

A STUDY OF HEAT TRANSFER IN FORCED CONVECTIVE DRYING WITH THE APPLICATION OF LASER-INDUCED FLUORESCENCE THERMOMETRY

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Ph.D.

Declaration

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Eoin Fanning, May 2016

Abstract

Industrial drying is a highly energy-intensive process accounting for up to 25% of energy consumption in some developed countries. While there is a wide range of literature on the topic of drying, the majority of studies focus on specific materials and processes. Furthermore, there are few studies which consider local heat transfer effects on drying. Recently, this has become a fast-developing topic of study in numerical work using two-dimensional and three-dimensional computational fluid dynamics simulations. However, there is not a wide range of experimental work in which the influence of local heat transfer behaviour is studied.

In order to study local heat transfer, complex arrangements of thermocouples or infra-red thermography are typically used to measure the surface temperature. A novel two-dimensional temperature measurement technique involves the use of laser-induced fluorescence in which temperature fields of the fluid can be inferred from fluorescence images. However, this technique has been employed scarcely in low temperature heat transfer problems involving gases. This study investigates two-dimensional heat transfer in drying and evaluates the use of LIF thermometry as a technique for convective drying as well as other low temperature heat transfer problems.

An experimental model was developed, using an instantaneous mass measurement system and infra-red thermography to study the time evolution of the mass (or the moisture content) and the surface temperature distribution of a saturated porous cube. A laser-induced fluorescence experimental setup has been designed to validate and calibrate the relationship between fluorescence intensity and temperature, for toluene vapour excited with a 266 nm laser sheet, over a small temperature scale. This experimental setup was then extended to a flow tunnel and used to measure temperature fields surrounding a heated cylinder (convective heat transfer) and then a drying porous cube (convective heat and mass transfer). This

technique was also tested using an emerging fluorescent tracer material, anisole, for which there are only a few studies to date.

The relationship between toluene fluorescence intensity and temperature was validated over a temperature scale significantly smaller than in previous studies. Despite some inaccuracy in inferring temperature measurements, it was found that laser-induced fluorescence thermometry can be applied successfully in low temperature scales and used as a tool for studying heat transfer distribution across solid geometries. Using the single colour technique, the error of temperature measurement was found to range between 2°C and 10°C for temperatures of 20°C and 95°C, respectively. This is comparable to the error ranges of previous studies. A novel method for increasing the temperature-sensitivity by time-resolving the fluorescence signal has also been proposed. This led to a 50% increase in the sensitivity of the temperature measurements. This technique has not been performed to date and is particularly useful for low temperature scales where the sensitivity between fluorescence intensity and temperature is limited.

During the constant drying rate period of the drying experiments, determination of heat and mass transfer coefficients meant that the analogy between heat and mass transfer in drying could be examined. These results indicated that the analogy between heat and mass transfer is not likely to hold for intense drying conditions (flow rate and temperature). This is an important finding since many numerical drying models rely on the analogy to calculate heat and mass transfer coefficients. It is hypothesized that this may be related to an imbalance between the rate of convective heat transfer and moisture supply to the cube edges, which the temperature distributions across the surface suggest. The laser-induced fluorescence temperature measurements demonstrate a thin thermal boundary layer at the corners facing the flow which supports this theory. These results demonstrate the value of two-dimensional temperature measurements for developing a deeper understanding of heat and mass transfer behaviour in drying.

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Nomenclature

Symbol	Units	Description
A_{ds}	[m ²]	Area of drier solid (drying solid during CRP where temperature is 10% above the CRP average)
A_{mr}	[–]	Mass amplification due to moment ratio
A_{rod}	[–]	Cross-sectional area of suspension rod
A_s	[m ²]	Surface area
Bi_m	[–]	Biot number for mass transfer, $Bi_m = h_m L / D$
C	[mol/m ³]	Tracer concentration
c_p	[J/kgK]	Specific heat capacity
C_{hm}	[–]	Heat and mass transfer analogy factor
CRP		Constant-drying rate period
D	[m ² /s]	Mass diffusivity
D_{cyl}	[m]	Cylinder diameter
D_{H_2O-air}	[m ² /s]	Diffusivity of water vapour in air
$\dot{E}_0$	[W]	Laser power
f	[–]	Non-dimensional drying rate, $f = \dot{m} / \dot{m}_i$
F_{ds}	[%]	Fraction of drier solid, $F_{ds} = A_{ds} / A_s$
FRP		Falling-drying rate period
g	[m/s ²]	Acceleration due to gravity
gd	[ns]	Intensifier gate delay
Gr	[–]	Grashof number, $Gr = (g\beta(T_s - T_\infty)D_{cyl}^3) / \nu^2$
gw	[ns]	Intensifier gate width
H	[m]	Laser sheet height
h	[W/m ² K]	Mass transfer coefficient
h_m	[m/s]	Mass transfer coefficient
h_{ds}	[J/kg]	Specific enthalpy
h_{lat}	[J/kg]	Specific enthalpy of vaporization
I	[W/m ³]	Fluorescence intensity

I^*	[—]	Temperature-normalized fluorescence intensity, $I^* = I/I(T_{ref})$
I_0	[W/m ²]	Laser excitation intensity (or pulse energy)
K	[m ²]	Intrinsic permeability
k	[W/mK]	Thermal conductivity
K_g, K_l	[—]	Relative permeability of gas and liquid
L	[m]	Length of cube (characteristic length)
Le	[—]	Lewis number, $Le = Sc/Pr = \alpha/D$
M	[kg/kg]	Moisture content (dry basis), $M = (m - m_e)/m_e$
m	[g]	Mass of drying solid
$m_{balance}$	[g]	Mass recorded by balance
M_{crit}	[kg/kg]	Critical moisture content (where FRP begins)
$\dot{m}$	[kg/s]	Rate of mass transfer (or drying rate)
n_r	[—]	Refractive index
n_{re}, n_{ro}	[—]	Extraordinary and ordinary refractive indices
Nu	[—]	Nusselt number, $Nu = hk/L$
Nu_{av}	[—]	Average Nusselt number during CRP
P	[pa]	Pressure
P_c	[pa]	Capillary pressure
P_{sat}	[pa]	Saturation pressure
P_v	[pa]	Vapour pressure
Pr	[—]	Prandtl number
q	[m ³ /s]	Volumetric flow rate of gas
$\dot{q}$	[W/m ²]	Heat flux, $\dot{q} = \dot{Q}/A_s$
$\dot{Q}$	[W]	Rate of heat transfer
$\dot{Q}_{cond,rod}$	[W]	Rate of conductive heat transfer from suspension rod to drying cube
$\dot{Q}_{conv}$	[W]	Rate of convective heat transfer
$\dot{Q}_{lat}$	[W]	Rate of latent heat transfer
$\dot{Q}_{rad}$	[W]	Rate of radiation heat transfer
$\dot{Q}_{st}$	[W]	Rate of sensible heat transfer
$\dot{Q}_{cond,tc}$	[W]	Rate of conductive heat transfer from thermocou- ple to drying cube
r	[m]	Radius
R_h	[m ² K/W]	Resistance to heat transfer
R_I	[—]	Signal ratio, $R_I = I_{blue}/I_{red}$
R_m	[s/m]	Resistance to mass transfer
r_t	[μm]	Pore throat radius

R_v	[J/kgK]	Gas constant for water vapour, $R_v = 461.5$ J/kgK
Ra	[—]	Rayleigh number, $Ra = GrPr$
Re	[—]	Reynolds number, $Re = (\rho V_\infty L)/\mu = (V_\infty L)/\nu$
RH	%	Relative Humidity
$S(X)$	[—]	Standard deviation of measurement series, X
Sc	[—]	Schmidt number
Sh	[—]	Sherwood number, $Sh = h_m L/D$
Sh_{av}	[—]	Average Sherwood number during CRP
w	[m]	Width of drying tunnel
w_{ans}, w_{tol}	[%]	Mass percentage of anisole and toluene
T	[°C]	Temperature
t	[s]	Time
T_{crp}	[°C]	Average surface temperature during CRP
T_{df}	[°C]	Temperature at drying front
T_{ref}	[°C]	Reference temperature
T_s	[°C]	Surface temperature (spatial average for porous cube and maximum for heated cylinder)
T_{tc}	[°C]	Surface temperature reference measurement from thermocouple
T_{vessel}	[°C]	LIF vessel temperature
T_∞	[°C]	Free-stream temperature
u	[J/kg]	Internal energy
V	[m ³]	Volume
$\mathbf{v}$	[m/s]	Fluid velocity (in vector form), $\mathbf{v} = v(x, y, z)$
V_∞	[m/s]	Free-stream velocity of gas (in y direction)
ΔX	[—]	Total uncertainty of a measurement series, $\Delta X = \sqrt{\Delta X_r^2 + \Delta X_b^2}$
$\Delta X_b, \Delta X_r$	[—]	Systematic (or bias) and random uncertainty of a measurement series, X
ΔY	[—]	Total uncertainty of a calculation, Y
Greek		
α	[m ² /s]	Thermal diffusivity
α_{crit}	[°]	Critical angle of incidence
ε	[—]	Emissivity
Φ	[%]	Fluorescence quantum yield
ϕ	[—]	Non-dimensional moisture content, $\phi = (M - M_e)/(M_i - M_e)$

φ	$[-]$	Non-dimensional moisture content based on critical moisture content, $\varphi = (M - M_e)/(M_{crit} - M_e)$
γ	$[\%]$	Total conversion efficiency of imaging setup
γ_R	$[-]$	Ratio of imaging efficiencies through red and blue channel, $\gamma_R = \gamma_{red}/\gamma_{blue}$
η	$[\%]$	Relative depth of drying front
λ	$[\text{nm}]$	Wavelength
μ	$[\text{Ns/m}^2]$	Dynamic viscosity
ν	$[\text{m}^2/\text{s}]$	Kinematic viscosity
θ	$[-]$	Non-dimensional surface temperature, $\theta = (T_s - T_{s,i})/(T_\infty - T_{s,i})$
ρ	$[\text{kg/m}^3]$	Density
ρ_{vs}	$[\text{kg/m}^3]$	Vapour concentration at surface
$\rho_{v\infty}$	$[\text{kg/m}^3]$	Vapour concentration of free-stream gas
σ	$[\text{m}^2/\text{mol}]$	Molar absorption cross section
σ_b	$[\text{W/m}^2 \text{ K}^4]$	Stefan-Boltzmann constant, $\sigma_b = 5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$
τ	$[-]$	Non-dimensional drying time, $\tau = t/t_e$
ω	$[\text{Hz}]$	Angular frequency
ψ	$[-]$	Volume-averaged variable
Ψ	$[\text{N/m}^2]$	Stress tensor describing viscous forces in momentum equation
ζ	$[\text{N/m}]$	Surface tension

Subscripts

∞	Free-stream
a	Air
blue	Blue channel fluorescence
crit	Critical
cf	Carbon fibre
crp	Beginning of CRP
e	Equilibrium
frp	Beginning of FRP
g	Gas
i	Initial
l	Liquid
max	Maximum
red	Red channel fluorescence

s	Surface
sd	Solid
v	Vapour

Chapter 1

Introduction

1.1 Background

Convective drying is an essential production process in a wide range of industries such as food processing, textiles, pharmaceuticals and is also a common process in farming and refinement of raw materials. In recent years, convective drying has become an increasingly important process in the treatment of biofuels and a common heat use for combined heat and power (CHP) plants, particularly biomass-fuelled CHP plants. Low moisture content (MC) in biofuels (e.g. willow chips, miscanthus) increases the efficiency of combustion and the net calorific value of fuel gas produced in gasification, reduces microbial growth during storage enabling longer storage periods and also reduces CO and volatile emissions during combustion [1–3].

The process of biomass feedstock drying, however, can be quite energy intensive due to the high enthalpy of vaporization for water evaporation [3–6]. Industrial drying, as a whole, has contributed up to 20 - 25% of industrial energy consumption in Germany and Denmark in the past [7]. Therefore, it is necessary to find ways to increase the thermal efficiency of the drying process. In practice, the effectiveness of the drying process depends on a wide range of operational parameters (e.g. air temperature, air flow rate, relative humidity, material size and arrangement, etc.). For example, in the natural drying of a bed of biomass pellets, Gigler et al. [8] reported that the size of pellets must be small enough to limit the distance of internal diffusion of water but large enough to provide gaps between pellets for the air to convect heat

effectively. It has also been reported that, if the relative humidity of the air used for drying is too low and its temperature is too high, the drying product can experience non-uniform drying. A large difference in the drying rates between the surface and core of the drying solid can lead to a hard dry layer near the surface, caused by shrinkage, and a moist core [9]. Therefore, drying is a complex industrial process involving many constraints and factors which affect the efficiency of the process and the quality of the dried product.

Drying is regarded as a highly complex heat and mass transfer problem and has been extensively investigated in the past few decades [5, 10–21]. Drying is still regarded by many authors as an open question which has no consistent or definitive theoretical model [22–25]. Given the large contribution to energy consumption of industrial drying and its importance in determining the quality of important products such as biomass fuels, food and pharmaceutical products, it is important to fully understand the kinetics of the drying process.

In practice, there is a large variety of configurations for convective drying ranging across different drying technologies, e.g. belt/moving bed drying, rotary drying, fluidized bed drying and pneumatic drying. In each of these configurations, the flow configuration can be quite different. On the scale of a single drying particle, however, the convective heat and mass transfer process is similar. Therefore, the approach taken for this present study is to focus on the drying kinetics of individual moist or initially saturated solids in a forced convective flow regime.

1.2 Research objectives

The fundamental theory of heat and mass transfer in convective drying has been extensively reviewed over the past century. The rate of drying is dependent on both internal conditions of the moist drying solid and external conditions in the drying gas. Most literature to-date reveals that there is a need to understand, in more depth, the influence of local surface convective heat transfer on the drying rate and the spatial non-uniformities associated with this [26, 27]. Moreover, the material structure and hence internal conditions generally cannot be controlled or optimized in practice, whereas the external conditions can be. Hence, it is important to develop an in-depth understanding of the external convection of heat and vapour in drying so that geometries and flow configurations can be optimized in the design

and operation of industrial dryers.

A laser-imaging technique, known as laser-induced fluorescence (LIF) has become an emerging field of research in engineering problems that involve fluid flow, heat transfer and mass transfer. It involves the determination of a two-dimensional temperature field in a fluid from a fluorescent image formed by a laser sheet (a detailed overview is given in chapter 3). The use of LIF for temperature field measurements is of particular interest in the field of convective drying in order to understand the behaviour of convective heat transfer and evaporation on the surface of a drying body. Convective drying typically involves heat transfer at low temperature ranges where the technique has been employed scarcely to date. For instance, the majority of studies involving LIF thermometry in gaseous flows have been performed in intensive heat transfer flows such as combustion engines [28–30], shock tubes [31, 32] and high temperature turbulent jets [33–35].

Therefore the objectives of this study are as follows:

1. The **first objective** is to develop an experimental model which facilitates simultaneous measurements of the moisture content and the surface temperature distributions of a saturated capillary-porous cube in a forced convective drying configuration, by means of a precision mass balance and infra-red thermography. Note, a cube-shaped drying solid is initially chosen for its regular shape and also since it is more suited to thermal imaging. Rectangular bodies such as slabs, blocks and cubes have also been chosen in many of the most recent two-dimensional and three-dimensional numerical models (discussed in the section 2.3.1), allowing more scope for comparison and verification.
2. The **second objective** of this project is to evaluate the use of LIF temperature field measurements in low temperature gaseous convective heat transfer problems. Ultimately, the aim is to use this technique to measure the temperature fields surrounding a drying solid and to develop a more in-depth understanding of local convective heat transfer in drying.

Chapter 2

Literature Review: Heat and mass transfer in convective drying

2.1 Principles of the drying process

Prommas et al. [36] regarded thermal drying as a result of two simultaneous processes: a heat transfer process in which the moisture content of a solid is reduced and a mass transfer process in which fluid is displaced within the structure of a solid towards its surface. In general, the main driving force which influences the rate of evaporation in a drying process is a gradient in the vapour pressure or vapour concentration between the free stream or ambient air and the near-surface region of the moist body [10, 37]. If the body is initially fully-saturated, and hence the vapour pressure near the surface is saturated, and if the free-stream air has zero vapor content or relative humidity (RH), the gradient in vapour pressure is maximized [37]. As the evaporation occurs, an internal moisture gradient develops between the centre and the surface of the moist body.

Fig. 2.1 shows a cross-sectional illustration of forced convective heat transfer to a dry cylindrical/spherical solid (fig. 2.1a) and forced convective drying (heat and mass transfer) of a saturated capillary-porous cylindrical/spherical solid (fig. 2.1b). For convective heat transfer only, Fig. 2.1a, while the free-stream air velocity, V_∞ , can increase the rate of heat transfer, $\dot{Q}$, the driving force is the temperature difference between the solid surface and free-stream air temperatures, $(T_s - T_\infty)$. For convective drying, Fig. 2.1b, the vapour concentration difference

between the saturated vapour at the material surface and the non-vapour-saturated free-stream air ($\rho_{vs} - \rho_{v\infty}$) is the driving force for evaporation and hence mass transfer.

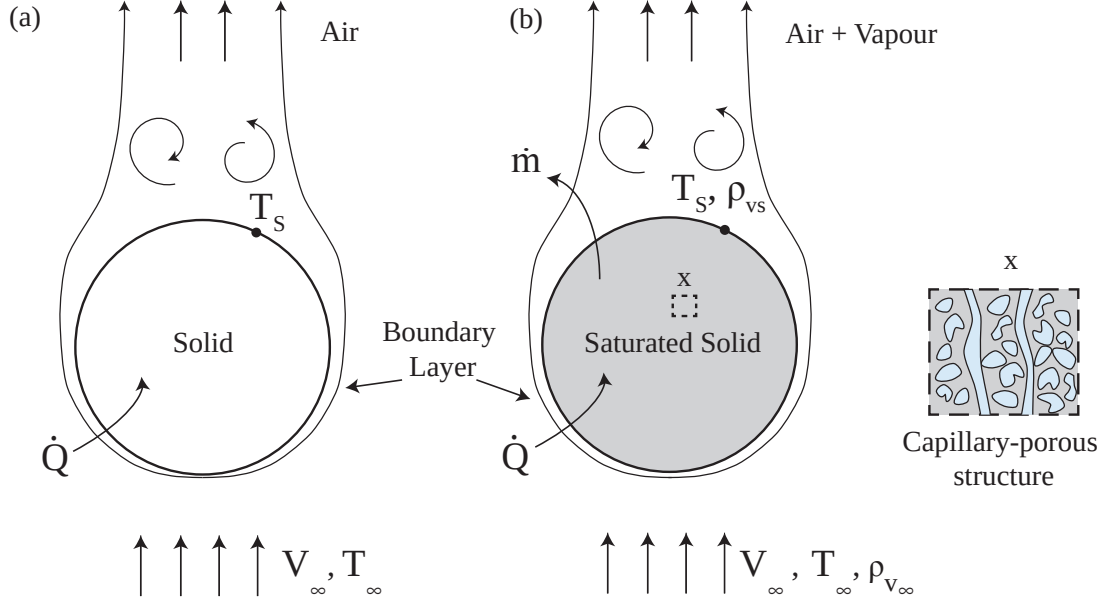


Figure 2.1: Forced convective heat and mass transfer to a sphere for (a) a dry solid sphere (i.e. forced convective heat transfer) and (b) a saturated capillary-porous sphere (i.e. forced convective drying). The typical capillary-porous structure of the saturated material is shown in section x. The blue irregular shapes and streaks are pores and capillaries that bound the water and the remaining grey section is the solid.

The air flow conditions also heavily influence the external and internal moisture gradients that lead to drying. A raised free-stream temperature, T_∞ , results in a larger vapour pressure or vapour concentration difference between the surface and the free-stream air ($\rho_{vs} - \rho_{v\infty}$). Increasing the free-stream velocity of the air, V_∞ , reduces the thickness of the vapour boundary layer. The internal transport of moisture depends on the capillary structure and density of the material. However, a raised free-stream temperature, T_∞ , will also increase the internal gradient from the surface of the body to the core of the body.

2.1.1 Bonding and transport of moisture in solids

Solids are often divided into colloidal bodies, porous bodies and capillary-porous bodies [7, 9]. Porous bodies can be defined as bodies with a structure that contains void spaces of dimensions exceeding the molecule radius of $0.5 \times 10^{-9}m$. It is reported by Kowalski [9] that

porous bodies can be completely dried whereas some bodies contain micro-capillaries (between 0.5×10^{-7} and $0.5 \times 10^{-9}m$) whose capillary forces are greater than the gravitational forces and so cannot be completely dried. These bodies are generally called capillary-porous bodies. Another important property is hygroscopy. Hygroscopic materials such as clay or wood can retain or hold a lot of moisture physically, undergoing some physical changes (e.g. expansion) to accommodate the moisture. Non-hygroscopic materials such as sand and crushed minerals do not physically retain moisture or the amount held is negligible. Colloidal bodies, such as soap or glue, are bodies which contain no pores and are completely hygroscopic. This means they change their dimensions significantly during drying (i.e. the behaviour is elastic).

Bound moisture within a porous solid can be divided into three main bond types: chemical bonds, physical-chemical (PC) bonds and physical-mechanical (PM) bonds [9]. Only moisture in PC bonds and PM bonds can be removed by drying. PC bonds consist of adsorptive, osmotic and structural bonds which are strong bonds found between colloidal particles, in colloidal particles and in the structural cage, respectively. PM bonds are the weakest bonds, most often found in capillary-porous or porous bodies. These consist of bonds of water found in capillaries and spaces between grains, and also free water in pore volumes or the liquid coating the solid surface.

Kowalski [9] also summarized the internal transport of moisture in the form of liquid or gas during drying. Free water and capillary water transports to the surface of the porous solid in liquid form where it evaporates and escapes as vapour to the free stream air. Adsorptive water first evaporates in a pore space and then diffuses in vapour form to the surface. This means that if the porous material is fully saturated, liquid transport is dominant but if it is only partially-saturated, gaseous (vapour) transport is dominant.

2.1.2 Evolution of mass and temperature in drying

In the analysis of the transient drying process, most literature has identified two main periods or stages: A constant drying rate period (CRP) and a falling-drying rate period (FRP) [10, 12, 14, 16, 22, 38–40]. In cases where the solid being dried is initially at a lower temperature than the free-stream air, there is also a pre-heating period in which the solid

surface conditions come into equilibrium with the drying air [9, 17]. Fig. 2.2 below illustrates the evolution of moisture content, M , and temperature, T , of a porous body during a typical drying process.

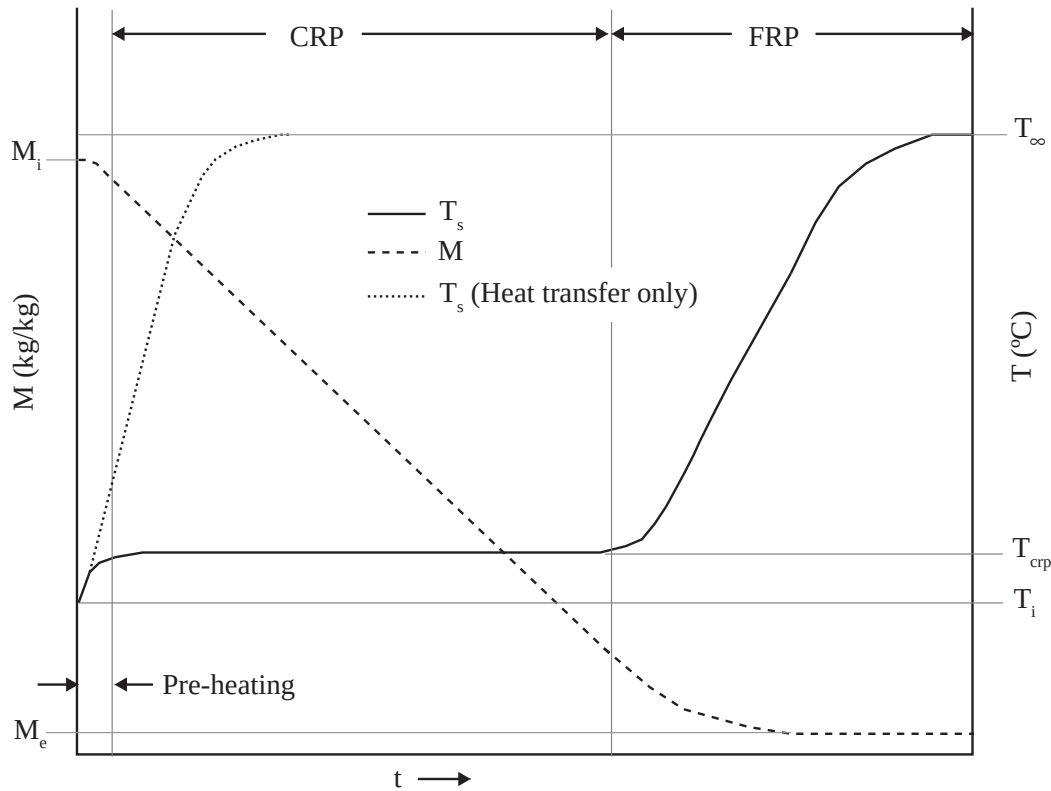


Figure 2.2: Typical curve of evolution of temperature, T , and moisture content, M , for a capillary-porous body. The curve consists of two main stages: A constant-drying rate period (CRP) and a falling-drying rate period (FRP). Also shown is the pre-heating stage where the solid is pre-heated to the steady temperature of the CRP. The dotted line represents the temperature curve of a solid with no moisture (i.e. heat transfer only) being heated to the same ambient or free-stream temperature, T_∞ .

When the surface conditions reach an equilibrium with the drying air, constant evaporation occurs at the surface (the surface remains saturated) and the surface temperature approaches the wet bulb temperature since the vapour at the surface-air interface is saturated. This is where the constant drying rate period begins, as shown in fig. 2.2. This period is often described as a pseudo or quasi steady-state period since the surface temperature, which now equals the wet bulb temperature, remains constant as does the rate of change of mass or rate of drying [39, 41]. This is because the moisture difference between the surface, which remains saturated, and the free-stream air is constant. In cases where there is shrinkage (e.g.

highly hygroscopic materials), since the surface area is changing and hence the evaporation flux is changing, the drying rate will not be constant [42]. The rate of drying in the CRP depends primarily on external conditions (i.e. free stream air temperature, relative humidity and velocity) and evaporation takes place mostly at the surface [9]. The surface remains saturated throughout this period as free water and capillary water is transported from the core to the surface.

Eventually, the rate of evaporation at the surface exceeds the rate of moisture transport from the core to surface and so the surface is no longer saturated [14, 17, 40]. This is when the falling-drying rate period begins. Since the surface is no longer saturated, the surface temperature, as shown in fig 2.2, approaches that of the free-stream air. In the falling-drying rate period, the rate of change of mass or the drying rate becomes dependent on the internal moisture transport conditions rather than the free-stream conditions [9]. Ultimately, the drying rate is reduced since diffusion and evaporation of bound moisture occurs at the interior matrices of the material and hence transport of free water or capillary water is reduced [14, 40]. Eventually, the material reaches zero moisture content or equilibrium moisture content, depending on the material structure (e.g. materials with micro-capillaries or chemically-bound water would not reach zero moisture content).

2.1.3 Simultaneous heat and mass transfer in drying

Unless the free-stream air has a relative humidity of 100% and is at exactly the same temperature as the bound moisture of the solid, simultaneous heat and mass transfer will occur [37]. In order to vaporize, the moisture must absorb the latent heat of vaporization. Therefore, assuming the material surface is saturated with water, as in the constant-drying rate period, a simple energy balance can be described between the rate of latent heat transfer, $\dot{Q}_{lat}$, and the rate of sensible heat transfer at the material surface, such that

$$\dot{Q} = \dot{Q}_{lat} \quad \text{or} \quad \dot{Q} = \dot{m}h_{lat} \quad (2.1)$$

where $\dot{m}$ is the rate of evaporation of water and h_{lat} is the specific enthalpy of vaporization [37, 39]. The sensible heat transfer, $\dot{Q}$, is in the form of convection from the free-stream air and radiation from the surroundings. An additional heat storage term $\dot{Q}_{st} = m\partial h_{ds}/\partial t$

(where m and h_{ds} are the mass and specific enthalpy of the drying solid, respectively and t is the drying time) should be included on the right hand side of equation 2.1 when applied to the pre-heating and the falling drying rate period since the porous body and bound moisture is being heated to the wet-bulb and the free-stream temperature, respectively, during these periods. During the constant drying rate period, however, since the material is at a constant wet bulb temperature, we can ignore the heat storage term, $\dot{Q}_{st}$. The convective heat and mass transfer rates, $\dot{Q}_{conv}$ and $\dot{m}$, respectively, are given by

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) \quad (2.2)$$

and

$$\dot{m} = h_mA_s(\rho_{vs} - \rho_{v\infty}) \quad (2.3)$$

where h and h_m are the heat and mass transfer coefficients, respectively. A_s is the surface area on which heat and mass transfer is taking place. Assuming heat transfer due to radiation is negligible, such that $\dot{Q} = \dot{Q}_{conv}$ for the constant-drying rate period, equations 2.1 - 2.3 combine to give

$$\frac{h}{h_m} = \frac{h_{lat}(\rho_{vs} - \rho_{v\infty})}{T_\infty - T_s} \quad (2.4)$$

Dimensionless analysis of heat and mass transfer by convection demonstrates many analogies between the two transfer processes and this is particularly relevant in the context of drying [37]. The heat and mass transfer coefficients can be represented non-dimensionally as the Nusselt number, $Nu = hL/k$, and the Sherwood number, $Sh = h_mL/D$, respectively. k is the thermal conductivity of the free-stream air and D is the mass (or moisture) diffusivity (or diffusion coefficient). L is the characteristic length of the drying solid (e.g. cylinder diameter, sphere diameter, cube side length, etc.), Nu represents the ratio of convective to conductive heat transfer while Sh represents the ratio of convective mass transfer to diffusive mass transfer. The ratio of momentum diffusivity (kinematic viscosity, ν) to thermal diffusivity, α , of a fluid is represented by the Prandtl number, $Pr = \nu/\alpha$. In the same way, for convective mass transfer, the Schmidt number, $Sc = \nu/D$, represents the ratio of momentum diffusivity

to mass diffusivity. The Lewis number, $Le = Sc/Pr = \alpha/D$, therefore represents the ratio of thermal to mass diffusivity. The Lewis number, Le , indicates the ratio of thermal and concentration boundary layer thicknesses at the drying material surface.

The Chilton-Colburn analogy of heat and mass transfer is a condition in which the molecular diffusivities of heat and mass are identical. This means that the normalized temperature and concentration profiles in their respective boundary layers will coincide. The Chilton-Colburn analogy can be expressed in terms of the ratio of convective heat and mass transfer coefficients, such that

$$\frac{h}{h_m} = \rho_a c_{p,a} \left(\frac{Sc}{Pr} \right)^{2/3} = \rho_a c_{p,a} Le^{2/3} \quad (2.5)$$

for a Prandtl number range of $0.6 < Pr < 60$ and a Schmidt number range of $0.6 < Sc < 3000$ [17, 37, 43]. ρ_a and $c_{p,a}$ are the density and specific heat capacity of the air which are usually evaluated at the mean film temperature but for low vapour mass flux and at atmospheric conditions they can be evaluated from the gas properties with reasonable accuracy [37]. The analogy is typically limited to problems in which the mass flux of the fluid of interest is low relative to the mass flux of the free-stream or bulk fluid [37, 43].

Drying is an example of such a scenario, since the bound liquid is typically evaporating at a low mass flux compared to the mass flux of the drying air. This has been applied to the constant-drying rate period of a convective drying process [15, 17, 27, 44]. Unless all the bound water approaches the saturation temperature, this condition holds in most low temperature (i.e. below the boiling point) steady forced convective drying processes over most regular shaped geometries such as flat plates, cylinders, spheres and liquid surfaces [15, 17, 37]. This analogy has been used in more recent studies involving CFD simulations of convective drying of moist solids [19, 20]. The significance of this is that estimating the Nusselt number, Nu , and Sherwood number, Sh , due to simultaneous heat and mass transfer in drying can be simplified. Combining equation 2.4 and 2.5, the analogy can also be used to determine the surface temperature T_s , such that

$$T_s = T_\infty - \frac{h_{lat}(\rho_{vs} - \rho_{v\infty})}{\rho_a c_{p,a} Le^{2/3}} \quad (2.6)$$

Provided the free-stream temperature, T_∞ , and relative humidity are known, equation 2.6

can be used to estimate the surface temperature and hence heat and mass transfer coefficients during the constant drying rate period of forced convective drying processes over flat plates, circular cylinders, spheres and liquid surfaces [17]. The analogy has also been applied in studies involving rectangular and cube shaped drying solids [19, 20, 45].

This section has provided a summary of the heat and mass transfer process of drying. Section 2.2 reviews in more detail the theories that explain the transient heat and mass transfer phenomena in forced convective drying. Section 2.3 then reviews some of the numerical and experimental techniques developed to understand local heat and mass transfer in drying and the influence of spatial heterogeneities for different geometries.

2.2 Theories of transient heat and mass transfer in the drying of porous materials

In this section, literature is reviewed which examines more closely the evolution of heat and mass transfer during drying. Some of the common theories and analytical models for transient heat and mass transfer in drying are included here.

The evolution of mass and temperature in a typical drying process for constant free-stream conditions and for an initially saturated material was summarised in section 2.1.2. In general, during the two main periods (the constant-drying rate period and the falling-drying rate period) of the typical drying process, different mechanisms of mass transfer are dominant. In the constant-drying-rate period (CRP), assuming a saturated surface, the drying rate $\dot{m}$ is controlled by the rate of surface evaporation [7, 9]. The external conditions therefore have a significant influence in this stage.

At a critical moisture content, M_{crit} , the surface is no longer saturated and the falling-drying-rate period (FRP) begins. In this period, the drying rate is more dependent on the internal conditions (moisture diffusivity, moisture transfer coefficient, water activity, structure and composition, etc. [46]). The heat transfer now consists of forced convective heat transfer to the surface and conduction to the core of the material. Mujumdar [7] explains that in the FRP, as the product becomes drier near the surface, the saturated moisture level becomes progressively deeper and hence the drying rate becomes increasingly influenced by heat

conduction. Furthermore, if the material has a relatively high bulk density with very small pores, the drying rate in this period is more dependent on resistance to moisture diffusion within the material. Both of these factors contribute to a decrease in the rate of drying.

These are the main features of heat and mass transfer during a typical drying process. However, the dominance of the mechanisms and periods of drying described here varies significantly with material structure, making the drying process a complex heat and mass transfer problem [12].

2.2.1 Classic theoretical modelling of transient heat and mass transfer in drying

Most theoretical models of transient heat and mass transfer in drying involve simplification of the heat, mass and momentum conservation equations with realistic assumptions and boundary conditions. The first of the governing equations, the continuity (mass conservation) equation can be expressed as

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0 \quad (2.7)$$

where $\mathbf{v}$ is the fluid velocity (a vector comprising of velocity in x, y and z directions) and t is time. The momentum conservation (Navier-Stokes) equation can be expressed in a shortened form as

$$\rho \left(\frac{\partial \mathbf{v}}{\partial t} + \mathbf{v} \cdot \nabla \mathbf{v} \right) = \rho \mathbf{g} - \nabla P + \nabla \cdot \Psi \quad (2.8)$$

where $\mathbf{g}$ is acceleration due to gravity, P is pressure and Ψ is the stress tensor that describes viscous forces. The energy conservation equation is expressed as

$$\rho \left(\frac{\partial h_{ds}}{\partial t} + h_{ds} \cdot \nabla \mathbf{v} \right) = -\nabla \cdot \dot{\mathbf{q}} + \rho \left(\frac{\partial p}{\partial t} + P \cdot \nabla P \right) + \Theta \quad (2.9)$$

where $\dot{\mathbf{q}}$ is the heat flux (a vector comprising of the heat flux in x, y and z directions) and Θ is the viscous dissipation term, i.e. the rate at which kinetic energy is irreversibly converted to thermal energy due to viscous effects in the fluid.

Some of the earliest theories of drying were hypothesized by Lewis and Sherwood [47–

49] in which the mass transfer inside the drying solid was considered as a purely diffusional process before later work attempted to incorporate the capillary forces of liquid transport [50–52]. A theoretical study by Whitaker [12] can be regarded as an important foundation for many of the common theoretical models used to predict the drying process today. This study considered three phases within a rigid porous medium: A gas phase (vapour and air), a liquid phase (water) and a rigid solid matrix phase (drying solid). By making some assumptions, equations 2.7, 2.8 and 2.9 were simplified and volume-averaged in order to solve moisture content and temperature as a function of time and space during a drying process. The volume-averaging approach means integrating some variable (such as velocity, temperature or pressure), ψ , which is a function of space, over an averaging or representative volume, V , and dividing by the volume of a phase, such that

$$\bar{\psi}_i = \frac{1}{V_i} \int_V \psi dV \quad (2.10)$$

where the subscript i represents the phase, either l , g or sd for liquid, gas (air and vapour) and solid, respectively. This expression gives the intrinsic phase average of a quantity, ψ_i . An illustration of a typical volume element is shown in fig. 2.3. Representative or averaging volumes need to be sufficiently larger than the average pore size to allow representative averaging of the material but also sufficiently small to avoid variations of quantities such as temperature due to macroscopic gradients [53]. The spatial average over the whole averaging volume is given by

$$\bar{\psi} = \frac{1}{V} \int_V \psi dV = \varepsilon_g \bar{\psi}_g + \varepsilon_l \bar{\psi}_l + \varepsilon_{sd} \bar{\psi}_{sd} \quad (2.11)$$

where ε_i is the volume fraction of a phase within a volume. The volume fractions of gas and liquid are functions of time (i.e. $\varepsilon_g = \varepsilon_g(t)$ and $\varepsilon_l = \varepsilon_l(t)$) whereas it may be assumed the volume fraction of solid, ε_{sd} , remains constant (assuming negligible shrinkage of the solid).

