperature from the decreasing thermal resistance through the droplet. This results in greater thermal energy dissipation from the boundary despite the decreasing surface area. Thus, spraying onto a perfectly wetting substrate in the evaporative regime is more advantageous for optimum heat transfer.

6.4 Summary

Droplet evaporation for different surface wettability conditions was investigated using thin foil thermography. Pinned and stick-slip evaporation were observed for partial wetting and partial non-wetting heated substrates respectively. Thermal energy and heat flux was noted to marginally increase by 5.5% for the pinned evaporation case for a 16% and 31% decrease in the liquid-gas surface area and droplet height respectively. Whereas, for the stick-slip regime with its receding contact line and increasing CLL (0.97 – 2.09 mm^{-1}), thermal cooling decreased by 41% while heat flux increased by 67%. This was for a 79% and 76% decrease

6.4. SUMMARY

in the solid-liquid and liquid-gas surface areas.

Comparing droplet evaporation for different time points and substrate wettability conditions, by scaling the results to achieve similar wetted areas, the importance of CLL and contact angle was highlighted. A 2.7 times increase in cooling was calculated by replacing a singular large partial non-wetting droplet with several (4.66) smaller partial non-wetting droplets of the same total wetted area. The multiple smaller droplets resulted in an increase in CLL by 2.15 times and a decrease in droplet height by 44%, for similar contact angle, liquid-gas and solid-liquid surface areas. A 4.87 times increase in heat transfer was noted by comparing a singular large partial wetting droplet with several (16.61) smaller partial non-wetting droplets of the same total wetted area. The multiple smaller droplets resulted in an increase in CLL and liquid-gas surface area by 4.08 times and 4.48 times respectively. The droplet height and wetted area were comparable across this range.

These results for heat transfer and CLL give greater understanding to the heat transfer mechanisms for evaporative and pool electrospray cooling. They correlate well and further develop the observations of Horacek et al. [73] and Sodtke and Stephan [72]. From the presented results, considering real world conditions, the optimum cooling case is achieved for the smallest droplets (largest CLL) operating in the evaporative regime just prior to CTP on a substrate with high wettability for the applied coolant.

Chapter 7

Droplet Evaporation in an Electric Field

Sobac and Brutin [136] noted that augmentation of contact angle and wetted contact area impact the average heat transfer to an evaporating droplet. This was achieved by varying the substrate wettability properties through different surface coatings. These coatings can be expensive and are not always permanent [165]. Application of an applied electric field has been shown to augment the morphology and wetted area of a sessile droplet [180]. The research in this chapter aims to investigate the impact of an applied electric field on an evaporating droplet on a heated substrate. This work will build on the work presented in Chapter 6. The applied electric field in this study can be equated to the point source-ground

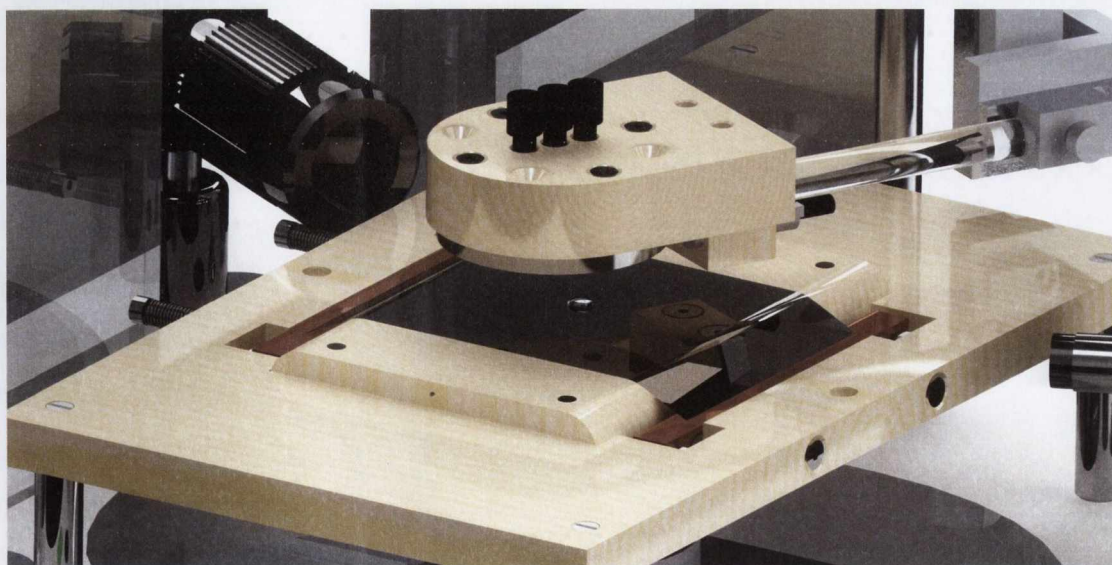


Figure 7.1: Electric field droplet evaporation test facility.

plate electric field from Chapter 4, or that of the driving electric field from a three electrode electrospray configuration observed in Chapter 5. As reviewed in Section 2.2.6 impacted droplets lose their charge to the grounded target surface during quasi-steady evaporation [70]. In the evaporative regime, sprayed impinging droplets on the grounded target surface are subjected to an applied electric field during evaporation in this condition. Whereas, in pool cooling the quasi-steady macro droplet on the heated substrate surface experiences the electric field. Partial wetting and partial non-wetting 80 μL droplets were evaporated in an electric field of varied strength and the results were compared with their no electric field counterparts discussed in Chapter 6.

7.1 Partial Wetting Droplet Evaporation in an Electric Field

Figure 7.2 qualitatively illustrates the impact of an applied electric field on the evaporating partially wetting droplet at $t = 2$ min. The applied electric field acts to augment the liquid-gas interface into the shape of a cone. If a sufficiently strong field was applied to a suitable working fluid the droplet would form into a Taylor cone and the cone-jet regime of electrospraying could be temporarily established. However for our case with water in ambient surroundings, coronal discharge would occur prior to the onset of electrospraying due to the working fluid properties. The highest local heat flux for all applied electric field cases was noted for the region around the droplet triple line. Droplet evaporation for all cases proceeded in a similar manner to that of the no electric field case (see Figure 6.4). The droplets initially evaporated in a CCR regime ($t = 2 - 8$ min), the triple line then unpins in a stochastic manner evaporating in an oblate stick-slip regime ($t = 10 - 17$ min) before finally regaining its axisymmetric thermal profile and evaporating completely in the stick-slip regime ($t = 18 - 21$ min). Droplet evaporation for $t < 2$ min is not considered here as the system behaves in a transient manner before this point as a result of the step input of heat into the foil and subsequent rapid transients in temperature. Of particular interest is the evaporation behaviour of the CCR regime. As such only the initial pinned evaporation ($t = 2 - 8$ min) will be considered for further analysis. Moving triple line evaporation behaviour in an applied

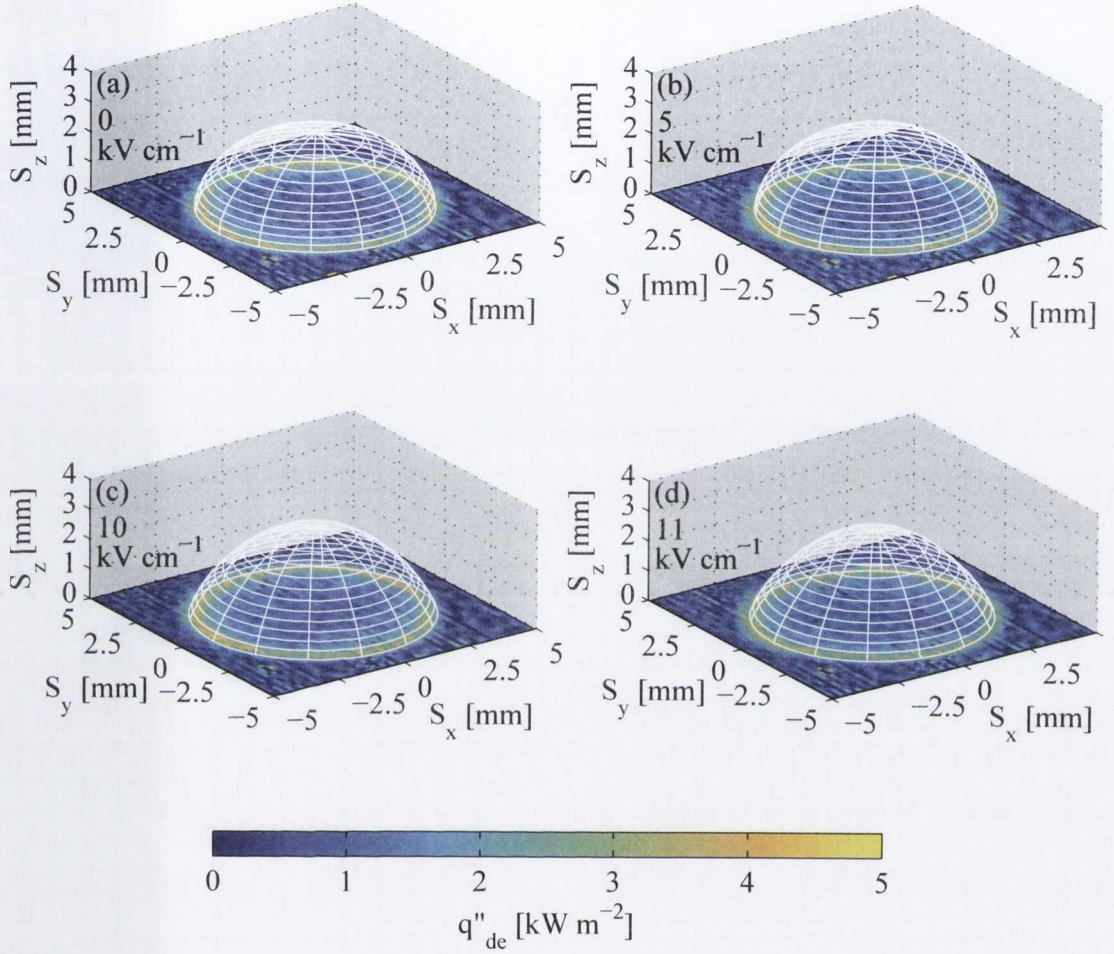


Figure 7.2: Partial wetting droplet evaporation in a varied electric field. (a) 0 kV cm^{-1} , (b) 5 kV cm^{-1} , (c) 10 kV cm^{-1} , and (d) 11 kV cm^{-1} . Total evaporation time $\approx 21 \text{ min}$, $t = 2 \text{ min}$, $V_{d,0} = 80 \text{ }\mu\text{L}$, $\theta_0 = 79^\circ$, $E_{\text{field}} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{\text{gen}} = 912 \text{ W m}^{-2}$.

electric field will be detailed for the coated substrate in Section 7.2.

Figure 7.3 compares the droplet properties of the applied electric fields over the CCR evaporation. A linear decrease in droplet volume is observed for all cases (Figure 7.3a). Apparent droplet contact angle is smaller for increased values of applied electric field with a decreasing change noted as the droplet evaporation process proceeds (Figure 7.3b). A peak change of 15% decrease is observed in contact angle at $t = 2 \text{ min}$, by $t = 8 \text{ min}$ the reduction in contact angle is $\approx 10\%$. This is due to the decreasing height of the evaporating droplet through time (Figure 7.3d). The droplet is in contact with a pseudo ground (heated substrate), thus electrically grounding the droplet. As the droplet evaporates its height decreases and

7.1. PARTIAL WETTING DROPLET EVAPORATION IN AN ELECTRIC FIELD

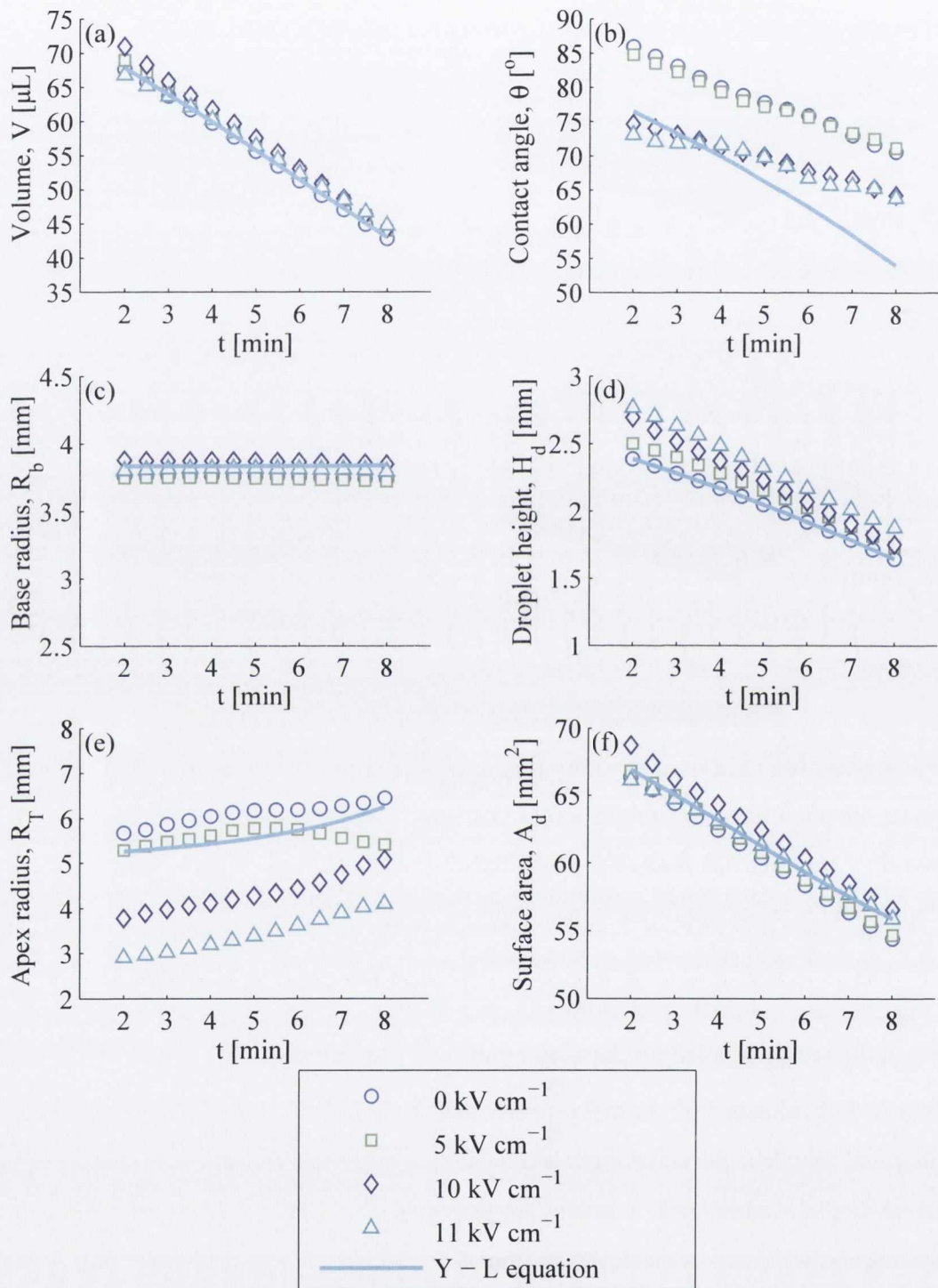


Figure 7.3: Partial wetting droplet geometric properties in a varied electric field through time. (a) Volume, (b) contact angle, (c) base radius, (d) height, (e) apex principal radii, and (f) liquid-gas surface area. $t = 2 - 8$ min, $V_{d,0} = 80 \text{ } \mu\text{L}$, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

this increases the distance between the charged electrode and the grounded droplet resulting in a weaker local electric field, which in turn diminishes the impact of the fixed applied electric field on the evaporating droplet. The droplet base radius was similar in magnitude and constant over the evaporation period for all cases (Figure 7.3c). This results highlights the CCR regime of evaporation for all cases over this evaporation period. Droplet height decreases monotonically over the CCR evaporation (Figure 7.3d), with higher heights noted for increasing electric fields. The droplet apex radii was observed to generally grow as the droplet evaporates and lower apex radii were noted for increasing applied electric fields (Figure 7.3e). The highest liquid-gas surface area A_d was noted for the second strongest applied electric field of 10 kV cm^{-1} . Surface area decreased for all cases as the droplets

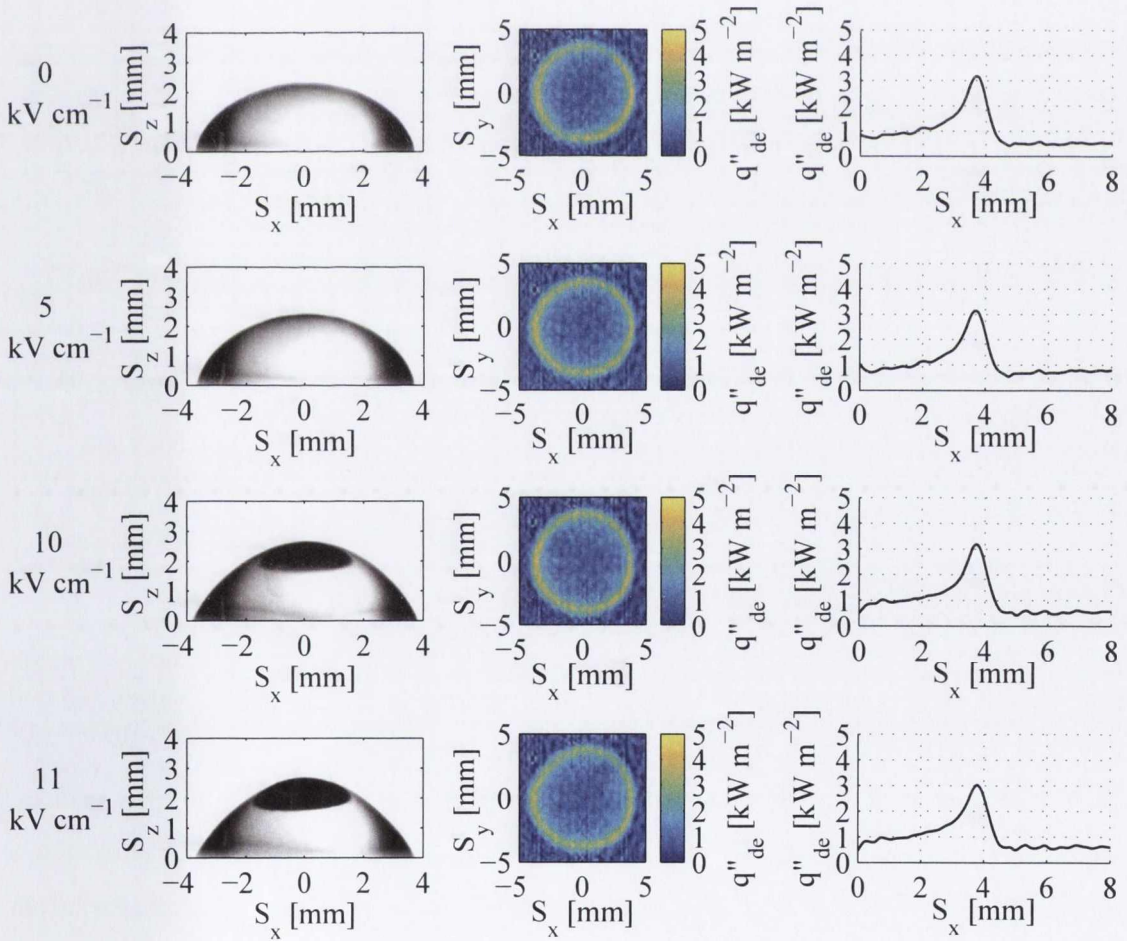


Figure 7.4: Partial wetting droplet evaporation, shape, heat flux distribution and radial heat flux profile for varied electric field. $t = 2 \text{ min}$, $V_{d,0} = 80 \text{ }\mu\text{L}$, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

evaporated (Figure 7.3f).

Figure 7.4 shows the droplet shape (column one), local heat flux distribution (column two) and heat flux radial profile (column three) for varied electric field strength at $t = 2$ min. Change in droplet shape is qualitatively shown in the optical images. No substantial change in the local heat flux distribution and radial profile shape or magnitude is distinguishable for increasing electric field strength. This result suggests that despite a clear change in the apparent droplet contact angle no change in the thin film transition region occurs due to the near constant heat flux distribution. Figure 7.5 highlight the subtle changes in the liquid-gas interface (Figure 7.5a) and near constant radial heat flux profile (Figure 7.5b) for varied electric field strength at $t = 2$ min.

Figure 7.6a and b shows the magnitude and location of the peak local heat flux of the

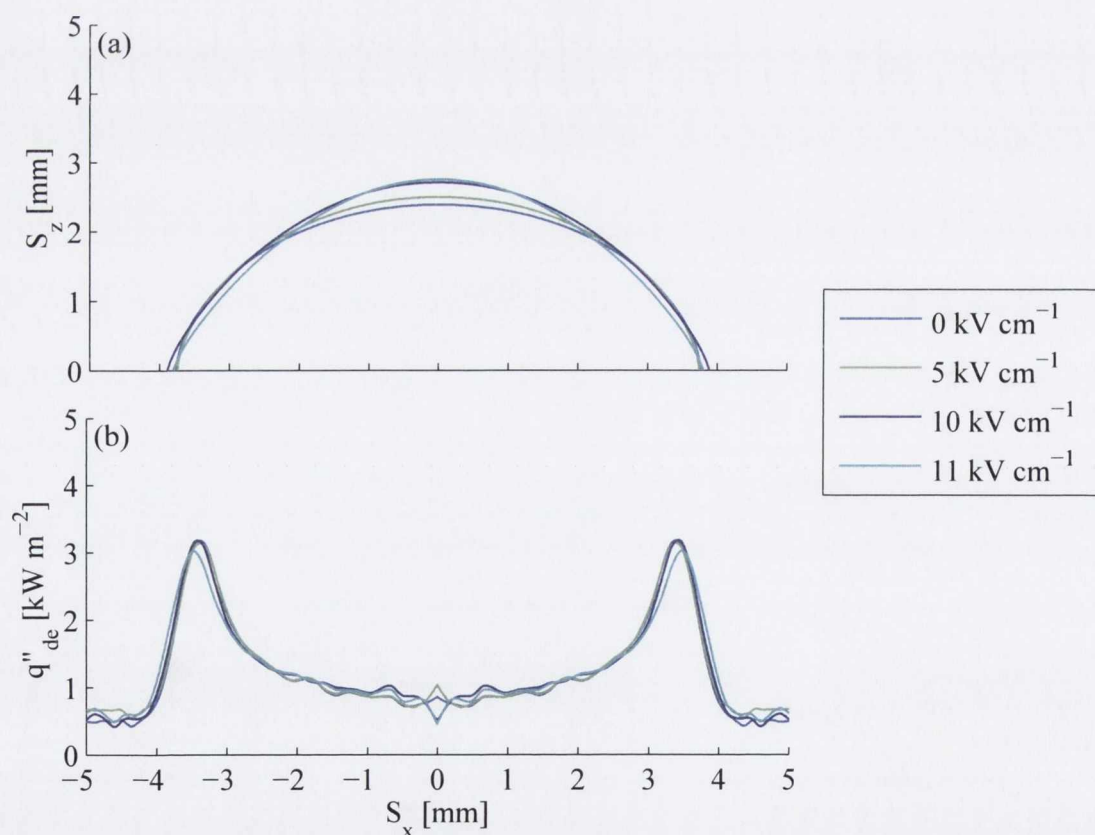


Figure 7.5: Partial wetting droplet evaporation in a varied electric field. (a) liquid-gas interface, (b) radial heat flux profile. $t = 2$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

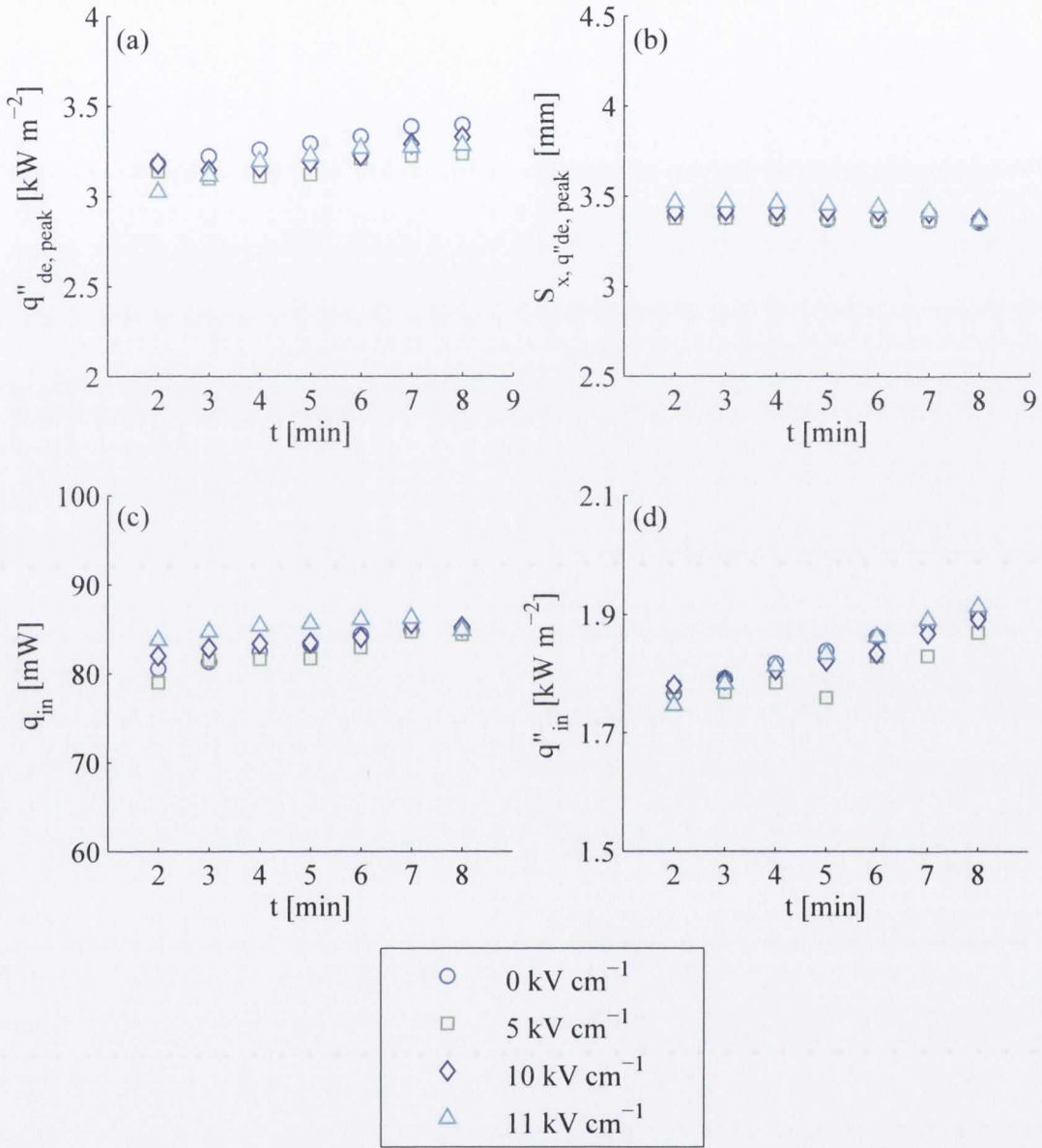


Figure 7.6: Partial wetting droplet evaporation heat transfer for varied applied electric fields. (a) Peak heat flux magnitude, (b) location, (c) total power into droplet, and (d) average heat flux. $t = 2 - 8$ min, $V_{d,0} = 80 \text{ }\mu\text{L}$, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

evaporating droplet for varied electric field strength. A marginally increasing $q''_{de, peak}$ and near constant position of this $q''_{de, peak}$ is observed for all cases. Figure 7.6c and d demonstrates the total power and the average heat flux into the droplet across the solid-liquid interface. A generally increasing value for q_{in} and q''_{in} is noted for all cases. As the droplets evaporate in the CCR regime the interface area between the solid substrate and liquid droplet remains

constant (fixed CLL), while the thickness of the droplet decreases. This reduced thermal resistance across the droplet resulting in an increase in temperature of the liquid at the liquid-gas interface. This causes an increase in the heat flux across this boundary resulting in an increase in $q''_{de,peak}$, q_{in} and q''_{in} . This is despite the decreasing surface area of the liquid-gas interface across this CCR evaporative regime.

Figure 7.7 highlights the bulk forces acting on the evaporating droplet in the vertical direction. Analysis of the force balance acting on an evaporating droplet enables one to quantify the impact of an applied electric, and can thus be linked to any change in heat and mass transfer. Figure 7.7a plots the bulk forces for the Young-Laplace solution. The net force of the Young-Laplace solution is 0 mN for all time points. The contact pressure force, F_{cp} , decrease as the droplet evaporates due to the increasing radius of the apex radii and decreasing droplet height. The weight force, F_w , and capillary force, F_c , acting downward, decrease in magnitude over the course of the evaporation due to the decreasing droplet volume and contact angle respectively. A net force, F_{net} , and electrical force, F_e , equal to zero is expected as all forces are accounted for in the case of a slowly evaporating droplet in no applied electric field. Figure 7.7b shows the tabulated bulk forces calculated using values obtained from the filtered 0 kV cm^{-1} boundary. The resultant net force is $\sum F_y \approx 0$ with any discrepancies due to difficulties measuring the liquid-gas interface in the region around the triple line. Figure 7.7c, for the 5 kV cm^{-1} experiment, shows no easily observable impact of the applied electric field. This is most likely due to any augmentation of the electric field being on the same scale as the residual error caused by the triple line measurement inaccuracies. Figure 7.7d and e correspond to the 10 and 11 kV cm^{-1} electric field cases. A clear impact of the applied electric field is noted in both cases with a negative electrical force that decreases as the droplet evaporates. Electrical force magnitude increases with applied electric field strength. As the electric field acts to shape the droplet into the morphology of a cone, it decreases the droplet apex radius and contact angle while increasing the droplet height, with greater impact observed for higher applied electric fields (Figure 7.3c and e). This increases the magnitude of the contact pressure force and marginally decreases the capillary force resulting in a negative electrical force to ensure equilibrium. The electric field force decreases as the droplet evaporates due to the decreasing droplet height over this period resulting in a greater distance between the charged electrode and pseudo ground.