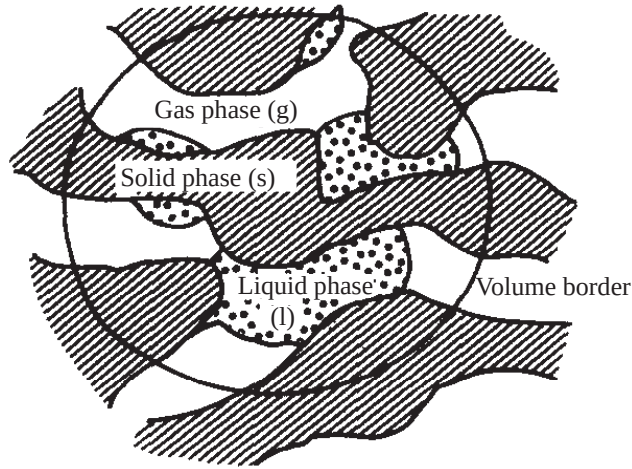


Figure 2.3: Macroscopic volume-averaging element of a rigid porous drying medium containing three phases as applied by Whitaker [12].

By solving volume-averaged expressions of the governing equations 2.7, 2.8 and 2.9, Nasrallah and Perre [15] provided a one-dimensional model of the drying of an infinitely long porous slab (as shown in fig. 2.4), impermeable on one side ($x = 0$) and exposed to air flow (convection) on the other side ($x = L$). The transport of liquid by capillary forces during the constant-drying rate period was modeled by applying Darcy's law:

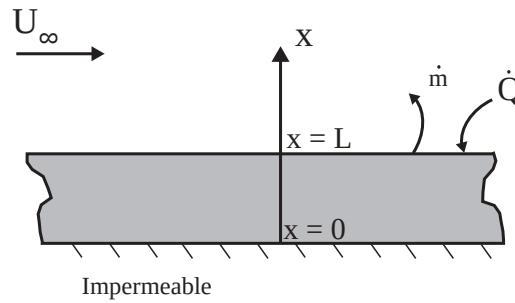


Figure 2.4: Illustration of the geometry considered in the one-dimensional theoretical model of an infinite slab with one permeable side ($x = 0$) and one side exposed to air flow ($x = L$) and hence convective drying [15].

$$\bar{v} = -\frac{K}{\mu_l} \frac{\partial P}{\partial x} \quad (2.12)$$

This represents a proportionality between the velocity of capillary flow $\bar{v}$ and the pressure drop per distance, $\partial P / \partial x$ (in this case x represents the direction normal to the slab surface [15]).

K is the intrinsic permeability of the porous medium and μ_l is the dynamic viscosity of the liquid (water). Nasrallah and Perre [15] applied equation 2.12 separately to the liquid ($\bar{v}_l$) and gaseous ($\bar{v}_g$) phases and the permeability of each is multiplied by their respective relative permeabilities, K_l and K_g . Liquid and vapour permeabilities are typically found for a material from experimental data [38, 54]. The mass conservation equation (equation 2.7) can then be expressed as

$$\frac{\partial \bar{\rho}_l}{\partial t} + \frac{\partial (\bar{\rho}_l \bar{v}_l)}{\partial x} = -\dot{m} \quad (2.13)$$

for the liquid phase, where $\dot{m}$ is the rate of evaporation, $\bar{\rho}_i$ is the volume-averaged density of the phase i (not constant across all phases due to compressibility) and

$$\frac{\partial \bar{\rho}_g}{\partial t} + \frac{\partial (\bar{\rho}_g \bar{v}_g)}{\partial x} = \dot{m} \quad (2.14)$$

for the gaseous phase. Equation 2.14 is applied to the vapour alone, giving

$$\frac{\partial \bar{\rho}_v}{\partial t} + \frac{\partial (\bar{\rho}_v \bar{v}_v)}{\partial x} = \dot{m} \quad (2.15)$$

where the term $\bar{\rho}_v \bar{v}_v$ is the mass flux of vapour which is found from the mass flux of the gas (air and vapour) phase minus the mass flux of vapour diffusion, such that

$$\bar{\rho}_v \bar{v}_v = \bar{\rho}_g \bar{v}_g - \bar{\rho}_g D_{eff} \frac{\partial (\bar{\rho}_v / \bar{\rho}_g)}{\partial x} \quad (2.16)$$

where the mass flux of vapour is due to diffusion and hence expressed using Fick's law of diffusion ($\bar{\rho}_g D_{eff} \partial (\bar{\rho}_v / \bar{\rho}_g) / \partial x$). D_{eff} is the effective diffusion coefficient of the vapour in the porous medium.

Equations 2.13 - 2.15 can be used to model the mass transport within the drying solid. The energy conservation equation (equation 2.9) can be used similarly to model the heat transfer within the drying solid, assuming constant specific heat capacities [15]. It is assumed as a boundary condition that at the impermeable side ($x = 0$), the fluxes of heat and mass transfer are zero. On the exposed side ($x = L$), the heat and mass fluxes are expressed as

$$\bar{\rho}_l \bar{v}_l + \bar{\rho}_v \bar{v}_v = h_m (\rho_{vs} - \rho_{v\infty}) \quad (2.17)$$

$$k \frac{\partial T}{\partial x} + h_{lat} \bar{\rho}_l \bar{v}_l = h(T_\infty - T) \quad (2.18)$$

Nasrallah and Perre [15] applied these equations to model the drying of brick and wood. This was achieved by integrating over control domains, Δx , located at different points in space along a grid or mesh which represented the slab thickness, L . This was applied for a time interval, Δt . Fig 2.5 demonstrates the evolution of average moisture content (in a normalized form, ϕ) and the drying rate (in terms of mass flux, $\dot{m}/A_s$) of wood calculated using this model for a free-stream temperature of $T_\infty = 80^\circ\text{C}$ and relative humidity of $RH_\infty = 60\%$.

Nasrallah and Perre [15] also investigated the effect of applying different heat and mass transfer coefficients to the boundary conditions (equation 2.17 and 2.18) during the CRP. As shown in fig 2.5, they found that the rate of drying depended more significantly on the heat transfer coefficient than the mass transfer coefficient for these free-stream conditions. However, for lower free-stream temperatures, T_∞ (and hence wet-bulb temperature), both heat and mass transfer coefficients have a significant influence and so the analogy between heat and mass transfer is more likely to hold.

The same approach was later adopted by Ilic and Turner [55] and applied to the drying of brick. Their results agreed well with experimental work and with that of Nasrallah and Perre [15]. These particular models have many limitations, however, as listed by Whitaker [12]. Some important limitations include: The assumption that the three phase system is in local thermal equilibrium which is not the case in reality, particularly during the FRP where significant local temperature gradients can exist. A homogeneous porous medium is assumed which doesn't account for the random distribution of liquid, vapour and solid that exists within an volume-averaging element. The models are also quite material-specific, since they require empirically determined coefficients such as diffusivity and permeability. Finally, the models are one-dimensional and do not take into account heterogeneities associated with 2D and 3D models, such as the spatially non-uniform heat and mass transfer along the boundary layers or lateral transport of internal moisture. Nevertheless, these early studies provide important foundations for modelling of transient heat and mass transfer in drying.

Peishi and Pei [56] developed a generalized model of drying using finite element analysis that could be applied to various drying materials. They considered early theories about drying

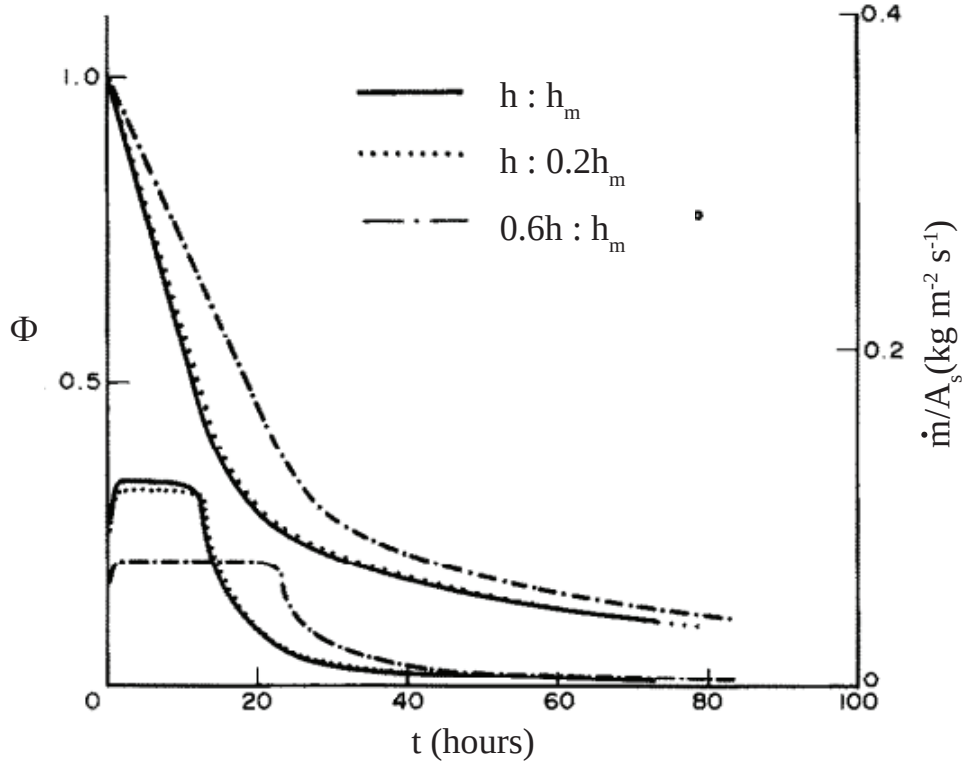


Figure 2.5: Evolution of normalized average moisture content, ϕ , and drying rate (in terms of mass flux, $\dot{m}/A_s$) of wood versus time for three different ratios of heat to mass transfer coefficient ($h : h_m$) [15]. The free-stream conditions in this case are $T_\infty = 80^\circ\text{C}$ and $RH_\infty = 60\%$

that assume discontinuous wetting of the solid surface (patches of wet and dry solid) after a certain critical moisture content [57, 58]. The area of "wetted" surface (i.e. containing free water) was represented as a function of moisture content (and hence time). A percolation theory was used to predict how the fraction of wet area would vary with surface moisture content. There exists a critical free water content above which there is a continuous layer of free water at the surface (constant-drying rate period) and below which the free water at the surface becomes discontinuous (falling-drying rate period). Peishi and Pei [56] reported that this critical value is about 30% of the saturated free water content, for a three-dimensional porous medium.

Later work by Prat [59] and Masmoudi and Prat [16] applied a similar theoretical model to a two-dimensional porous rectangular body exposed to a parallel flow at one surface only and with an impermeable wall at the other surfaces (i.e. zero mass flux). They reported that the typical method of applying spatially-averaged heat and mass transfer coefficients as boundary conditions at the drying solid-air interface (or drying solid surface) is not realistic

since they will vary across the interface. In particular, the mass transfer coefficient will vary since strong moisture content heterogeneities exist at the surface in the falling-drying rate period. This highlights a need for more generalized experimental studies of local surface heat and mass transfer.

2.2.2 The retreating drying front

A popular theoretical concept, often used to explain the modelling of the falling drying rate period, is known as the retreating drying (or moisture) front. It has been reported that in some porous materials a retreating moisture front or evaporation plane is often observed when capillary forces cannot sustain liquid flow to the surface to maintain its saturation [10, 14, 15, 21, 42, 54, 60]. This is illustrated in fig. 2.6.

Schadler and Kast [14] and Yoshida et al. [61] used this theory to model the moisture transport during the FRP for non-hygroscopic porous bodies. A critical moisture content M_{crit} , where the FRP begins, is assumed below which ($M < M_{crit}$) moisture transport takes place by capillary liquid flow to the surface to maintain saturation. Above this value ($M > M_{crit}$), the plane of evaporation begins to retreat inward through the material leaving an unsaturated drier porous zone (often called the sorption zone) through which vapour is diffused. Heat transfer to the evaporation plane is due to conduction through the sorption zone. As the drier sorption zone thickens, the resistance to moisture transport increases and hence causes the drying rate reduction seen in the FRP. Schadler and Kast [14] showed that the heat and mass transfer resistances inside the solid body during the FRP are dependent on the relative depth of the drying front, η (see fig. 2.6). The relative depth of the drying front, for cylinders and spheres, can be related to the dimensionless moisture content (in this case defined as $\phi = M/M_{crit}$). It is reported that for a cylinder, the following expression is valid

$$1 - \eta = \phi^{1/2} \quad (2.19)$$

and for a sphere:

$$1 - \eta = \phi^{1/3} \quad (2.20)$$

2.2. THEORIES OF TRANSIENT HEAT AND MASS TRANSFER IN THE DRYING OF POROUS MATERIALS

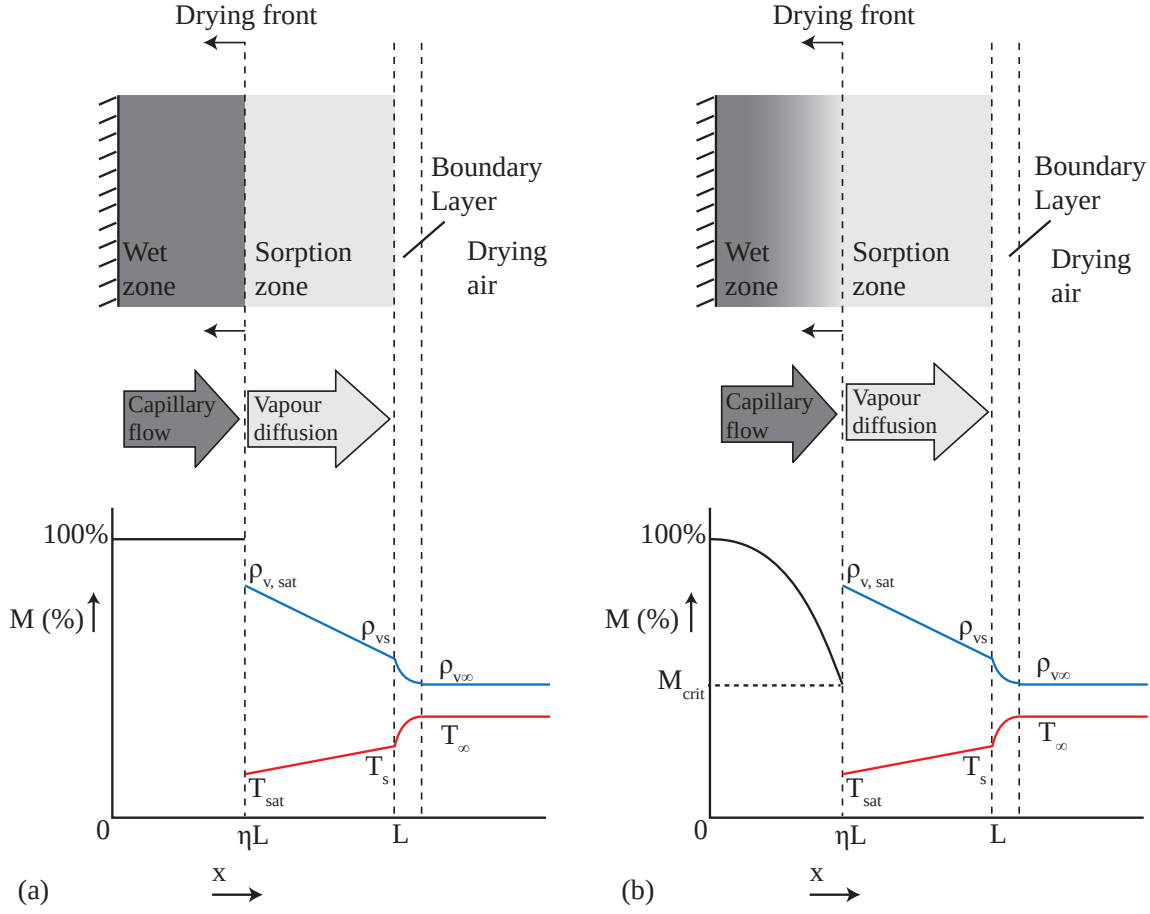


Figure 2.6: Illustration of the retreating drying front in a section of a drying porous medium. The shaded sections at the top represent the wet (liquid capillary flow) and sorption (vapour diffusing) zones either side of the front as it retreats towards an impermeable wall. The blue and red lines are the vapour concentration and temperature profiles, respectively. L is the characteristic length of the solid. η is the relative depth of the moisture front. (a) represents a uniformly retreating moisture front whereas (b) represents a moisture front that retreats when a critical moisture content M_{crit} is reached and which has a resistance to liquid flow (illustrated by the moisture gradient)

This gives the following relationships for heat and mass transfer resistances R_h and R_m , respectively, for a cylinder:

$$[R_h, R_m] \propto \left(\ln \frac{1}{\phi^{1/2}} \right) (1 - \phi); \quad (2.21)$$

and for a sphere:

$$[R_h, R_m] \propto \left(\frac{1}{\phi^{1/3}} - 1 \right) (1 - \phi); \quad (2.22)$$

These expressions tell us that the heat and mass transfer resistances are zero at the critical

moisture ($\phi = 1$) content and tend to infinity as the material becomes fully dried ($\phi = 0$). Schadler and Kast [14] used this to estimate drying rate $\dot{m}$ and their theoretical curves agreed well with experimental data, however, this was for specific materials and drying conditions. They also developed an expression for the surface temperature during the FRP which can be expressed as:

$$T_s = \frac{T_\infty + \frac{1}{hR_h} T_{df}}{1 + \frac{1}{hR_h}} = \frac{hR_h T_\infty + T_{df}}{hR_h + 1} \quad (2.23)$$

The drying or moisture front temperature, T_{df} , was calculated iteratively as a function of ϕ by equating energy balances during the FRP. An additional radiation heat transfer term can be included but, assuming the radiation heat transfer is small compared to convective heat transfer, it can be neglected. Equation 2.23 shows that for a constant convective heat transfer coefficient h , at the beginning of the FRP ($\phi = 1$), R_h tends to zero and hence $T_s = T_{df}$ and at the end of the FRP ($\phi = 0$) R_h tends to infinity and hence $T_s = T_\infty$. It is worth mentioning that this temperature estimation is limited to one dimensional problems and a specific set of materials. It does, however, provide a good demonstration of the use of the retreating drying front theory.

2.2.3 The characteristic drying curve

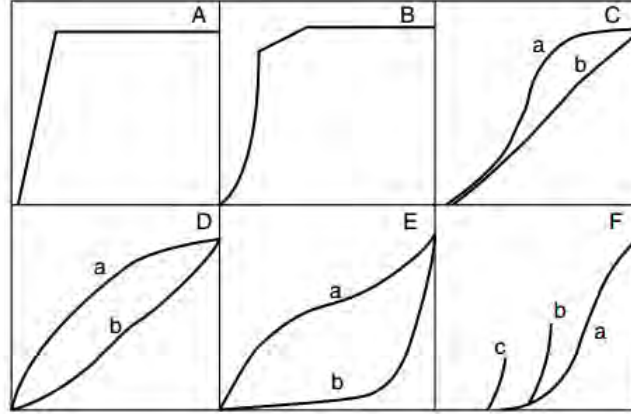


Figure 2.7: Typical curves of normalized drying rate f versus normalized moisture content ϕ for different material types. (A) sand, clay, silica gel, paper pulp, leather; (B) sand, plastic-clay mix, silica-brick mix, ceramic plate, leather; (C) [a] fir wood and [b] Cyprus wood; (D) [a] paper, wool, and [b] potatoes, tapioca tuber, rice flour; (E) [a] rye bread, yeast, and [b] butter and margarine; (F) [a] wheat corns, [b] and [c] represent curves at lower values of initial moisture [7].

The curves in fig. 2.7 represent "characteristic curves" of the normalized drying rate versus normalized moisture content for different material types [7]. It is clear from these curves that there is no single characteristic drying curve, independent of the material properties and differences in the internal structure, and hence resistance to moisture diffusion of the drying solid can strongly influence the evolution of the drying rate with drying time or with moisture content of the solid.

Keey and Suzuki [10] developed the theoretical concept of a characteristic drying curve in order to predict the evolution of drying rates of specific materials based on known drying conditions and material properties. Curves of the normalized drying rate, $f = \dot{m}/\dot{m}_i$, as a function of the normalized moisture content, $\phi = (M - M_e)/(M_{crit} - M_e)$, were plotted for a porous, non-hygroscopic slab of infinite extent. In this case, $\dot{m}_i$ is the initial rate of drying, M_e is the equilibrium moisture content and M is the average moisture content. The critical moisture content, M_{crit} , was assumed to be an invariant value and independent of the initial moisture content and external conditions. The curves are shown in fig. 2.8. Keey and Suzuki [10] noted that the relationship between f and ϕ becomes more linear for lower relative drying intensity, represented as N in fig. 2.8. N is a complicated mathematical expression which is a measure of the steepness of the internal moisture gradient during the FRP. It can be seen that for an increased mass transfer Biot number, $Bi_m = h_m b/D$ (where b

is material thickness and D is its diffusivity), the curve or relationship is almost completely linear at low drying intensity and almost exponential at high intensity. This suggests that for a single material, the trend of the dimensionless relationships of f with ϕ can vary quite significantly for different drying conditions (i.e. N , Bi_m).

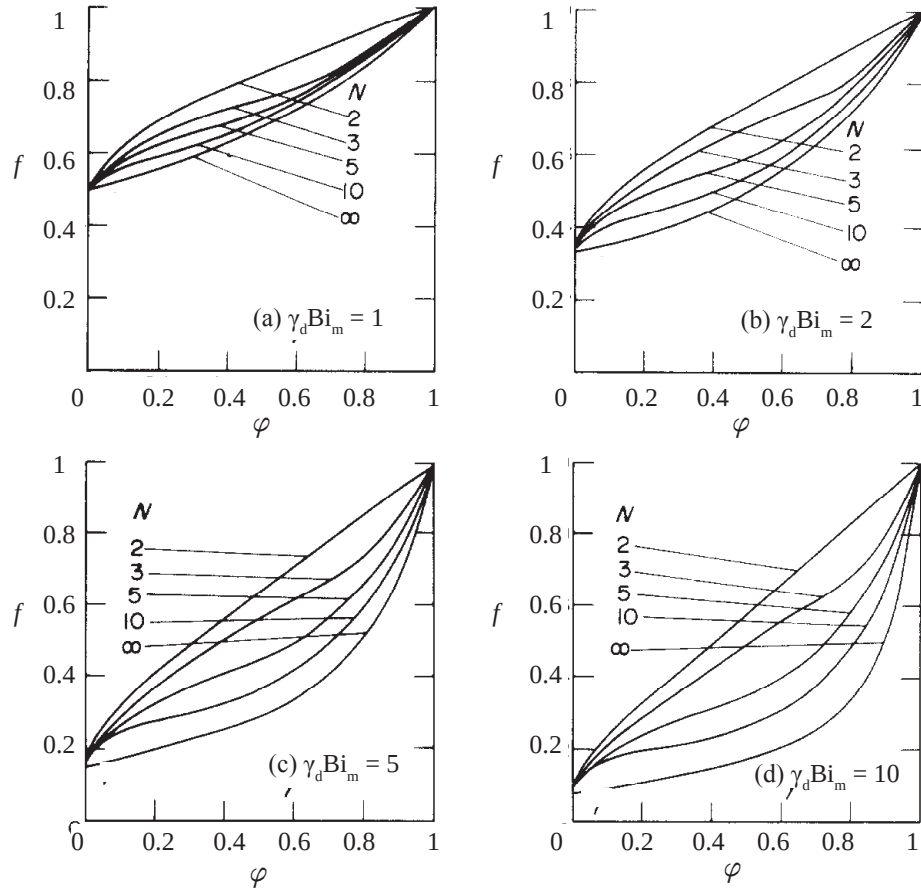


Figure 2.8: Theoretical characteristic drying curve of normalized drying rate f versus normalized moisture content ϕ as defined by Keey and Suzuki [10]. N is a non-dimensional term described as the relative drying intensity which measures the severity of the internal moisture gradient. γ_d is a complex thermophysical coefficient but is usually a constant value between approximately 0.5 and 0.1, for drying wet bulb temperatures between 20 and 100°C, respectively. Bi_m is the mass transfer Biot number.

Mujumdar [7] and Coumans [42] reported that this characteristic curve has a weak theoretical basis but can be used for interpolations and prediction of dryer performance for very specific drying conditions and materials. The overlying problem in normalizing the drying rate into a universal curve is the complex dependence of moisture diffusivity, D , on the local concentration of moisture, which decreases at different rates depending on the drying conditions and material structure[42]. The complex dependence of diffusivity on moisture

content is often highlighted by authors as a key challenge to modelling internal mass transfer accurately [14, 44, 62, 63].

2.2.4 Pore-scale drying

In the macroscopic drying models developed by Whitaker [12] (see section 2.2.1), assumptions were typically made about the phase-distribution in the averaging elements. Later theoretical models of the drying process considered the moisture transport phenomena on a pore-scale in order to understand more fundamentally the moisture transport mechanisms that govern drying. The earliest pore-scale models used pore-networks to represent the porous structure of a solid [64–66]. These networks consist of pores and throats which link neighbouring pores. This concept is demonstrated in fig. 2.9. Though in reality the networks have random irregular coordination, most early models assumed a regular structure with the same mean coordination (i.e. number of throats and pores and average throat and pore size) in order to simplify the model.

These pore-scale models [64, 65] were applied to the scale of a typical macroscopic averaging-volume element in order to estimate more accurately parameters such as permeability, capillary pressure and the effective mass diffusion coefficient as a function of liquid saturation and drying rate. A set of simplified equations based on mass continuity, Darcy's law and Fick's law was used to simulate moisture transport in a pore network using an evaporative flux on one side of the network as a boundary condition. These models assumed negligible external temperature gradients and hence isothermal conditions within the pore networks for further simplification.

A novel approach to modelling the transient moisture transport using pore networks was suggested by Shaw [67] using the theory of invasion percolation. Invasion percolation is the slow replacement (or invasion) of a wetting fluid by a non-wetting one in a porous medium. The technique, typically used to model processes like drainage, was applied to drying where the wetting fluid, in this case, represents liquid water and the non-wetting fluid represents the gas mixture of vapour and air. The receding of liquid or invasion of liquid by vapour and air is in many ways a pore-scale description of the retreating drying front which begins in the falling-drying rate period (see section 2.2.2). A network of pores and throats is considered,

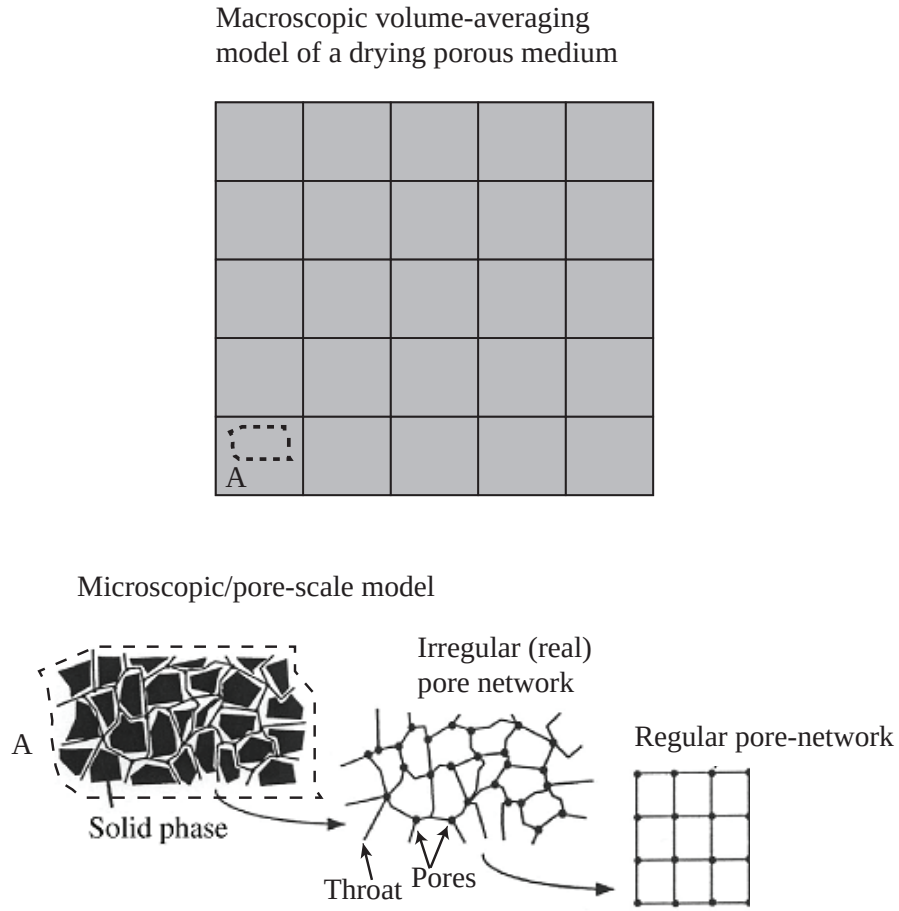


Figure 2.9: Pore network for modelling moisture transport on the pore-scale (i.e. section A shown) where the top illustration represents the scale of a macroscopic drying model and the small squares represent the averaging-volumes (or finite elements) of the porous medium [66].

initially completely filled with liquid with one side of the network exposed to evaporation. This side is the entrance side where the non-wetting (vapour and air) invades from. The throat radii, r_t , are distributed randomly and have a threshold capillary pressure of

$$P_c = \frac{2\zeta}{r_t} \quad (2.24)$$

where ζ is the surface tension of the water. The modelling procedure uses an algorithm as follows: Firstly all the throats at the entrance side are identified. Next, the throat with the lowest threshold capillary pressure (i.e. where capillary forces are weakest) amongst all the throats at the entrance side is selected. This throat is first invaded along with its adjacent pore. The invasion process finishes when the invading fluid (vapour/air) reaches a pore on

the exit side (i.e. the opposite side to the entrance side). The resulting phase distribution can be seen in fig. 2.10a.

This algorithm was later refined by Prat [66] for drying so that clusters of isolated liquid which can remain in drainage processes are allowed to evaporate. This was done by including steps in the algorithm to apply evaporation fluxes at the appropriate throats. The boundary conditions were also changed so that transport of vapor (i.e. bulk evaporation) only took place at the entrance side of the invading fluid. The resulting phase-distribution is shown in fig. 2.10b. Note that the drying process, in this case, occurred at a slow rate with negligible thermal gradients in order to simplify the model. As a result the invasion front appears quite spatially non-uniform in fig. 2.10b. Later studies showed that the front can become more uniform depending on the thermal gradients and distribution of pore sizes [68–70]. Hence, it becomes more representative of a retreating drying front described in section 2.2.2.

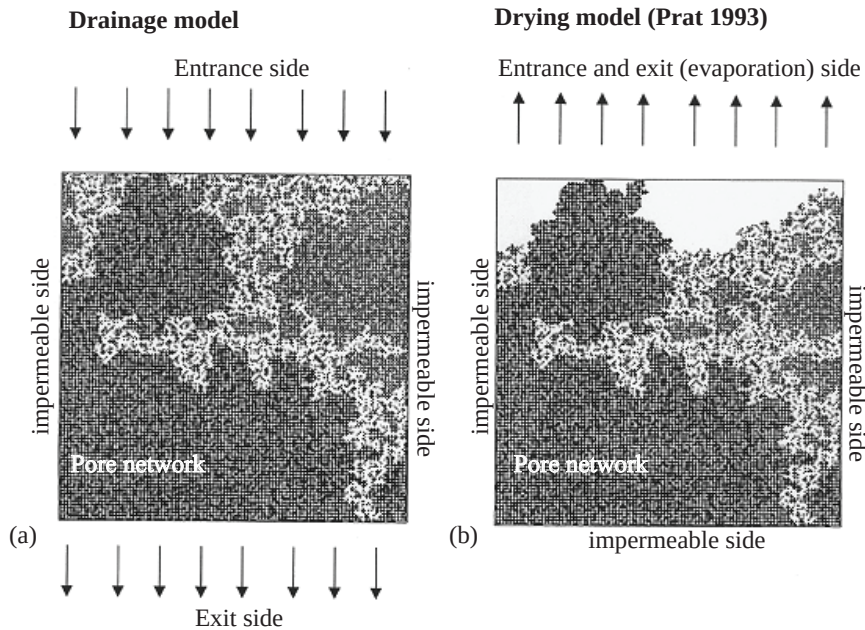


Figure 2.10: Phase distribution of a porous medium predicted by Prat [66] using (a) original invasion percolation algorithm for the drainage process and (b) a modified algorithm for drying [66]. The liquid phase and gas phase (vapour and air) are represented by black and white, respectively. The remaining isolated clusters of liquid in the drainage model (a) are allowed to evaporate in the drying model (b). A dry zone near the top surface can be observed in (b). The non-uniformity of its depth illustrates how wet and dry patches may occur at the surface of a drying porous medium.

Interestingly, non-uniformly distributed wet and dry patches on the surface of the drying material were observed, even for a macroscopically homogenous porous structure. This

phenomenon is consistent with early hypotheses about discontinuous wetting at the drying solid surface [56–58]. A major advantage of the invasion percolation theory is that it leads to realistic phase distributions as a function of drying time unlike some of the conventional models discussed in section 2.2.1 where approximations and assumptions are made about phase distributions.

Pore-scale modelling of drying has developed further over the last decade or so. Huinink et al. [68] studied the effect of thermal gradients on the invasion percolation model for the same geometry as in fig. 2.10b. It was shown that thermal gradients can accelerate the invasion process (or retreating of the drying front) by causing vapour transport deeper within the pore network, beyond the drying front, due to thermally-induced density gradients. Similar results were demonstrated by Plourde and Prat [69] who also demonstrated that the invasion patterns for different thermal gradients become almost identical as the disorder of the pore-network is increased. Metzger and Tsotsas [70] studied in detail the effect of pore size distribution. Their results suggested that a porous medium with a broad distribution of micro and macro pores will have an extended constant-drying rate period.

Yiotis et al. [71] extended the model by Prat [66] to account for the effects of capillary liquid film flows that can exist in throats and pores. For instance, liquid films may develop along the corners of more duct-shaped throats. It was observed that the liquid films which can remain in gas-invaded pores and throats can be transported to the exposed drying surface by strong capillary flows which significantly increase the drying rate and contribute to a longer constant-drying rate period. The influence of film flows and the geometry of the pores and throats on the drying rate was studied in more detail by Prat [23]. Though naturally the pores and throats may have much more complicated shapes, this study suggests that the pore shape can significantly influence the drying behaviour. For instance, it was demonstrated that the contact angle which the meniscus makes in a throat or capillary may have a critical angle (which depends on the shape of the throat cross-section) above which film flows cannot develop. This reduces the capillary transport to the surface significantly and hence the drying rate.

A later experimental study by Chauvet et al. [72] agreed with these predictions using infrared thermography to study the evaporation in a capillary tube. Their study, however, was limited to just a single capillary tube of square cross-section. Hexagonal tubes and

interconnected tubes may yield much different results. They also noted that their model led to a non-zero film thickness at the tip of the receding film where it should tend to zero and remarked that the film flow modelling was crude and would require a full detailed analysis. These limitations highlight the complex and highly variable conditions of moisture transport in a single capillary tube, let alone a porous material.

Further refined pore-network models which adopt the invasion percolation theory have been developed in recent years giving more accurate results [73, 74]. Most recently, the predictions of non-isothermal drying on the invasion percolation process [68, 69] have been confirmed experimentally by Vorhauer et al. [75], using high resolution imaging of a pore network constructed from borosilicate glass with interconnected etched channels. This allowed visualization and measurement of the invasion process. The development of pore-scale models using invasion percolation theory has undoubtedly led to a deeper understanding of the mechanisms of mass transfer which shape the drying process. It has also led to realistic estimations of the evolution of phase distribution in porous media during drying compared to the empirical and volume-averaged assumptions of phase-distribution in the classic drying models. However, the models have rarely been used within the framework of typical industrial drying processes. In other words, there aren't many studies in which the pore-scale theories are incorporated into an experiment involving a macroscopic drying body. There are also few studies on the influence of typical operating conditions - such as air temperature, flow rate and geometry - on the pore-scale moisture transport.

2.3 Local heat and mass transfer in drying

In order to fully understand the simultaneous heat and mass transfer in the drying process, there is a need to determine the temperature and moisture concentration distributions across the solid geometry and at the interface between the drying solid and the drying gas (i.e. where the thermal and concentration boundary layers form). Section 2.2 outlined some of the main heat and mass transfer theories used to characterize the drying process as a function of time. It is clear that the drying rate can be quite dependent on internal heat and mass transfer within the drying solid, particularly during the falling-drying rate period. In practice, however, only the external conditions can really be controlled or optimized. It is therefore also important to understand in detail the external convection of heat and mass in drying as this can lead to the optimization of geometry and flow configuration for different industrial drying technologies.

Furthermore, in a comprehensive review of advanced computational models, Defraeye [27] highlighted the importance of explicitly resolving the surface convective heat transfer coefficients both spatially and temporally, particularly in non-isothermal drying (where significant thermal gradients exist). In such drying problems, the heat transfer coefficient greatly affects the latent heat supply to the evaporating liquid. Also, the boundary layer thermal resistance can be quite dominant compared to the internal/porous material resistance (i.e. the Biot number is much less than 1). This is unlike the situation with mass transfer where internal resistance is often bigger (i.e. the Biot number is closer to or larger than 1).

This section provides a review of some of the existing numerical and experimental drying models that involve spatially and temporally determined heat and mass transfer coefficients.

2.3.1 Numerical modelling of local heat and mass transfer in drying

In recent years there have been many theoretical studies of convective drying involving solutions of the heat and mass transfer evolution in the drying solid and the drying air. This is achieved using boundary-layer equations or Navier-Stokes equations by means of CFD simulations [19, 20, 24, 45, 76–81]. Such models are often described as conjugate or semi-conjugate. In conjugate models, local temperature and moisture fields are solved simultaneously and transiently in both the air and the porous material. Semi-conjugate models involve separate CFD simulations of air when spatial-variations are important or

incorporating dependency on moisture content when time-variations are important. Many of these studies have allowed spatially resolved temperature and moisture concentration fields to be calculated as a function of time. Therefore the local heat and mass transfer behaviour can be characterized with a good degree of accuracy. However, this does require reliable experimental data to be available for validation and for information on the material properties. While experimental data of spatially-averaged drying curves may be widely available for common industrial drying materials, validation of spatial distributions are far less common, particularly temperature distributions.

Most of the theoretical models discussed in section 2.2 can be described as non-conjugate as they use well-established correlations for simple cases (e.g. convection from a flat plate) to apply heat and mass transfer coefficients as boundary conditions. The heat and mass transfer analogy is often used to calculate the unknown coefficient (i.e. h_m is typically found from h using the analogy) in non-conjugate (and some semi-conjugate) models (see for example Kaya et al. [19, 45] or Chandra Mohan and Talukdar [20]).

Murugesan et al. [76] performed a semi-conjugate simulation to investigate the convective drying of a saturated rectangular brick (aspect ratio of 2) sitting on a flat plate. Navier-Stokes equations were used to solve the flow field of the drying gas while energy and mass transport equations were solved for the brick. The resultant transient heat and moisture fluxes were applied as boundary conditions and hence distributions of temperature and moisture content on the drying solid surface and in the flow field could be resolved. The moisture gradients along the surface showed that the leading side of the brick dries at a faster rate compared to the rear. This can also be seen in fig. 2.11 which demonstrates the contours of vapour concentration in the drying air surrounding the concrete brick.

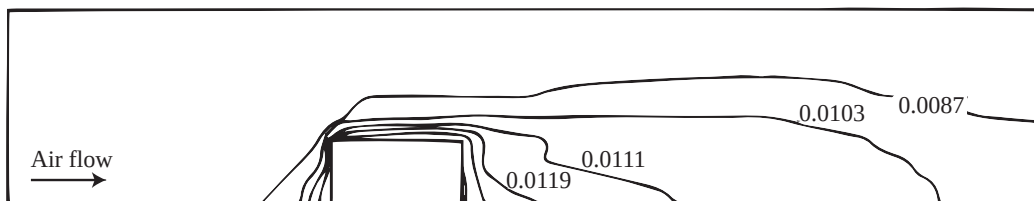


Figure 2.11: Contours of moisture content (kg/kg of dry air) surrounding a rectangular brick solid during the constant-drying rate period [76]

Kaya et al. [19] performed a similar study for a fully exposed rectangular object of various aspect ratios and also resolved the internal temperature and moisture concentration fields inside the solid. Upon solving the temperature fields in the drying gas and at the drying solid surface, the heat and mass transfer coefficient across the rectangular surface was then calculated. Their results showed that the heat and mass transfer coefficients at the leading surface of the rectangular solid were higher than at the other sides for all aspect ratios. De Bonis and Ruocco [77] performed a fully coupled (conjugate) CFD simulation of the convective drying of rectangular slices of carrot in the same flow orientation as Murugesan et al. [76]. These authors also witnessed the largest drying rate at the leading edge and presented spatially resolved vapor fields surrounding the drying solid. These showed significantly higher vapour concentration in the vicinity of the leading edge compared to the top and rear edge. Fig 2.12 below demonstrates the concentration fields surrounding the drying sample with a steeper vapour gradient evident near the leading edge.



Figure 2.12: Vapour concentration (mol/m^3) field surrounding a rectangular drying solid during the falling-drying rate period [77]. The greyscale ranges from 6.05 to 6.20 mol/m^3 .

This higher drying rate at the leading edge was also seen by Lamnatou et al. [24] using a conjugate model for a rectangular moist solid body with an aspect ratio of 5. They also compared this flow configuration to one with the same Reynolds numbers except with a flow divider blocking the front side in order to cause stagnation on the top and bottom surfaces rather than the leading side normal to the flow. This caused more uniform drying and significantly higher Nusselt number on the bottom and top surfaces but the case with the front side exposed displayed a higher overall drying rate. Chandra Mohan and Talukdar [20] performed similar simulations of a rectangular block in a three-dimensional flow, though their model was not fully conjugate. Their results also showed higher heat and mass transfer coefficients at the leading edge and near the corners of the block, which can be seen in the heat transfer coefficient profiles shown in fig. 2.13.

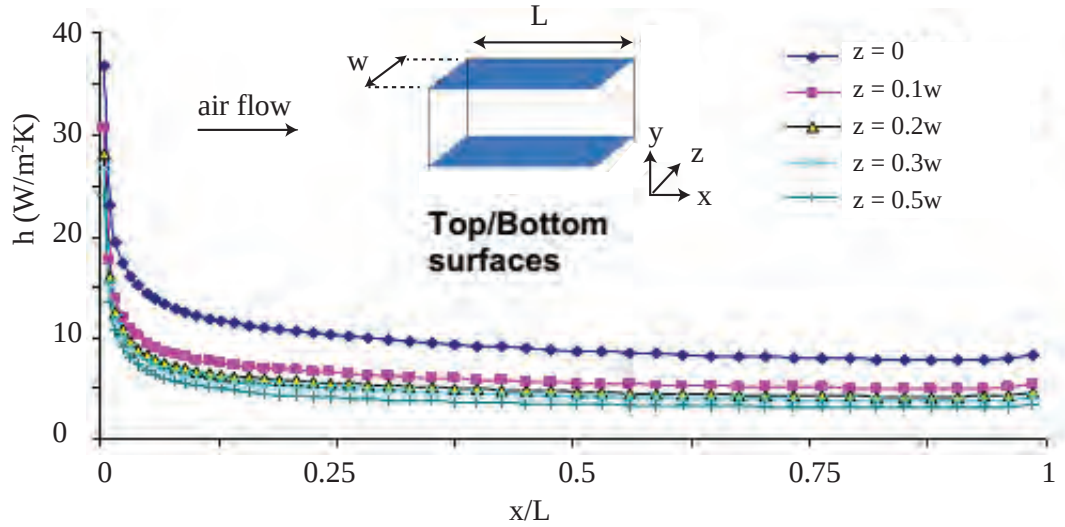


Figure 2.13: Surface heat transfer coefficient profiles of a rectangular drying solid [20].

Defraeye et al. [79] used a conjugate drying simulation to model more closely the spatial heterogeneities of moisture and temperature along a porous flat plate. Their study revealed a reduction in the length of the CRP compared to when constant heat and mass transfer coefficients are applied. This was a result of leading edge effects, meaning the drying rate decreases with distance from the leading edge as shown in fig. 2.14. This demonstrated the two dimensional nature of the drying of a flat porous slab contrary to the one dimensional analysis in older models [15, 55]. Also revealed in fig. 2.14 is a maximum drying rate at the point of transition from CRP to FRP (as the surface begins to dry out). More recently, Lemus-Mondaca et al. [80] have extended fully conjugate drying simulations to a three dimensional rectangular drying solid, demonstrating non-uniform temperature and moisture fields in all directions. Van Belleghem et al. [82] have applied a conjugate model to the drying of a ceramic slab yielding good agreement with experimental data and demonstrating a faster drying rate upstream near the leading edge.

2.3.2 Experimental modelling of local heat and mass transfer in drying

The advantage of many of the theoretical models, discussed in the previous section, is their ability to resolve temperature and moisture concentration fields both in the drying gas, drying solid surface and internally in the drying solid. Achieving this with experimental models

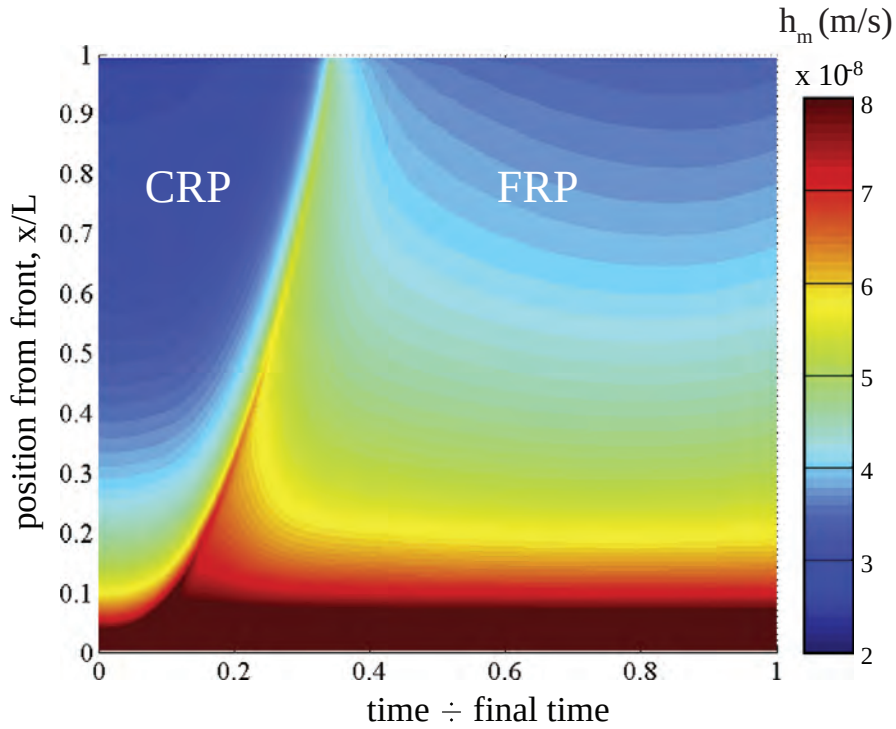


Figure 2.14: Plot of mass transfer coefficient, h_m , at the surface of a flat porous plate versus time and space (position from the front of plate, x , divided by the plate length, L) [79].

is a more difficult task due to the complex nature of the problem. Defraeye [27] remarked that there is a general need for more detailed experimental data for validation of advanced numerical simulations.

Moisture measurement

There have been some recent advances in non-intrusive experimental techniques used for measuring local moisture content in a drying solid which involve the use of x-ray imaging and magnetic resonance imaging (MRI). Fig 2.15 shows examples of some of the moisture profiles measured within drying solids [62, 83, 84].

X-ray attenuation from a drying solid can depend on its moisture content and so this dependence can be used to measure local moisture content. Leonard et al. [83] used an x-ray micro-tomography technique to measure radial moisture profiles of cylindrical moist solids at different stages of overall dryness (drying time). This is shown in fig 2.15a. Their results demonstrated flat moisture gradients in the early stages of drying and progressively steeper

moisture gradients, decreasing from core to surface, as the drying time increased. Aregawi et al. [85] observed similar effects for forced convective drying of apple cylinders (for a Reynolds number of approximately $Re \cong 1000$) using x-ray technology.

Baettig et al. [86] used x-ray measurements to determine moisture profiles along the thickness of boards of wood, demonstrating similar gradients. However, some non-uniformity was present at the interface between the sapwood (the outer layer of the tree trunk), and heartwood (the inner layer of the tree trunk) which interestingly disagreed with a well-established theoretical model used for the same material [18]. X-ray measurements of moisture content in wood drying have also been reported by Alkan et al. [87] and Watanabe et al. [88]. Shokri and Sahimi [89] have recently applied x-ray measurements to pore-scale drying, demonstrating 3D drying fronts (or retreating moisture front) on the pore-scale by taking cross-sectional scans along different depths of a saturated coarse sand column. They demonstrated clear fluctuations of the moisture gradients (from surface to core), due to the irregularities of the pore-scale (and hence moisture transport). A consistent correlation was developed between the fractal distribution of the liquid phase in the porous medium and the water content.

Over the past few decades MRI techniques have also been used for 2D and 3D moisture measurement in a diverse range of drying studies [90–94]. More recently, Koptug et al. [62] performed an in-depth analysis of the evolution of moisture distribution during the drying of porous solids using MRI measurements. The moisture profiles demonstrated a flat profile at the early stages of the drying experiment before evolving into more curved profiles (moisture gradients) in the later stages, as shown in fig 2.15b. They demonstrated the decrease of diffusivity (D) with the decrease of overall moisture content and highlighted how this dependence is closely linked to the pore size distribution. Frías et al. [84] performed MRI experiments demonstrating the evolution of moisture profiles in rice under convective drying, shown in fig 2.15c. Koptug [95] reviewed extensively the application of MRI in drying and sorption processes of porous media. It is clear from the wide range of studies that the technique can provide valuable 2D and 3D models of moisture transfer in solids. The review also highlighted some challenges in quantifying moisture transport on the pore-scale due to resolution and temporal limitations of the technique.

Many of the studies mentioned above using x-ray imaging and MRI have provided

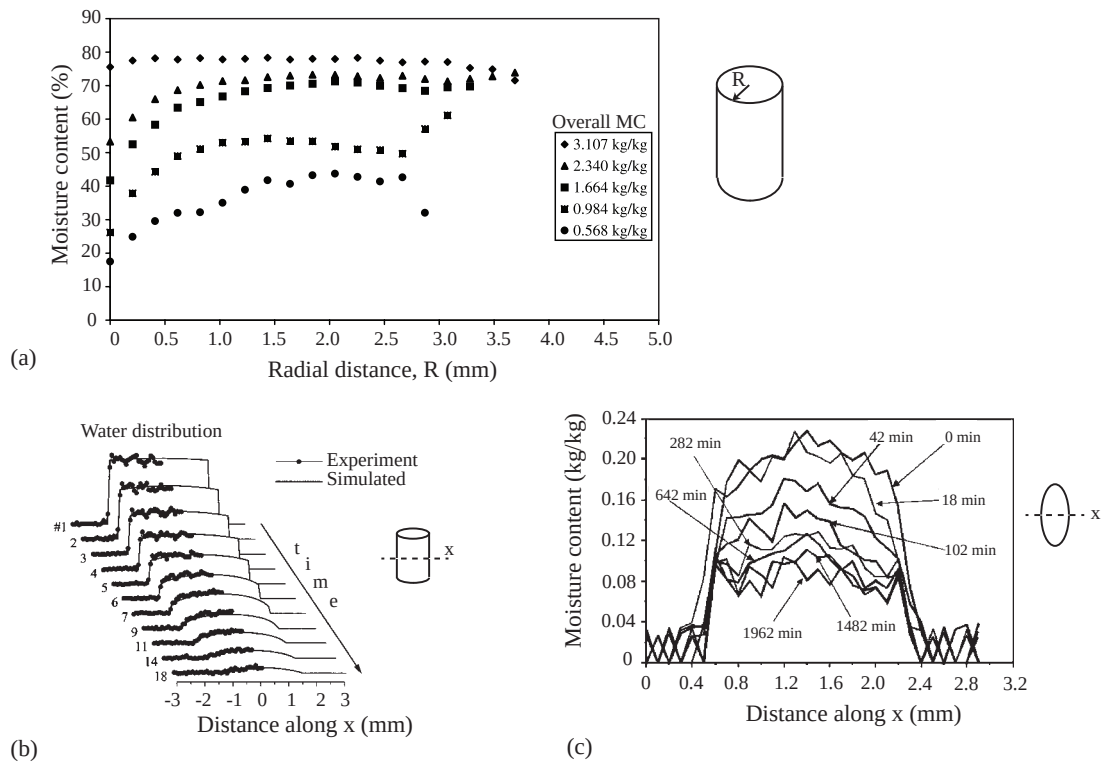


Figure 2.15: Measurements of internal moisture profiles during the drying of (a & b) porous cylinder and (c) rice using (a) X-ray and (b & c) MRI techniques as a function of time or overall moisture content [62, 83, 84]

valuable experimental data which has validated some of the earliest drying experiments and further developed understanding of mass transfer within drying solids. As shown in fig 2.15, most of the moisture profiles demonstrate gradients between the solid core and its surface. This is consistent with the theories of moisture transport that were discussed in section 2.2. For high initial moisture contents (long CRP), the gradients represent transport of free or capillary liquid to the surface from the core for evaporation. For low initial moisture contents (short CRP or FRP only), the gradients may represent a retreating moisture front or diffusion of vapour to the surface where vapour concentration is lower.

Temperature measurement

Non-intrusive 2D and 3D temperature (and hence heat transfer) measurement appears to be less common in drying experiments to-date. Koptuyug [95], remarked that temperature measurement in combination with MRI studies are rare and Defraeye [27] reported on the

2.3. LOCAL HEAT AND MASS TRANSFER IN DRYING

need for more experimental determination of spatially resolved heat transfer coefficients.

There are many drying experiments involving temperature measurements but most of these involve point temperature measurements or small Biot numbers and temperature difference, such that the process can be assumed isothermal. Kondjoyan and Daudin [17] took point temperature measurements along the surface of a saturated porous cylinder at different angular positions from the front stagnation point. This is shown in fig 2.16. They found that while the average surface temperature remained constant during the CRP, the temperature did increase locally from the stagnation point to a maximum near the separation point (approximately 135°) and then decreased again between 135° and 180° . For higher air velocities, the local maximum temperature shifted to 90° from the stagnation point and a lower surface temperature was observed. This was due to radiation heat transfer being more influential at lower velocities. The same study verified the validity of the Chilton-Colburn heat and mass transfer analogy during the CRP. Sun and Marrero [39] also observed this analogy in the drying of cylindrical solids; they noted that the water distribution over the surface was not perfectly uniform and used a term to account for this which yielded more accurate prediction of average surface temperature during the CRP.

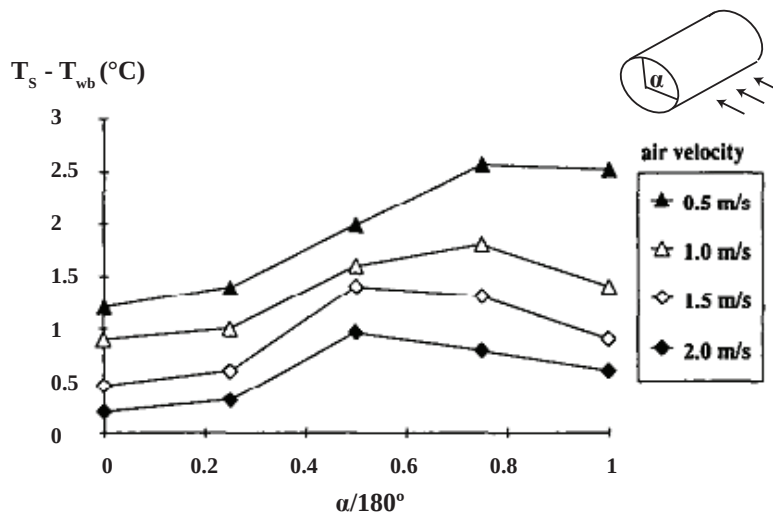


Figure 2.16: Temperature difference between wet bulb and surface temperature (for a dry bulb/air temperature of 40°C at different angles from the stagnation point of a cylindrical drying solid during the CRP [17])

Velić et al. [96] measured instantaneously the surface temperature and average moisture content for forced convective drying of rectangular shaped apple slices. The temperature was measured with thermocouples and the surface temperature was assumed to be uniform. Ruiz-López et al. [63] and Kaya et al. [97] carried out a similar experimental procedure in the study of convective drying of rectangular carrot slices and assumed a uniform temperature in the drying solid. Lu et al. [98] determined, experimentally, the temperature as a function of depth for the drying of a long slab by placing thermocouples at various depths. This experimental model was a one-dimensional study and the surface temperature distribution (and hence the temperature non-uniformity due to leading-edge effects) was not considered. More recently, Van Belleghem et al. [82] validated a conjugate CFD model of the drying of a ceramic slab (configuration shown in fig. 2.17a) with experimental data in which temperature was measured at different points along the surface parallel to the flow. The results, as demonstrated in fig. 2.17b, showed faster temperature evolution upstream near the leading edge. Chandra Mohan and Talukdar [40], Zlatanović et al. [99] and Zhu and Shen [100] all recently developed accurate experimental drying models for rectangular shaped food solids. However, these studies did not consider the influence of surface temperature. There is a large variety of other drying experiments in which temperature or heat transfer evolution has been studied but most of these involve one-dimensional or point measurements of temperature [14, 36, 80, 101–103].

There have been some spatial measurements of temperature during studies of drying using infra-red thermography. However, many of these have been for process-specific rather than fundamental studies [104–109]. That is to say the experiments involve specific drying materials of irregular shape (e.g. apple slices [104], pine veneer [109]) placed in a configuration particular to a drying process (e.g. microwave drying [108], jet impingement drying [105]). Kowalski et al. [110] used infra-red thermography to compare the temperature distributions in a clay cylinder during microwave-assisted drying and natural convective drying. The measurements were not instantaneous or non-intrusive since the cylindrical sample was removed from the oven and promptly sliced in half for imaging along its mid-plane, as shown in fig. 2.18a. Nevertheless, the measurements provided an interesting qualitative comparison of the thermal gradient induced by both drying configurations. Fig. 2.18b shows that for the microwave-assisted drying, the gradient is in the opposite direction (centre-to-surface)

2.3. LOCAL HEAT AND MASS TRANSFER IN DRYING

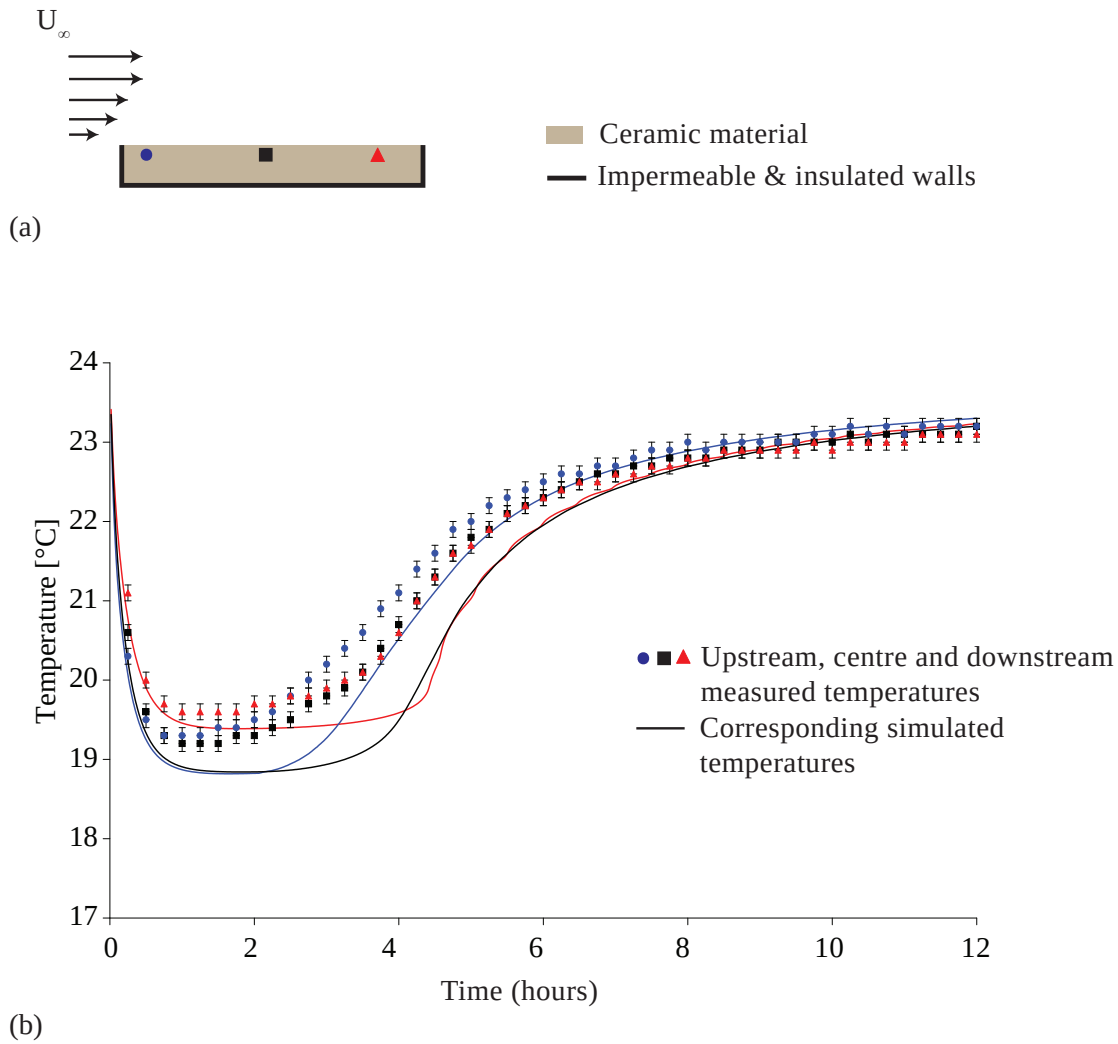


Figure 2.17: Temperature evolution at different points from the leading edge of a drying slab dried at an air or free-stream temperature of 23°C and relative humidity of 40% [82]. For the illustrated drying configuration (a), the evolution of the upstream, centre and downstream temperatures versus drying time are plotted (b).

compared to that of a typical convective drying process (surface-to-centre) shown in fig. 2.18c. This is due to the heat flux acting radially outward from the centre where the microwave interference generates heat. Note one end of the cylinder was insulated with an impermeable wall, which is why the temperature distribution appears asymmetric across the cylinder height, H , in fig. 2.18c.

There have been some recent studies involving infra-red thermography which have focused more fundamentally on the two-dimensional or spatial evolution of heat transfer in drying bodies. Recent work by Aminzadeh and Or [111] studied the effect of heterogeneous

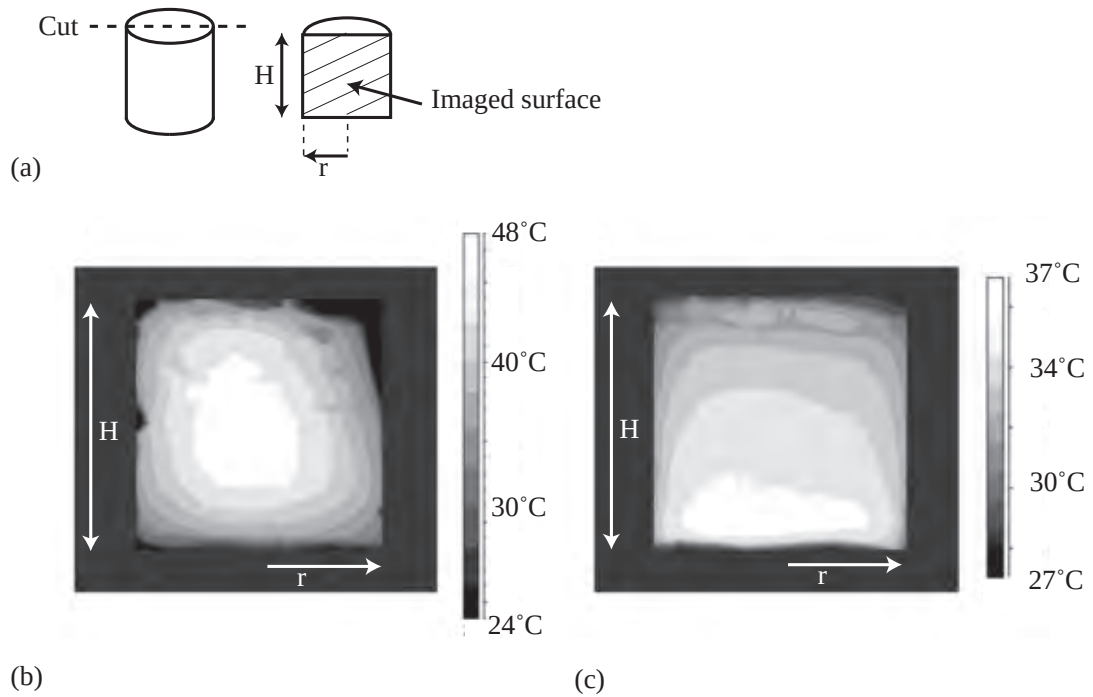


Figure 2.18: Thermal image of temperature distribution in the cross-section - as depicted in (a) - of a clay cylinder due to microwave-assisted natural convective drying (b) and natural convective drying (c) at [110]. The upper face of the cylinder was insulated with an impermeable wall.

porous materials on the temperature distribution and evaporation rate in drying. Thermal imaging was used to measure the spatial and temporal evolution of temperature across the surface of a block of saturated coarse sand containing three cylindrical inclusions of fine sand. This is illustrated in fig. 2.19a. It was found that the finer sand entered the FRP a lot later than the coarser sand. Fig. 2.19b shows how the temperature distributions evolve at different rates across the surface.

Haghighi and Or [26] used high speed thermal imaging in their study of momentum transfer and evaporation at the surface of a thin porous layer. They first developed a theoretical model based on the viscous-sub layer under the 'footprint' of a turbulent eddy (i.e. the thin layer of laminar flow between an individual eddy/vortex and the surface that must hold due to the no slip condition). In this region, an energy balance was used to develop a simplified expression for the local temperature fluctuation due to turbulent flow. High speed thermal images of a thin porous layer of cotton saturated in acetone, where the thermal capacity was low and the evaporation rate was high (i.e. a short evaporation time scale comparable to

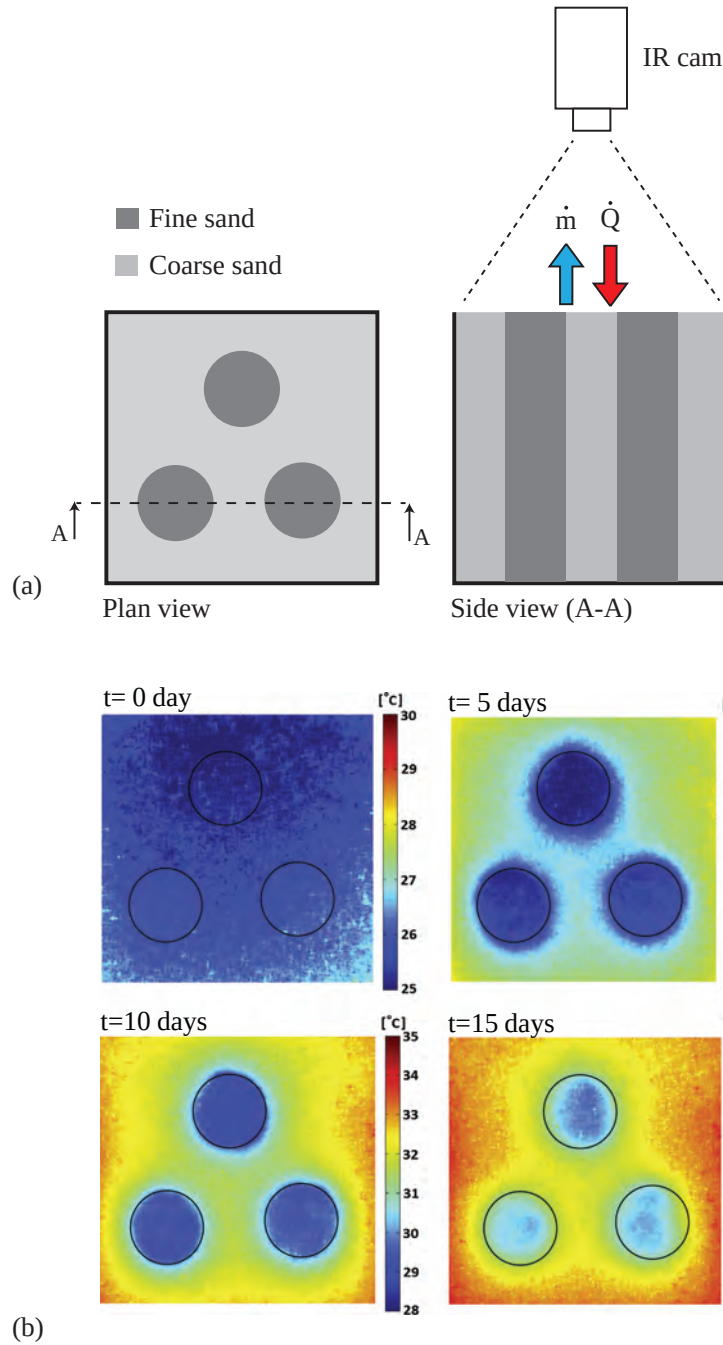


Figure 2.19: Schematic of IR thermography measurement of temperature from the surface of a drying coarse sand with fine sand cylindrical inclusions (a) and plots of the temperature distributions at four different stages of drying time (b) by Aminzadeh and Or [111].

that of the momentum time scale), allowed measurement of the temperature fluctuations. From the temperature residence time and magnitude of the temperature fluctuations, it was possible to infer turbulent flow field statistical information such as the probability distribution

of velocity fluctuations and eddy residence times. This technique also provided the means to infer the evaporation or drying rate from the thermal images.

The interesting insights provided by this recent work suggest that thermal imaging of drying surfaces is a promising tool for understanding the convective exchange of heat and mass in the boundary layers. There are few other experimental studies that focus fundamentally on the spatial, as well as temporal, distribution of heat transfer from drying solids.

2.4 Chapter summary

This chapter has aimed to provide a thorough overview of the main theories and experimental observations of heat and mass transfer in convective drying to date. Drying is a complex process involving simultaneous heat and mass transfer which typically has two main stages:

- A constant-drying rate period marked by a steady drying rate, a constant surface temperature and capillary transport of liquid within the material.
- A falling-drying rate period marked by a decrease in the drying rate, an increase in the surface temperature and the evaporation taking place progressively deeper within the solid.

The earliest theories considered drying as a purely diffusive process. Later this was extended to include capillary transport of liquid in one-dimensional volume-averaging models. These models can provide accurate predictions of the evolution of spatially-averaged moisture content and temperature although are highly dependent on empirical information about the drying material. The falling-drying rate period was modelled more accurately by introducing the concept of a retreating drying front. In recent decades, by examining the wetting of porous materials on a pore-scale, a novel concept called invasion percolation was used to model, more realistically, the transport of vapour through the anisotropic pore structure.

It is important to understand, in detail, the local surface heat and mass transfer in drying since, in practice, only the external conditions can be controlled. The internal transfer mechanisms are governed by the material. In the past decade or so, through the use of CFD simulations, drying models have been extended to predict the two dimensional and three

dimensional spatial distributions of moisture and temperature in drying bodies. Advanced imaging techniques, in recent years, have been used to experimentally determine moisture profiles as a function of drying. However, there are fewer experimental studies in which the evolution of surface temperature or heat transfer distribution is explored. Of those few studies, many are quite process-specific and so there is a need for more generalised studies of the influence of local convective heat transfer.

Typically in drying experiments (or any convective heat transfer experiment), to understand and quantify convective heat transfer, the approach is to measure the surface temperature distribution and calculate local heat transfer coefficients. Measurement of the instantaneous temperature fields surrounding a drying solid would be hugely valuable in the study of local heat and mass transfer phenomena. In particular, measurement of temperature gradients in the thermal boundary layer and study of their evolution during the constant-rate period and falling-rate period of the drying process would provide an in-depth understanding of how local convective heat transfer influences mass transfer during the drying process. Due to the typically intrusive nature of temperature measurement (e.g. thermocouples, temperature probes, etc.) and non-uniform distribution of temperature within the thermal boundary layer, reliable measurement of temperature fields in gases is a difficult task. In the next chapter, the use of a novel laser-imaging technique, laser-induced fluorescence (LIF), for temperature field measurement in a gas is investigated as a potential technique.

Chapter 3

Literature Review: Application of laser-induced fluorescence for fluid temperature field measurements

3.1 Introduction

Laser-induced fluorescence (LIF) is a laser-imaging technique which can be used to study the structure of molecules and for the detection of selected species. It is also used as a fluid flow measurement technique by calibration of the fluorescence signal to fluid quantities such as density, temperature, species concentration and species detection in combustion flows. This makes it a powerful tool for studying engineering problems that involve fluid flow, heat transfer and mass transfer.