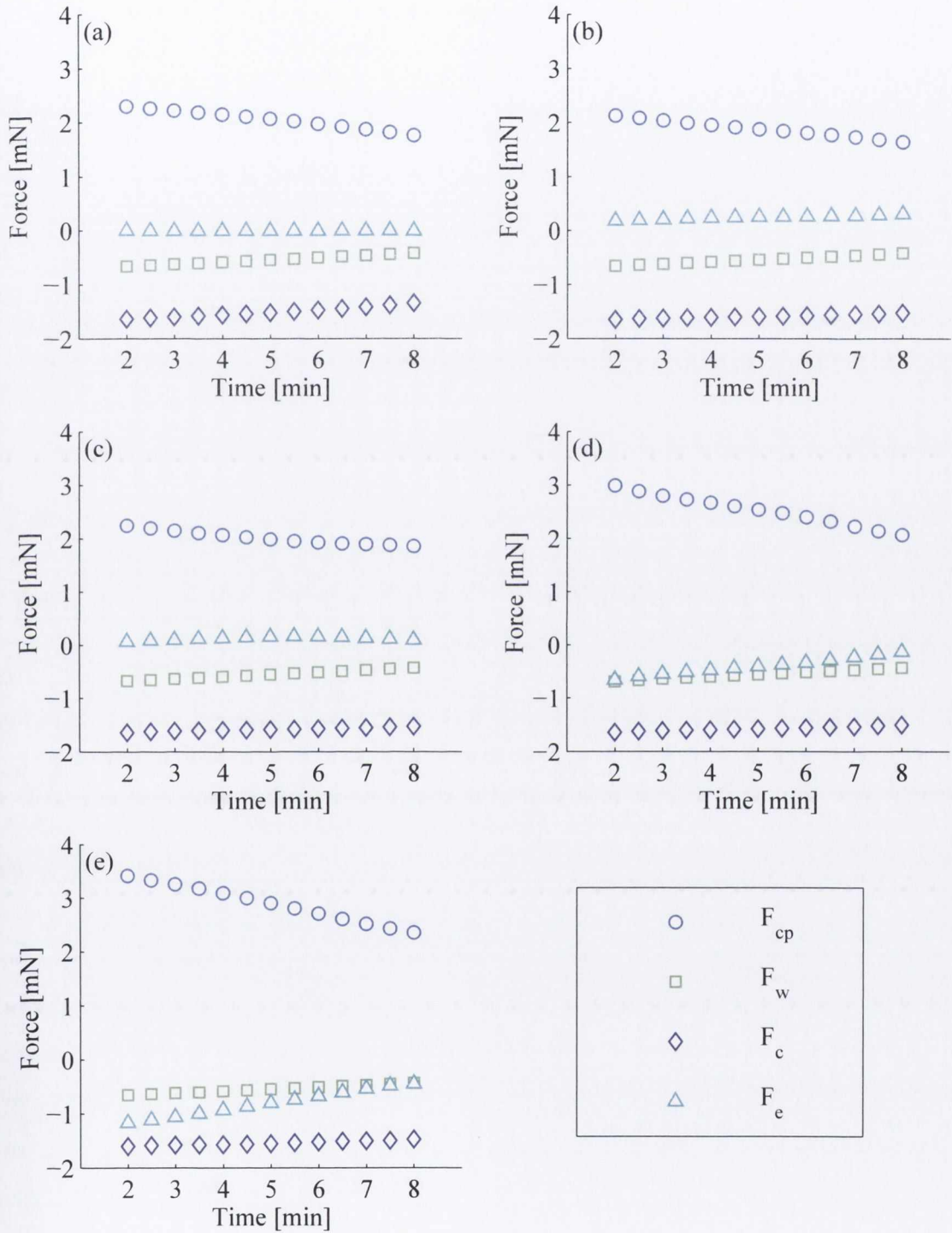


Figure 7.7: Partial wetting droplet bulk force analysis. (a) Young-Laplace, (b) 0 kV cm^{-1} , (c) 5 kV cm^{-1} , (d) 10 kV cm^{-1} , and (e) 11 kV cm^{-1} . $t = 2 - 8 \text{ min}$, $V_{d,0} = 80 \text{ }\mu\text{L}$, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

7.1. PARTIAL WETTING DROPLET EVAPORATION IN AN ELECTRIC FIELD

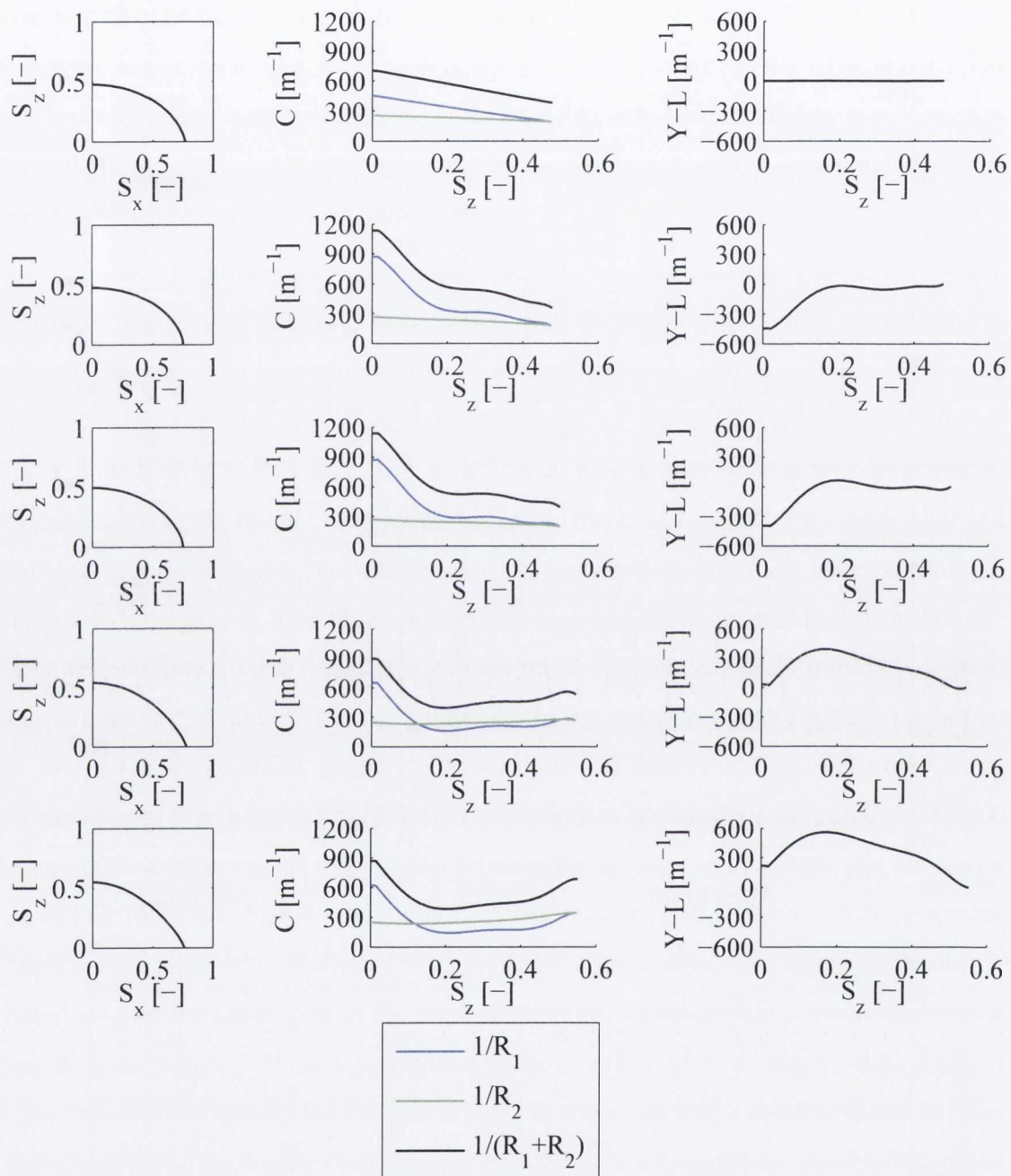


Figure 7.8: Partial wetting droplet local curvature analysis. $t = 2$ min, $V_{d,0} = 80$ μ L, $\theta_0 = 79^\circ$, $E_{field} = 0 - 11$ kV cm⁻¹ and $q''_{gen} = 912$ W m⁻².

Figure 7.8 highlights the impact the electric field has on the local droplet curvature across the liquid-gas interface. The first column of Figure 7.8 is the liquid-gas interface made dimensionless by a fixed length scale. Column two plots the reciprocal of the first and second principal radii and the total curvature at a given point. The third column shows the local

solution to the Young-Laplace equation. The first row corresponds to the boundary predicted from the Young-Laplace solution derived from inputs given by the filtered droplet boundary. Droplet curvature linearly decreases as S_z increases. The solution to the Young-Laplace equation is equal to zero for all points. Row two relates to the 0 kV cm^{-1} filtered boundary case. The filtered boundary demonstrates good correlation with the Young-Laplace solution for $S_z > 0.15$. For values of $S_z < 0.15$ the inaccuracies caused by the difficulties measuring the droplet boundary near the contact line are apparent. Rows 3, 4 and 5 correspond to the 5, 10 and 11 kV cm^{-1} tests respectively. Analysing the data for only $S_z > 0.15$, the greatest curvature modification is observed at lower values of S_z for all electric field cases. With the highest curvature augmentation noted for increasing applied electric fields.

7.2 Partial Non-wetting Droplet Evaporation in an Electric Field

The next droplet evaporation case investigated was that of the coated stainless steel surface in an electric field of varied strength. Figure 7.9 qualitatively demonstrates the influence of an applied electric field on the evaporating partially non-wetting droplet at $t = 2 \text{ min}$. The applied electric field again acts to augment the droplet shape into a cone morphology. Droplet evaporation for all cases proceeded in a similar manner to that of the no electric field partial non-wetting case (Figure 6.10). The droplets initially evaporated in a stick-slip manner ($t = 2 - 26 \text{ min}$), after which the rate at which the triple line sticks and slips begins to slow with the triple line unpinning in a stochastic manner to final complete droplet evaporation ($t = 27 - 31 \text{ min}$). Droplet evaporation for $t < 2 \text{ min}$ is not considered as the system behaves in a transient manner before this point as a results of the rapid change in temperature when joule heating to the thin foil is applied at $t = 0 \text{ min}$. The stick-slip evaporation behaviour in an applied electric field is of particular interest and is in contrast to the pinned electric field evaporation observed in the partial wetting case. Only data on the stick - slip mode of evaporation ($t = 2 - 26 \text{ min}$) will be discussed here.

Figure 7.10 compares the droplet geometric properties of the applied electric fields over the stick-slip evaporation period. A monotonic decrease in droplet volume is observed for

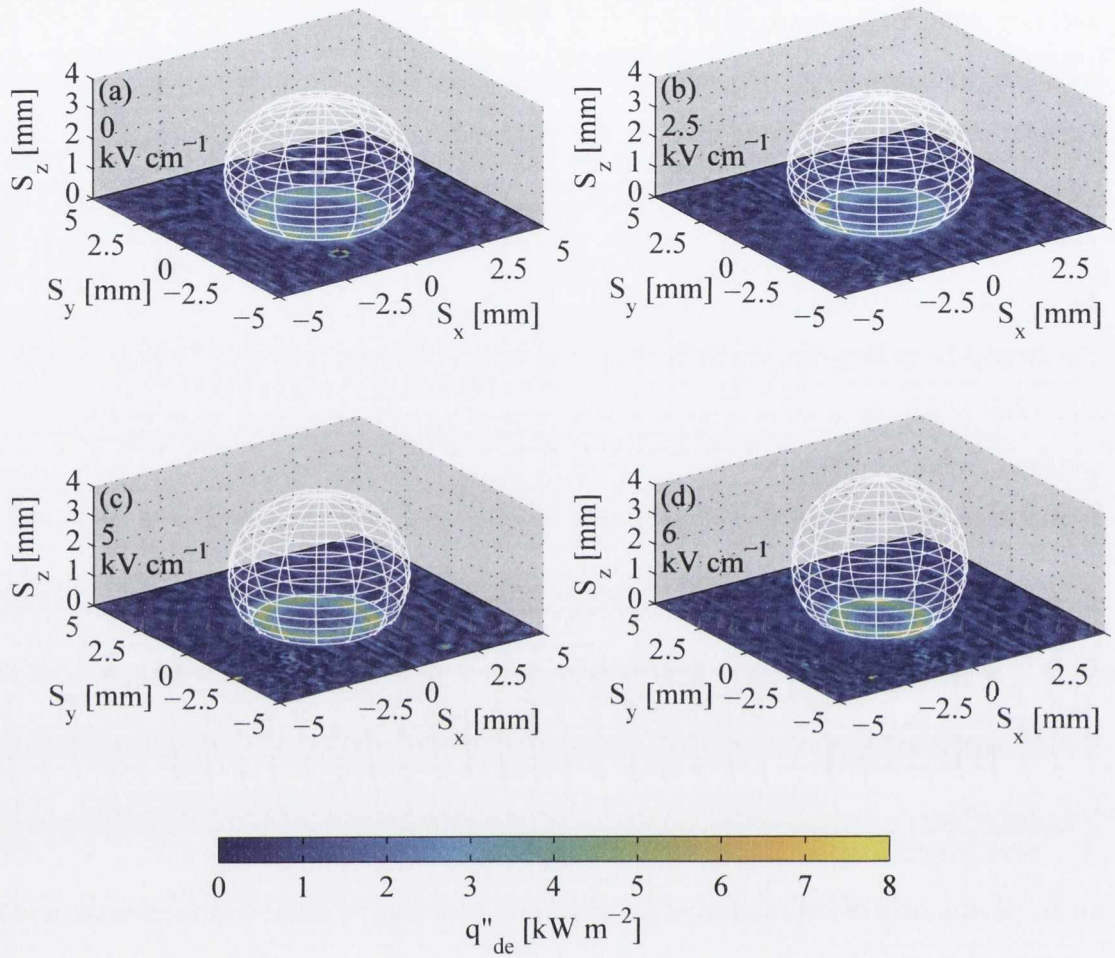


Figure 7.9: Partial non-wetting droplet evaporation in a varied electric field. (a) 0 kV cm⁻¹, (b) 2.5 kV cm⁻¹, (c) 5 kV cm⁻¹, and (d) 6 kV cm⁻¹. Total evaporation time ≈ 31 min, $t = 2$ min, $V_{d,0} = 80$ μ L, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6$ kV cm⁻¹ and $q''_{gen} = 912$ W m⁻².