The majority of studies involving LIF thermometry in gaseous flows have been performed in intensive heat transfer flows such as combustion engines [28–30], shock tubes [31, 32] and high temperature turbulent jets [33–35]. The objective here is to investigate and evaluate the use of LIF thermometry techniques for studying low temperature convective heat transfer problems and hence to perform measurements of temperature fields in the gas flow of a convective drying experiment.

The principle of LIF is that a chemical species - such as Rhodamine B [112], Acetone [113] or 3 pentatone [114] - can naturally fluoresce upon absorbing high-energy photons

at a resonant frequency (i.e. a laser tuned to the resonant frequency of that molecule). The fluorescence signal can depend on the concentration, pressure and temperature of the species, allowing these quantities to be determined.

Fluorescence on the molecular scale

The fundamentals of the fluorescence process can be understood by considering the molecular scale. More detailed overviews of the spectroscopic process of fluorescence can be found in references [115–117]. When fluorescent tracer molecules absorb photons, they undergo a transition from a ground electronic state to an excited electronic state. This transition between ground and excited state can be understood by first considering the atomic structure. An atom will have different electronic energy levels which represent the amount of energy (eV) an electron has, as it orbits the nucleus. In a ground state, electrons will orbit the nuclei at the lowest energy level and a particular element will have the ‘default’ number of electrons in its first orbit, second orbit and so on. Electrons can escape shorter orbits to longer ones as shown in fig. 3.1. When this occurs, the electrons are in an excited state. An electron can be promoted to a higher energy level or excited state by absorbing a photon of energy, $E = h_{plk}c/\lambda$, where h_{plk} is Planck’s constant, c is the speed of light and λ is the wavelength.

However, the magnitude of the energy levels depends on the atom. For instance, the first four electronic energy levels of a hydrogen atom are precisely -13.6eV , -3.39eV , -1.51eV and -0.85eV . Therefore, in order for electrons to be excited to a higher energy level, the energy of the photon, E , must equal the difference in energy between the two levels, ΔE_{el} . The wavelength at which this will occur can be found by $\lambda = \Delta E_{el}/h_{plk}c$. This corresponds to the resonant wavelength of the tuned laser. Unless the atom becomes ionised - where the electrons receive enough energy (e.g. from a low wavelength source like an X-ray) to escape all orbits or bonds - the electrons will always return to the ground state. When this occurs, a photon will be emitted, such that energy is conserved. This emission of a photon is known as radiative relaxation.

The excitation process shown in fig. 3.1 is a very simple case of an electron in an atom absorbing and emitting a photon at the same energy and hence wavelength. The energy exchange is shown in the simple Jablonski diagram in fig 3.2a. The excitation of molecules,

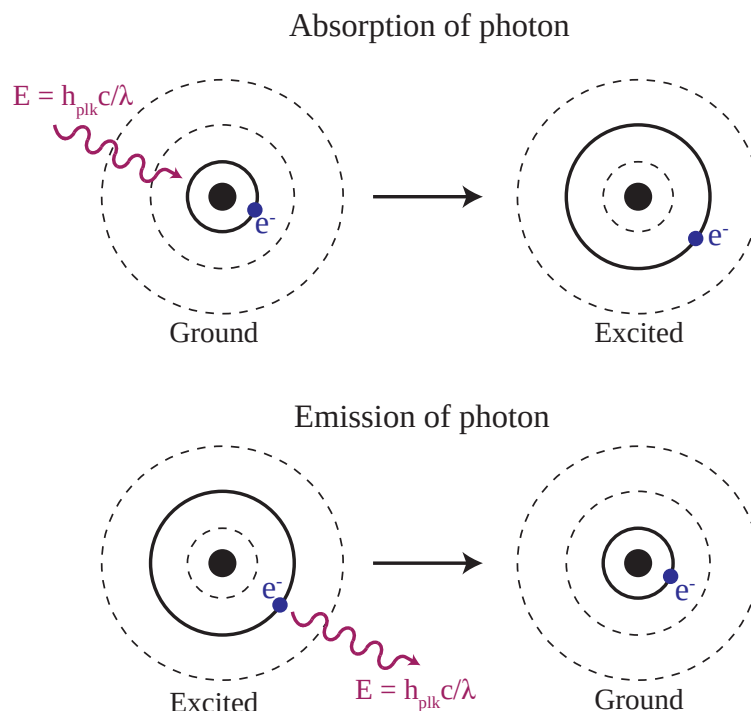


Figure 3.1: Simple illustration of excitation process on the atomic scale. At a resonant wavelength (λ), the energy of the photon, E , is equal to the energy required to promote an electron to the excited state (outer orbit). As the electron undergoes relaxation back to the ground state (inner orbit), a photon of energy, E , is emitted.

however, is generally characterized by the emission of photons at a lower energy or higher wavelength as the molecule relaxes to a ground state. This shift in wavelength is known as the Stokes shift and is due to non-radiative energy transfer during the relaxation period. The molecule not only transfers energy through radiation (fluorescence) but also by non-radiative means. This is shown in fig. 3.2b.

Non-radiative relaxation in molecules is normally due to vibrational and rotational energy exchange. This is represented in fig. 3.2b by the additional energy levels at the excited state. Vibrational and rotational degrees of freedom exist in molecules due to the bonds between atoms. It is often convenient to consider the bonds between adjacent atoms as springs for vibrational energy and as rigid rods for rotational energy, as shown in fig. 3.3. This gives rise to oscillatory and inertial motion of bonded atoms. Therefore the electronic energy levels (or electronic states) are further subdivided into a number of vibrational and rotational energy levels, since some of the energy is converted to vibrational and rotational motion. This is shown in fig. 3.4 which is a plot of the potential energy versus the internuclear distance of the

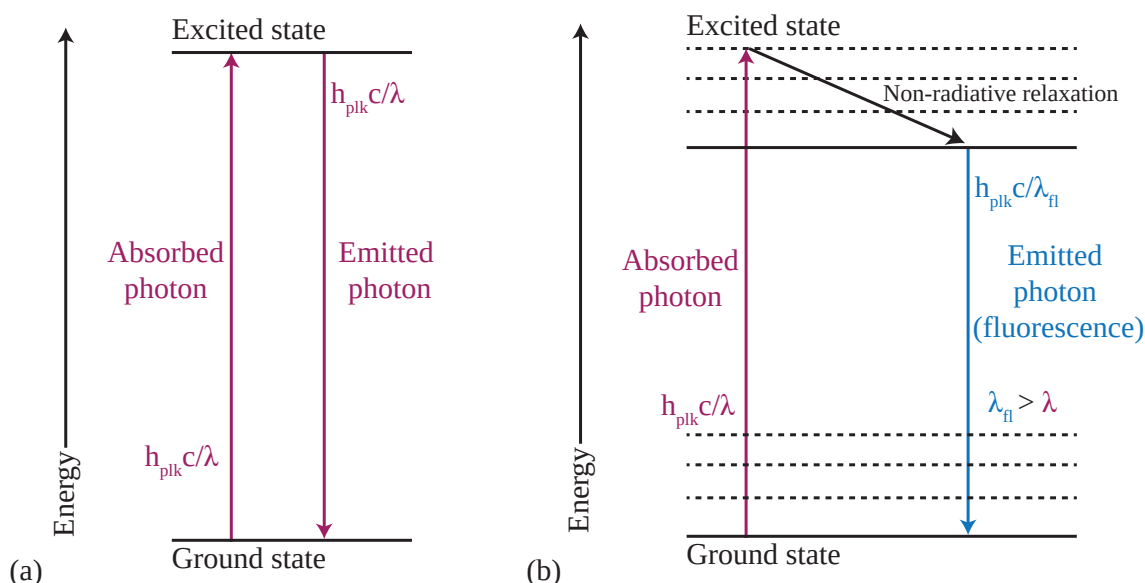


Figure 3.2: Simple Jablonski diagram of the absorption and emission processes for (a) an electron in an atom (e.g. a hydrogen atom) and (b) a fluorescent tracer molecule. The horizontal lines represent the energy levels in the atom or molecule. The additional dashed lines in the Jablonski diagram for a molecule (b) represent the sub-levels of vibrational and rotational energy (this is illustrated in more detail in fig. 3.4)

molecule (i.e. distance between the nuclei of two bonded atoms). A ground and an excited electronic state with vibrational (and rotational) levels are shown here. The shape of the curves resembles that of simple harmonic motion, where the potential energy is highest at the smallest and largest internuclear distances. Since the curves are not symmetric, they actually represent anharmonic motion.

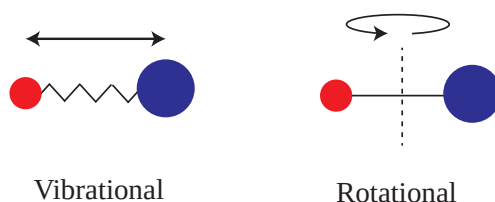


Figure 3.3: Illustration of vibrational and rotational motion between bonded atoms in a molecule.

Typically, electronic energy is significantly larger than vibrational energy and vibrational energy is typically very large compared to rotational energy, as illustrated by the scale of the energy levels in fig. 3.4. This means that higher wavelength electromagnetic radiation

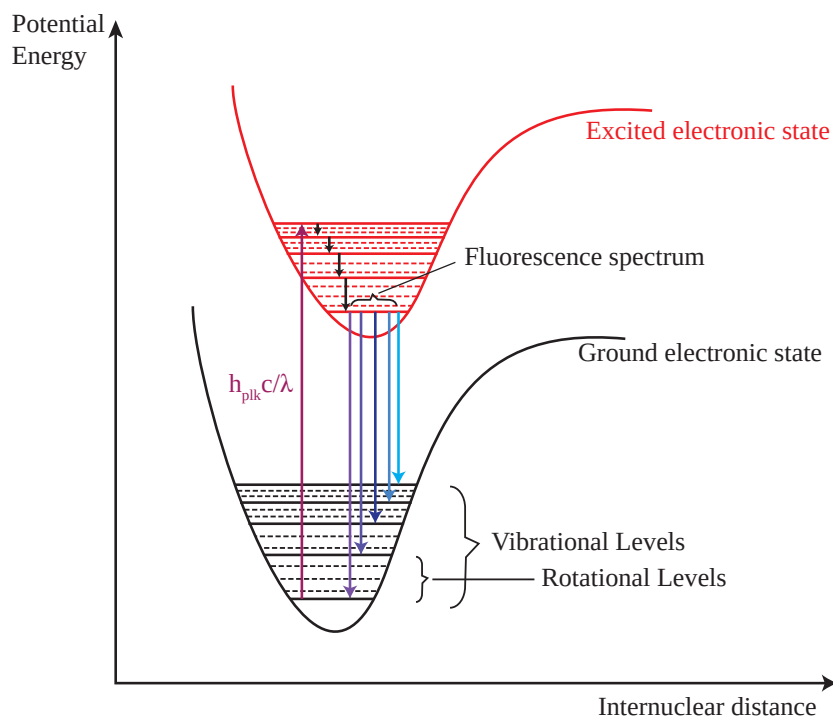


Figure 3.4: Potential energy versus internuclear distance of a molecule. The curves of an electronic ground state and electronic excited state are shown with their corresponding sub-levels of vibrational and rotational energy.

like far infra-red light and microwaves can cause vibrational or rotational excitation without electronic excitation.

When a particular molecule at a ground state absorbs a sufficiently high energy photon, it undergoes electronic excitation. After a molecule is excited to a higher electronic level by a photon, the energy is first exchanged as vibrational and rotational energy (non-radiative transfer). Then once the molecule has reached the lowest vibrational energy level within the excited electronic state, it then relaxes to the ground electronic state in the form of radiative relaxation. The radiative energy transfer takes the form of fluorescence. This involves the emission of a lower energy (or higher wavelength) photon, since some of the excitation energy was converted to vibrational and rotational (non-radiative) energy in the excited state. The excited molecule may return to any of the possible vibrational sub-levels in the ground state. As shown in fig. 3.4, this leads to a fluorescence spectrum since photons with a range of energies and hence wavelengths are emitted, depending on which vibrational level the corresponding molecules return to. The duration of the relaxation period from an excited to a

ground state is known as the fluorescence lifetime. This length of time is typically very large compared to non-radiative relaxation.

Note that there are other competing methods (excluded in fig. 3.4 for simplicity) by which a molecule can relax to the ground state. These include vibrational and rotational energy exchange with the surrounding medium (internal conversion) and collisional quenching in which the excited molecule comes into contact with a different molecule dissolved in the medium. Another method is called intersystem crossing which leads to phosphorescence. More detail on these methods can be found in reference [115]. These methods can be important in the context of LIF since they reduce the intensity of the fluorescence signal.

LIF intensity

During a typical LIF experiment, the main quantity which is measured is the fluorescence intensity, I . Equation 3.1 is a general equation which represents the fluorescence intensity emitted per unit volume, $I(W/m^3)$, such that

$$I = I_0 \Phi C \sigma \gamma \quad (3.1)$$

where I_0 is the intensity of the excitation laser sheet (W/m^2), Φ is the quantum efficiency or yield, defined as the ratio between total energy emitted per quantum and energy absorbed per molecule (ratio of emitted photons to absorbed photons) [32, 33, 112, 118–120]. C is the concentration of the fluorescent chemical (mol/m^3) and σ is its molar absorptivity or absorption cross section (m^2/mol). γ is the total conversion efficiency of the optics used to measure the fluorescence signal.

The practical significance of laser-induced fluorescence, in relation to studying convective heat transfer phenomena, is that by forming a laser sheet (typically from a pulsed laser) and exciting a plane of interest (i.e. a flow field), containing some fluorescent dye or tracer, the corresponding plane of fluorescence can be used as a temperature map or temperature field. This can be achieved since the quantum yield, Φ , and absorption cross section, σ (and hence the signal intensity, I) of some fluorescent dyes or tracers are functions of temperature [32, 33, 118–120]). Using high-sensitivity imaging devices, such as CCD cameras, the fluorescent signal level at any point in the excitation plane (laser-sheet) can be

measured. Hence, by calibrating the local fluorescence signal at all points in the plane to temperature, a temperature field or map can be measured.

Two-dimensional mapping of temperature (or other quantities such as concentration, pressure, density, etc.) with LIF is often referred to as planar laser-induced fluorescence (PLIF). An illustration of the basic experimental principle of PLIF is shown in fig. 3.5 for measuring the 2D temperature fields arising from natural convection of a fluid (seeded with the fluorescent tracer) around a cylindrical solid.

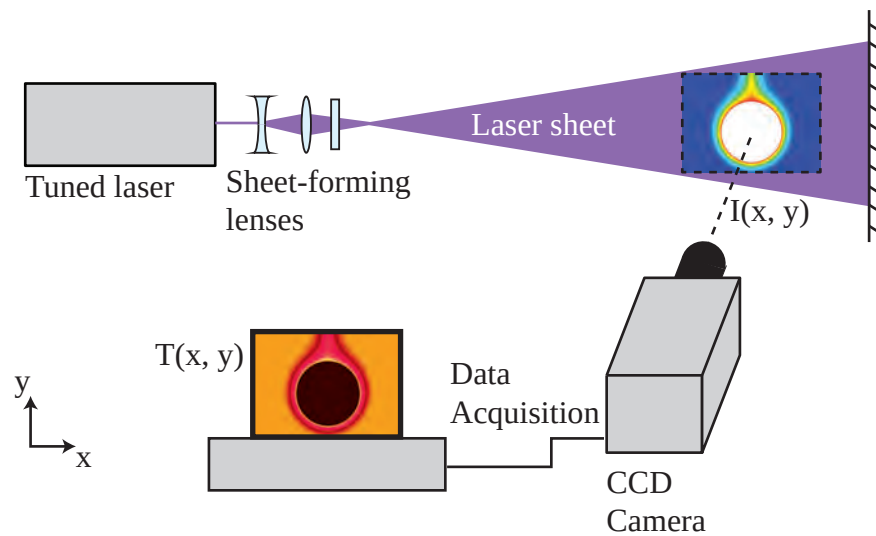


Figure 3.5: Basic experimental principle of planar laser-induced fluorescence (PLIF). The rectangular border within the laser sheet represents a convection flow field seeded with a fluorescent dye.

3.2 Temperature measurement with LIF

It has been demonstrated by many authors [32, 33, 118–121], that for certain tracer materials, the quantum efficiency (Φ) and the absorption cross section (σ) are functions of temperature. This relationship is hypothesised to be due to higher vibrational and collisional quenching of fluorescence (i.e. higher non-radiative relaxation) for higher temperatures [118, 121]. This generally means that for an increasing temperature, there is a reduction in fluorescence intensity.

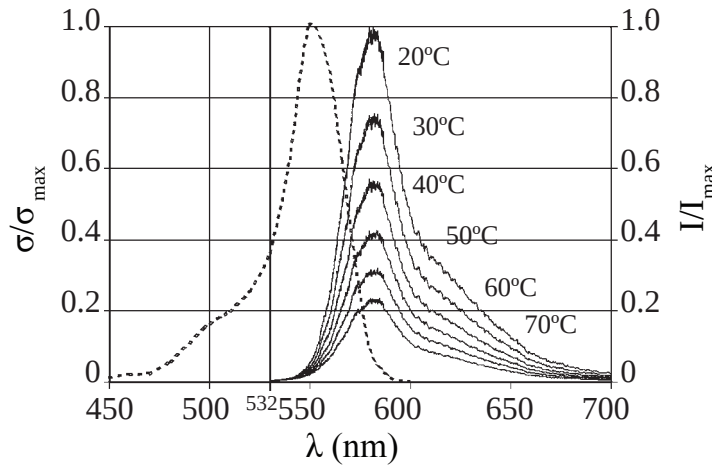


Figure 3.6: Absorption (---) and emission (—) spectra of Rhodamine B for 20, 30, 40, 50, 60 and 70°C excited by an Nd:YAG pulsed laser with a wavelength of 532nm [118]

Fig. 3.6 shows how fluorescence intensity, I , decreases as temperature increases (due to a decrease in quantum yield Φ) for rhodamine B. This tracer is ideally suited for temperature field measurement in liquids since it has a very high quantum yield which is quite sensitive to temperature change and is dissolved easily in water, meaning little or no variation in local concentration of the tracer [112, 118, 119, 122–124]. The absorption spectrum of rhodamine B is also shown in fig. 3.6. Coolen et al. [118] accessed this absorption spectrum using a 532nm Nd:YAG pulsed laser to excite the rhodamine B. A CCD camera, with a long pass filter to cut off any laser scatter, was used to record the emitted fluorescence signal which could be calibrated to temperature. This was used to study natural convection heat transfer phenomena in a cavity containing water. A typical curve of fluorescence intensity (normalized with the value of I at a reference temperature) is shown in fig. 3.7 for rhodamine B [118].

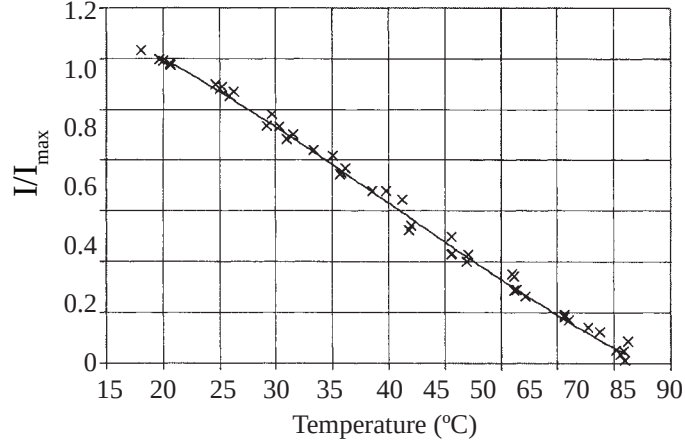


Figure 3.7: Relation between the fluorescence intensity (normalized with maximum $I(T)$) and temperature for rhodamine B [118]

This LIF thermometry technique is often known as the single-colour technique, since only one spectral range of fluorescence or emission spectrum is measured (e.g. $> 550nm$ for Rhodamine B). Another LIF thermometry technique known as the two-colour technique has been applied in many liquid flow experiments to-date [123–130]. The two-colour technique has also been applied successfully in gaseous flows and this is discussed in more detail in section 3.3. It involves two fluorescence signals, I_1 and I_2 , with different spectral ranges which are measured with separate filters. Their respective quantum yields, Φ_1 and Φ_2 , respond differently to temperature changes (i.e. one signal is significantly more temperature sensitive than the other). The ratio of the two signals, $R_I = I_1/I_2$, is independent of concentration and laser intensity, such that

$$R_I = \frac{I_0 C \Phi_1(T) \sigma}{I_0 C \Phi_2(T) \sigma} = \frac{\Phi_1(T)}{\Phi_2(T)} \quad (3.2)$$

Therefore, the signal ratio is solely a function of temperature, eliminating the dependence on dye/tracer concentration, laser pulse energy (which is known to vary from pulse-to-pulse) and laser energy distribution in the laser sheet. This technique is particularly useful in conditions where the concentration of the tracer can vary, such as turbulent fluid flows and compressible fluid flows. The two-colour technique can be performed by using a mixture of two dye materials with different temperature responses. For instance Sakakibara and Adrian [123] and Kim et al. [125] both mixed Rhodamine B with Rhodamine 110 which has

a much lower temperature sensitivity than Rhodamine B. Natrajan and Christensen [130] mixed a similarly temperature insensitive dye, Sulforhodamine-101, with Rhodamine B. This ensured a good dependence of the signal ratio, R_I , of the two fluorescent signals on temperature. Two-colour LIF can also be performed with a single dye by measuring different spectral bands of its fluorescence signal which have different temperature responses [124, 126, 128, 129]. For some dyes, the emission spectrum is found to be significantly less temperature sensitive over certain portions of the fluorescence wavelength range. For example, Lavieille et al. [124] found this to be the case for Rhodamine B from about 520 to 540 nm. A simple illustration of the two-colour technique is given in fig. 3.8 which shows how two emission spectral ranges (two dyes in fig. 3.8a and single dye in fig. 3.8b) with different temperature sensitivities can be used to develop a calibration between signal ratio and temperature.

The current research requires temperature measurements in a gaseous flow field (i.e. a drying gas). Though Rhodamine materials are very useful fluorescent tracers for liquid flow thermometry, it is not feasible to dissolve them into a gas [131]. Nevertheless, PLIF using Rhodamine materials, provides a basis for understanding how the laser-induced fluorescence technique can be used for two-dimensional temperature measurement in a fluid flow. The next section examines the application of LIF temperature measurements in gaseous flows.

3.3 Temperature measurement in gaseous flows

As equation 3.1 shows, in order to reduce fluorescence intensity to being a function of temperature only and hence resolve the temperature field in a flow (i.e. single colour LIF), we require a constant dye concentration or mole fraction. This means that the application of LIF in gaseous flows can be more difficult because of the compressible nature of gases.

In recent decades, the application of LIF to gaseous flow thermometry has been an active area of research. By seeding certain non-reactive materials into gaseous flows, the same techniques as discussed in section 3.2 can be used. Some of the early seeding materials include nitric oxide (NO) [132–135] and naturally occurring species such as iodine (I_2) [136, 137] and oxygen (O_2) [138].

PLIF temperature field measurement using NO as a tracer has been demonstrated by Seitzman et al. [132] by seeding the reactant gas mixture of a methane and air flame. This

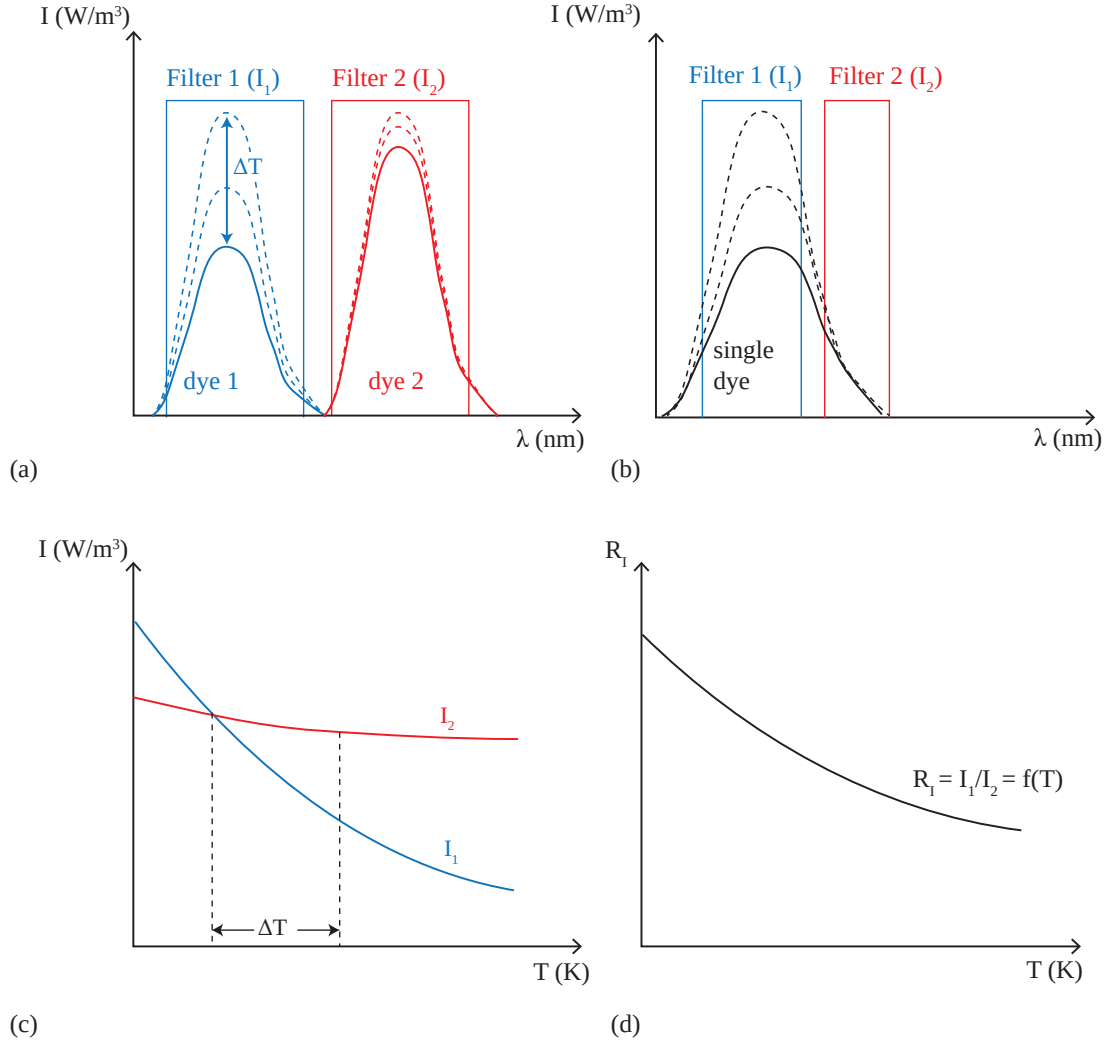


Figure 3.8: Illustration of the two-colour technique using two dyes (a) and a single dye (b). As the temperature is increased by ΔT , the fluorescent signal decreases at different rates for both emission spectral ranges (c). The ratio of these signals is dependent on temperature only (d).

was performed for extremely high temperature ranges and the uncertainty of temperature measurement was particularly high. Lee et al. [138] applied a similar LIF technique by using O_2 as the seeding choice for its stability, low reactivity and for its natural abundance. This method was limited to an accuracy of $\pm 50K$. Studies involving iodine LIF were applied to sub-zero temperature ranges since the fluorescent signal's temperature dependence reduces significantly close to room temperature [136, 137].

A seeding material that has demonstrated good properties as a tracer for the LIF technique is acetone (CH_3COCH_3). Research has shown that it has a wide wavelength range of

3.3. TEMPERATURE MEASUREMENT IN GASEOUS FLOWS

absorption (225 – 320nm), allowing fluorescence to be detected with low-noise CCD non-intensified cameras [120, 139]. Other advantages include low toxicity and low cost, high vapour pressure permitting seeding in high concentrations [139] as well as being less prone to collisional quenching than NO and OH [120]. Note, collisional quenching occurs when the fluorescent molecule is brought to a ground state without fluorescence due to contact or collision with another molecule in the gas. Oxygen (O_2) is an example of such a molecule.

A significant advantage in relation to LIF thermometry is that acetone is reported to have a high temperature dependence which varies with the wavelength of excitation [114, 140]. Thurber et al. [139] used two different lasers (pulsed-dye laser and an excimer laser), with a combination of optics to obtain harmonic beams, in order to generate beams of different excitation wavelengths for fluorescence of acetone. The beams were directed at the same point and their corresponding fluorescence signals, $I(\lambda, T)$, were measured and characterized for different laser/excitation wavelengths, λ_e . The temperature dependency of different excitation wavelengths is illustrated in fig. 3.9a. This is presented as fluorescence intensity, normalized for a reference temperature of 295 K ($I^* = I/I(295K)$), versus temperature. The ratio of the fluorescence intensity resulting from different excitation wavelength combinations was then plotted as a function of temperature. This can be seen in fig. 3.9b.

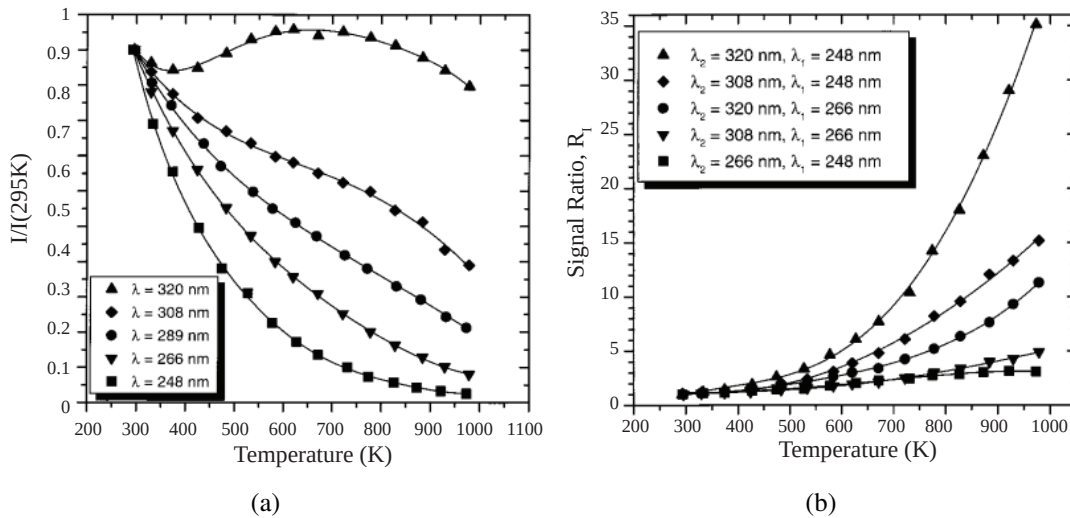


Figure 3.9: Acetone fluorescence as a function of temperature [139]: (a) Fluorescence intensity, normalized to the room-temperature measurement, versus temperature for various excitation wavelengths. (b) Fluorescence signal ratios for different excitation wavelength combinations versus temperature. Each data point represents the ratio of the longer wavelength fluorescence signal at a given temperature to the shorter-wavelength signal at the same temperature.

These authors concluded from their results that there existed two possible methods for temperature measurement of the flow field:

1. Single wavelength method - This single-colour method involves using a single excitation wavelength that is highly sensitive with temperature. This method involves a less complex experimental setup since it requires only a single laser and simpler optical equipment. A single laser, however, requires constant seeding concentration or mole fraction throughout the flow field. An appropriate excitation and fluorescence wavelength with high temperature sensitivity can be chosen from fig 3.9a.
2. Dual wavelength method - This involves using two different excitation wavelengths (e.g. from those shown in fig. 3.9b), i.e. two laser sheets in the same position or plane and at the same instant in time. This is another two-colour technique in which the ratio of the two resulting fluorescence signals (which will also have a different wavelength and hence can be separated with two separate filter-camera combinations) is calculated. This does not require constant concentration or laser pulse intensity since the dependency (in equation 3.1) is canceled out (when the signal intensities at any point in the flow field are divided by one another). It involves a complex experimental setup since two aligned laser sheets are required but it allows temperature distributions in the flow field to be calculated using the relationships from fig. 3.9b.

In a later study by Thurber and Hanson [120], the dual-wavelength method was applied to a heated air-jet and surrounding co-flow seeded with 9% and 3% acetone, respectively. 2D temperature maps of the jet and the co-flow (fig. 3.10) were measured using the dual-wavelength method. A map of the mole fraction or concentration in the jet was also inferred from the resulting temperature images. Reference fluorescence images at different temperatures but constant concentration for one of the excitation wavelengths were used to convert an image of the jet for the same excitation wavelength into a map or image of mole fraction, using tabulated data of that fluorescence signal versus temperature for constant acetone seeding.

In recent years, toluene (C_7H_8) has been investigated as a tracer for gaseous PLIF thermometry since its fluorescent signal has a high quantum efficiency and has been shown

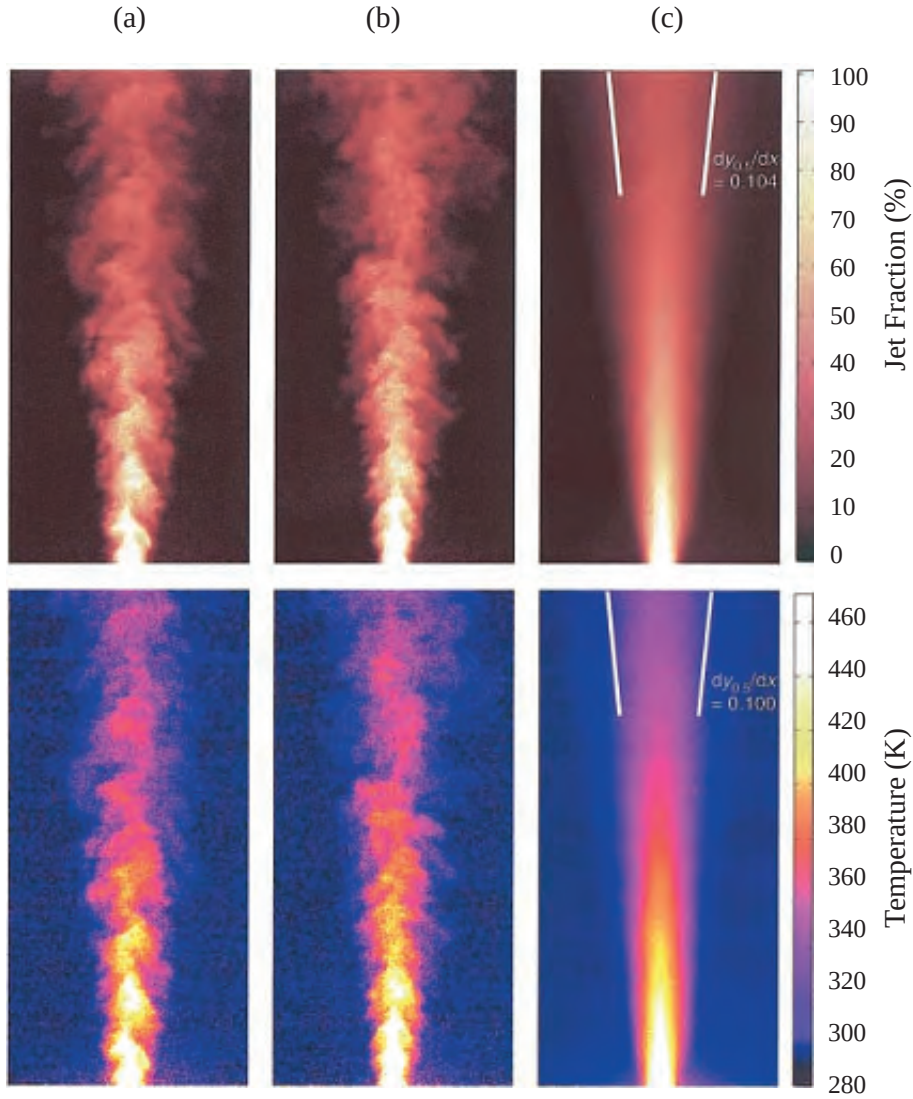


Figure 3.10: Mole fraction (top) and temperature (bottom) distribution images of a turbulent heated air jet ($Re = 5100$) and surrounding co-flow seeded with 3% and 9% acetone, respectively. Images (a) and (b) are instantaneous and (c) is averaged over 100 frames [139]

to be highly sensitive to temperature [141–143]. The fluorescence quantum yield, Φ , for toluene as a function of temperature was measured for both 248 nm and 266 nm excitation in nitrogen by Koch et al. [144] and Koban et al. [141]. Their results indicated a significant temperature dependence. This is shown in fig. 3.11.

Another important property of toluene in relation to temperature field measurements with PLIF is that its fluorescence signal undergoes a ‘red-shift’ with increasing temperature. This means that as the temperature increases, the emission wavelength spectrum of the

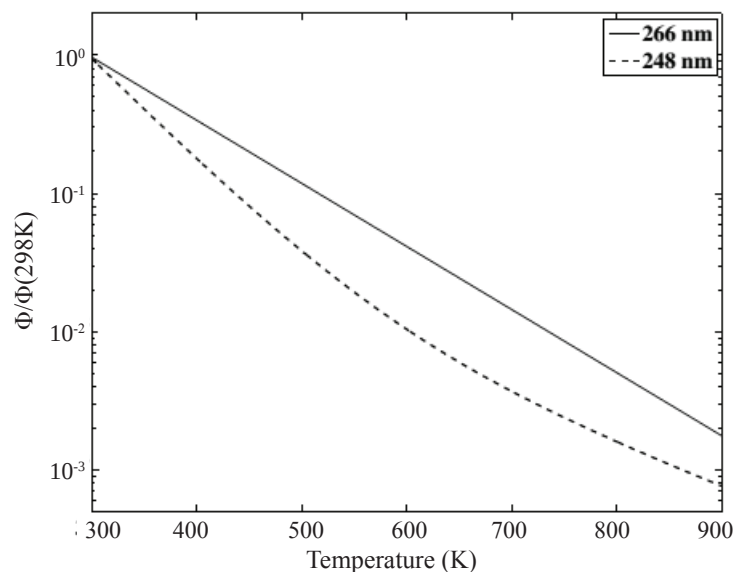


Figure 3.11: The relative (normalized) fluorescence quantum yield, $\Phi(T)/\Phi(298K)$, for toluene mixed in nitrogen as a function of temperature for both 248nm (–) and 266nm (–) excitation [141, 144].

fluorescence signal broadens or shifts in the direction of red light. The fluorescence spectrum for various temperatures is shown in fig. 3.12a from data by Zegers et al. [145] for 248 nm excitation. By normalizing the signal with respect to the peak wavelength as shown in fig. 3.12b, this broadening or shifting of the fluorescence spectrum to larger wavelengths can be illustrated.

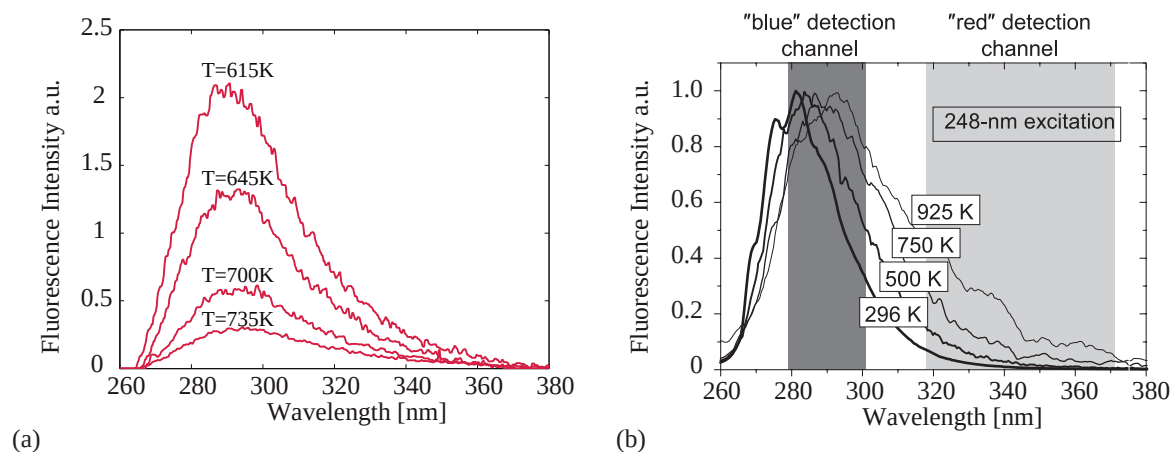


Figure 3.12: Toluene fluorescence spectrum for various temperatures excited at a wavelength of 248nm normalized with (a) temperature [145] and with (b) wavelength [28, 142]. The ‘blue’ and ‘red’ channels (in b) represent the two wavelength ranges which the fluorescence signal was split into by Luong et al. [28] using filters.

This red-shift displayed by toluene makes it convenient for the two-colour thermometry

technique. Thus, the fluorescence signal can be split into a lower wavelength ('blue') signal and a higher wavelength ('red') signal of the same plane (or field of view) and their ratio can be calibrated directly to temperature [33]. Similarly to the dual-wavelength method for LIF thermometry with acetone, the ratio of one fluorescence signal to the other at any point is independent of seeding concentration or laser energy. However, this is achieved without the need for an additional laser sheet or tracer. The local signal or intensity ratio, R_I , which is dependent on temperature, T , across an x-y plane (i.e. the camera image) can be expressed as

$$R_I = \frac{I_{red}(x, y, T)}{I_{blue}(x, y, T)} = \frac{I_0(x, y)C(x, y)\Phi_{red}(T(x, y))\sigma(T(x, y))\gamma_{red}}{I_0(x, y)C(x, y)\Phi_{blue}(T(x, y))\sigma(T(x, y))\gamma_{blue}} \quad (3.3)$$

which can be expressed as the ratio of blue to red also. At any point (x, y) , the concentration, $C(x, y)$, laser intensity, $I_0(x, y)$, and absorption cross-section, $\sigma(T(x, y))$, are equal. Equation 3.3 therefore simplifies to

$$R_I = \gamma_R \frac{\Phi_{red}(T(x, y))}{\Phi_{blue}(T(x, y))} = f(T(x, y)) \quad (3.4)$$

where γ_R is the ratio of the imaging optics efficiencies of both channels, such that $\gamma_R = \gamma_{red}/\gamma_{blue}$. R_I is a direct function of temperature at any point (x, y) . Therefore, as the temperature increases and the fluorescence signal shifts towards higher wavelengths, the intensity ratio, R_I , should increase also. This has been applied successfully by Luong et al. [33] in the measurement of temperature fields across the flow field of a heated turbulent air jet. The same authors later measured the temperature fields inside a combustion chamber (without fuel injection) seeded with a mixture of toluene and air [28]. Their measurements demonstrated a one standard deviation precision of ± 34 K for single-pulse measurements but averaged over several frames this improved to ± 7 K.

Trunk et al. [34] performed high speed (10 kHz laser pulse repetition rate) measurements of temperature fields in the boundary layer of an impinging nitrogen jet seeded with 2.75% toluene. A nitrogen jet was used rather than air, since collisional quenching of oxygen can reduce the fluorescence intensity significantly. Thus, Luong et al. [33] reported that the sensitivity of temperature measurements increases by more than 3 orders of magnitude for toluene mixed in nitrogen compared to toluene mixed in air. Strozzi et al. [29] reported that in the absence of quenching, the fluorescence efficiency of toluene is two orders of magnitude

greater than that of acetone.

There have been several other recent successful applications of LIF temperature field measurement using toluene, for instance in combustion engines [29, 30, 145–147], shock tubes [148] and high temperature jets [149]. Most of these studies deal with experiments involving intensive heat transfer with very large temperature ranges. However, Trunk et al. [34] found that the sensitivity of the technique increases from $\pm 25^{\circ}\text{C}$ to $\pm 8^{\circ}\text{C}$ as the temperature is lowered from 500°C to 100°C , indicating that the use of toluene is better suited for lower temperature scales. Miller et al. [31] demonstrated ways of optimizing the temperature sensitivity, and hence the precision of the technique by selecting appropriate filters for the blue and red channels. In general, by increasing the spectral gap between the two filters/channels the temperature sensitivity is increased, however, the absolute signal (and hence signal-to-noise ratio) is reduced. It was also demonstrated how this trade-off can be optimized through filter selection. Quite recently, Jainski et al. [150] seeded a nitrogen impinging jet with toluene and were able to visualize and measure thermal boundary layers at the impingement surface and compare this with velocity measurements using particle image velocimetry (PIV). They were able to identify a region of hot fluid ejection called a ‘flare’ in the boundary layer and relate it to vorticity effects illustrated by the simultaneous velocity measurements. This is shown in fig. 3.13. The temperature range in the jet flow field of $20 - 227^{\circ}\text{C}$, though still relatively high, is significantly lower than that of most toluene PLIF studies to date. A single shot accuracy of 9 K was reported. For low speed or steady-state problems, averaging over several shots would presumably yield quite good accuracy. This study is further evidence of the suitability of this technique in low temperature environments.

This work suggests that PLIF, particularly with the toluene two-colour method, is a promising tool for temperature measurement in gaseous heat transfer experiments. The literature survey, however, reveals very few studies where the LIF thermometry techniques have been applied to engineering problems with relatively small temperature differences such as those found in the convective drying literature outlined in chapter 2. Table 3.1 lists the majority of the studies in this review which involve tracers suitable for gaseous LIF thermometry and their respective temperature scales. Also included in table 3.1 are some studies using emerging gas flow tracers, anisole and naphthalene [149, 151–153]. These materials, anisole in particular, have recently demonstrated comparable fluorescent

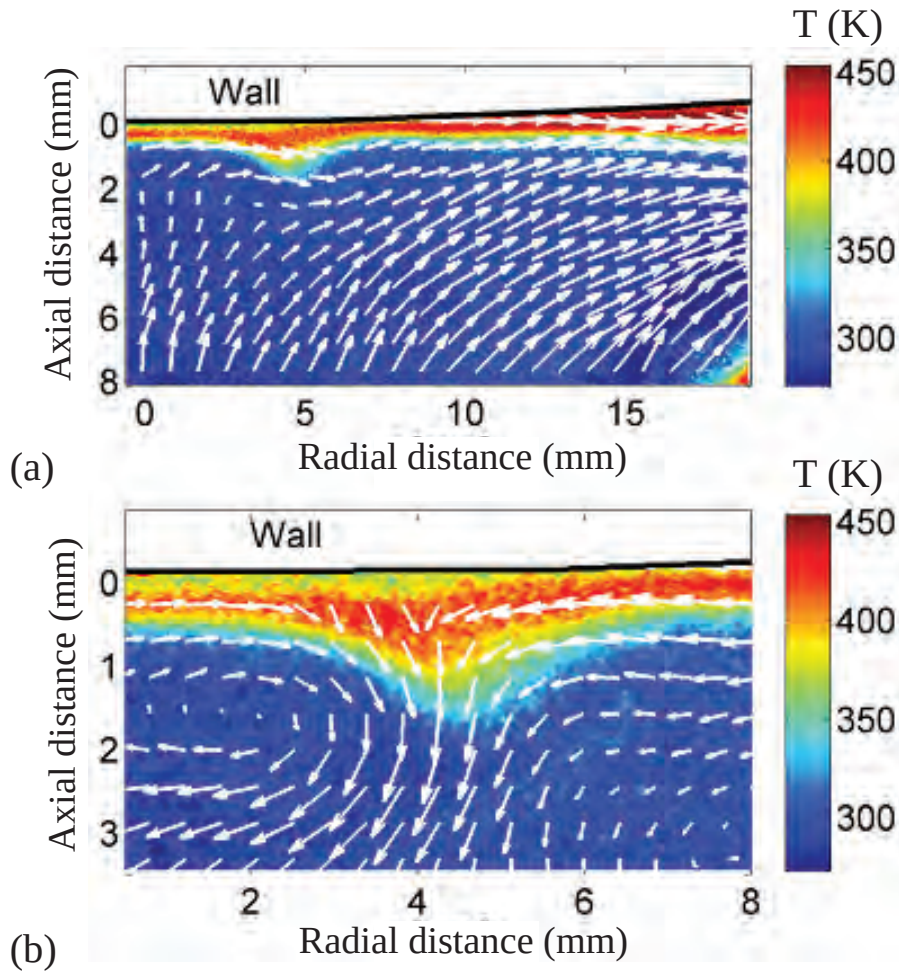


Figure 3.13: Toluene PLIF measurement of (a) instantaneous temperature and velocity vector field at the impingement zone of a nitrogen jet and (b) close-up view with instantaneous velocity vector field minus the time-averaged velocity vector field (in order to illustrate vortices) [150]. The temperature field shows evidence of a ‘flare’ which is a region of hot fluid ejection in the boundary layer for large Reynolds numbers. The combination of PLIF and PIV measurements made it possible to demonstrate that this heat transfer phenomenon was due to vorticity effects.

properties to that of toluene which make them very suitable for LIF thermometry. The lowest temperature range studied, for the relevant LIF tracers, of approximately $10 - 100^{\circ}\text{C}$ was from a study by Kearney and Reyes [131] in which single-colour acetone PLIF was used to study Rayleigh-Bénard convection. In general, table 3.1 suggests that there is a need to extend LIF thermometry to low temperature environments and less intensive heat transfer problems.

Therefore, a key objective of the current study is to evaluate the use of PLIF in a lower temperature range. Toluene was chosen as the fluorescent tracer for its high temperature

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Table 3.1: List of studies using LIF thermometry in gaseous studies and their respective temperature range and interval of calibration.

Author	Tracer(s)	Application(s)	Range*	Interval**
Lee et al. [138]	O ₂	Impinging jet	227 – 827°C	20 – 40°C
Seitzman et al. [132]	NO	Methane flame	227 – 1527°C	N/A
Lee et al. [134]	NO	Laminar jet and supersonic jet	23 – 927°C	100 – 200°C
McMillin et al. [135]	NO	Supersonic jet cross flow	27 – 1227°C	N/A
Tamura et al. [133]	NO	Methane flame	27 – 2727°C	N/A
Thurber et al. [154]	Acetone	Heated cylinder, free air jet	27 – 273°C	40 – 50°C
Thurber et al. [139]	Acetone	Heated flow cell	22 – 727°C	40 – 50°C
Thurber et al. [120]	Acetone	Turbulent jet	24 – 210°C	40 – 50°C
Kearney et al. [131]	Acetone	Rayleigh-Bénard convection	10 – 100°C	20°C
Shi et al. [155]	Acetone	Parallel plate heat exchanger	20 – 200°C	20°C
Löffler et al. [113]	Acetone	High pressure sealed vessel	22 – 455°C	50°C
Koban et al. [141]	Toluene	Heated flow cell	23 – 677°C	20 – 50°C
Koban et al. [143]	Toluene	Heated flow cell	27 – 573°C	75°C
Luong et al. [33]	Toluene	Turbulent jet	27 – 487°C	50 – 100°C
Luong et al. [28]	Toluene	Combustion engine	25 – 427°C	N/A
Strozzi et al. [29]	Toluene	Combustion engine	27 – 677°C	N/A
Yoo et al. [148]	Toluene	Shock tube	27 – 627°C	N/A
Trunk et al. [34]	Toluene	Boundary layer of an impinging jet	100 – 600°C	100°C
Cundy et al. [35]	Toluene	Boundary layer of an impinging jet	100 – 600°C	100°C
Faust et al. [156]	Toluene	Heated flow cell	24 – 652°C	50°C
Miller et al. [31]	Toluene	Expansion tube (supersonic flows)	25 – 677°C	50 – 100°C
Kaiser et al. [30]	Toluene	Combustion engine	27 – 527°C	N/A
Peterson et al. [147]	Toluene	Combustion engine	22 – 300°C	N/A
Jainski et al. [150]	Toluene	Impinging jet	20 – 227°C	90°C
Faust et al. [152]	Napthalene and toluene	Heated flow cell	24 – 602°C	50°C
Faust et al. [153]	Anisole	Heated flow cell	24 – 677°C	30 – 50°C
Tran et al. [151]	Anisole	High pressure sealed vessel	200 – 550°C	50 – 100°C
Faust et al. [149]	Anisole, napthalene, toluene and acetone	Impinging jet	24 – 652°C	25 – 50°C

*Temperature range of experiment

**Approximate interval of calibration for I or R_I vs T . N/A indicates that this information is not available in the literature

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dependence, red-shift with temperature and also its relatively high quantum efficiency. Some additional experiments were performed in the later stages of this project using anisole, since anisole has only recently emerged as another promising tracer. The first step of the current study is to develop a simple calibration experiment to verify if repeatable and precise trends between fluorescence intensity and temperature can be attained. The next step is to apply this technique to study the temperature fields in the gas of a convective drying experiment in order to develop a deeper understanding of the local heat and mass transfer phenomena in drying.

Chapter 4

Experimental Methodology

Given the range of experimental objectives outlined in section 1.2, this study involves three main phases of experimental design:

1. **Convective drying of a porous cube** - Firstly, an experimental model is developed in which the moisture content and surface temperature distribution as a function of drying time is investigated using non-intrusive instantaneous mass measurements and infra-red thermography.
2. **Validation of gaseous LIF thermometry at low temperature ranges** - A LIF calibration experiment is designed which validates and characterizes the trends of toluene fluorescence intensity and signal ratio versus temperature over small temperature scales.
3. **Application of LIF thermometry to convective heat transfer and drying** - Finally, an experiment is designed in which the temperature fields surrounding a drying solid are measured using LIF thermometry, thereby combining the experimental work in phase 1 and 2. This is initially tested with a heated copper cylinder (heat transfer only) before progressing it to a drying solid (heat and mass transfer).

The strategy of this approach is outlined in the flow chart in fig. 4.1. Depending on the feasibility of the LIF thermometry technique (phase 2), the temperature field surrounding a drying body is studied using LIF or the influence of surface convection is studied using an

alternative technique such as schlieren imaging, particle image velocimetry (PIV), etc. Note that since phase 2 was successful, the latter approach was not pursued.

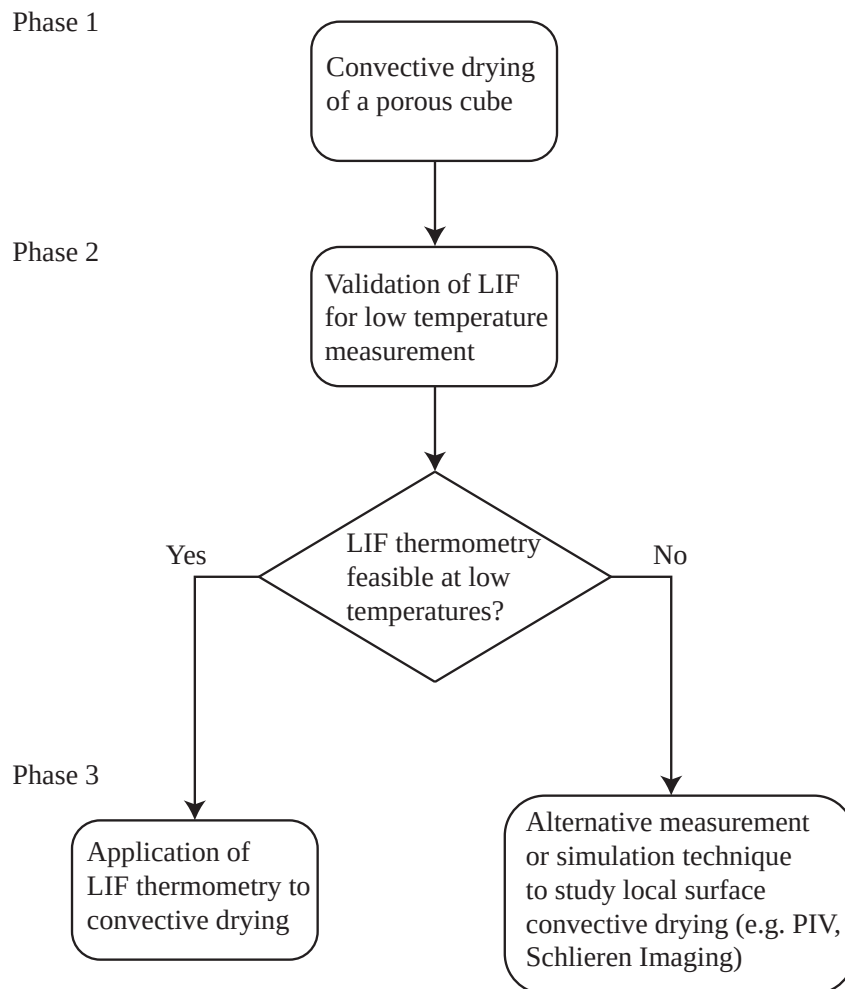


Figure 4.1: Flow chart of experimental strategy, outlining the three main phases of experimental work. Note that since phase 2 was successful, the alternative approach for phase 3 was not pursued.

4.1 Convective drying of a porous cube

4.1.1 Experimental set-up

An experimental testing facility is set up to non-intrusively measure the instantaneous mass of a porous cube and the surface temperature distribution of one of the cube sides. The main components of the experimental setup are illustrated in fig. 4.2. It consists of a drying tunnel with a square cross-sectional area $w \times w = 100 \text{ cm}^2$ through which heated air flows at controlled flow rates and temperatures. The inlet air flow is monitored using a digital flow meter (FMA-1613) and the inlet and exit temperature in the chamber are measured using T-type thermocouples. The inlet temperature measurement is taken after the air passes over the heating element and through the mixing screens, at a sufficient distance such that the flow is fully developed. This provides the free-stream temperature, T_∞ . The inlet air which comes from a compressed air supply has a very low relative humidity typically between 1-3%. This is measured using a relative humidity sensor (Omega HX15) at the same point where T_∞ is measured. It is verified using calculations from the wet-bulb and dry-bulb (free-stream air) temperature [157].

The sample which is dried consists of a capillary porous cube of side length, $L = 1 \text{ cm}$. The material chosen is a phenolic floral foam for its strong absorption and capillarity of water and its high porosity. The sample is initially saturated in a beaker of water at room temperature, giving an initial moisture content of $M_i = 58 \text{ kg/kg dry basis}$ (or $M_i = 98\% \text{ wet basis}$).

4.1.2 Porous cube moisture content and temperature measurements

The porous cube is supported on a thin rigid suspension rod (diameter of 0.5 mm or $0.05L$) made of carbon fibre for its high stiffness and relatively low thermal conductivity. The rod is suspended in the chamber from a balanced lever with a knife-edge pivot to minimize friction. The weight at the point of support of the lever is recorded on a digital mass balance (Kern KB-360-3N) that has a precision of 0.001 g. A data sheet of the balance is provided in Appendix B3. Hence, the instantaneous mass of the sample, $m(t)$ (or moisture content of the sample, $M(t)$), during the drying process can be determined with minimum intrusion.

4.1. CONVECTIVE DRYING OF A POROUS CUBE

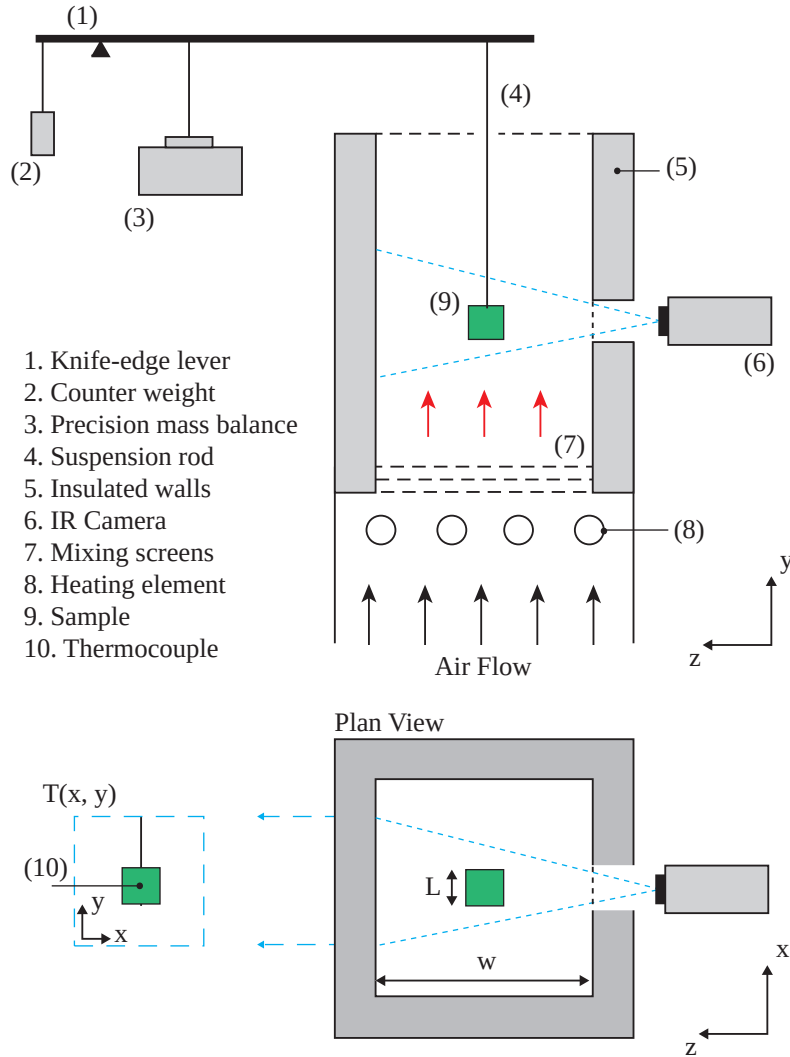


Figure 4.2: Experimental set-up

Maximizing the moment of force ratio between the point of sample suspension and the point of support at the balance maximizes the sensitivity. That is to say that for a change of mass, Δm , of the cube, the digital balance will register a change of $A_{mr}\Delta m$, where A_{mr} is the amplification due to the moment ratio. The use of a counterweight on the opposite side of the lever shifts the offset and maintains measurement within the range of the digital balance (0.001 – 360g).

The instantaneous temperature distribution $T(x, y; t)$ over the surface area (x, y) of one of the porous cube sides (parallel to the direction of air flow), is measured using an infra-red camera (A-40 Flir Thermovision) which views the sample through an infra-red transparent window. An ultra-thin (0.08 mm diameter) T-type thermocouple is also placed at the surface

centre as a reference measurement, T_{tc} . An offset is usually found between the thermocouple measurement and that of the infra-red camera. This is due to the high concentration of vapour near the surface which affects the propagation of infra-red radiation towards the camera and hence its accuracy. A correction factor which is a function of drying time (and hence surface vapour concentration) is used to correct for this effect and is applied to each pixel of the camera. A detailed explanation and derivation of the correction factor is provided in Appendix B1. The spatially-averaged surface temperature, T_s , is found by averaging across a square border (illustrated in figs. 7.1 and 7.2) inside the cube surface perimeter and centered at its origin $(x,y) = (0, 0)$. The average surface temperature can be expressed non-dimensionally as

$$\theta = \frac{T_s - T_{s,i}}{T_\infty - T_{s,i}}. \quad (4.1)$$

and the non-dimensional moisture content can be expressed as

$$\phi = \frac{M - M_e}{M_i - M_e}. \quad (4.2)$$

where the subscripts i and e represent the initial and the equilibrium (or final) values, respectively. Both θ and ϕ are plotted as a function of non-dimensional drying time, $\tau = t/t_e$ in section 7.1.

4.1.3 Heat and mass transfer coefficient estimation

During the constant-drying rate period, the drying rate, $\dot{m}$, is constant with respect to time and the average surface temperature, T_s , is almost constant with respect to time (see fig. 7.3). This quasi-steady state heat and mass transfer process makes it possible to estimate the heat transfer coefficient, h , from an energy balance between the rate of convective heat transfer to the cube and the rate of latent heat release from evaporation (as in equation 2.1 in section 2.1.3). In most experiments, there is a slight increase in the average surface temperature over the duration of the CRP and hence some sensible heat transfer occurs, raising the temperature of the cube. The rate of sensible heat transfer, $\dot{Q}_{st}$, is calculated as the change in internal energy over time during the CRP, such that

$$\dot{Q}_{st} = \frac{d(mu)}{dt} = u(t) \frac{dm}{dt} + m(t) \frac{du}{dt} \quad (4.3)$$

where u is the internal energy of the cube (kJ/kg). The internal energy is changing slightly with time, such that, $u = u(t)$. Note, the internal energy, u , can be taken as that of water only since the ratio of water to solid (M) is significantly large throughout the CRP. Another form of heat transfer that must be accounted for is that due to radiation from the surroundings inside the chamber, $\dot{Q}_{rad}$. This is estimated as

$$\dot{Q}_{rad} = \varepsilon \sigma_b A_s (T_\infty^4 - T_s^4) \quad (4.4)$$

where ε is the emissivity, σ_b is the Stefan-Boltzmann constant and A_s is the surface area of the cube. The surface of the surrounding walls is made of black card with an emissivity determined to be 0.94. The emissivity of the phenolic foam which the drying cube is made of is determined to be 0.93.

A small amount of conduction from the carbon fibre suspension rod into the cube, $\dot{Q}_{cond,rod}$, is also accounted for. This can be determined from the IR images which allow the temperature gradient, dT/dy , to be estimated as the change in temperature per length of rod, $\Delta T/\Delta y$. $\dot{Q}_{cond,rod}$ is therefore calculated from Fourier's law as

$$\dot{Q}_{cond,rod} = -k_{cf} A_{rod} \frac{dT}{dy} \cong -k_{cf} A_{rod} \frac{\Delta T}{\Delta y} \quad (4.5)$$

where k_{cf} is the thermal conductivity of carbon fibre and A_{rod} is the cross-sectional area of the rod. $\dot{Q}_{cond,rod}$ is typically small compared to the other heat transfer terms, and becomes almost negligible for higher Reynolds numbers where the rate of convective heat transfer is much larger. The rate of conductive heat transfer from the thermocouple to the cube, $\dot{Q}_{cond,tc}$, is calculated in the same way. This term is found to be negligible compared to the other heat transfer terms since the thermocouple's metallic wire section is only 0.08 mm in diameter and its insulation material is made from Perfluoroalkoxy (PFA) which has a very low thermal conductivity.

The heat transfer coefficient, h , during the CRP is hence calculated as

$$h = \frac{\dot{Q}_{conv}}{A_s(T_\infty - T_s)} = \frac{\dot{Q}_{lat} + \dot{Q}_{st} - \dot{Q}_{rad} - \dot{Q}_{cond,rod} - \dot{Q}_{cond,tc}}{A_s(T_\infty - T_s)} \quad \text{for} \quad t_{crp} \leq t \leq t_{frp} \quad (4.6)$$

and the rate of latent heat transfer for evaporation, $\dot{Q}_{lat}$, can be found by

$$\dot{Q}_{lat} = \dot{m}h_{lat} \quad (4.7)$$

where h_{lat} is the enthalpy of vaporization of water and the drying rate, $\dot{m}$ (or mass flow rate of the vapour from the cube surface (kg/s)), is determined from the slope of mass versus time (dm/dt) during the CRP. The ratio of the radiative heat transfer rate, $\dot{Q}_{rad}$ to latent heat transfer, $\dot{Q}_{lat}$, ranges from 25% to 12% as the Reynolds number is increased from 180 to 1000, respectively. Therefore, at lower Reynolds numbers, the radiation heat transfer can be quite significant.

The mass transfer coefficient, h_m , during the CRP can be calculated from the expression for mass convection

$$h_m = \frac{\dot{m}}{A_s(\rho_{vs} - \rho_{v\infty})} \quad (4.8)$$

where $\rho_{v\infty}$, the vapour mass concentration in the free-stream air, is determined from the relative humidity measurements, RH_∞ , of the inlet air (measured after the mixing screens and heating elements) in the drying chamber. The water vapour pressure, $P_{v,\infty}$, can be calculated from $P_{v,\infty} = RH_\infty \times P_{sat}(T_\infty)$, where $P_{sat}(T_\infty)$ is the saturation pressure at the free-stream temperature. Hence, $\rho_{v\infty}$ can be calculated from the ideal gas relationship $\rho_{v\infty} = P_{v,\infty}/R_v T_\infty$, where R_v is the gas constant for water vapour. Since the vapour pressures are sufficiently low, the ideal gas law can be applied with negligible error [37]. The vapour mass concentration at the surface, ρ_{vs} , corresponds to a relative humidity of 100% assuming the surface is saturated during the CRP.

4.2. INVESTIGATION OF GASEOUS LIF THERMOMETRY AT LOW TEMPERATURE RANGES

The Nusselt number, Nu , and Sherwood number, Sh , are calculated as

$$Nu = hL/k \quad (4.9)$$

and

$$Sh = h_m L / D_v \quad (4.10)$$

where the thermal conductivity, k , is evaluated at the film temperature between the air and cube surface and L is the characteristic length of the cube. D_v is the vapor diffusivity which is calculated from an empirical expression for diffusivity of water vapor in air, D_{H_2O-air} , proposed by Marrero and Mason [158] such that

$$D_v = D_{H_2O-air} = 1.87 \times 10^{-10} \left(\frac{T_\infty^{2.072}}{P} \right). \quad (4.11)$$

where P is the absolute pressure of the drying air in atm. The Nusselt number and Sherwood number represent the ratios of heat convection to conduction and mass convection to diffusivity, respectively. This allows for a generalized comparison of the heat and mass transfer as a function of Reynolds number and free-stream temperature. It also provides insight into whether the heat convection and mass (or vapour) convection from the cube surface are analogous processes.

4.2 Investigation of gaseous LIF thermometry at low temperature ranges

A laser-imaging experimental setup has been designed to investigate the use of laser-induced fluorescence for temperature measurements in a gas, using toluene as the tracer material. Thus, a 266 nm laser is used to excite controlled concentrations of toluene at controlled temperatures in a specifically designed calibration test section or sealed container (which is adapted later as the test section of the forced convective drying LIF experiment). The signal intensity, I , and signal ratio, R_I , of the resulting fluorescence signal is measured, by means of a CCD camera and an intensified CCD camera, for a small range of gas temperatures

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(20 – 60°C). All of the optical components of the experimental setup are shown in fig. 4.3 and a photograph of the setup is shown in fig. 4.4. The three main components of the experimental setup are the UV laser setup (#1-6 in fig. 4.3), the test section (#7 in fig. 4.3) and the fluorescence imaging setup (#8-13 in fig. 4.3). Each of these components are discussed in detail here.

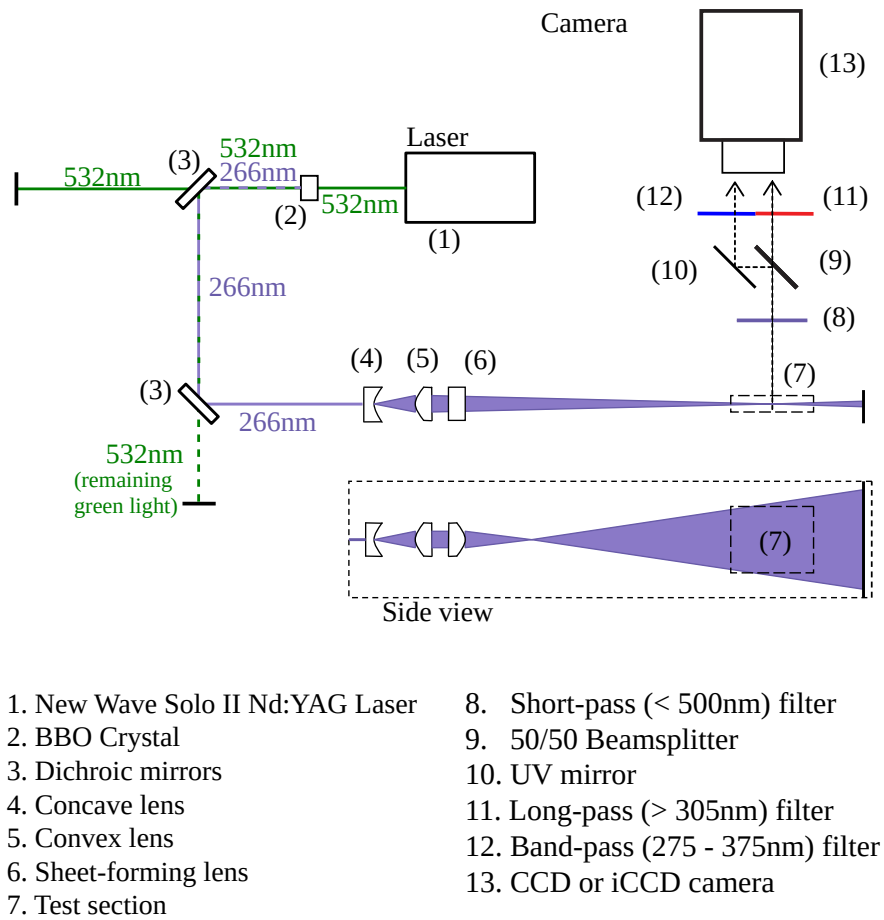


Figure 4.3: Schematic of LIF experimental setup with list of components

4.2.1 UV laser setup

Excitation of toluene occurs within an absorption spectrum of approximately 240 – 270 nm as demonstrated in figure 4.5 which shows the absorption cross-section as a function of wavelength at room temperature and atmospheric pressure. The two most commonly available laser wavelengths in this range are 248 nm and 266 nm which have been used in most toluene LIF experimental studies to date. In this study a 266 nm laser sheet is used. An

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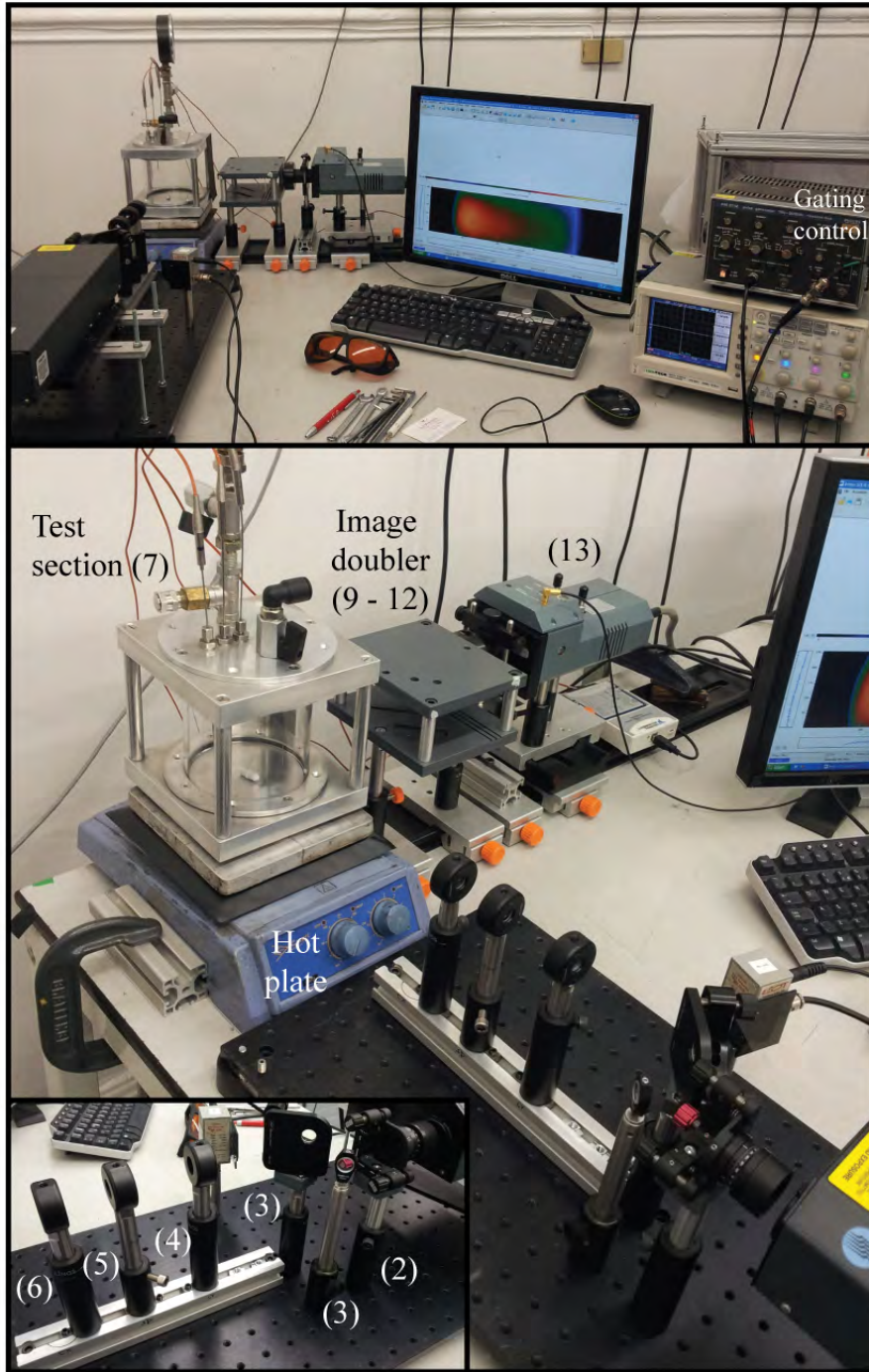


Figure 4.4: Photographs of LIF experimental setup. Numbers correspond to components in fig 4.3

important advantage of excitation at 266 nm is that its fluorescence quantum yield, Φ , is up to 3.4 times greater than that of 248 nm near room temperature and atmospheric pressure [32].