all cases (Figure 7.10a). Apparent droplet contact angle is generally smaller for increased values of applied electric field with a decreasing change noted as the droplet evaporation process proceeds (Figure 7.10b). The droplet base radius decreased as the evaporation process proceeded with lower values observed for higher applied electric fields (Figure 7.10c). Droplet height decreased over the evaporation period for all case with greater droplet heights noted for increasing value of the applied electric field (Figure 7.10d). The results and trends of Figures 7.10b, c and d characterise the stick-slip regime of evaporation. Figure 7.10e shows that the droplet apex radius decreases as the droplet evaporated for all cases, lower values are noted for higher magnitudes of the applied electric field. The liquid-gas interface surface

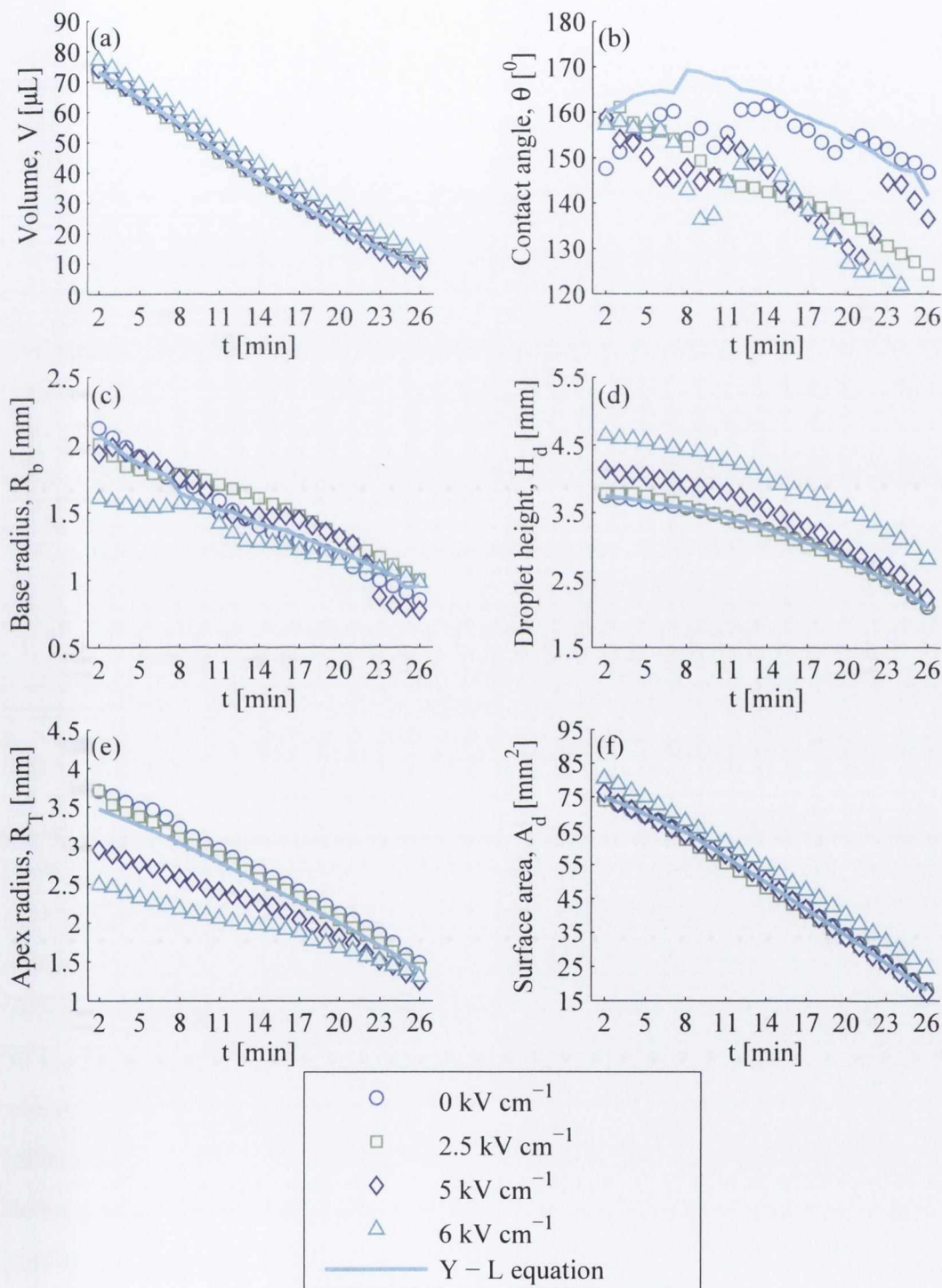


Figure 7.10: Partial non-wetting droplet geometric properties in a varied electric field through time. (a) Volume, (b) contact angle, (c) base radius, (d) height, (e) apex principal radii, and (f) liquid-gas surface area. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

7.2. PARTIAL NON-WETTING DROPLET EVAPORATION IN AN ELECTRIC FIELD

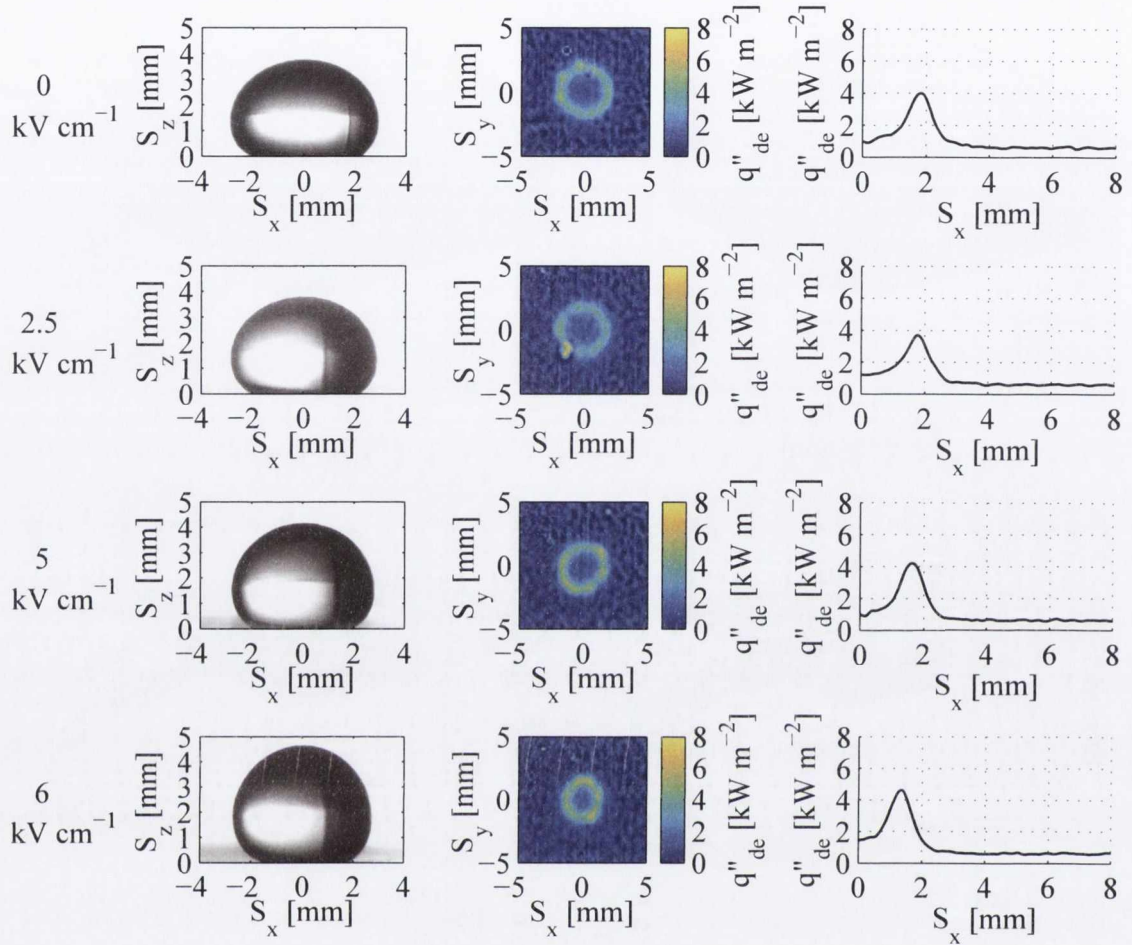


Figure 7.11: Partial non-wetting droplet evaporation, shape, heat flux distribution and radial heat flux profile for varied electric field. $t = 2$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

area decrease monotonically for all cases as the droplet evaporates. An average increased surface area of 11% is observed for the highest applied electric field case of 6 kV cm^{-1} over the no electric field test over the course of the evaporation.

Figure 7.11 shows the partially non-wetting droplet shape (column one), local heat flux distribution (column two) and heat flux radial profile (column three) for varied electric field at $t = 2$ min. Change in droplet shape is qualitatively shown in the optical images, with the applied electric field acting to elongate the droplet vertically. All droplet footprints appear axisymmetric from the heat flux distribution data. A decreasing droplet radius and increasing peak heat flux is noted for an increasing applied electric field. This point is demonstrated further in Figure 7.12. Figure 7.12 shows the droplet liquid-gas interface (Figure 7.12a)

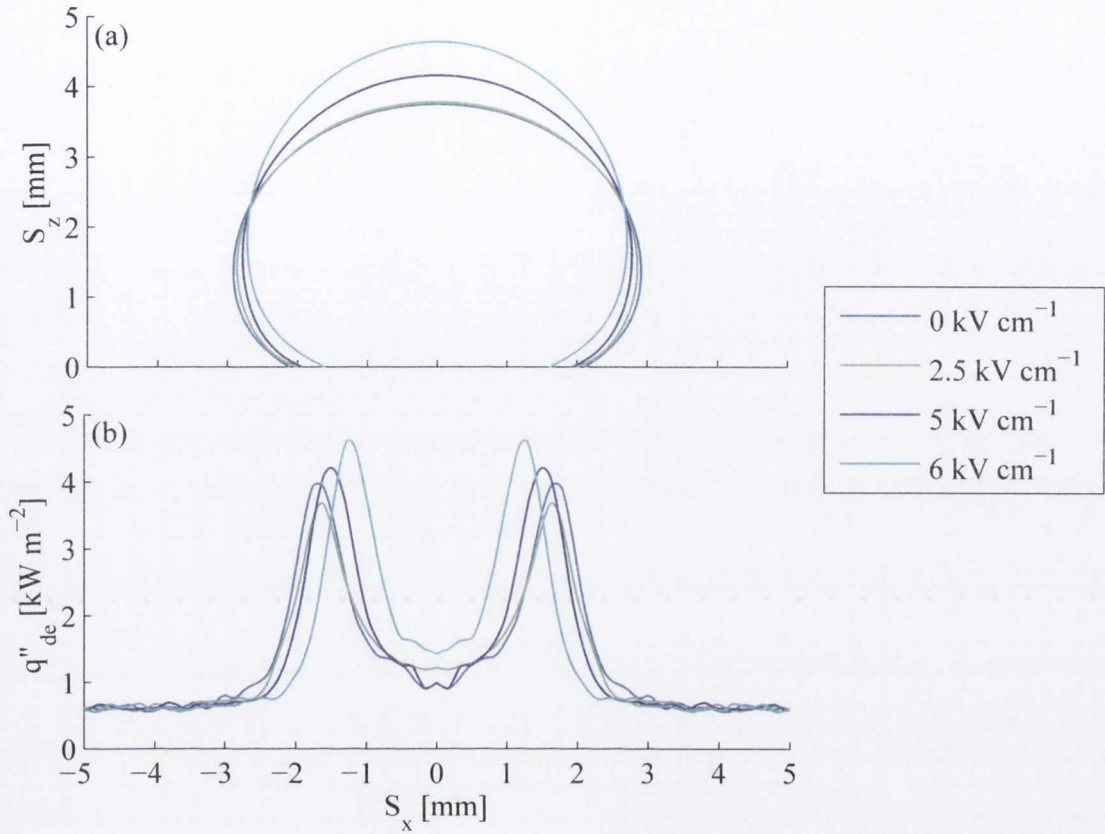


Figure 7.12: Partial non-wetting droplet evaporation in a varied electric field. (a) liquid-gas interface, (b) radial heat flux profile. $t = 2$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

and radial heat flux profile (Figure 7.12b) for varied electric field strengths collapsed onto individual graphs for comparison at $t = 2$ min. Due to the applied electric field the droplet is elongated in the direction of the charged electrode (Figure 7.12a). This is similar to the results of Di Bari and Robinson [182], albeit for bubbles, not liquid droplets. For their research simulations showed that the electric field was most intense at the bubble equator, resulting in electrical pressure acting inward, on the bubble causing it to deform and have the appearance of being stretched. For increasing electric field strength this apparent inward pressing reduces the contact area between the solid-liquid interface at the droplet base. This reduction in base diameter can be attributed to the fact that at higher applied electric fields, the droplet contact angle reduces to less than that of the receding contact angle of the droplet - substrate condition, resulting in a decreasing triple line radius and increasing CLL. This results in higher peak heat flux acting on a smaller area for the 5 and 6 kV cm^{-1} cases (Figure 7.12b).

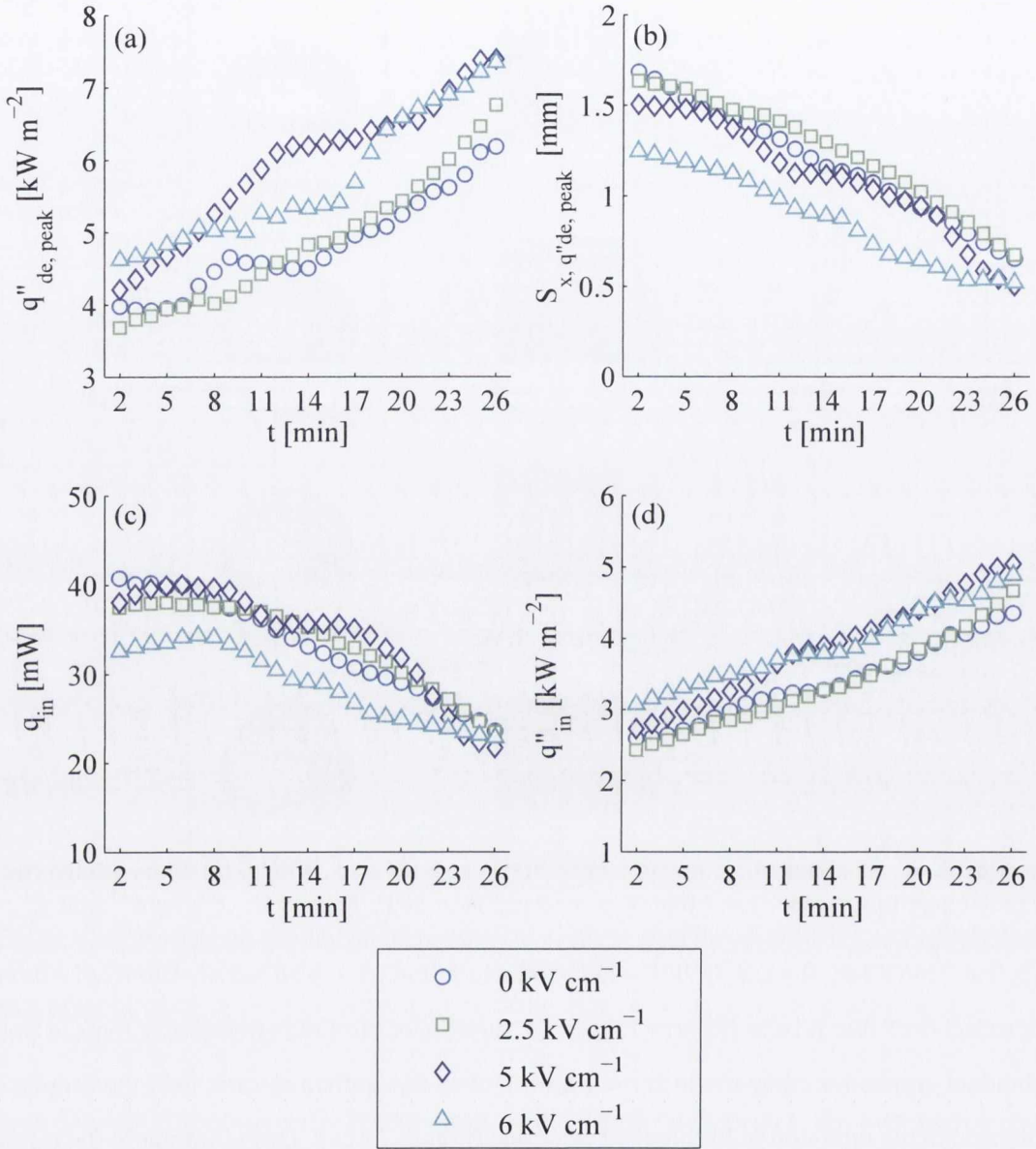


Figure 7.13: Partial non-wetting droplet evaporation heat transfer for varied applied electric fields. (a) Peak heat flux magnitude, (b) location, (c) total power into droplet, and (d) average heat flux. $t = 2 - 26$ min, $V_{d,0} = 80$ μ L, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6$ kV cm $^{-1}$ and $q''_{gen} = 912$ W m $^{-2}$.

This electric field effect is further shown in Figure 7.13.

Figure 7.13a and b plot the magnitude and location of the peak local heat flux of the evaporating droplet. A general increasing $q''_{de,peak}$ and decreasing $S_{x, q''_{de,peak}}$ is observed for all electric field cases. A increased value for $q''_{de,peak}$ is noted for the higher electric field cases with a lower $S_{x, q''_{de,peak}}$ over the course of the evaporation process. Consistent with the

earlier findings for no electric field partial non-wetting droplet evaporation, decreasing the base radius tends to enhance the convective heat flux at the triple line due to the increased CLL.

Figure 7.13c and d demonstrates the total power and the average heat flux into the droplet across the solid liquid interface for all electric field tests. A decreasing value for q_{in} and an increasing value for q''_{in} are observed for all tests. As the droplet evaporates in the stick-slip regime the interface area between the solid substrate and liquid droplet decreases. As this occurs the thickness of the droplet also decreases. The lower total thermal energy at higher electric field is homogeneous to the comparison between the partial wetting and partial non-wetting comparison discussed in Section 6.3. Where a greater solid-liquid surface area and contact line length results in a larger total thermal energy dissipated. This is due to a greater total area of transition region at the triple line and a reduced thermal resistance across the droplet, which in turn enables a lower thermal resistance across the liquid-gas boundary as the boundary is at a higher temperature creating a greater ΔT to ambient surroundings. A higher q''_{in} is noted for increasing values of applied electric field due to the increasing CLL as a result of the augmented triple line.

Figure 7.14 shows the vertical forces acting on the partially non-wetting evaporating droplet in a varied strength electric field. Figure 7.14a plots the forces for the Young-Laplace solution in the vertical direction. The net force of the Young-Laplace solution is 0 mN for all time points. The upward directed contact pressure force, F_{cp} , decrease as the droplet evaporates due to the decreasing base radius and droplet height. The weight force, F_w , and capillary force, F_c , acting in the downward direction, decrease in magnitude over the course of the evaporation due to the decreasing droplet volume and base radius respectively. The electrical force, F_e , is sensibly zero for all time points. Figure 7.14b shows the vertical forces calculated from the measured 0 kV cm^{-1} boundary. The resultant net force is $\sum F_y \approx 0$ with any discrepancies due to inaccuracies associated with measuring the liquid-gas interface in the region around the triple line. Figure 7.14c and d shows the 2.5 and 5 kV cm^{-1} experimental cases. No clear impact of the applied electric field can be seen. This is most likely due to any modification due to the electric field being on the same scale as the residual error caused by the triple line measurement inaccuracies. Figure 7.14e correspond to the 6 kV cm^{-1} electric field cases. A positive resultant electrical force is calculated that decreases

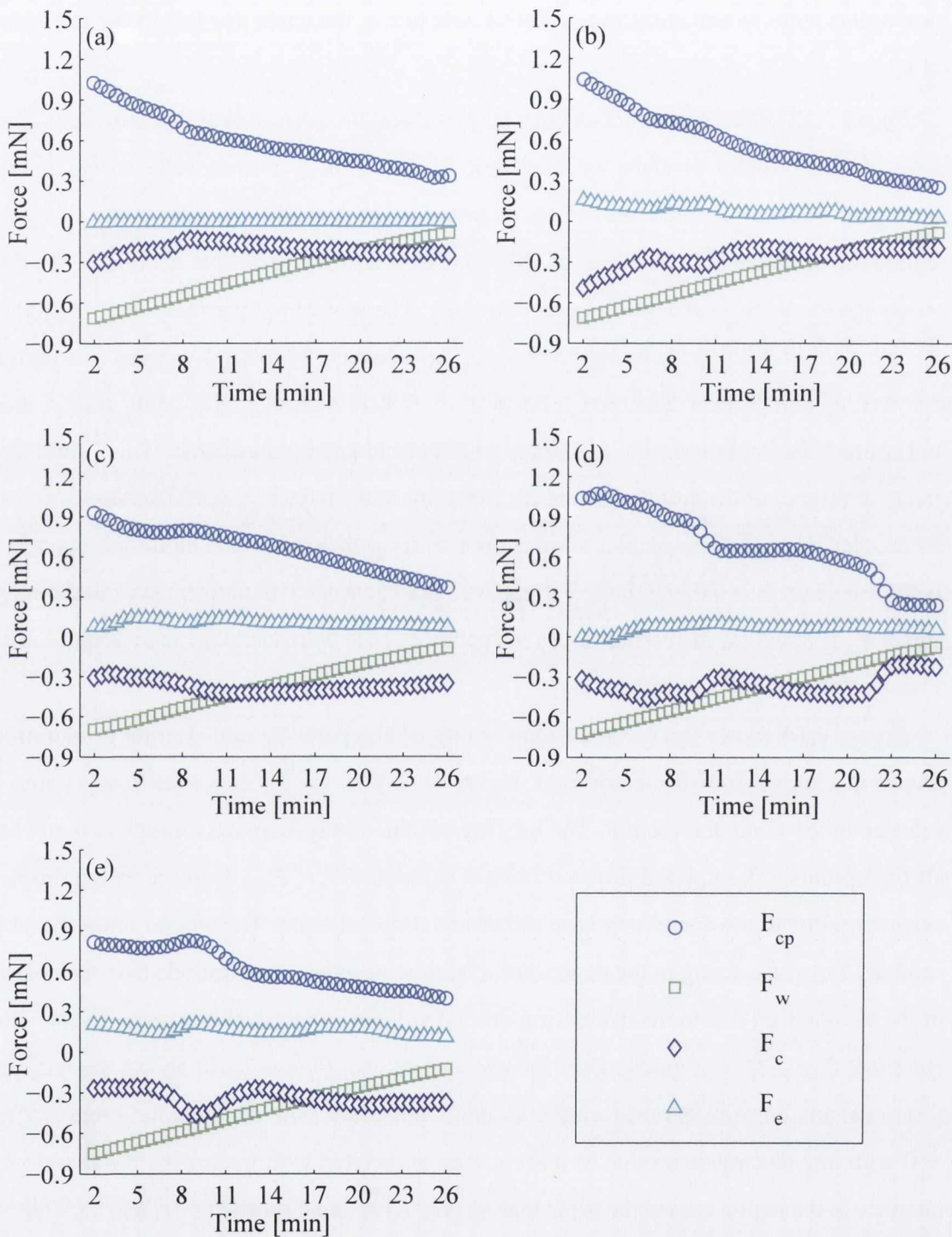


Figure 7.14: Partial non-wetting droplet bulk force analysis. (a) Young-Laplace, (b) 0 kV cm⁻¹, (c) 2.5 kV cm⁻¹, (d) 5 kV cm⁻¹, and (e) 6 kV cm⁻¹. $t = 2 - 26$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6$ kV cm⁻¹ and $q''_{gen} = 912 \text{ W m}^{-2}$.

in magnitude as the droplet evaporates. As the electric field acts to shape the droplet into the morphology of a cone, the decreasing base radius impacts the contact pressure force and to a lesser extent the capillary force. Highest reduction in the base radius is observed for the highest electric field case (Figure 7.10c and Figure 7.13b). Since $F_{cp} \propto R_b^2$ large change in

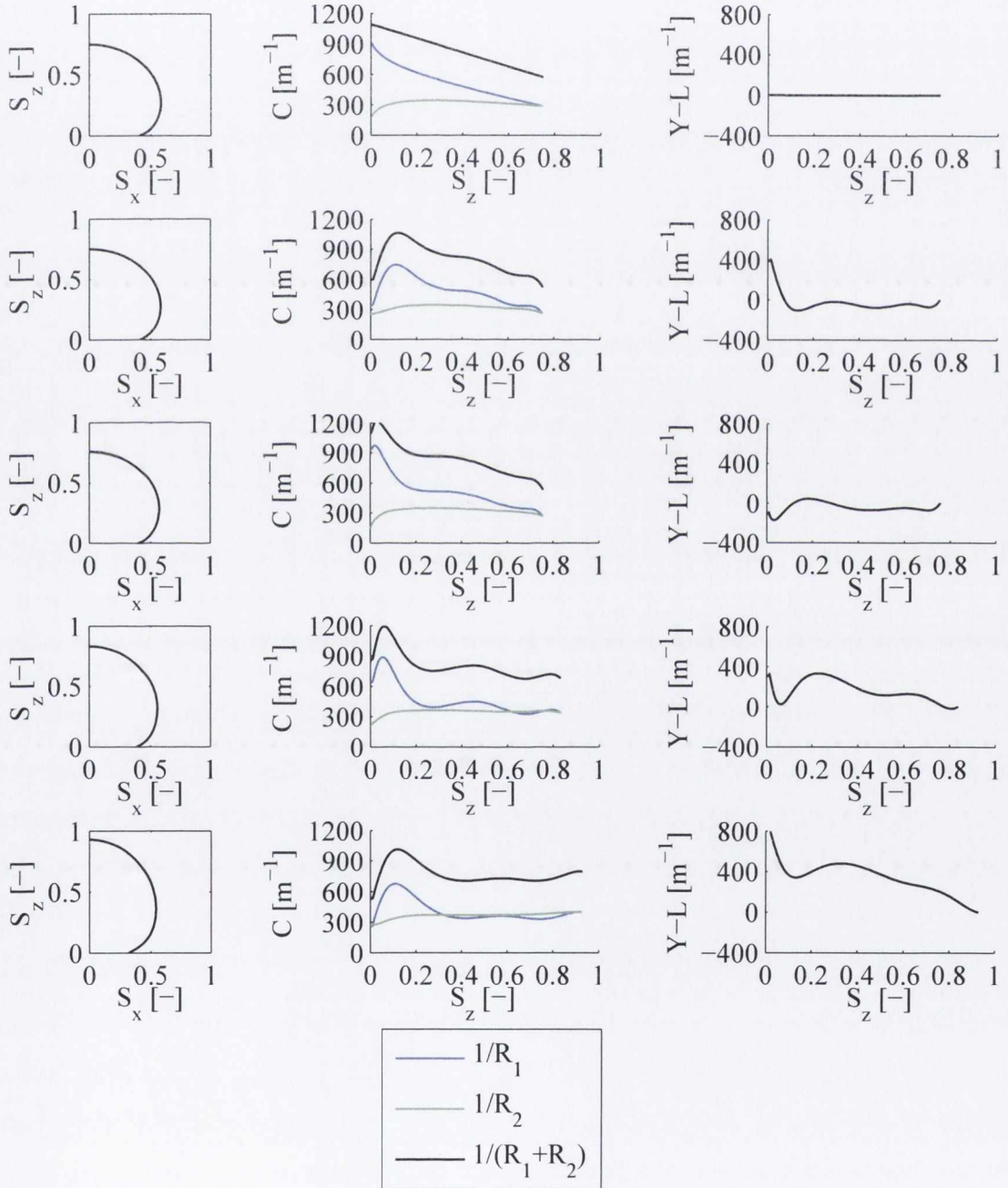


Figure 7.15: Partial non-wetting droplet local curvature analysis. $t = 2$ min, $V_{d,0} = 80$ μ L, $\theta_0 = 160^\circ$, $E_{field} = 0 - 6$ kV cm⁻¹ and $q''_{gen} = 912$ W m⁻².

the contact pressure force is noted which is balanced by a net upward electrical force.

Figure 7.15 highlights the impact the electric field has on the local droplet curvature of the partially non-wetting droplet at $t = 2$ min. The first column of Figure 7.15 is the liquid-gas interface made dimensionless by a fixed length scale. Column two plots the reciprocal of the first and second principal radii and the total curvature at a given point. The third column shows the local solution to the Young-Laplace equation. The first row corresponds to the boundary predicted from the Young-Laplace solution derived from inputs given by the filtered droplet boundary. The total curvature is noted to decrease linearly as S_z increases. The solution to the Young-Laplace equation is equal to zero for all points. Row two relates to the 0 kV cm^{-1} filtered boundary case. The filtered boundary demonstrates good correlation with the Young-Laplace solution for $S_z > 0.15$. For values of $S_z < 0.15$ the inaccuracies caused by the difficulties measuring the droplet boundary near the contact line are apparent. Rows 3, 4 and 5 correspond to the 2.5, 5 and 6 kV cm^{-1} tests respectively. Analysing the data for only $S_z > 0.15$, the greatest curvature modification is observed at lower values of S_z for all electric field cases. With the highest curvature modification noted for increasing electric fields.