The available New Wave Solo II Nd:YAG pulsed laser has a pulse energy of 30 mJ and a wavelength of 532 nm and so it needs to be frequency doubled to a 266 nm ultra-violet laser

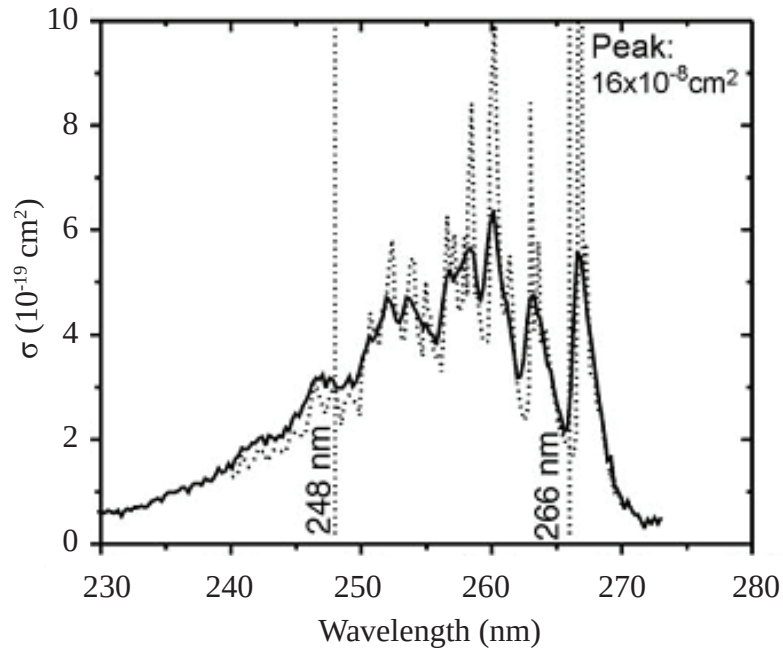


Figure 4.5: Absorption spectrum of toluene from spectrometer measurements with a resolution of (dotted) 0.003 nm [159] and (solid line) 0.25 nm [141]

pulse. Frequency doubling is achieved using a Beta Barium Borate (BBO) crystal (supplied by Photonics Solutions). This process, also known as second harmonic generation (SHG), is a non-linear optical process. A brief summary of the process is given here. A more detailed explanation of the process can be found in reference [160].

Second harmonic generation

When light travels through a medium linearly, polarization is directly proportional to the electric field. When light of substantial intensity or wave amplitude (e.g. a pulsed laser) passes through a non-linear medium (e.g. a BBO crystal), this relationship can become non-linear. The electric field of the light can be expressed as a Taylor expansion consisting of linear, quadratic, cubic components. It can be demonstrated through Taylor summation that if the angular frequency, ω , of a sinusoidal electric field (i.e. that of a polarized light wave) undergoes a non-linear response, the higher order terms correspond to harmonic angular frequencies. For example, the second order term is a function of 2ω . Ultimately, this means that light leaving the non-linear medium can contain harmonic frequencies or wavelengths of the initial light source.

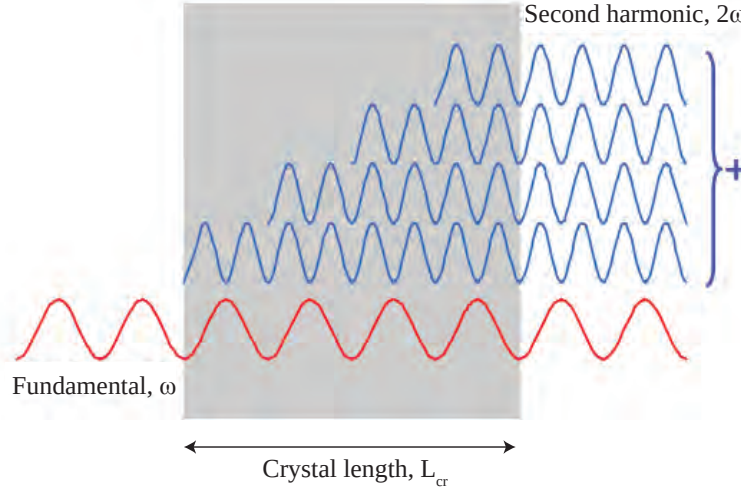


Figure 4.6: Simple illustration of SHG in a non-linear crystal of length, L_{cr} . The secondary harmonics are phase-matched so that their peak amplitudes interfere constructively to form the maximum output laser beam intensity. Sketch taken from [161].

In general, the first harmonic or fundamental frequency (ω) of the output light dominates and the light intensity becomes negligible for increasing harmonics. However, the intensity of the lower harmonics can be made sufficient by using a high energy laser pulse and making the propagation length of the non-linear crystal as long as possible. Another crucial requirement is for the harmonics to be phase-matched so that the parallel light waves can constructively interact to produce maximum output laser energy. This is illustrated in fig. 4.6. For phase-matching to occur, the refractive indices, n_r , for the fundamental and second harmonic must be equal, such that:

$$n_r(\omega) = n_r(2\omega) \quad (4.12)$$

Since refractive indices vary with wavelength, this is impossible in most materials. BBO crystal, however, is a birefringent material. This means the refractive index depends on the polarization also. One plane or axis of polarization results in an ordinary refractive index, n_{ro} , which remains constant and the orthogonal axis of polarization results in an extraordinary refractive index, n_{re} , which depends on the angle of incidence of the input light or laser pulse, such that

$$n_{re}(\omega) = n_{ro}(2\omega) \quad (4.13)$$

for a critical angle of incidence, α_{crit} . In this study, this critical angle is fine-tuned by placing the BBO crystal in a kinematic and rotatable mount (see #2 in fig. 4.4). The 532 nm laser beam passes through the BBO crystal leaving as 532 nm and 266 nm harmonics. A dichroic mirror (CVI Melles Griot BSR-21-1260M) transmits the 532 nm beam to a beam dump and reflects the output 266 nm harmonic beam which is then passed through a second dichroic mirror (Thorlabs HBSY0534) to further separate the 266 nm beam. The output laser power and hence the average pulse energy of the 266 nm beam was determined, using a power meter (Coherent PM10). The average pulse energy was found to be between 3.2 and 3.6 mJ/pulse, varying slightly with each setup of the mount. This value was carefully monitored throughout the experiment to ensure a constant laser power.

Finally, to form the laser sheet a concave lens, convex lens and sheet-forming lens, all made of 266 nm-transparent fused silica, are placed in series to form a narrow (about 1 mm thick) focused laser sheet (i.e. plane or field of excitation).

4.2.2 Test section

A detailed sketch of the test section (#7 in fig. 4.3 and fig. 4.4) is shown in fig. 4.7. It consists of a UV transparent (> 250 nm) fused quartz tube (inner and outer diameters of 100 mm and 105 mm, respectively) sealed in an aluminum frame. The laser sheet passes through the central axis of the tube and a camera is then aligned normal to the laser sheet and measures the fluorescence signal, $I(x,y)$, in the plane of the laser sheet. The use of a quartz tube (instead of rectangular quartz windows) simplifies the design and manufacture of the test section and provides greater optical accessibility for the laser and camera.

The test section holds a gas mixture (800 mL) of toluene (C_7H_8) and nitrogen (N_2) which are fed in through valves at the top of the test section. Carefully-measured amounts of liquid toluene are injected with a micro-litre gas-tight syringe and allowed to evaporate and diffuse. The toluene is mixed with nitrogen rather than air, since the presence of oxygen molecules causes collisional quenching of the fluorescence signal, which has been shown to reduce the signal amplitude significantly and hence the temperature sensitivity by about 3 orders of magnitude [33, 34, 141, 144]. The test section is sealed using viton rubber o-rings as shown in section B of fig. 4.7. A heat flux is applied to the aluminum base (using a hot plate) in

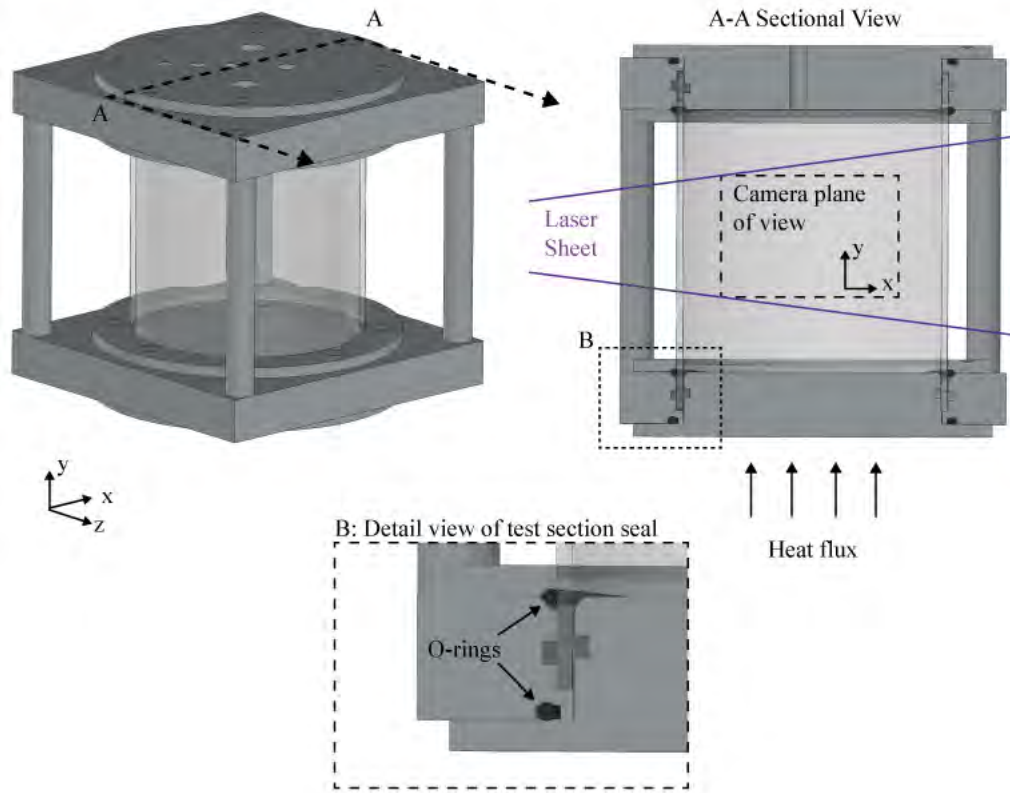


Figure 4.7: Sketch of LIF test section. Left hand side shows an isometric view of the vessel or chamber and right hand side shows a sectional view through the centre of the test section (A-A). Section B shows a detail view of the o-ring seal of the vessel.

order to heat the gas mixture to a desired temperature, which is monitored at different points in the container by means of T-type thermocouples to ensure good temperature uniformity across the test section volume. A magnetic stirrer is also placed at the base of the sealed container to prevent non-uniformities in toluene concentration as well as in temperature. The top of the test section also contains a pressure regulator and relief valve (which can be seen in fig. 4.4). The test section is designed such that the aluminium caps that form the bottom and top seal can be removed easily, making it adaptable to a wind tunnel for the forced convective drying experiment. This is discussed in more detail in section 4.3.

4.2.3 Fluorescence imaging setup

A CCD camera (PCO Sensicam) with a sensitivity range extending down to 275 nm was initially used to measure the fluorescence signal, $I(x, y)$, across the plane of fluorescence

4.2. INVESTIGATION OF GASEOUS LIF THERMOMETRY AT LOW TEMPERATURE RANGES

(i.e. where the laser sheet passes through the test section). Regular CCD cameras are typically not sensitive enough to measure effectively the toluene fluorescence signal, due to its low wavelength and amplitude. For instance, the quantum efficiency of this CCD sensor is between 0 and 10% between 275 and 300 nm, meaning it has particularly low sensitivity within the fluorescence emission spectrum of toluene. The CCD camera used here recorded signals less than 2.5% of its upper (saturation) limit. Theoretical estimations of the fluorescence signal using the CCD camera yielded a similar value (see Appendix C1 for detail). Intensified CCD (iCCD) cameras which amplify the signal and are more sensitive at lower wavelengths are typically used in studies of this nature [31, 33, 34, 141, 144]. After performing the preliminary experiment with the regular CCD camera, iCCD cameras were used (Andor DH520-18T and an iStar 334T) to provide more sensitive measurements of the fluorescence signal. Therefore, the temperature sensitivity of signal measurement could be compared for both CCD and iCCD cameras.

By making use of the red-shift toluene displays with increased temperature (discussed in section 3.3), the signal ratio can be calibrated directly to temperature (two-colour approach). An image doubling system (#8-12 in fig. 4.3 and fig. 4.4) was designed in order to separate the fluorescence signal into two channels, a ‘blue’ (lower wavelength) channel and a ‘red’ (higher wavelength) channel; these signals could then be measured on a single camera rather than on two separate cameras. A detailed schematic is shown in fig. 4.8. This design is similar to the image doubling technique used by Luong et al. [28]. The image doubling system consists of a 50/50 beam splitter (Edmund Optics NT48-193) which is positioned at a 45° angle to the camera and laser sheet. The beam splitter transmits half of the light signal and reflects the other half. The transmitted signal (red channel) passes through a long pass (> 305 nm) filter (CVI Melles Griot WG-305) and is imaged through a UV fused silica lens combination (Edmund Optics NT84-339 and NT48-322) onto the camera sensor. The reflected portion (blue channel) is reflected back towards the camera by means of a UV mirror (Edmund Optics NT43-876) and passes through a UV band pass 275 – 375 nm filter (Thorlabs FGUV11S). An additional short pass (< 500 nm) filter (Edmund Optics NT64-663) is used to suppress any 532 nm laser scatter. This allows two identical instantaneous images of the fluorescence signal, but within different wavelength ranges, to be captured on the same camera. Dividing the two images provides the local signal ratio, $R_I(x,y)$, which is a function

4.2. INVESTIGATION OF GASEOUS LIF THERMOMETRY AT LOW TEMPERATURE RANGES

of temperature, $T(x,y)$, and independent of both local concentration and laser pulse energy (see equation 3.4 in section 3.3).

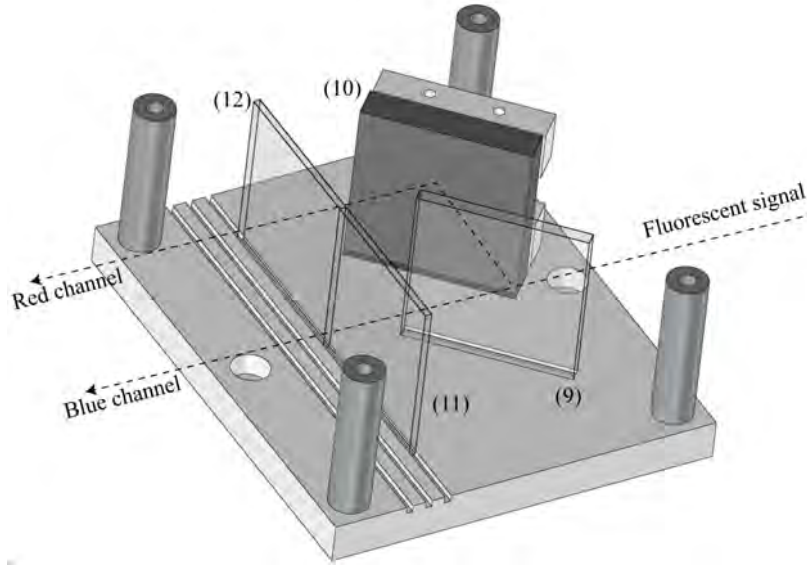


Figure 4.8: 3D schematic of image doubling system. The top platform is suppressed to show the optical components. Numbers correspond to components in fig 4.3

The gas mixture in the vessel is heated to different temperatures. Once the temperature in the vessel has stabilized to a uniform temperature across the plane of view, $I(x,y)$ is measured and hence the average value of I across the plane can be calibrated to the average temperature across the plane. This value is then averaged over 500 frames for the CCD camera due to its low sensitivity. The iCCD camera detected the signal more effectively giving higher measured signals. Therefore a lower number of averages of between 10-30 was used. For this calibration experiment, in some cases the average signal ratio, R_I , is measured by simply taking separate measurements with the blue and red channel filters placed in front of the camera rather than using the image doubling system. This is acceptable for simply finding an average value of R_I , since the temperature within the vessel is stabilized and uniform and a sufficient number of averages are taken. When applied to heat transfer experiments in which the temperature field is unsteady, the image doubling system is required since the local instantaneous signal ratio, $R_I(x,y;t)$, is required (i.e. an identical image is required through both channels).

The CCD camera was triggered and synchronized to the laser pulse using a La Vision programmable timing unit (PTU). For the iCCD camera, a TTL signal was taken from the

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laser Q-switch and sent to a delay generator to control the intensifier gating before triggering the camera. Using a photo-diode, the laser pulse signal could be viewed on an oscilloscope along with the gating of the intensifier, in order to ensure correct synchronization. The effective lifetime of the toluene fluorescence signal (i.e. the time taken for most of the fluorescence signal to decay or for the molecules to reach a grounded state) is reported to be 50 ns at a temperature of 298 K and a pressure of 1 bar of pressure and decreases slightly with temperature [156]. It is also found to increase slightly with pressure [162]. To ensure the signal was captured correctly, three different gate settings are examined, as shown in fig. 4.9. The first setting is identical to that of the minimum CCD exposure time (1000 ns gate width), while the second setting has a short gate (100 ns) and a delay from the laser Q-switch in order to just cut off the laser pulse and only capture the fluorescence signal. The third setting has an intermediate gate time (250 ns) and a short delay from the Q-switch, which ensured all of the fluorescence signal was captured but minimized any laser scatter or measurement of noise. The intermediate gate was used for most experiments.

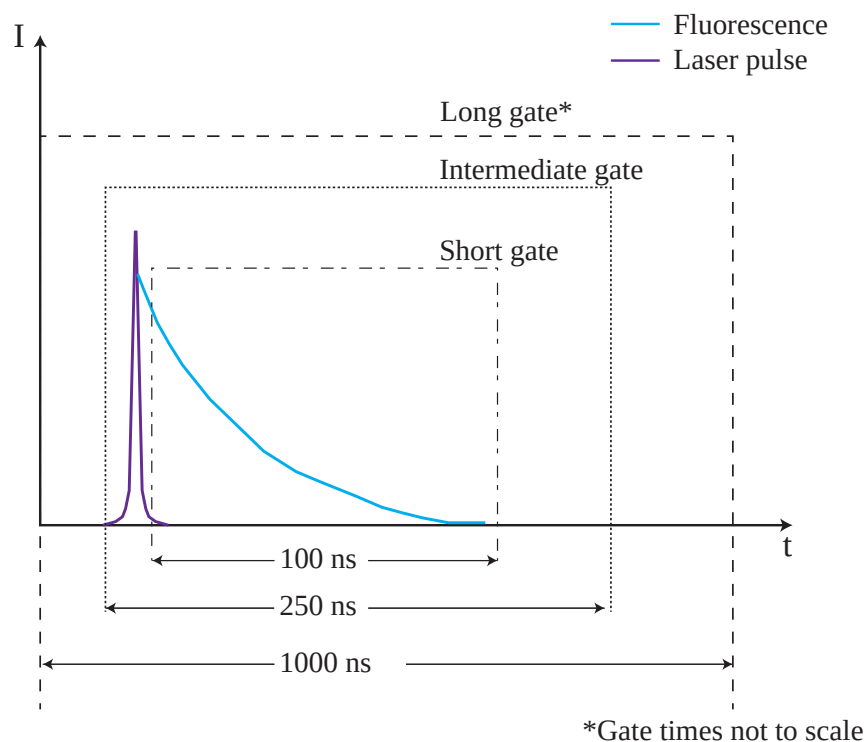


Figure 4.9: Diagram of gate settings. The beginning ($t = 0$) marks the time of the laser Q-switch triggering. This is followed by a short delay (≈ 30 ns) before the pulse is fired. The longest gate setting is the same as that used for the CCD camera.

4.3 Application of LIF thermometry to convective heat transfer and drying

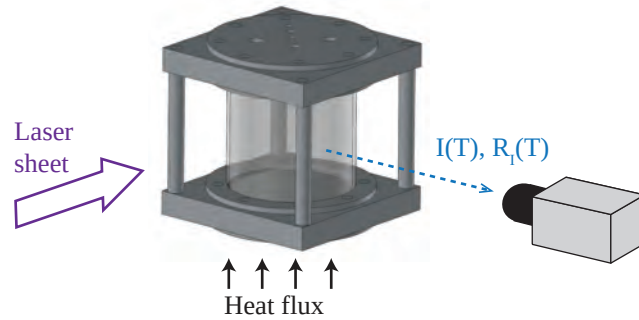
Upon completing a successful investigation of the relationship between toluene fluorescence intensity and temperature, the final phase of experimentation was designed. This consists of the same optical setup as described in section 4.2 but with a test section designed for convective heat transfer and drying experiments. Thus, the test section described in section 4.2.2 is adapted to a drying experiment by removing the top and bottom caps and connecting a sealed heated flow loop or tunnel as shown in the schematic in fig. 4.10 and the photographs in figs. 4.11 and 4.12.

The heated flow tunnel consists of u-pvc piping with the same inner diameter as the fused quartz tube. The top cap from the test section is placed at the top of the three-way junction directly above the test section. Nitrogen and toluene are fed through the same inlets as before (see section 4.2.2) to produce a gas mixture. The gas is circulated with an in-line fan at controlled flow rates. A fixed fan blade is placed in front of the fan in order to remove the swirl from the flow. Before reaching the test section, the gas is forced through a honeycomb flow straightener to produce a uniform velocity profile. The flow rate of the gas, generated by the fan, is controlled by a variable power supply. The velocity, V_{∞} , and its maximum fluctuation, at the centre of the pipe cross-section was measured with a velocity probe (Testo 425) for different values of fan power. A basic calibration was made between velocity, V_{∞} , and fan power. The velocity ranges from 0 – 2 m/s, with an error of $+/- 12 - 20\%$ respectively.

The gas flows across a filament heating element with adjustable power and is heated to a desired free-stream temperature, T_{∞} . Dessicant is placed in the tunnel to minimize the relative humidity of the gas. The relative humidity inside the tunnel is measured using a precision relative humidity sensor (Omega HX15) at one of the corners of the tunnel (see the bottom photograph in fig. 4.12). T-type thermocouples were placed in the caps of the tunnel to measure the free-stream gas temperature.

A porous cube made of phenolic floral foam, the same material used in the earlier drying experiment (see section 4.1), is suspended in the test section on a long narrow stainless steel syringe needle. The needle has an outer and inner diameter of 1 mm and 0.7 mm, respectively

Validation and Calibration of LIF thermometry (phase 2):



Application to convective drying (phase 3):

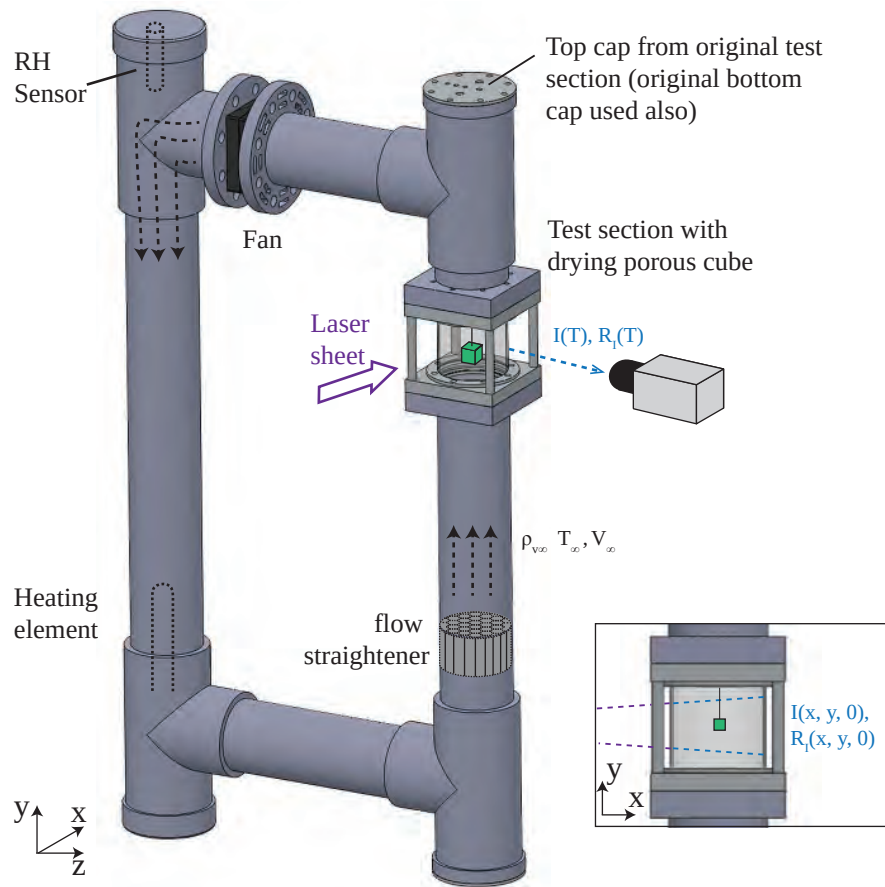


Figure 4.10: Experimental setup for studying the temperature fields surrounding a drying porous cube with LIF. The top and bottom caps of the test section (#7 in fig. 4.3 and fig. 4.4) are removed and a heated flow tunnel is connected. A porous cube is suspended in the test section and the plane parallel to its central axis $(x, y, 0)$ is excited with the laser sheet, as detailed in the inset.

and is connected through the top cap with a sealed connection. This allows water to be supplied to the porous cube from outside the sealed tunnel. Once the free-stream temperature has stabilized inside the tunnel, a measured quantity of water is injected which is very quickly absorbed by the foam and the drying experiment begins. The fluorescence signal in the plane

4.3. APPLICATION OF LIF THERMOMETRY TO CONVECTIVE HEAT TRANSFER AND DRYING

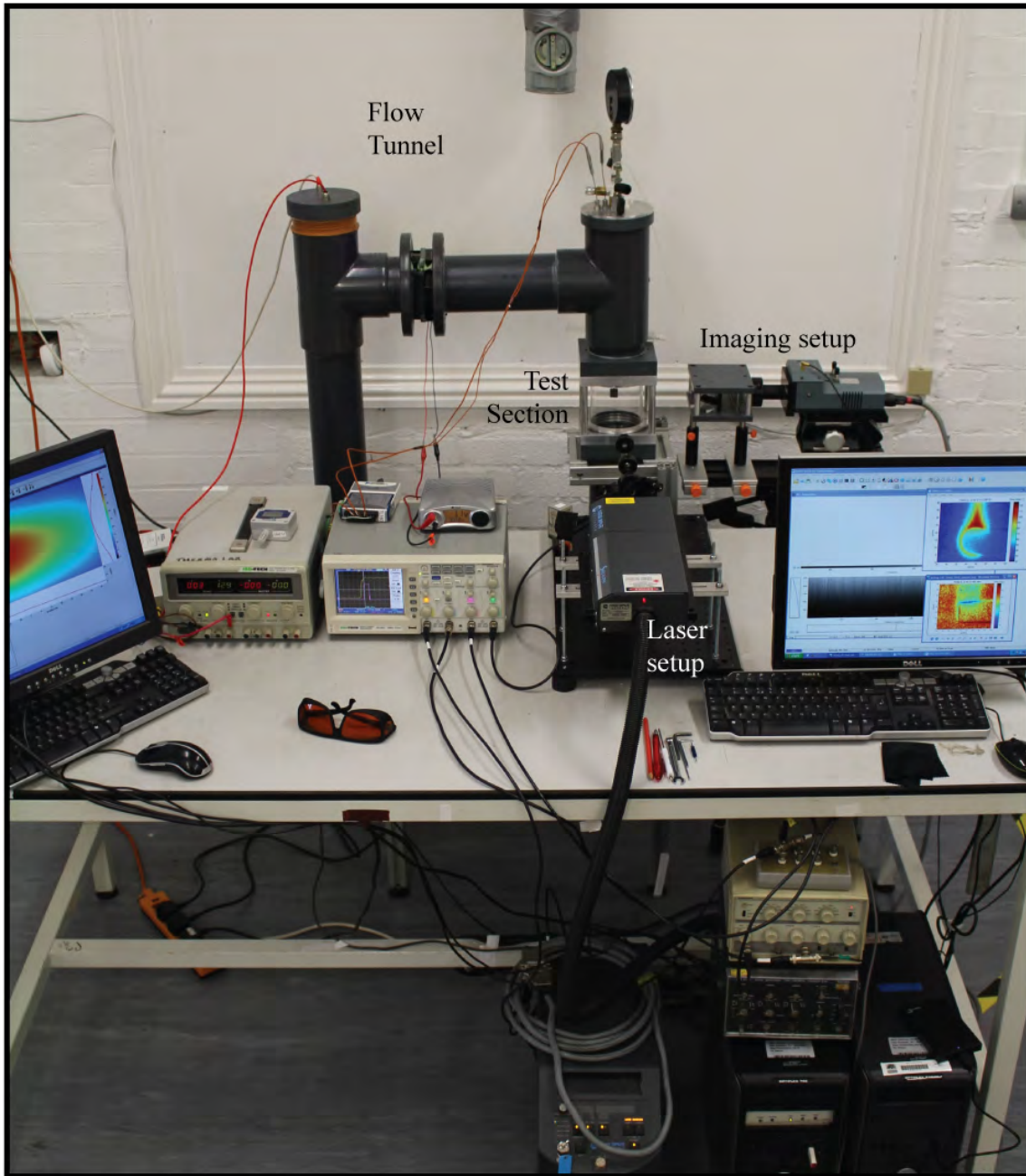


Figure 4.11: Photograph of experimental setup for studying the temperature fields surrounding a drying porous cube with LIF, with all components shown.

parallel to the cube's central axis, $I(x, y, 0)$ (and the signal ratio, $R_I(x, y, 0)$), is measured at different points of drying time.

Before applying the LIF technique to forced convective drying, the technique is applied to a more simple convection problem, that of convective heat transfer from a cylinder. A copper cylinder was suspended in the same position as the porous cube and heated up electrically

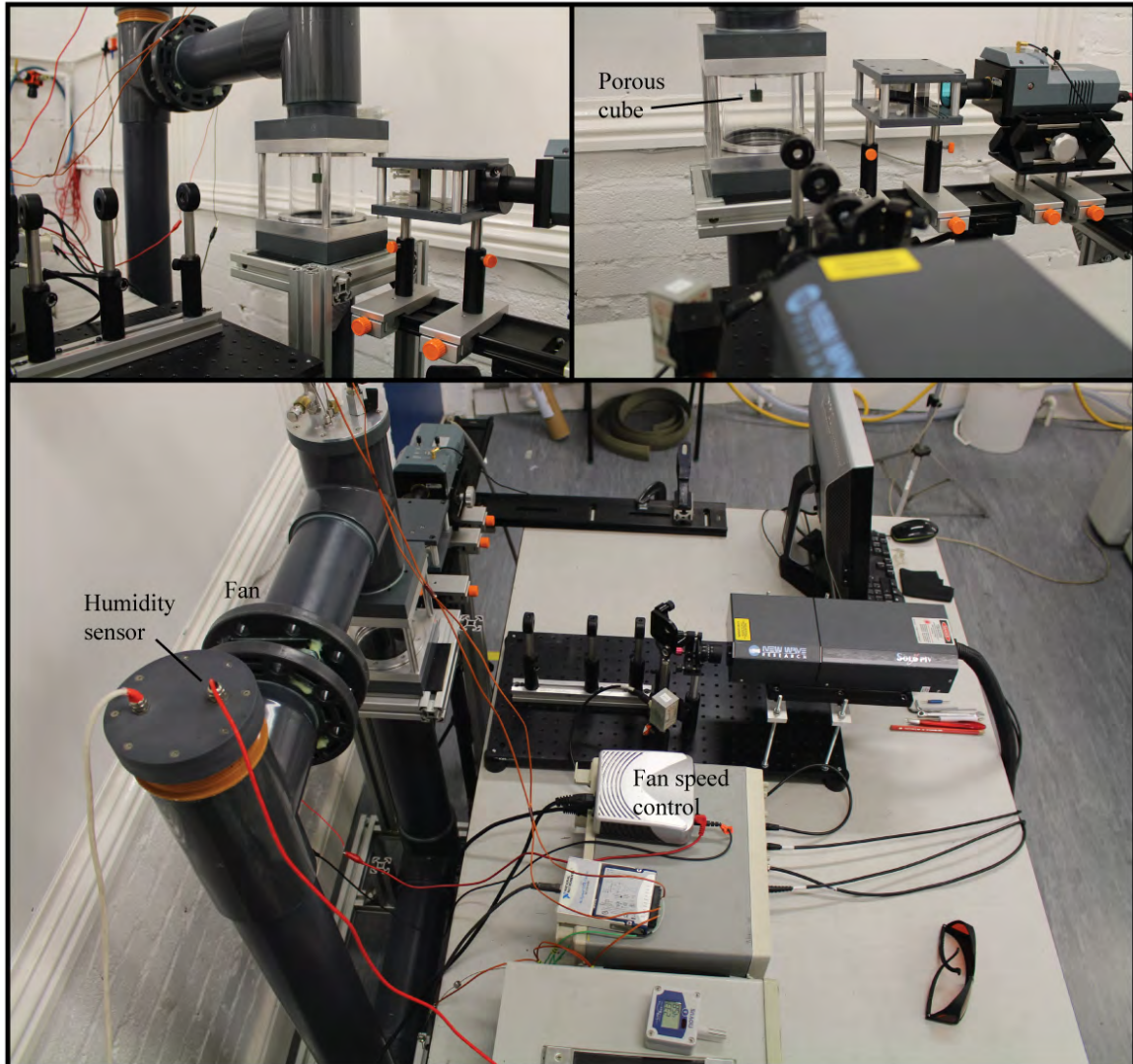


Figure 4.12: Detailed views of the experimental setup from fig. 4.11.

to various temperatures. This is achieved by sliding a high power cylindrical resistor inside a cylindrical copper sleeve. Photographs are shown in fig. 4.13. The outer diameter of the combined cylinder is 15 mm and the length is 20 mm. The inner diameter of the hole through which the resistor slides is 6 mm. A tight fit is ensured between the resistor and copper cylinder and they are assembled with thermal paste to minimize resistance to conduction at the interface. A copper rod acts as a suspension rod and also as the positive electrode while a thin wire, neatly coiled around the rod (to minimize intrusion of the flow), connects the resistor to the negative terminal of a power supply outside of the flow tunnel. Similarly the copper rod is connected to the positive terminal. Both electrodes are carefully sealed

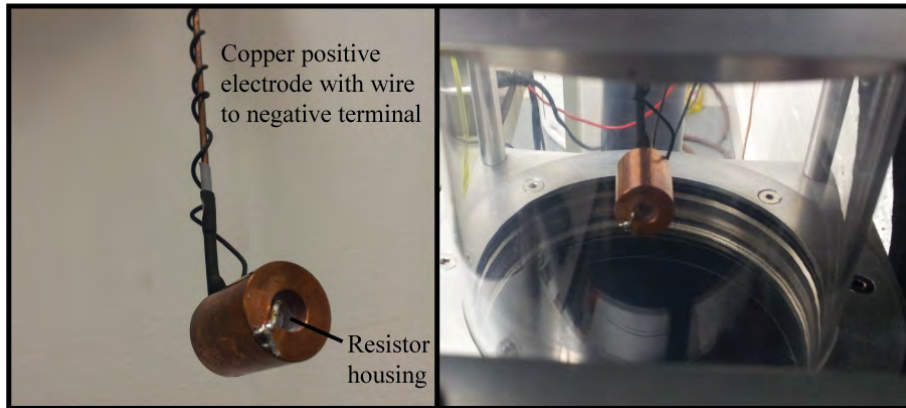


Figure 4.13: Photograph of heated copper cylinder which is suspended in the test section, as shown on the right. The LIF thermometry technique is first tested on convective heat transfer from a cylinder before applying it to forced convective drying (heat and mass transfer).

and insulated to prevent any leaks or shorting of the circuit. A K-type thermocouple is placed inside the copper cylinder. A constant heat flux is applied to the cylinder. Once the temperature of the cylinder stabilizes, $I(x, y, 0)$ and $R_I(x, y, 0)$ are measured.