7.3 Electric field Evaporation Comparison

Figure 7.16 contrasts the partial wetting and partial non-wetting evaporation modes in and out of an applied electric field in terms of interface profile, radial heat flux distribution and vertical electric force. Figures 7.16a, c, and e shows that the heat flux distribution is invariant to the applied electric field despite clear morphology augmentation resulting from the downward electrical force. Figures 7.16b, d, and f show that if applied electrical force acts to reduce the droplet base a subsequent increase in the heat flux distribution is observed but over a smaller area. These relationships are explored further in Figure 7.17. Figure 7.17 shows droplet volume (Figure 7.17a), thermal power (Figure 7.17b), base radius (Figure 7.17c) and the average heat flux (Figure 7.17d) over similar volumetric evaporation periods for both the partial wetting and partial non-wetting in extremum electric field cases. These results show that if the objective is to achieve droplet evaporation in the shortest time period then droplet evaporation in no electric field on the partial wetting substrate in the

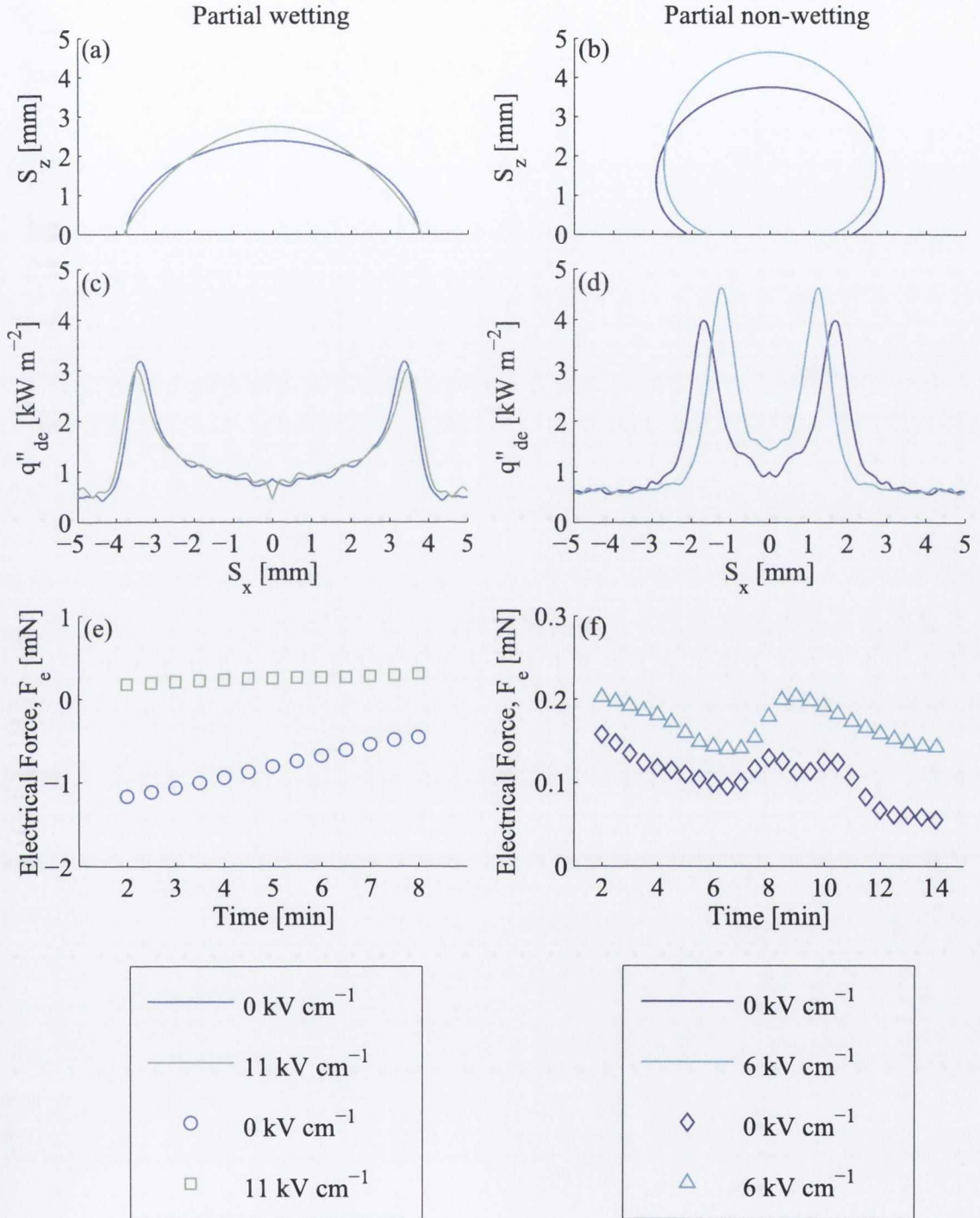


Figure 7.16: Partial wetting and partial non-wetting droplet comparison in and out of an applied electric field. (a) - (b) Interface (c) - (d) local convective heat flux distribution, and (e) - (f) electrical force. $t = 2 - 14 \text{ min}$, $V_{d,0} = 80 \text{ }\mu\text{L}$, $\theta_0 = 79 - 160^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

pinned regime is optimum. As it supports the highest thermal power into the droplet due to its larger triple line transition region area, and thus lower thermal resistance [KW $^{-1}$].

7.3. ELECTRIC FIELD EVAPORATION COMPARISON

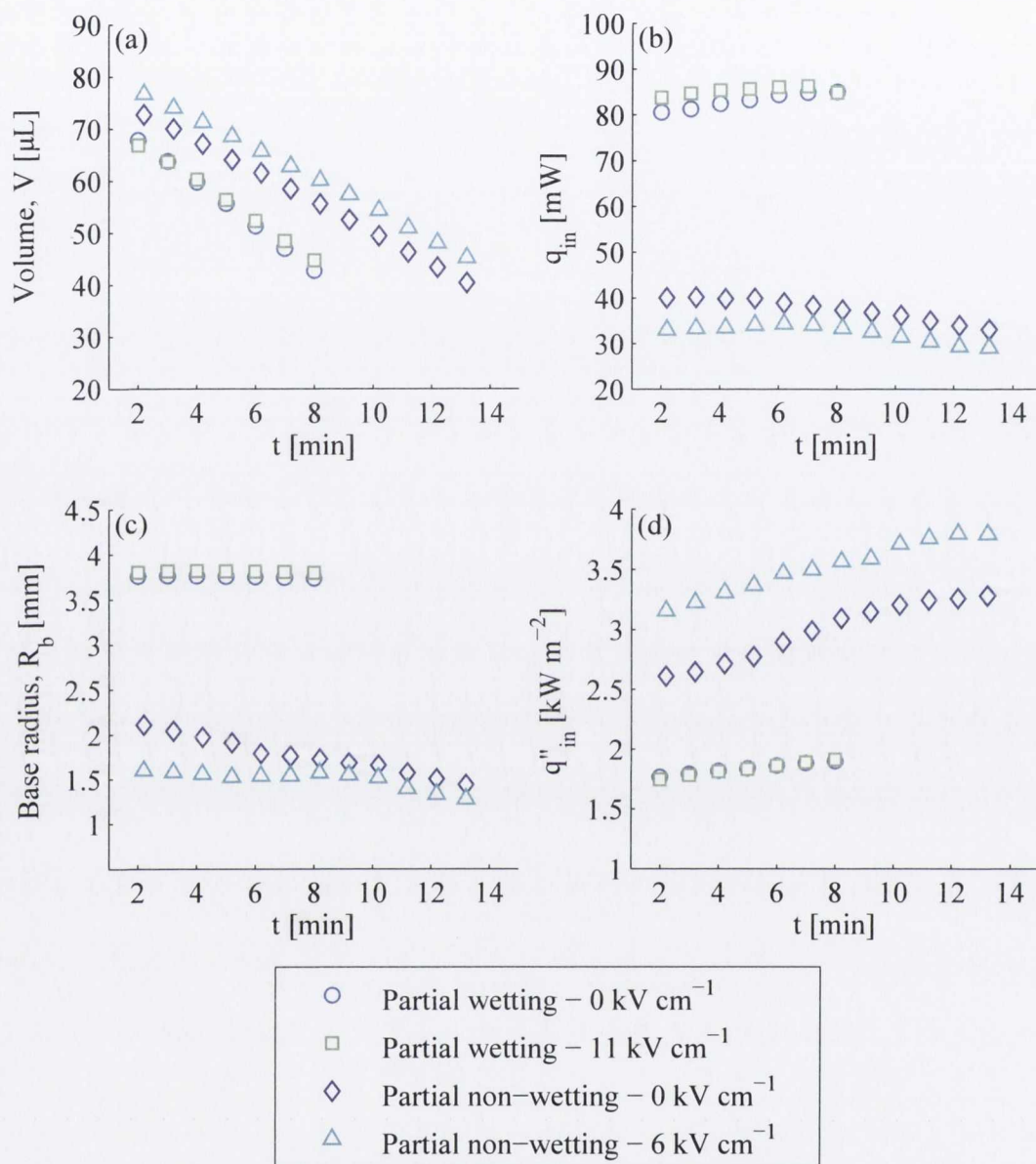


Figure 7.17: Partial wetting and partial non-wetting comparison in and out of an applied electric field. (a) Volume, (b) thermal power, (c) base radius, and (d) average convective heat flux. $t = 2 - 14$ min, $V_{d,0} = 80 \mu\text{L}$, $\theta_0 = 79 - 160^\circ$, $E_{field} = 0 - 11 \text{ kV cm}^{-1}$ and $q''_{gen} = 912 \text{ W m}^{-2}$.

However if the application is in achieving peak cooling of a thermal exchange surface then the maximum applied electric field on the partial non-wetting substrate in the stick-slip regime of evaporation is the most advantageous option, due to its larger CLL and thus lower specific thermal resistance [cm K W^{-1}].

Comparing the partial non-wetting droplet at $t = 2$ min for the 0 and 6 kV cm^{-1} cases

Table 7.1: Wetted area electric field comparison.

	0 kV cm ⁻¹	6 kV cm ⁻¹	6 kV cm ⁻¹ scaled	
Time	2	2	2	[min]
Volume	73.17	77.13	135.08	[μL]
Contact angle	147.6	157.2	157.2	[°]
Height	3.78	4.64	4.64	[mm]
Liquid-gas interface	74.91	80.62	141.19	[mm ²]
Wetted area	14.17	8.09	14.17	[mm ²]
CLL	0.94	1.25	1.25	[mm ⁻¹]
q_{in}	40.68	32.54	56.98	[mW]
q''_{in}	2.60	3.08	3.08	[kW m ⁻²]

over a similar wetted area highlights the impact of a receding contact line due to the applied electric field. A 1.75 times greater wetted area (solid-liquid interfacial area) is noted for the 0 kV cm⁻¹ case at $t = 2$ min. Linear scaling of the partial non-wetting droplet properties is assumed, such that 1.75 partial non-wetting droplets at $t = 2$ min are evaporating on a heated substrate, to compare the cases over a similar wetted areas. The relevant parameters of the two cases and the calculated scaled 6 kV cm⁻¹ results are shown in Table 7.1. The fundamental comparison is the thermal energy dissipated from the substrate. 1.4 times more thermal energy (40.68 – 56.99 mW) is dissipated by having multiple smaller droplets in an applied electric field in comparison to a singular large droplet in no electric field. This increase in q_{in} is achieved for an 84% and 88% increase in volume and liquid-gas surface area for the multiple droplet case in an applied electric field. The CLL increases by 1.32 times from 0.94 – 1.25 mm⁻¹. Similar to the conclusions of Section 6.3, these results demonstrate that the optimum heat transfer for fixed wetted area is achieved for the evaporation that gives the greatest CLL. For a receding contact line on a partially wetting surface this is achieved at the highest applied electric field.

7.4 Summary

Droplet evaporation in an electric field for different surface wettability conditions was investigated using thin foil thermography. Similar to the results of Chapter 6, both pinned and stick-slip evaporation were observed for partial wetting and partial non-wetting heated

7.4. SUMMARY

substrates respectively for all applied electric fields. Comparable trends in q_{in} and q''_{in} were observed for all applied electric field cases as their no electric field counterparts from Chapter 6. A peak electric field was noted at the start of each evaporation ($t = 2$ min), decreasing in magnitude as the evaporation process proceeded due to the increasing separation height between the electrode and the droplet as the height decreased. A resultant downward force was calculated for applied electric field for the partial wetting case. A peak value of 1.17 mN was found for the maximum applied field of 11 kV cm^{-1} at $t = 2$ min. In contrast, an upward resultant force was calculated for the applied electric field for the partial non-wetting case. A peak magnitude of 0.20 mN was calculated for the maximum applied field of 6 kV cm^{-1} at $t = 2$ min. The applied electric field resulted in clear augmentation of the droplet morphology with greater change noted with increasing applied electric fields for both wettability conditions. A reduction of 49% and 24% was noted in the apex radii for the partial wetting and partial non-wetting cases respectively at their maximum applied electric fields at $t = 2$ min in comparison to their no electric field cases. A 15% decrease in the contact angle with no change in CLL or wetted area was observed for the partial wetting substrate at $t = 2$ min, comparing the 0 and 11 kV cm^{-1} cases. In contrast, a 6.5% and 33% increase in droplet contact angle and CLL respectively, with a 43% decreases in wetted area was shown for the partial non-wetting case at $t = 2$ min, comparing the 0 and 6 kV cm^{-1} cases. This result is due to unpinning and receding of the triple line as a result of the applied 6 kV cm^{-1} electric field.

Despite clear change in the droplet shape, the near constant CLL and wetted area resulted in an approximately unchanged q_{in} and q''_{in} across the evaporation period for the partial wetting droplet for all applied electric fields. In contrast, the receding contact line, and subsequent increasing CLL and decreasing wetted area, due to the 6 kV cm^{-1} applied electric field results in a 20% decrease in q_{in} in comparison to the 0 kV cm^{-1} case at $t = 2$ min for the partial non-wetting substrate. While a 18.5% and 7.6% increase in q''_{in} and the liquid-gas surface area was observed respectively. Comparing the partial non-wetting droplet at $t = 2$ min for the 0 and 6 kV cm^{-1} cases by scaling the results to achieve the same wetted area highlights the influence of the applied electric field. A 1.4 times increase in cooling was calculated by replacing a singular partial non-wetting droplet in no applied electric field, with several (1.75) partial non-wetting droplets in the maximum applied field for the same total

wetted area. The multiple droplets resulted in an increase in CLL and liquid-gas surface area by 1.32 times and 1.88 times respectively.