4.3.1 Single colour & two colour measurement techniques

When applying the LIF technique to measure temperature fields, $T(x, y)$, there are two main methods of determining local temperature: the single colour technique and the two-colour technique. These concepts are outlined in section 3.2. A flow chart to summarize how both techniques are applied is provided in fig. 4.14.

The single colour technique typically involves measuring the fluorescence intensity, I , of the whole fluorescence spectrum (i.e. using a single filter or ‘colour’). It often involves calculating the temperature explicitly by solving equation 3.1. The temperature, $T(x, y)$, can also be found by calibrating the intensity normalized with a reference temperature, $I^* = I/I(T_{ref})$ to temperature, T . An I^* versus T calibration is performed in the validation experiment (phase 2, see fig. 4.1). This approach is used in this study by measuring the fluorescence intensity, $I(x, y)$, for an isothermal temperature field at a reference temperature, T_{ref} , at all points (x, y) . The fluorescence intensity of the temperature field of interest is then divided by the isothermal image. This gives

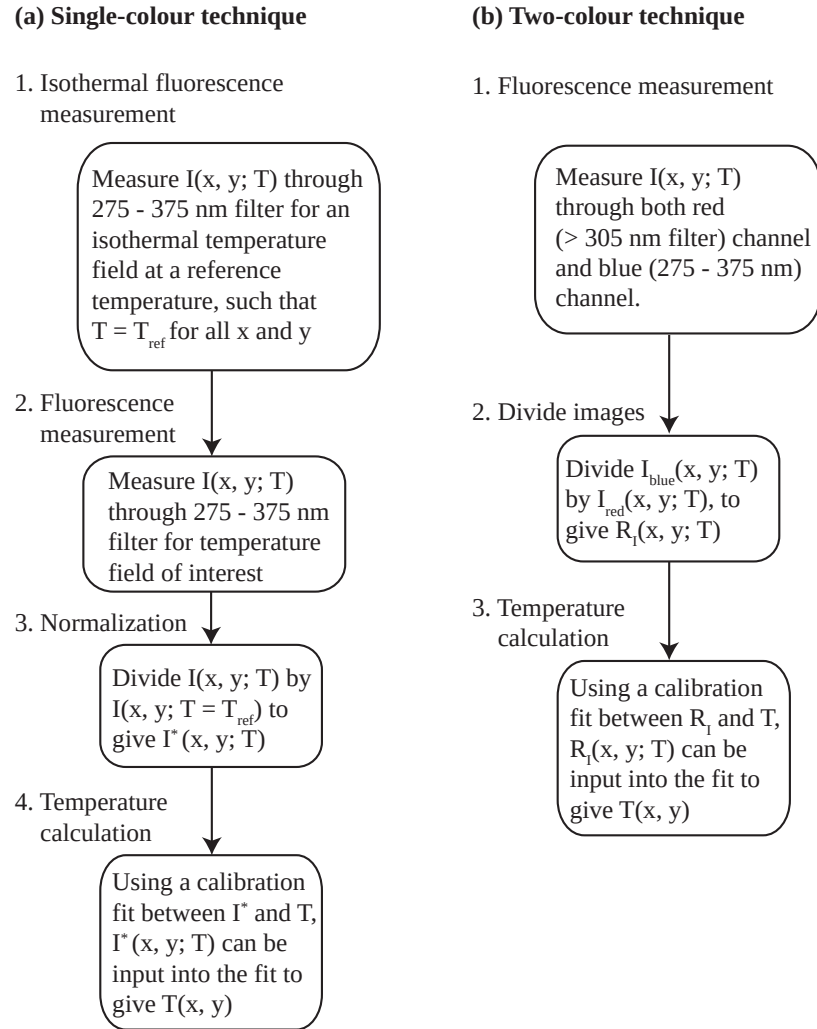


Figure 4.14: Flow chart illustrating the measurement sequence for the (a) single-colour and (b) two-colour technique.

$$I^*(x, y; T) = \frac{I(x, y; T)}{I(x, y; T_{ref})} = \frac{I_0(x, y)C(x, y; T)\Phi(T(x, y))\sigma(T(x, y))\gamma}{I_0(x, y)C(x, y; T_{ref})\Phi(T_{ref})\sigma(T_{ref})\gamma} \quad (4.14)$$

which simplifies to

$$I^*(x, y; T) = \frac{C(x, y; T)\Phi(T(x, y))\sigma(T(x, y))}{C(x, y; T_{ref})\Phi(T_{ref})\sigma(T_{ref})} = f(T(x, y), C(x, y)) \quad (4.15)$$

The temperature-normalized intensity is thus dependent on temperature, T , and tracer concentration, C , at all points, (x, y) . It is acceptable to assume that the toluene tracer is well-distributed and given that several averages are taken, C will not vary much spatially.

The only significant local variation in tracer concentration will be due to an increase in local temperature. However, this variation is built into the calibration between I^* and T , performed in the validation experiment (phase 2, see fig. 4.1). Therefore, $I^*(x, y; T)$ is dependent on just local temperature, $T(x, y)$, which can be calculated by multiplying the local fluorescence intensity at each pixel by the calibration fit.

The two-colour technique typically involves using two dyes with different fluorescence spectra or splitting up the fluorescence spectrum into two portions, I_1 and I_2 , using the necessary optical filters and then taking the ratio of two different signals, $R_I = I_1/I_2$, which respond differently to temperature. See fig. 3.8 for an outline of the concept. In this experiment, a ratio of the higher wavelength (red) portion, 310 – 340 nm, and the lower wavelength (blue) portion 270 – 310 nm (or the full spectrum, 270 – 340 nm) which decrease at different rates with temperature is found, such that

$$R_I = \frac{I_{blue}(x, y, T)}{I_{red}(x, y, T)} = \frac{I_0(x, y)C(x, y)\Phi_{blue}(T(x, y))\sigma(T(x, y))\gamma_{blue}}{I_0(x, y)C(x, y)\Phi_{red}(T(x, y))\sigma(T(x, y))\gamma_{red}} \quad (4.16)$$

which simplifies to

$$R_I = \gamma_R \frac{\Phi_{red}(T(x, y))}{\Phi_{blue}(T(x, y))} = f(T(x, y)) \quad (4.17)$$

which is a direct function of temperature, $T(x, y)$. In cases where different iCCD camera gains are required for the blue and red channel (and hence their signals are being amplified differently), their respective signal intensities can be normalized before division. This additional normalization step is adopted in most studies [28, 31, 150]. A calibration fit between R_I and T also performed in the validation experiment (phase 2, see fig. 4.1) can then be used to determine $T(x, y)$.

Chapter 5

Experimental Accuracy

In this chapter, uncertainty estimates are presented for the measurements made and for calculations of derived quantities throughout the experimental work of this project. The main measured and calculated values from the convective drying experiments (phase 1) and the LIF experiments (phase 2 & 3) of the experimental work are presented in sections 5.1 and 5.2 respectively. Prior to this a brief summary of the method used for calculating uncertainties is provided. A more detailed explanation of these calculations can be found in the textbooks by Coleman and Steele [163] or Bendat and Piersol [164]. A concise outline of the uncertainty principles is also provided by Mills and Chang [165].

The uncertainty of a measured quantity due to random error (error that varies during a measurement), ΔX_r , can be quantified from the expression

$$\Delta X_r = 1.96S(X) \quad (5.1)$$

where X in this case represents a series of measurements taken at a steady value and $S(X)$ represents the standard deviation of the measurement series, X , due to random error. This expression assumes a normally distributed (Gaussian) measurement series, X . Therefore, equation 5.1 states that the true value must lie within ± 1.96 times the value $S(X)$ for a confidence level of 95% (i.e. the probability of a measurement lying within this uncertainty range is 0.95). The value 1.96 is for a series, X , with zero mean and unity standard.

The uncertainty of a measured quantity due to systematic or bias error (error that does not vary during a measurement and is steady-state), ΔX_b , can be quantified from the same

expression as equation 5.1. In this case, X would represent a series of measurements repeated with N different instruments of the exact same model. This, of course, is not feasible to determine here but most instrument manufacturers will supply this information.

The total uncertainty, ΔX of a measured value including both systematic and random error is found by

$$\Delta X = \sqrt{\Delta X_r^2 + \Delta X_b^2} \quad (5.2)$$

The uncertainty of a calculated value, Y , depends on the measured values, such that

$$\Delta Y = \Delta Y(\Delta X_1, \Delta X_2, \Delta X_3 \dots \Delta X_n) \quad (5.3)$$

where Y depends on n number of measured values, X . The uncertainty of the calculated value, ΔY , is determined using the Taylor series method for propagation of uncertainties. A full derivation can be found in Appendix B of Coleman and Steele [163]. A simplified expression of the Taylor series describing the propagation of measurement uncertainty is given by

$$\Delta Y = \left[\sum_{i=1}^n \left(\frac{\partial Y}{\partial X_i} \right)^2 (\Delta X_i)^2 \right]^{1/2} \quad (5.4)$$

where ΔX_i is the total (random and systematic) uncertainty of a measured value (which Y depends on), found from equation 5.2. Equation 5.4 provides an estimate of the uncertainty range which the true value of Y lies within, for a 95% confidence level.

5.1 Uncertainty estimates for convective drying experiments

Table 5.1 presents a list of the uncertainty estimates, δX , for the various raw measurements made during the convective drying experiment. Also included in table 5.1 are the relative random and systematic uncertainties. For some instruments, the uncertainty will depend on the magnitude of the value being measured. Thus for some uncertainty measurements, a range of the minimum to maximum uncertainty is presented in table 5.1. Note, the terms in the left column which have not been assigned symbols up until now are as follows: T_{tc} - the

5.1. UNCERTAINTY ESTIMATES FOR CONVECTIVE DRYING EXPERIMENTS

thermocouple temperature measurement at the cube surface; $m_{balance}$ - the mass recorded by the mass balance. Note this is not the mass of the drying cube but rather the mass read at the point where the lever is supported to which the cube mass is calibrated (see fig. 4.2); m_i - the initial or wet mass (used to calculate M); m_e - the equilibrium or dry mass (used to calculate M); q - the volumetric flow rate of the air entering the drying chamber;

Table 5.1: List of measurement uncertainties from convective drying experiment (phase 1 of experimental work (see fig. 4.1)). Included are the random, $\Delta X_r/\bar{X}$, systematic or bias $\Delta X_b/\bar{X}$, and total uncertainty, $\Delta X/\bar{X}$, of each measurement. The instrument used for each measurement is also listed in the second column.

Measurement	Instrument	δX_r	δX_b	δX
T_{tc} (°C)	T-type thermocouple	0.05	0.19	0.2
$T(x,y)$ (°C)	IR-camera	0.4	0.08	0.41
T_∞ (°C)	T-type thermocouple	0.04	0.19	0.19
$m_{balance}$ (g)	Digital mass balance	0.5	0.002	0.5
m_i (g)	Digital mass balance	0.008	0.002	0.008
m_e (g)	Digital mass balance	0.002	0.002	0.003
L (mm)	Micrometer	0.53	0.005	0.53
w (mm)	Bore gauge	8	0.04	8
q (L/min)	Digital flow meter	3.4 - 10	0.7 - 4	3.5 - 10
RH_∞ (%)	Humidity sensor	0.3	3	3

Each of the values in table 5.1 represent the sources of experimental error in the convective drying experiments. The largest source of uncertainty was found for the equilibrium or dry mass of the cube, m_e which results in a relative uncertainty of 16%. This in turn affects the uncertainty of the moisture content. However, from the point-of-view of heat and mass transfer analysis, this is not critical since all of the mass transfer quantities are calculated from the mass, m , which has much lower uncertainty (see table. 5.2). Therefore, the uncertainty of M won't propagate in other calculations. The absolute moisture content is more important in material-specific studies, where reliable empirical information is required to predict the moisture content of a certain drying material. In this study, it is mostly of importance from a qualitative point of view.

The uncertainty of the cube side length, L , is due to the difficulty in cutting the porous material precisely. This leads to some inaccuracy in estimating the surface area (see table. 5.2).

This error then propagates in the calculation of the rate of convective heat transfer. Therefore, it is important to find a method of ensuring precise side lengths. This inaccuracy can also be reduced by making larger cubes such that the relative uncertainty decreases. However, this lengthens the drying time considerably and requires scaling up of the tunnel size, which is not always practical.

The lever system (described in section 4.1.1) serves to reduce the uncertainty in measuring the cube mass. The mass balance measurement, $m_{balance}$, (where the lever is supported) has a greater random uncertainty compared to simply weighing a solid without the lever system. This is due to vibrations and friction in the lever (which are minimized by using vibration dampening pads and knife-edge supports). However, the amplification due to the moment ratio, A_{mr} , amplifies the scale or span of measurement. Furthermore, by operating near the upper limit of the mass balance range, the mean value $\bar{m}_{balance}$ is very large compared to the absolute uncertainty $\Delta m_{balance}$. This means the relative uncertainty, $\Delta m_{balance} / \bar{m}_{balance}$, is very small.

In order to determine the propagation of these uncertainties in the calculated values, equation 5.4 is used. It is found that in almost all cases, when the equation is expressed in terms of relative uncertainty, it simplifies to

$$\frac{\Delta Y}{Y} = \left[\sum_{i=1}^n \left(K_o \frac{\Delta X_i}{\bar{X}} \right)^2 \right]^{1/2} \quad (5.5)$$

where K_o is the order of the term before partial differentiation. This simplification occurs because the measurement uncertainties are independent of one another [166]. For example, consider the Reynolds number, $Re = V_{\infty} L / \nu$, which is a function of the free-stream velocity, V_{∞} , the cube length, L , and the kinematic viscosity, ν . The uncertainty of the Reynolds number, from equation 5.4, is given by

$$\Delta Re = \left[\left(\frac{\partial Re}{\partial V_{\infty}} \right)^2 (\Delta V_{\infty})^2 + \left(\frac{\partial Re}{\partial L} \right)^2 (\Delta L)^2 + \left(\frac{\partial Re}{\partial \nu} \right)^2 (\Delta \nu)^2 \right]^{1/2} \quad (5.6)$$

which after partial differentiation becomes

$$\Delta Re = \left[\left(\frac{L}{\nu} \right)^2 (\Delta V_{\infty})^2 + \left(\frac{V_{\infty}}{\nu} \right)^2 (\Delta L)^2 + \left(\frac{V_{\infty} L}{\nu^2} \right)^2 (\Delta \nu)^2 \right]^{1/2} \quad (5.7)$$

Then squaring both sides and dividing by Re^2 , gives

$$\left(\frac{\Delta Re}{Re}\right)^2 = \left[\left(\frac{\Delta V_\infty}{V_\infty}\right)^2 + \left(\frac{\Delta L}{L}\right)^2 + \left(\frac{\Delta v}{v}\right)^2\right] \quad (5.8)$$

or

$$\frac{\Delta Re}{Re} = \left[\left(\frac{\Delta V_\infty}{V_\infty}\right)^2 + \left(\frac{\Delta L}{L}\right)^2 + \left(\frac{\Delta v}{v}\right)^2\right]^{1/2} \quad (5.9)$$

which is the relative uncertainty of Re . Equation 5.9 is a simplified expression of the individual uncertainties in the form of equation 5.5. This can be shown for the other calculated values.

Table 5.2 provides a table of the relative uncertainty estimates of the calculated values. This uncertainty, $\Delta Y/Y$, represents the percentage of the value which the true value should lie within for a confidence level of 95%. As in table 5.1, some uncertainties are expressed as a range (i.e. the minimum to the maximum uncertainty) since some uncertainties depend on the magnitude of the value being measured. Note, the term C_{hm} in table 5.2 is the heat and mass transfer analogy factor (which tends to 1 for an analogous process). This is defined and described in more detail in section 7.4.

The uncertainty is at a reasonable level for most terms. The heat transfer coefficient, h , and Nusselt Number, Nu , appear to show the higher uncertainty, since they depend on multiple measured and calculated terms which leads to more error propagation. The radiation heat transfer term depends on the surface area, A_s . Therefore, h depends on two calculations of A_s causing further inaccuracy. The uncertainty of the drying chamber width, w , similarly effects the uncertainty of the Reynolds number, Re . There is additional uncertainty in the analogy factor, C_{hm} , since it depends on both the heat and mass transfer coefficients. Note that h and h_m can be time-averaged over the duration of the CRP (which is done in section 7.4). Ensemble averages of these values, for different experiments (at the same conditions), are also taken. This means the accuracy of the average heat and mass transfer coefficients during the CRP are slightly greater.

An aspect of experimental accuracy which is important to highlight here also is the repeatability of the time evolution curves of surface temperature and moisture content.

5.2. UNCERTAINTY ESTIMATES IN LIF EXPERIMENTS

Table 5.2: List of calculated uncertainties (expressed as relative uncertainty) from convective drying experiment (phase 1 of experimental work - see fig. 4.1).

Calculated value	$\Delta Y/Y$ (%)
m	5 - 8
M	16
$\dot{m}$	5 - 8
$\dot{Q}_{lat}$	5 - 8
ρ_{vs}	0.5 - 2.4
$\rho_{v\infty}$	3
A_s	10.4
V_∞	16.1
Re	17
$\dot{Q}_{rad}$	10.5 - 11.5
$\dot{Q}_{cond,rod}$	2.4 - 5
$\dot{Q}_{st}$	5.1 - 5.6
$\dot{Q}_{conv}$	13.7 - 13.9
h	17.2 - 17.6
Nu	17.2 - 17.6
h_m	12 - 13.7
Sh	13.12 - 14.7
Pr	0.3 - 0.5
Sc	0.3 - 0.5
C_{hm}	21 - 22

Fig. 5.1a compares the moisture content evolution curve for three experiments repeated at the same parameter setting of $Re = 200$ and $T_\infty = 200$. Fig. 5.1b compares the average surface temperature evolution curve of three experiments where $Re = 300$ and $T_\infty = 200$. Both plots indicate a good level of repeatability, especially considering that the suspension of the cube in the drying chamber is not an automated process and a new porous cube is suspended each time from the lever. This means the potential for error may be significant.

5.2 Uncertainty estimates in LIF experiments

The same approach to estimating measurement and calculation uncertainties is applied to the LIF experiments (phase 2 & 3). Table 5.3 provides a list of the uncertainty estimates of the measured values during this experimental work. The table is divided into those quantities calculated in phase 2, phase 3 and both phases of the experimental work. The terms in the left column which have not been assigned symbols up until now are as follows: T_{vessel} - the

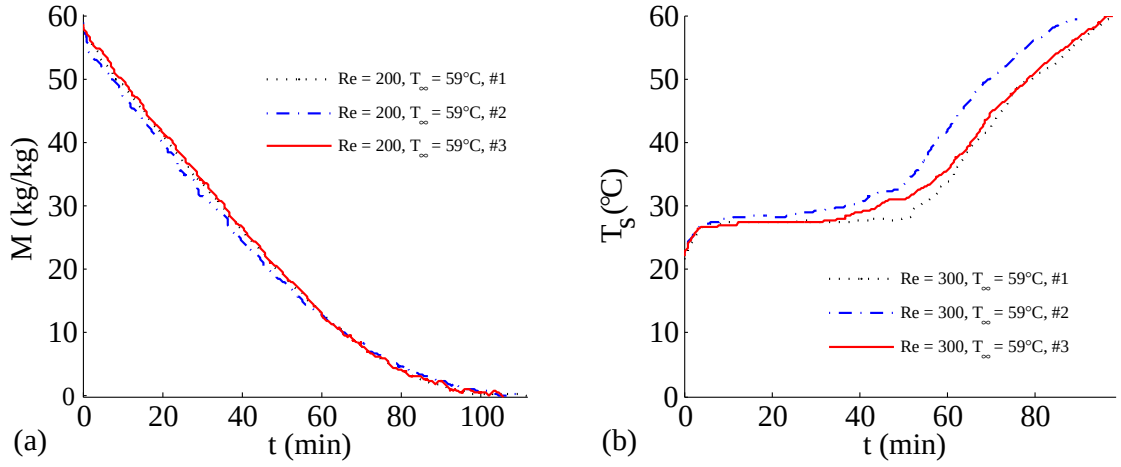


Figure 5.1: Repeatability of time evolution curves of (a) moisture content, M , and (b) average surface temperature, T_s . The Reynolds number, free-stream temperature and test number are shown in the legend.

temperature measurement inside the vessel measured using T-type thermocouples at different positions. $\dot{E}_0$ - the laser power, measured with a power meter over several laser pulses (i.e. time-averaged); V_{tol} and V_{ans} - the volume of liquid toluene and anisole, respectively, injected into the test section (the relative uncertainty of V_{tol} is smaller in phase 3 of experiments, since a larger amount is required for the flow tunnel and also a larger syringe is used); $\dot{Q}$ - heat input to the copper cylinder (dissipated in resistor). Note also that T_s in this case represents the surface temperature of the copper cylinder and the drying porous cube, measured with a K-type thermocouple.

As with the drying experiment measurements (see table 5.1), most of the measurement uncertainty is at an acceptable level. The uncertainty of the local fluorescence intensity, $I(x, y)$, is reasonable since it is averaged over several images. For a single image, however, it is 15.6% for the iCCD camera which is somewhat high. The UV lens used has not been optimized for this imaging device which leads to a reduced spatial resolution. It is also known that for iCCD cameras, though the sensitivity is increased this is at the expense of a reduced spatial resolution [167]. Since the fluorescence signal through the red channel (305 – 375 nm filter combination) is much lower, the uncertainty is a little greater for $I_{red}(x, y)$. For the validation experiments (phase 2), this is not too critical, since a spatial average, I_{avg} , is taken which increases the accuracy significantly (see table 5.4 for more detail). However, it does

5.2. UNCERTAINTY ESTIMATES IN LIF EXPERIMENTS

Table 5.3: List of measurement uncertainties from LIF experiments (phase 2 & 3 of experimental work (see fig. 4.1)). Included are the random, ΔX_r , systematic or bias, ΔX_b , and total uncertainty, ΔX , of each measurement. The instrument used for each measurement is also listed in the second column. Uncertainties for $I(x,y)$ and $I_{red}(x,y)$ are expressed as relative uncertainties ($\Delta X_r/\bar{X}$), since the electronic count levels can vary from experiment to experiment. The relative uncertainty does not vary significantly, so the largest or most conservative estimate is taken.

Measurement	Instrument	ΔX_r	ΔX_b	ΔX
Phase 2				
T_{vessel} (°C)	T-type thermocouples	0.04 - 0.24	0.16 - 0.18	0.17 - 0.3
V_{tol} (μL)	250μL syringe	10	1	10
Phase 3				
T_∞ (°C)	T-type thermocouples	0.04 - 0.24	0.16 - 0.18	0.17 - 0.3
T_s (°C)	K-type thermocouple	0.5	0.165 - 0.2	0.53
$D_{cyl}(mm)$	Micrometer	0.005	0.08	0.08
$\dot{Q}$ (W)	Digital power supply	0.05	0.007	0.05
V_∞ (m/s)	Velocity probe	0.04 - 0.29	0.08	0.09 - 0.3
V_{tol} (mL)	5 mL syringe	0.1	0.05	0.11
V_{ans} (μL)	250μL syringe	10	5	11
Phase 2 & 3				
$\dot{E}_0$ (mW)	Laser power meter	2	0.6	2.1
$I(x,y)$ (%)	iCCD	6.3	N/A	6.3
$I_{red}(x,y)$ (%)	iCCD	7.2	N/A	7.2

lead to some inaccuracy in phase 3 of the experiments where the technique is applied to study local convective heat transfer and drying. Solutions for reducing this uncertainty are discussed in section 6.2.

Another source of uncertainty which is somewhat high is the volume of toluene injected, V_{tol} , into the vessel or test section in the validation experiments. This is due to a small amount of liquid becoming trapped (due to surface tension) in the valve which may decrease the actual amount. This amount has been estimated conservatively based on the volume of the valve. This relative uncertainty reduces in phase 3 of the experiments since a larger amount is required for the flow tunnel. Also, the syringe used, and hence the pumping pressure, is larger and so this trapping effect is minimized.

Equation 5.5 can then be applied to these measured uncertainties in order to estimate the calculated uncertainties of the LIF experiments. These are listed in table 5.4. Like table 5.3, this is divided into those quantities calculated in phase 2, phase 3 and both phases of the

experimental work.

The terms in the left column which have not been introduced yet include: w_{tol} and w_{ans} - the mass percentages of toluene and anisole, respectively (i.e. the ratio of toluene or anisole mass to the N_2 mass); I_{avg} - the spatially averaged fluorescence signal; $I_{avg,red}$ - the spatially averaged fluorescence signal for the red channel (305 – 375 nm filter combination); I^* - the spatially-averaged fluorescence signal normalized for a reference temperature (and for the red channel, I_{red}^*); Φ^* - the fluorescence quantum yield normalized for a reference temperature; $T(x,y)_{I^*}$ and $T(x,y)_{R_I}$ - the local temperature determined from the local temperature-normalized fluorescence, $I^*(x,y)$ and the local signal ratio, $R_I(x,y)$, respectively; $\dot{q}$ - the heat flux supplied to the copper cylinder; h - the local heat transfer coefficient at the cylinder surface, estimated from the temperature field; Nu - the local Nusselt number ($Nu = hD_{cyl}/k$) at the cylinder surface. These values are all defined and explained in more detail in sections 6.1 and 6.2 and also in the nomenclature.

The most significant uncertainties from table 5.4 are the local normalized fluorescence intensities, $I^*(x,y)$ and $I_{red}^*(x,y)$, and in particular the local signal ratio, $R_I(x,y)$. Since these values depend on two measurements of fluorescence intensity, $I(x,y)$ (or $I_{red}(x,y)$), there is more propagation of uncertainty. However, since the fluorescence intensity is averaged over 10 images, the uncertainty is reduced to a reasonable amount. For a single image the uncertainty of $I^*(x,y)$ and $R_I(x,y)$ are 22% and 39%, respectively. The uncertainty of local fluorescence intensity is also reasonable since the laser pulse energy is low compared to most other studies (for example see [28, 29, 31, 33, 148, 150]). The signal ratio, $R_I(x,y)$, generally needs to be calculated from two normalized fluorescence intensities (one for the blue and one for the red channel). This leads to an increased uncertainty for the local signal ratio. As a result, the local temperature measurement is more accurate by using the single colour technique (i.e. determining $T(x,y)_{I^*}$), compared to the two colour technique (i.e. determining $T(x,y)_{R_I}$). This is seen in section 6.2.2 where it is difficult to infer accurate temperature fields with the two colour technique. Note that the uncertainties of the correlation or calibration fit between I^* and T (and R_I and T), are also included in the uncertainty propagation (equation 5.4) for $T(x,y)$.

In the validation experiment (phase 2), where the relationship between I and T is being investigated, the fluorescence intensity is spatially-averaged for a spatially-uniform tempera-

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Table 5.4: List of calculated uncertainties (expressed as relative uncertainty) from LIF experiment (phase 2 and 3 of experimental work - see fig. 4.1).

Calculated value	$\Delta Y/Y$ (%)
Phase 2	
w_{tol}	20
I_{avg}	1.6
$I_{avg,red}$	2
I^*	2.2
I_{red}^*	2.5
R_I	4
σ	5
Φ^*	7.5
Phase 3	
w_{tol}	13.7
w_{ans}	9.4
$I^*(x,y)$	9.2
$I_{red}^*(x,y)$	10.2
$R_I(x,y)$	14
$T(x,y)_{I^*}$	11
$T(x,y)_{R_I}$	16
A_s	0.75
$\dot{q}$	1.7 - 12.5
Re	14 - 20
h	17
Nu	17
Phase 2 & 3	
I_0	7.2

ture. This means that the uncertainty of I_{avg} (and hence I^*) is much less, since the number of samples, N , is equal to the number of pixels in the averaging border. This can be calculated from

$$\Delta I_{avg} = \frac{2S(I(x,y))}{\sqrt{N}} \quad (5.10)$$

which results in a much smaller uncertainty than that of the local fluorescence intensity, $I(x,y)$, as shown in table 5.4. Since there is typically a correlation between adjacent pixels up to some unknown distance, N is taken as the number of independent pixels rather than the total number of pixels in the averaging border. A conservative estimate of N is used. This is

equal to the number of pixels every 4 mm which is 16 times the intensifier tube resolution limit of 0.25 mm of the iCCD.

The absorption cross-section, σ , is determined in section 6.1.1 from a fit by Koch [168] as a function of temperature. The uncertainty of σ is based on the propagation of error from the vessel temperature measurement, T_{vessel} , and the 5% uncertainty of the correlation by Koch [168]. This is then combined with I^* , to find the normalized fluorescence quantum yield, Φ^* . More detail on the calculations is provided in section 6.1.1. Finally, the uncertainty of the laser pulse energy is calculated from the propagation of error from the laser power measurement, $\dot{E}_0$ and the pulse-to-pulse fluctuation. The magnitude of this fluctuation is not measured or provided in the Nd:Yag laser data sheet but, according to other literature [145, 155], it is typically found to be somewhere between 0.5% and 1%. The uncertainty of the laser pulse rate (i.e. the timing), which is used to calculate I_0 , is found to be negligible.

Chapter 6

Results and discussion: LIF thermometry at low temperature ranges

Before using the LIF temperature field measurement technique, an experiment is performed to verify the relationship and develop a calibration between toluene fluorescence and temperature over a low temperature range, as explained in chapter 4 (see fig. 4.1 for an outline of the experimental strategy). These results are discussed in section 6.1. The technique is then applied to convective heat transfer from a heated cylinder and tested at low temperatures. This well-studied heat transfer problem is a good starting point for applying the technique at low temperatures before extending it to forced convective drying of a porous cube (chapter 7) which is a more complex heat and mass transfer problem. These results are discussed in section 6.2.

6.1 Validation of gaseous LIF thermometry at low temperature ranges

This section presents and discusses the experimental results for validation and calibration of LIF thermometry at low temperature ranges. The experimental setup used is detailed in section 4.2. Results from experiments using both a regular CCD camera and an intensified CCD (iCCD) camera are presented.

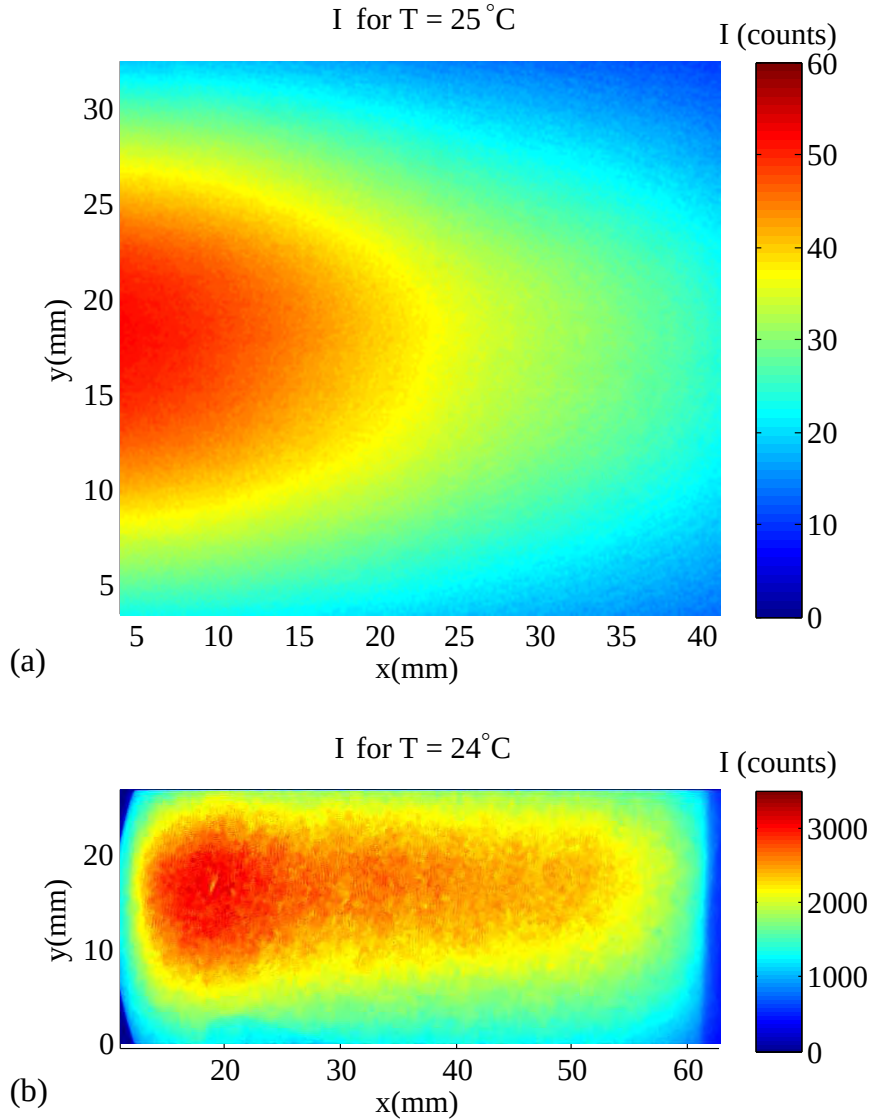


Figure 6.1: Sample image of fluorescence intensity, $I(x, y)$, using (a) the CCD camera averaged over 500 frames and (b) the iCCD camera averaged over 15 frames for similar temperatures of 25°C and 24°C, respectively.

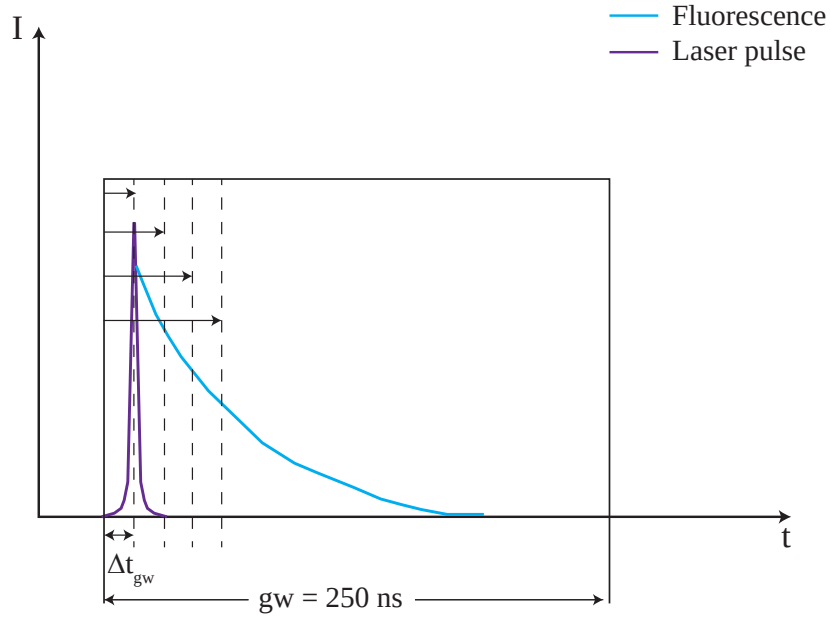
Fig. 6.1 demonstrates a sample image of the local fluorescent signal (through a 275 - 375 nm filter combination, enclosing the full fluorescence spectrum) plotted in counts (i.e. the digital value measured by the sensor proportional to the number of photo electrons) for a temperature of 25°C. Fig. 6.1a is for the CCD (PCO Sensicam) and fig. 6.1b is for the iCCD camera (Andor DH520-18T), averaged over 500 and 15 images, respectively. The images are taken a little left of the test section centre (see fig. 4.7 for a detailed sketch of test-section) where the laser sheet is less diverged and therefore has greater intensity. The decrease from left to right of the fluorescence is due to the laser energy being absorbed by the toluene as

it passes through the N₂-toluene mixture. The sheet also becomes more diverged and less focused as it passes through the test section meaning there is a decrease in the excitation intensity from left to right. This is a common effect that can be corrected by dividing with an image at a reference temperature (i.e. normalization of the fluorescence intensity across all pixels).