The results of this chapter highlight that droplet evaporation is independent of applied electric field if no change in the contact line occurs in comparison to the 0 kV cm^{-1} case, over the examined parameter space. However, clear dependency is shown on electric field if the applied field results in a moved contact line. In the case of spray cooling a correctly applied electric field could increase the CLL of an individually evaporating droplet enabling optimum cooling.

Chapter 8

Conclusions

Electrospray cooling and droplet evaporation on a heated substrate has been experimentally investigated. This involved the study of the heat and mass transfer mechanics at the solid - liquid interface. To aid in the interpretation of the results and to highlight the limitations of the current research in the field of spray cooling and droplet evaporation a comprehensive literature review was conducted. An experimental apparatus was designed and built that enabled characterisation of the local heat transfer from a heated substrate using thin foil thermography. Four experimental studies were carried out and extensive analysis was performed on all acquired data.

First, a parametric study of a single source electrospray was carried out using thin foil thermography to investigate the local convective heat transfer to a single source electrospray. Two distinct cooling regimes were observed; evaporative and pool electrospray cooling. The evaporative regime was shown to exist at greater separation heights and lower flow rates, with the converse true for pool cooling. Neither were shown to have a significant dependence on nozzle size. The CTP was shown to be a function of k_f , δ_f , q''_{gen} , j_m , T_∞ , and h_{fg} . The largest values of $q''_{ec,peak}$ were noted for the evaporative regime prior to CTP at the smallest separation heights. This point just prior to the transition to pool cooling corresponds to the impact of individual generated droplets, all with their own contact line, on the heated substrate to achieve the maximum CLL [72, 73, 154]. After this point the required evaporative heat flux exceeds that which can be supplied by q''_{gen} and q''_{lc} , resulting in coalescence of the generated droplets on the heated substrate forming a macro-sized droplet.

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The evaporation rate of this quasi-steady droplet is governed by its singular contact line. A peak heat flux of 18.87 kW m^{-2} was observed in the evaporative regime at $H = 2.5 \text{ mm}$, $Q = 4 \text{ }\mu\text{L min}^{-1}$, $D_i = 0.330 \text{ mm}$, $D_o = 0.629 \text{ mm}$, and $V_n = 2.03 \text{ kV}$. This corresponded to a 18.71 fold enhancement over natural convection. The results of this work highlight the importance of the local droplet mass flux on the convective heat flux as well as heat transfer regime. In general, any parameter change that tends to increase the droplet mass flux also increases the convective heat flux in the evaporative cooling regime due to the increasing CLL. However, due to the limited thermal energy that can be supplied due to the internal generation and lateral conduction, transition to pool cooling occurs when insufficient energy can be supplied to accommodate full evaporation of the impinging droplets. Both regimes are observed to be dependent on triple line heat transfer; evaporative due to the multiple contact lines induced by impacting and spreading of droplets, while pool cooling is governed by the much larger singular contact line established by the quasi-steady droplet formed on the heated substrate.

Second, the convective heat transfer to an electrospray array operating in the evaporative regime was parametrically investigated. The influence of the working fluid flow rate and driving electric field in this cooling regime was explored. Increasing peak, radial profile, and average convective heat flux was observed for increasing flow rate due to the increasing droplet mass flux and CLL. An increase in the peak, radial profile, and average heat flux was also noted for increasing driving electric fields. An 89.2% and 64.86% increase in the peak and average heat flux was noted between the lowest and highest applied electric fields for $E_d = 4 - 8 \text{ kV cm}^{-1}$, $Q = 3 \text{ }\mu\text{L min}^{-1} N_n^{-1}$, $D_i = 200 \text{ }\mu\text{m}$, $D_o = 400 \text{ }\mu\text{m}$, $H_1 = 0.5 \text{ mm}$, $H_2 = 7.5 \text{ mm}$, $V_n = 4.1 - 7.1 \text{ kV}$, $V_{ex} = 3 - 6 \text{ kV}$, and $q''_{gen} = 1,395 \text{ W m}^{-2}$. This was due to increasing droplet mass flux, resulting in an increasing CLL, from reduced droplet evaporation en-route to the target and narrowing of the spray plume. This narrowing spray plume, at higher driving electric fields, also resulted in a more defined radial heat flux profile. Comparing the arrays performance with that of a single source under similar experimental conditions showed a 1.14 times increase in average heat flux and a 31% lower wall temperature. This lower wall temperature was noted for a 29% decrease in the standard deviation of the wall temperature over the compared cooling area. The results of this work highlight the importance of electrospray arrays in achieving uniform cooling over large areas and demonstrates the role that the working fluid flow rate and driving electric field have in optimising thermal

dissipation.

Next, the local heat transfer to an evaporating droplet for different surface wettability conditions was examined. Pinned and stick-slip evaporation regimes were observed for partial wetting and partial non-wetting heated substrates respectively. Thermal energy and heat flux were noted to marginally increase by 5.5% for the pinned evaporation case for a 16% and 31% decrease in the liquid-gas surface area and droplet height respectively. Whereas, for the stick-slip regime with its receding contact line and increasing CLL ($0.97 - 2.09 \text{ mm}^{-1}$), thermal cooling decreased by 41% while heat flux increased by 67%. This was for a 79% and 76% decrease in the solid-liquid and liquid-gas surface areas. Comparing droplet evaporation for different time points and substrate wettability conditions, by scaling the results to achieve similar wetted areas, the importance of CLL and contact angle was highlighted. A 2.7 times increase in cooling was calculated by replacing a singular large partial non-wetting droplet with several (4.66) smaller partial non-wetting droplets of the same total wetted area. The multiple smaller droplets resulted in an increase in CLL by 2.15 times and a decrease in droplet height by 44%, for similar contact angle, liquid-gas and solid-liquid surface areas. A 4.87 times increase in heat transfer was noted by comparing a singular large partial wetting droplet with several (16.61) smaller partial non-wetting droplets for the same total wetted area. The multiple smaller droplets resulted in an increase in CLL and liquid-gas surface area by 4.08 times and 4.48 times respectively. The droplet height and wetted area were comparable across this range. These results for heat transfer and CLL give greater understanding to the heat transfer mechanisms for evaporative and pool electrospray cooling. They correlate well and further develop the observations of Horacek et al. [73] and Sodtke and Stephan [72]. From the presented results, considering real world conditions, the optimum cooling case is achieved for the smallest droplets (largest CLL) operating in the evaporative regime just prior to CTP on a substrate with high wettability for the applied coolant.

Finally, droplet evaporation in an electric field for different surface wettability conditions was explored. Similar to the results of Chapter 6, both pinned and stick-slip evaporation were observed for partial wetting and partial non-wetting heated substrates respectively for all applied electric fields. Analogous trends in q_{in} and q''_{in} were observed for all applied electric field cases as their no electric field counterparts. A peak electric field was noted at the start of each evaporation, decreasing in magnitude as the evaporation process proceeded due to the

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increasing separation height between the electrode and droplet as the droplet evaporated. A resultant downward force was calculated for the applied electric field for the partial wetting case. A peak value of 1.17 mN was found for the maximum applied field of 11 kV cm^{-1} at $t = 2 \text{ min}$. In contrast, an upward resultant force was calculated for the applied electric field for the partial non-wetting case. A peak magnitude of 0.20 mN was calculated for the maximum applied electric field of 6 kV cm^{-1} at $t = 2 \text{ min}$. The applied electric field resulted in clear augmentation of the droplet morphology with greater change noted with increasing applied electric fields for both wettability conditions. A reduction of 49% and 24% was noted in the apex radii for the partial wetting and partial non-wetting cases respectively at their maximum applied electric fields at $t = 2 \text{ min}$ in comparison to their no electric field cases. A 15% decrease in the contact angle with no change in CLL or wetted area was observed for the partial wetting substrate at $t = 2 \text{ min}$, comparing the 0 and 11 kV cm^{-1} cases. In contrast, for the partial non-wetting case, a 43% decrease in wetted area, with a 6.5% and 32% increase in droplet contact angle and CLL respectively was shown at $t = 2 \text{ min}$, comparing the 0 and 6 kV cm^{-1} cases. This result is due to unpinning and receding of the triple line as a result of the applied electric field.

Despite clear change in the droplet shape, the near constant CLL and wetted area resulted in an approximately unchanged q_{in} and q''_{in} across the evaporation period for the partial wetting droplet for all applied electric fields. In contrast, the receding contact line, and subsequent increasing CLL and decreasing wetted area, due to the 6 kV cm^{-1} applied electric field results in a 20% decrease in q_{in} in comparison to the 0 kV cm^{-1} case at $t = 2 \text{ min}$ for the partial non-wetting substrate. While a 18.5% and 7.6% increase in q''_{in} and the liquid-gas surface area was observed respectively. Comparing the partial non-wetting droplet at $t = 2 \text{ min}$ for the 0 and 6 kV cm^{-1} cases by scaling the results to achieve the same wetted area highlighted the influence of the applied electric field. A 1.4 times increase in cooling was calculated by replacing a singular partial non-wetting droplet in no applied electric field, with several (1.75) partial non-wetting droplets in the maximum applied field for the same total wetted area. The multiple droplets resulted in an increase in CLL and liquid-gas surface area by 1.32 times and 1.88 times respectively. The results of this chapter highlight that droplet evaporation is independent of applied electric field if no change in the contact line occurs in comparison to the no electric field case, over the examined parameter space. However,

clear dependency is shown on electric field if the applied field results in a moved contact line. In the case of spray cooling a correctly applied electric field could increase the CLL of an individually evaporating droplet allowing a greater overall CLL on the heated substrate.

This work has demonstrated the potential of two-phase electrospray cooling for future cooling applications. On a broader scale, the importance of heat and mass transfer at the contact line for spray cooling and droplet evaporation, and the influence of contact line length density, surface wettability and an applied electric field on subsequent heat transfer has been shown.

Chapter 9

Future Work

Although significant contributions have been made by the current work, suggestions for future work include:

- Local convective heat transfer investigation in a scaled up electrospray array of 19 or 37 sources in order to improve central plume confinement and thus eliminate plume displacement in the central ring.
- Local convective transient heat transfer study of impacting droplets on a substrate of varied wettability in and out of an applied electric field. This would enable an insight into the local heat flux distribution during droplet impact and the influence that surface wettability and an applied electric field has on this process.
- Microgravity droplet evaporation for varied surface wettability, in and out of an applied electric field. The local heat flux distribution of an evaporating droplet in space has not been investigated. This baseline case along with experimental droplet augmentation through surface wettability and applied electric field may lend an insight into the fundamentals of two phase heat transfer in space.
- Microgravity electrospray cooling. Electrospraying has minimal pressure, electrical and coolant volume requirement for operation. Generated droplets are propelled towards the target surface primarily due to electrostatic forces. The present research has shown that a near uniform heat flux removal may be achieved through electrospray

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arrays. These facts make electrospray cooling highly attractive as a potential thermal removal device in space applications.

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Appendix A

Lateral Conduction Modelling

COMSOL 4.3a was used to numerically model the heat and mass transfer mechanics during experimentation using its heat transfer module. This model was used to investigate the influence of the heated substrates thermal resistance on transition between evaporative and pool cooling.

A.1 Model Definition

Figure A.1 illustrates the model space of the numerical model. A list of the relevant input parameters is shown in Table A.1. Table A.2 lists the domain materials from Figure A.1a. Figure A.1b and Table A.3 outlines the boundaries of the numerical model. The model was axisymmetric about the z-axis at $S_x = 0$ mm. A generate heat flux of 1395 W m^{-2} is applied to domain 1 (stainless steel foil). Natural convection from an upward facing plate boundary condition (BC) is applied to boundary iv, from a vertical surface to boundaries ii, iii, and v, and from a downward facing plate to boundaries vi and vii. A diffuse surface BC is applied to

Table A.1: Model parameters.

Generated heat flux, q''_{gen}	1,395	W m^{-2}
Foil thickness, δ_f	25	μm
Paint thickness, δ_p	10	μm
Air gap thickness, δ_{ag}	7	mm
IR glass thickness, δ_{IR}	2	mm
Ambient temperature, T_∞	28	$^\circ\text{C}$

A.1. MODEL DEFINITION

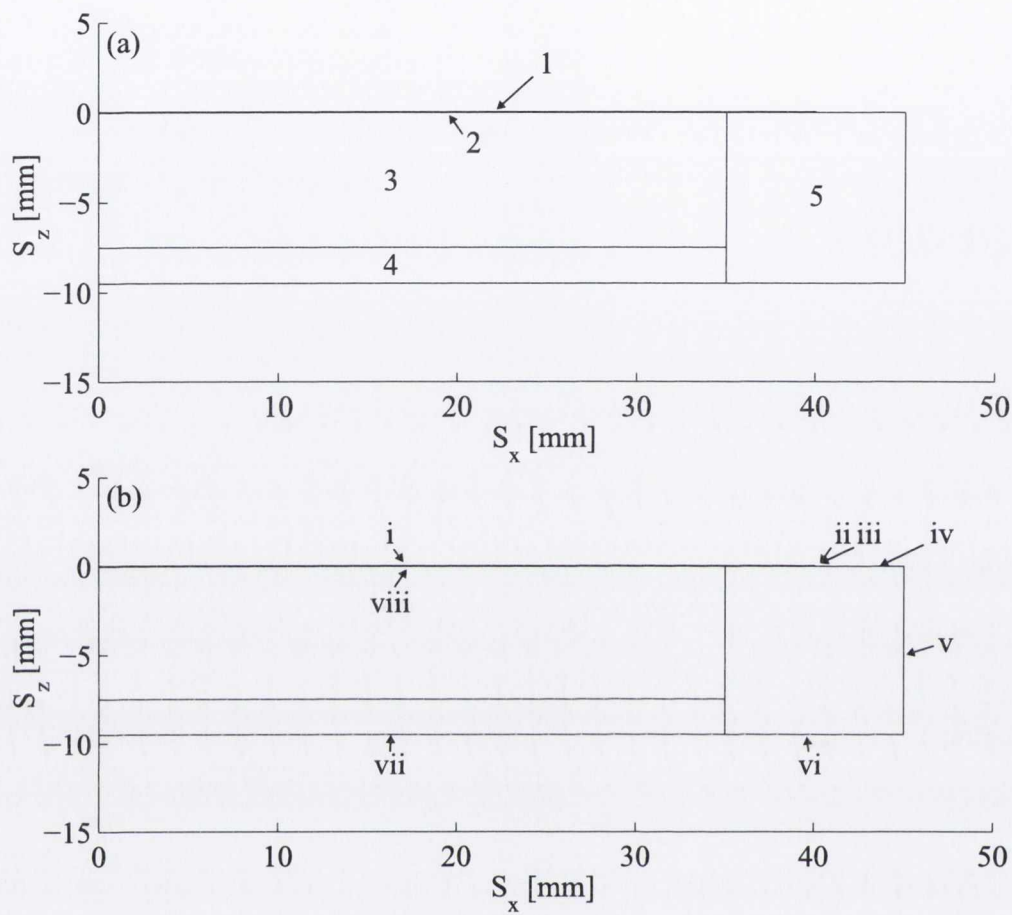


Figure A.1: COMSOL 4.3a model schematic. (a) model domains and (b) model boundaries.

boundaries i and viii. A radial heat transfer coefficient BC is placed also place on boundary i. The curve is based on a modified version of equation A.1:

$$h = \frac{x}{\sigma \sqrt{2\pi}} e^{-\frac{(S_x - \mu)^2}{2\sigma^2}} \tag{A.1}$$

where σ defines the radial spread of the curve, μ the location of the local maximum and x is

Table A.2: Model domain materials.

Domain	Material
1	Stainless steel 316
2	Paint
3	Air
4	CaF ₂
5	PEEK

Table A.3: Model boundary locations.

Boundary	Description
i	Top surface of foil
ii	Vertical edge of foil
iii	Vertical edge of paint
iv	Top surface of PEEK
v	Vertical surface of PEEK
vi	Bottom surface of PEEK
vii	Bottom surface of IR window
viii	Bottom surface of paint layer

a scalar that alters the magnitude of the applied heat transfer coefficient. This equation was modified to create a temperature profile that matched closely with an experimental case. This case corresponded to that of the radial evaporative energy balance outlined in Figure 4.2 at a scalar value x of 0.725. A comparison of the experimental and numerical results for radial temperature profile and lateral conduction are shown in Figure A.2a and b.

A.2 Model Results

Figure A.2a and b show good agreement between the experimental and numerical results. Using this result as a base case, the scalar value x is increased parametrically to increase the magnitude of the heat transfer coefficient h applied to the top surface of the heated substrate. Figure A.2c, d and e show the impact of this on the radial temperature profile (Figure A.2d) and subsequent radial lateral conduction (Figure A.2e) for increasing applied heat transfer coefficient (Figure A.2c). Figure A.2d shows that as h is increased, the temperature profile decrease up to the point where $T_s \approx T_\infty$. At this point h can no longer the foil temperature T_s at $S_x = 0$ any further. Subsequent increase in h results in an increase in the region around $S_x = 0$ mm where $T_s \approx T_\infty$.

It is at this point where T_s at $S_x = 0$ mm approaches T_∞ that the electrospray cooling transitions from evaporative to pool cooling. This is shown by Figure A.2e, as previously outlined, the transition in cooling regime occurs when the available energy at a point is insufficient to evaporate the mass flux being deposited on the surface. This available evaporative energy for our quasi-steady case is a combination of q''_{gen} the energy at a point, and q''_{lc} the energy able

A.2. MODEL RESULTS

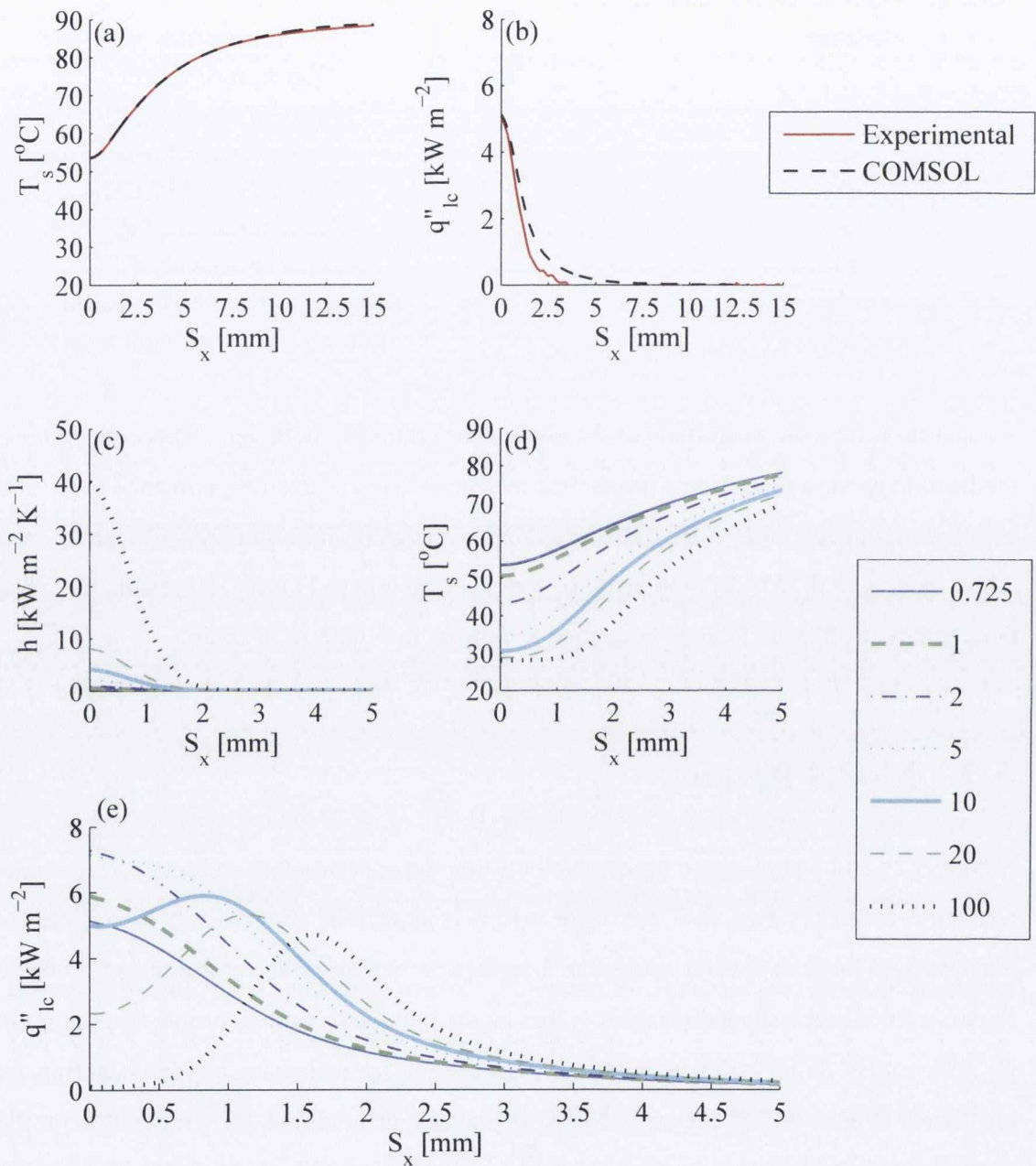


Figure A.2: (a) - (b) COMSOL - Experimental comparison for radial temperature profile and lateral conduction, (c) - (e) impact of increasing applied heat transfer coefficient on radial temperature profile and lateral conduction.

to move within the foil. As q''_{gen} does not change during our experimentation, the limiting case is when q''_{lc} can no longer supply the required energy from the far field to evaporate the working fluid.

Figure A.2e shows that as h increases, q''_{lc} increases to a point where $T_s \approx T_\infty$ at $S_x = 0$

mm, decreasing thereafter. It is at this “bottle neck” point that the onset of pool cooling occurs. Figure A.3 highlights the effect that an increasing h profile has on ΔT and q''_{lc} at $S_x = 0$ mm. ΔT is the difference in temperature between T_s and T_∞ . ΔT is shown to decrease in an asymptotic trend towards 0 K (T_∞). q''_{lc} is shown to have a local maximum between $x = 2$ and 5. It is between these two points that the thermal transport threshold is reached and the onset of pool cooling would occur. The local maximum is due to the flattening out of the radial T_s around $S_x = 0$ mm causing a subsequent drop in the q''_{lc} radial profile (see Figure A.2e). This transition point is highlighted by the vertical red line in the graph, with all points to the left denoting evaporative cooling and points on the right corresponding to the pool cooling regime. This conclusion is further corroborated by the experimental data tabulated in Figure A.4

Figure A.4 demonstrates the impact increasing the working fluid flow rate has during experimentation for a fixed nozzle size and separation heights. The vertical line denotes the transition from evaporative to pool cooling. This increasing working fluid is analogous to increasing the mass flux j_m . Figure A.4a – c shows the transition occurs at increasing values of Q for increasing values of H . This is due to the greater spreading of droplets due to the increased transit distance, resulting in a drop in j_m at a given flow rate. Regime transition does not occur for the $H = 17.5$ mm case. As with the results derived from the numerical model in Figure A.3, ΔT is shown to decrease asymptotically toward 0 K (T_∞) for increasing working fluid flow rate for all separation heights. Prior to transition, q''_{lc} is observed to increase, before decreasing after the onset of pool cooling (see Figure A.4b and

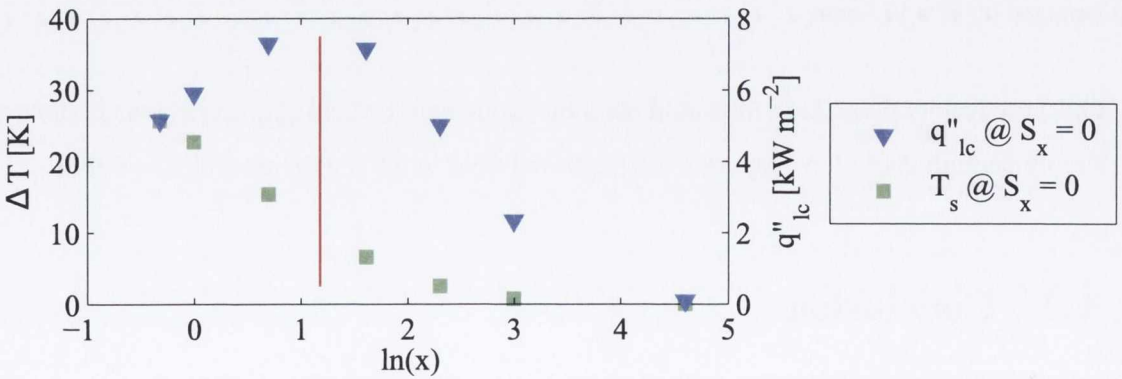


Figure A.3: COMSOL: Effect of increasing applied heat transfer coefficient on ΔT and q''_{lc} at $S_x = 0$ mm.

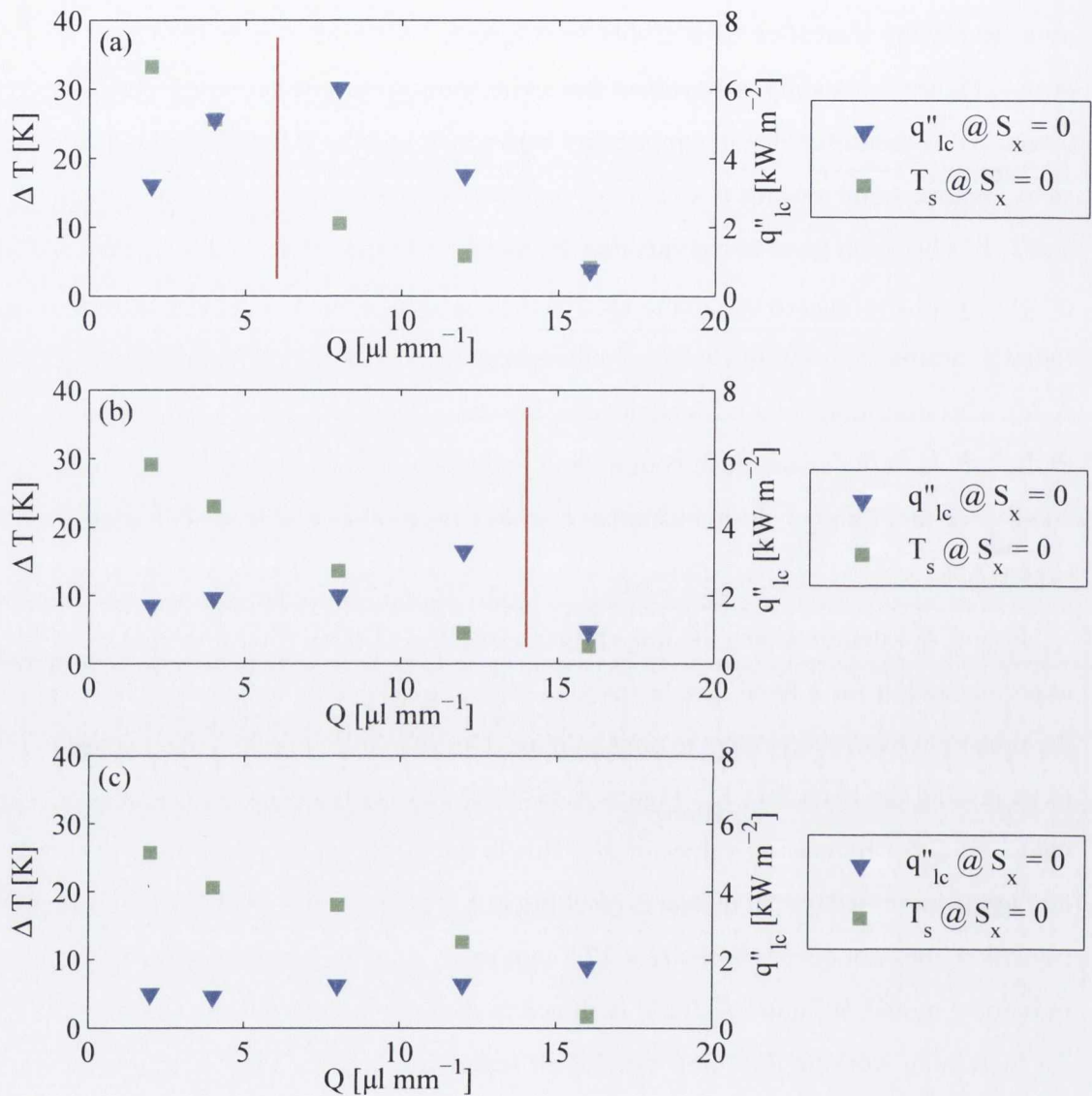


Figure A.4: Experimental: Effect of increasing working fluid flow rate on ΔT and q''_{lc} at $S_x = 0$ mm, $D_i = 0.330$ mm, and $q''_{gen} = 1,395$ W m^{-2} , for varied separation height, (a) $H = 7.5$ mm, (b) $H = 12.5$ mm, and (c) $H = 17.5$ mm.

c). These results show that a threshold mass flux point exists where q''_{gen} and q''_{lc} can no longer supply enough energy to evaporate the deposited fluid in the region around $S_x = 0$ mm.

A.3 Conclusion

From the presented experimental and numerical data presented it is clear that a threshold point between evaporative and pooling exists. This cooling transition point (CTP) is a function of

the heated substrate properties, applied generated power, the mass flux being deposited on the surface at the point in question, the ambient conditions and the latent heat of evaporation of the cooling fluid (h_{fg}):

$$CTP = f(k_f, \delta_f, q''_{gen}, j_m, T_\infty, h_{fg}) \quad (\text{A.2})$$