Fig. 6.1 shows that the intensified camera is significantly more sensitive with a signal intensity roughly 100 times greater. Note that the difference in the profile of the fluorescence for each camera is due to three causes: 1. The iCCD image has a wider field of view due to differences in the sensor position relative to the UV imaging lens, compared to that of the regular CCD camera. 2. The sensor of the iCCD is also detecting the left side of the imaging lens tube which results in the slight curve cutting off the far left side of the signal which can be seen in fig. 6.1b. 3. The background noise of the iCCD is slightly asymmetric in the y-direction leading to an apparent asymmetry in the y-direction in fig. 6.1, compared to fig. 6.1a which is symmetric.

In order to characterize the time response of the fluorescence and verify it agrees with that of previous work, some additional experiments were carried out in which the LIF intensity was measured for various intensifier gate widths. As illustrated in fig. 6.2, this can be done by setting the intensifier gate to open just before the fluorescence begins and measuring the fluorescence intensity, I . Then a measurement of I is taken for an increasingly smaller gate width by delaying the the rising edge by a time interval, Δt , so as to gradually cut-off more and more of the signal. This can also be done by bringing the falling edge progressively closer to the rising edge. The fluorescence intensity versus the delay to the gate rising edge is shown in fig. 6.3a along with an exponential fit to the data. This fit is also plotted in fig. 6.3b for fluorescence intensity normalized with respect to time, $I/I(t=0)$, illustrating the decay of fluorescence with time on a logarithmic scale. Also included is a fit by Faust et al. [156] for toluene excited at 266 nm at a temperature of 297 K and a pressure of 1 bar. A good agreement is demonstrated here.

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*Gate times not to scale

Figure 6.2: Illustration of method for determining fluorescence time response of toluene. The intensity is measured for a consecutively smaller gate width, $gw - \Delta t_{gw}$, and hence $I(t)$ can be determined.

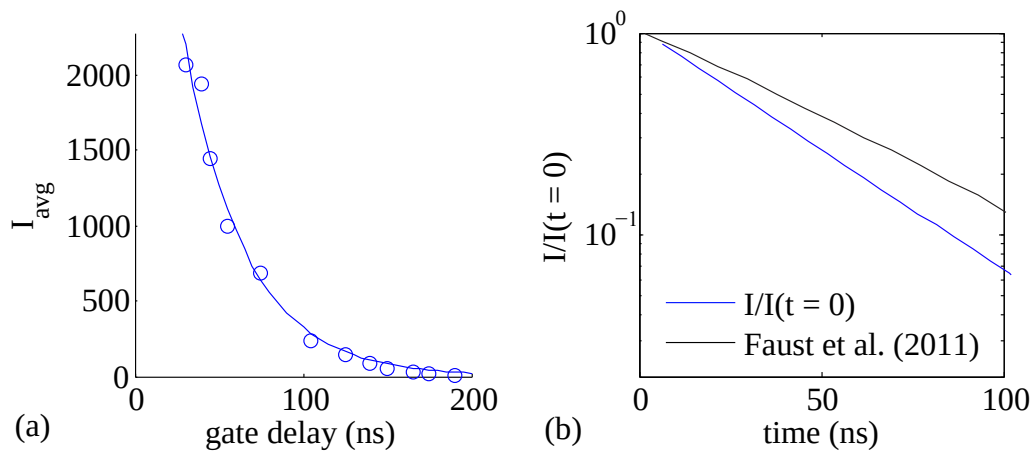


Figure 6.3: (a) Toluene fluorescence intensity, I , versus gate (rising edge) delay along with an exponential fit to the data. (b) The same exponential fit normalized with respect to time, $I/I(t=0)$, plotted on a logarithmic scale versus time along with a fit by Faust et al. [156].

6.1.1 Toluene LIF signal versus temperature

Table 6.1 lists the conditions and settings for both the CCD and iCCD cameras. w_{tol} is the mass percentage of toluene. A significantly larger amount of toluene ($w_{tol} = 12\%$) was used for imaging with the CCD camera, due to the sensitivity in the wavelength range of toluene fluorescence. Also, a significantly greater number of averages were taken for the CCD. The temperature intervals are slightly different in each experiment but within a similar temperature range. This is because the hot plate used to heat the sealed container did not allow precise specification of the temperature and also, the ambient or room temperature varied from day to day. However, the hot plate ensured a well-stabilized and uniform temperature within the test section which is most important. The pressure, P , was maintained above atmospheric pressure, at 1.4 bar. This allowed easy detection of any leaks. It also ensured that if a very small amount of oxygen was still present after purging the container, it would be negligible compared to the amount of nitrogen gas. Above 1 bar the fluorescence signal is found to be independent of gas pressure as shown by Strozzi et al. [29]. This was also verified by measuring fluorescence intensity for a few different pressures (see Appendix C2).

Table 6.1: Parameter settings for CCD and iCCD cameras. w_{tol} represents the mass percentage of toluene vapour in the test section, P is the absolute pressure in bar and I_0 is the laser pulse (excitation) energy in mJ. gd and gw are the intensifier gate delay and gate width, respectively.

Camera	$T(^{\circ}C)$	w_{tol} (%)	P (bar)	#Averages	gd, gw (nsec)	I_0 (mJ)
iCCD	24 – 56	3	1.4	15	30, 250	3.6
CCD	24 – 53	12	1.4	500	N/A	3.6

Fig. 6.4a-d shows the fluorescence intensity, $I(x, y)$, for a series of temperatures, using the iCCD camera and a 275 - 375 nm filter combination, and fig. 6.4e-h shows the same for the red channel (305 - 375 nm filter combination). The fluorescence intensity captured with the CCD camera for various temperatures is also shown in fig. 6.5. Fig. 6.4 and fig. 6.5 demonstrate a clear inverse relationship between fluorescence intensity, I , and temperature, T . As expected, the fluorescence portion through the red channel ($> 305\text{nm}$) has a significantly lower intensity. Note, this difference in magnitude is not as great for the CCD camera since the quantum efficiency of the sensor decreases dramatically below 300 nm. Therefore, the red portion is less affected by this deficiency.

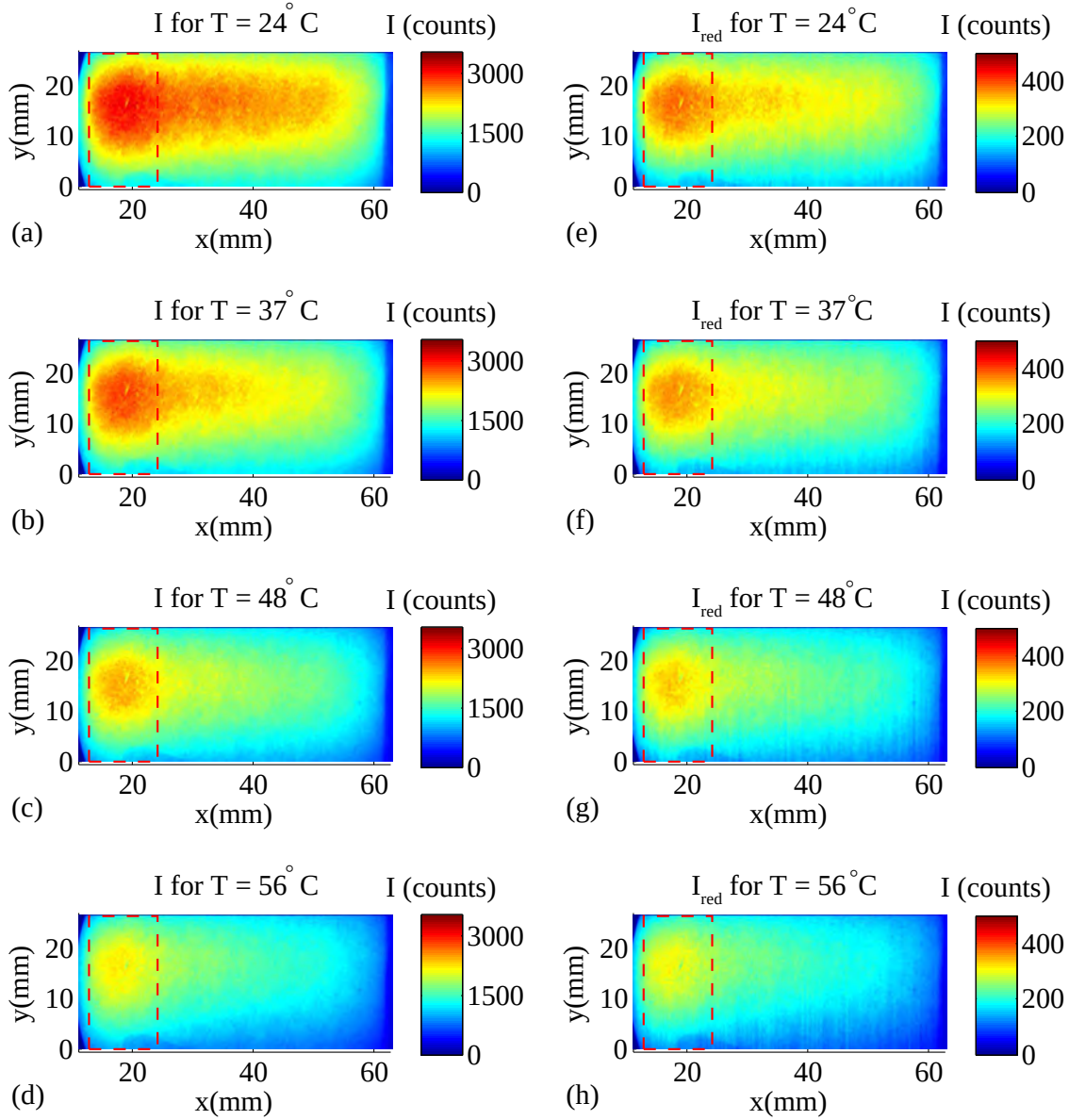


Figure 6.4: Fluorescence intensity, $I(x, y)$, measured with the iCCD camera at various temperatures for (a-d) the full fluorescence spectrum, using a 275 - 375 nm filter combination, and (e-h) the red channel (305 - 375 nm filter combination). The red dashed rectangle indicates the area over which the spatial-average, I_{avg} , of the fluorescence signal is calculated.

The red dashed rectangles indicate the border over which the spatial-average LIF intensity, I_{avg} , is calculated. A narrow averaging border on the left of the fluorescence images, where the laser sheet has just passed through the quartz window, was selected. The reason for this was due to some laser attenuation affecting the LIF images. This effect is most clear in the CCD camera images in fig. 6.5 where the significant decrease in fluorescence intensity from left to right becomes more pronounced with increasing temperature. Generally, the

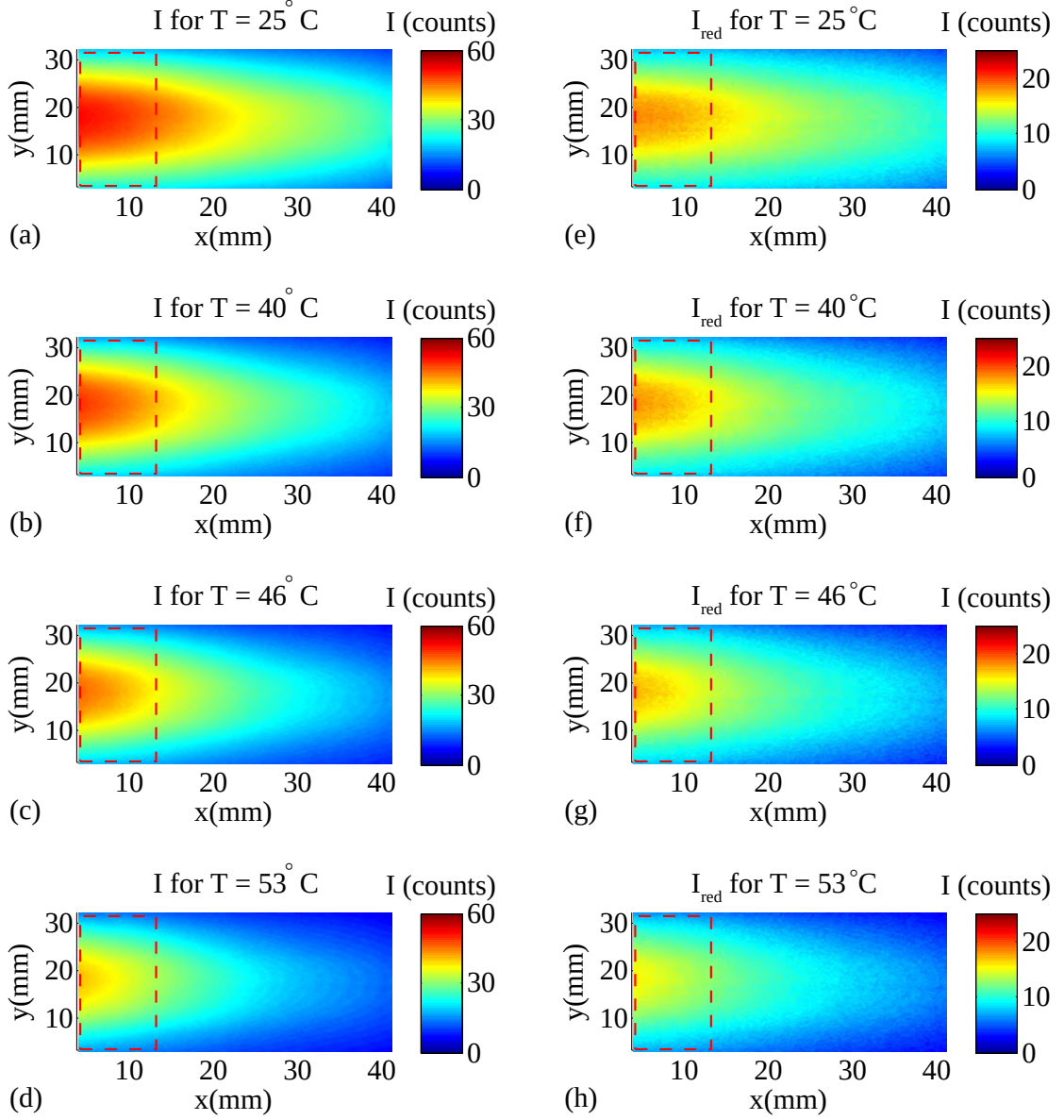


Figure 6.5: Fluorescence intensity, $I(x, y)$, measured with the CCD camera at various temperatures for (a-d) the full fluorescence spectrum, using a 275 - 375 nm filter combination, and (e-h) the red channel (305 - 375 nm filter combination). The red dashed rectangle indicates the area over which the spatial-average, I_{avg} , of the fluorescence signal is calculated.

laser intensity energy will decrease gradually from left to right as it passes through the test section and is absorbed by the tracer molecules. This attenuation can become significant, however, when large tracer concentrations are used [145, 149, 162, 167, 169]. Since the absorption cross-section, σ , increases with temperature the effect may be amplified at higher temperatures [35].

Since a significantly larger amount of toluene was used for the CCD camera, there

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is a more significant attenuation with temperature. This is illustrated in fig. 6.6 which contains profiles of fluorescence intensity along the mid-height of the laser sheet, where H represents the sheet height. The fluorescence intensity is normalized with the maximum fluorescence intensity along the profile, $I_{H/2,max}$. Comparing the profile from the iCCD camera measurements (fig. 6.6a) to that of the CCD camera (fig. 6.6b), it can be seen clearly that the laser sheet is more attenuated in the CCD camera measurements, especially as the temperature is increased. Therefore, it is important to average over a smaller border near the entry of the laser sheet in order for any trends between I_{avg} and T to be independent of this attenuation effect.

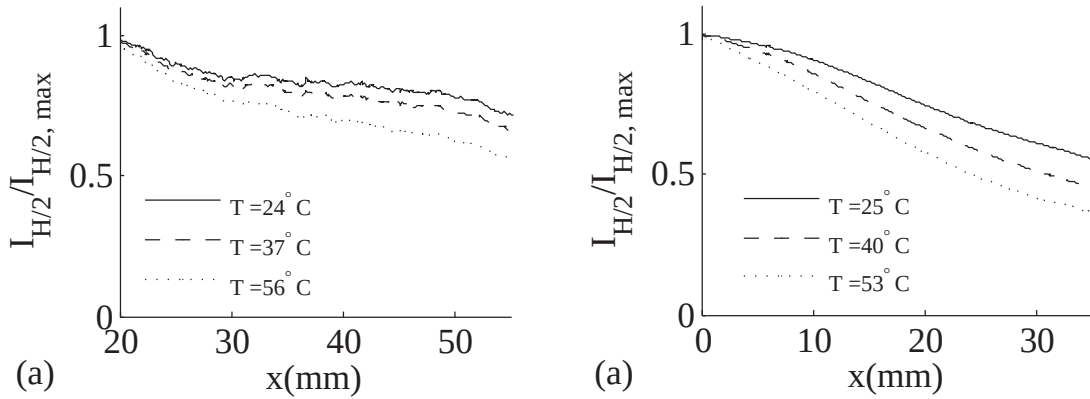


Figure 6.6: Profiles of fluorescence intensity along the mid-height of the laser sheet, $I_{H/2}$, where H represents the sheet height and normalized by the maximum fluorescence intensity along the profile, $I_{H/2,max}$. Profile (a) is for the iCCD camera measurements and (b) is for the CCD camera measurements.

The spatially-averaged fluorescence intensity, I_{avg} , versus temperature, T , is plotted for different experiments in fig. 6.7. Also included in each plot is the spatially-averaged fluorescence intensity through the red channel, $I_{avg,red}$. Fig. 6.7a & b are two experiments with the iCCD camera and fig. 6.7c & d are with the CCD camera. The decreasing trend of I_{avg} versus T is clear in each plot. The fluorescence through the red channel, $I_{avg,red}$, is decreasing more slowly with temperature which is consistent with the toluene fluorescence spectrum. As shown earlier in the fluorescence spectrum of toluene in fig. 3.12, above 300 nm the fluorescence becomes decreasingly less dependent on temperature.

A spectrum normalized to the wavelength of maximum or peak fluorescence, $I/I(\lambda_{max})$, from Miller and Gamba [31] is shown in fig. 6.8a and an exponential fit to this spectrum for

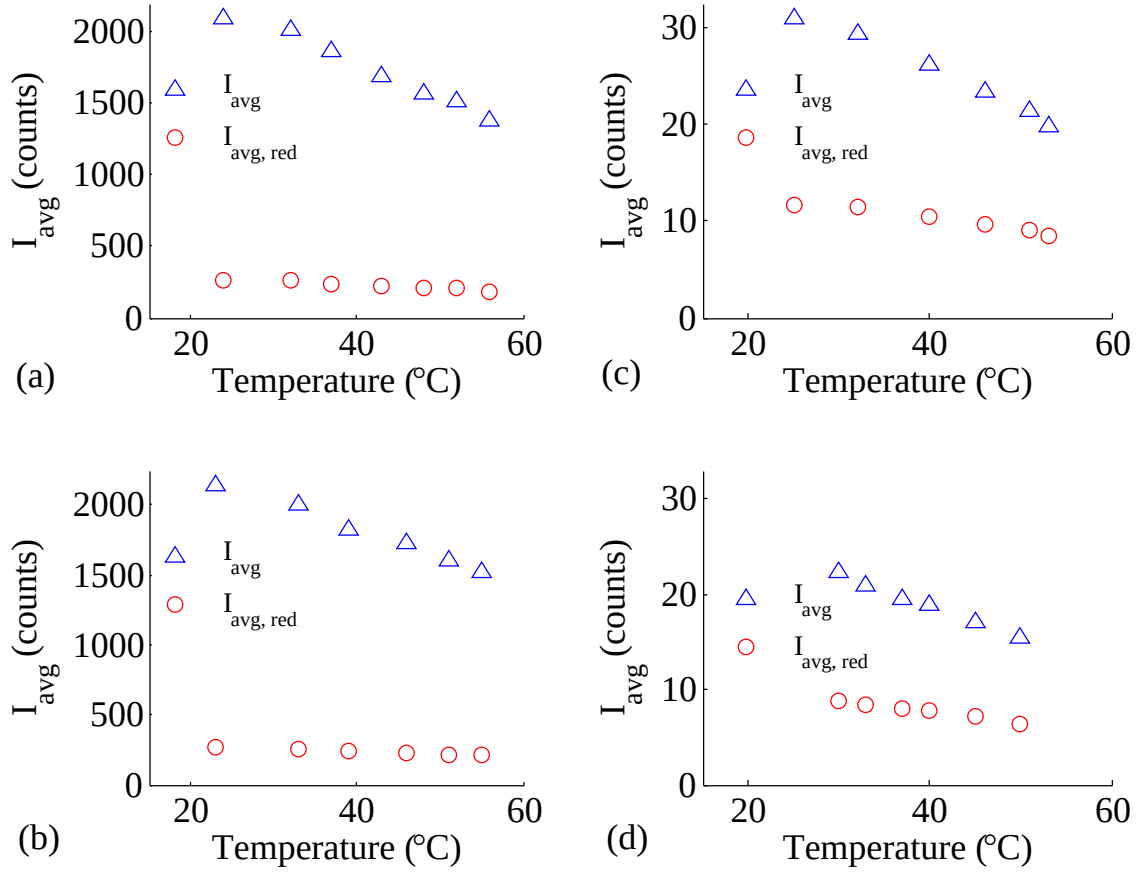


Figure 6.7: Spatial-average fluorescence signal, I_{avg} , versus temperature for two experiments using an iCCD camera (a & b) and two using a CCD camera (c & d). Also included in each plot is the spatial average through the red channel, $I_{avg, red}$.

$\lambda > 305$ nm is shown in fig. 6.8b. For the current data, at 24°C the ratio of maximum fluorescence through the red channel, $I_{max, red}$ to that of the full spectrum, I_{max} , is $I_{max, red}/I_{max} = 0.13$. This lies within the expected values of fig. 6.8b where the fluorescence above 300 nm should be somewhere between 0 and 0.2 times the maximum fluorescence. Integrating the fit for 25°C in fig. 6.8b, gives a value of 0.16. Considering that the transmission fraction of the 305 nm filter (red line in fig. 6.8a) increases from 0.7 to 0.92 between 305 and 340 nm (and hence the integrated value should be a little lower than 0.16), these values match very closely.

The sensitivity and consistence of the relationship between I and T can be examined by normalizing the data as a function of temperature. The average LIF intensity, normalized for a reference temperature, I^* , can be expressed as

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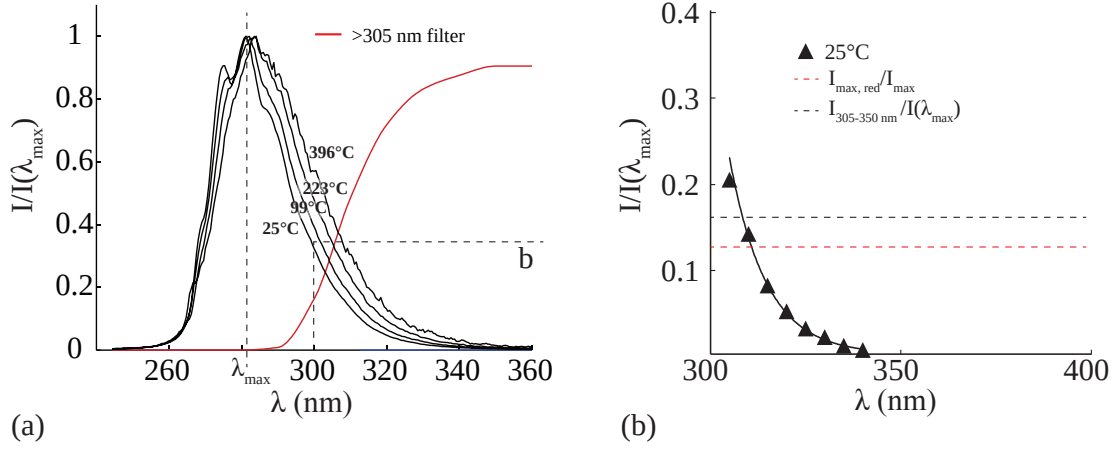


Figure 6.8: (a) Toluene fluorescence (excited at 266 nm) spectrum normalized to the wavelength of maximum or peak fluorescence ($I/I(\lambda_{max})$) from Miller and Gamba [31] for different temperatures and (b) an exponential fit to the spectrum at 25°C for $\lambda > 305$. Also shown is the transmission fraction of the long pass (> 305 nm) filter used by Miller and Gamba [31] which is the same as that used in this current study. The ratio of maximum fluorescence through the red channel, $I_{max,red}$ to that of the full spectrum, I_{max} , determined in this study, for a similar temperature of 24°C and the integrated value from the fit, $I_{305-350nm}/I(\lambda_{max})$ are shown in b.

$$I^* = \frac{I_{avg}(T)}{I_{avg}(T_{ref})} = \frac{I_0 C \Phi(T) \sigma(T) \gamma}{I_0 C \Phi(T_{ref}) \sigma(T_{ref}) \gamma} = \frac{\Phi(T) \sigma(T)}{\Phi(T_{ref}) \sigma(T_{ref})} = f(T) \quad (6.1)$$

where $T_{ref} \cong 24^\circ\text{C}$. Fig.6.9a shows a plot of I^* versus temperature for various experiments with the iCCD and CCD camera. A reasonable level of consistency is demonstrated here. Also included in this plot are error bars which indicate the range of uncertainty of I^* based on the estimates from section 5.2. It is worth noting that this temperature scale is significantly smaller than most LIF studies with toluene to date (see Table 3.1) and there is good sensitivity between I and T . The intensity seems to decrease more for the CCD camera but it is suspected that this is due to some minor laser sheet attenuation.

An original study was performed by Koban et al. [141] investigating the fluorescent properties of toluene as a function of temperature. The authors observed the following trend between normalized fluorescence quantum yield, $\Phi^* = \Phi(T)/\Phi(296K)$, for the same excitation wavelength of 266 nm:

$$\Phi^* = 22.5e^{-0.0105T} \quad (6.2)$$

This trend can be compared with the results of the current study. Rearranging equation 6.1

in terms of quantum yield gives

$$\Phi^* = \Phi(T)/\Phi(T_{ref}) = \frac{I(T)\sigma(T_{ref})}{I(T_{ref})\sigma(T)} \quad (6.3)$$

A best fit to the absorption cross-section as a function of temperature, $\sigma(T)$, is provided by Koch [168], which is

$$\frac{\sigma(T)}{10^{-19}} = -3.57 + 0.022T - 1.22 \times 10^{-5}T^2 \quad (6.4)$$

Therefore, Φ^* can be determined from the measurements of fluorescence using equations 6.3 and 6.4. The normalized fluorescence quantum yield, Φ^* , is plotted versus temperature (in Kelvin) for various experiments in fig. 6.9b and also versus the best fit correlation for Φ^* by Koban et al. [141]. Considering the difference in temperature scale at which the experiments were performed, there is some good agreement here. It appears, however, that the toluene fluorescence quantum yield measured from the current work is more sensitive to temperature compared to that of Koban et al. [141]. It is possible that the fit by Koban et al. [141] and the fit of $\sigma(T)$ (equation 6.4) by Koch [168] don't hold as well for a lower temperatures.

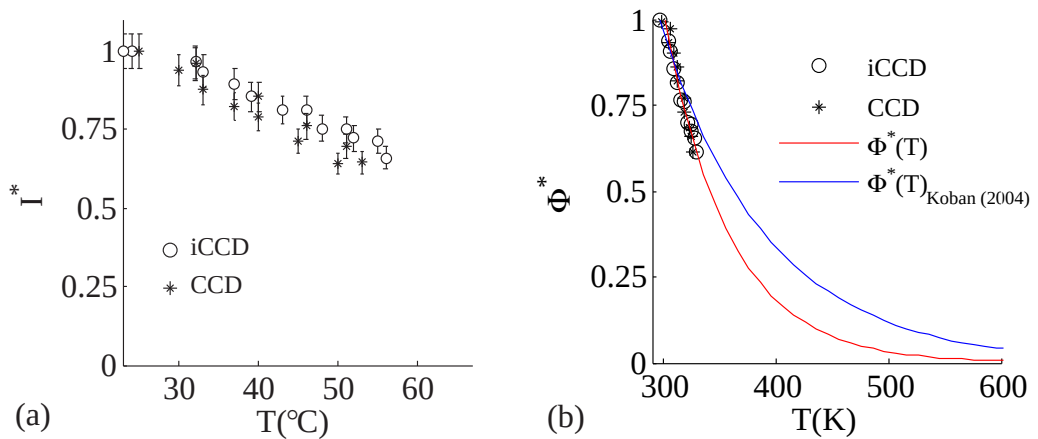


Figure 6.9: (a) Normalized fluorescence intensity, $I^* = I_{avg}(T)/I_{avg}(T_{ref})$, and (b) normalized fluorescence quantum yield, $\Phi^* = \Phi(T)/\Phi(T_{ref})$, versus temperature for various experiments with the iCCD and CCD camera. Also included in (a) are error bars which indicate the range of uncertainty for the I^* measurements. Φ^* (b) is plotted with an exponential fit, $\Phi^*(T)$, from the current data and a fit (equation 6.2) by Koban et al. [141], $\Phi^*(T)_{Koban(2004)}$.

The signal ratio, R_I , can be found by dividing the images through the ‘blue channel’ by the images through the ‘red channel’ (fig. 6.5e - h and fig. 6.4e - h). The blue channel, in this case, is the 275 - 375 nm filter combination (fig. 6.5a - d and fig. 6.4a - d) minus the 305 - 375 nm filter, giving a wavelength band of 275 - 305 nm. Note the same intensifier gain was used for blue and red channels here so this subtraction can be made. The fluorescence intensity through the blue channel, $I_{blue}(x,y)$, and through the red, $I_{red}(x,y)$, are first normalized for a reference temperature. This gives

$$R_I(x,y;T) = \frac{I_{blue}(x,y;T)/I_{blue}(x,y;T_{ref})}{I_{red}(x,y;T)/I_{red}(x,y;T_{ref})} \quad (6.5)$$

which is solely a function of temperature, T . Since the temperature in the test section is uniform, such that $T(x,y) = T$, the plots of $R_I(x,y)$ are effectively flat or plain (see Appendix C3) such that $R_I = R_I(x,y)$. Therefore R_I can be taken as the spatial-average of $R_I(x,y)$ for this experiment.

The signal ratio, R_I , is plotted versus temperature (in Kelvin) for several experiments in fig 6.10. Also included in the plot is a 2nd order polynomial fit made from a curve of signal ratio versus temperature by Jainski et al. [150] for toluene fluorescence excited at 266 nm. Over the temperature range of this study, this relationship is effectively linear. While the error bars, determined from section 5.2, show a larger range of uncertainty, there is an evident inverse relationship between R_I and T . The data also agrees well with a fit between R_I and T by Jainski et al. [150]. Though the signal ratio, R_I , is calculated from spatially-averaged values of fluorescence ($I_{blue,avg}$, and $I_{red,avg}$), these results suggests that two-dimensional maps of temperature, $T(x,y)$, may be determined for low temperature ranges such as those in this experiment. By optimizing the imaging sensitivity and efficiency and by maximizing the excitation/laser energy, the inaccuracy of the temperature measurements can be minimized.

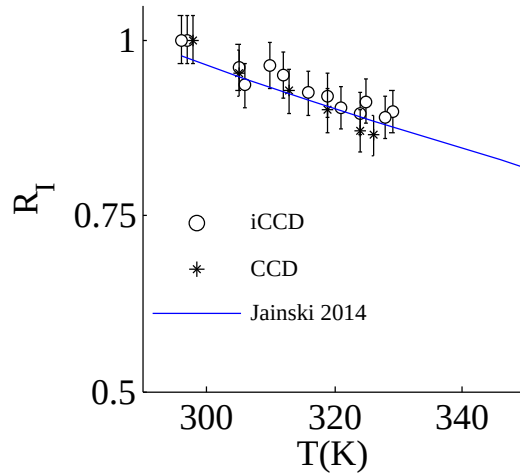


Figure 6.10: The signal ratio, R_I , versus temperature (in Kelvin) for a collection of a data using the iCCD and CCD cameras. The error bars indicate the expected uncertainty range for the R_I measurements. Also included in the plot for comparison, is a 2nd order polynomial fit between signal ratio and temperature by Jaini et al. [150] for toluene fluorescence excited at 266 nm.

6.2 Application of LIF temperature field measurement: Convection from a heated cylinder

This section presents temperature fields around a heated cylinder for both natural and forced convection. The technique is examined for a range of temperature differences, gas flow rates and also the intensifier gate settings are varied, with a view to determining how the technique can be optimized for low temperature ranges. Temperature fields surrounding a drying porous cube are later presented in chapter 7. These experiments have also been performed using anisole as a fluorescent tracer material. Anisole has only recently been applied as a gaseous LIF tracer [149, 151, 153, 170]. Its suitability as an alternative tracer to toluene for low temperature ranges is also discussed.

A copper cylinder (described in section 4.3 in more detail) is heated to surface temperatures ranging between $T_s = 30 - 130^\circ\text{C}$. The free-stream temperature of the nitrogen gas is typically about $19 - 23^\circ\text{C}$. Therefore, the temperature difference between the copper cylinder surface and the free-stream gas, $\Delta T = T_s - T_\infty$, ranged from $7 - 110^\circ\text{C}$.

The experiments were performed for toluene and anisole seeding concentrations (mass percentage) of $w_{tol} = 1 - 3\%$ and $w_{ans} = 0.5 - 0.9\%$, respectively. Since anisole has a much lower vapour pressure, lower concentrations were required to avoid condensation. For

natural convection experiments, the Rayleigh number ranged from $Ra = 3600 - 18500$. The Rayleigh number is the non-dimensional term that governs natural convection, such that

$$Ra = GrPr = \frac{g\beta(T_s - T_\infty)D_{cyl}^3}{\nu^2}Pr \quad (6.6)$$

where Gr is the Grashof number which represents the ratio of buoyant to viscous forces. β is the coefficient of volume expansion. This is simply $1/T_{film}$ for ideal gases, where $T_{film} = (T_s + T_\infty)/2$ is the mean film temperature. D_{cyl} is the diameter of the cylinder.

The forced convective experiments were performed at Reynolds numbers of $Re = V_\infty D_{cyl}/\nu = 530, 1150$ and 1460 , where V_∞ is the average free-stream gas velocity.

6.2.1 Single-colour technique

First, determination of temperature fields using a single wavelength channel (single colour) is discussed. Fig. 6.11a shows a sample plot of fluorescence (through the 275 - 375 nm filter combination) intensity, $I(x,y)$, in the flow field of natural convection ($Re = 0$) from a cylinder. The fluorescence intensity measured here is averaged over 10 images. The dashed circle marks the position of the cylinder cross section. To the right of this, a shadow can be seen where the laser sheet is blocked by the cylinder.

This image is for one of the larger heat inputs (in the range of this study) to the cylinder of $\dot{Q} = 2.15$ W. This produces a stabilized cylinder surface temperature of $T_s = 94.8^\circ\text{C}$ (measured by the thermocouple positioned inside cylinder) and hence a temperature difference of $\Delta T = 73.8^\circ\text{C}$ (or Rayleigh number, $Ra = 15200$). It can be seen in fig. 6.11a that the fluorescence intensity decreases in the higher temperature region above the cylinder where there is a buoyancy-induced convection current. An isothermal image ($\dot{Q} = 0\text{W}$) is shown in fig. 6.11b for comparison. This effect is more clear in fig. 6.11c which demonstrates the fluorescence intensity normalized for a reference temperature, $I^*(x,y) = I(x,y)/I(T_{ref})$. This normalization is achieved by dividing the fluorescence image by the isothermal image, fig. 6.11b, where $I(x,y) = I(T_{ref})$ at all points (x,y) .

A temperature field, $T(x,y)$ can be determined from $I^*(x,y)$ by using the relationship between temperature and normalized fluorescence (see equation 6.1). A second-order polynomial fit between I^* and T made from the data in fig. 6.9 is used to calculate $T(x,y)$. For

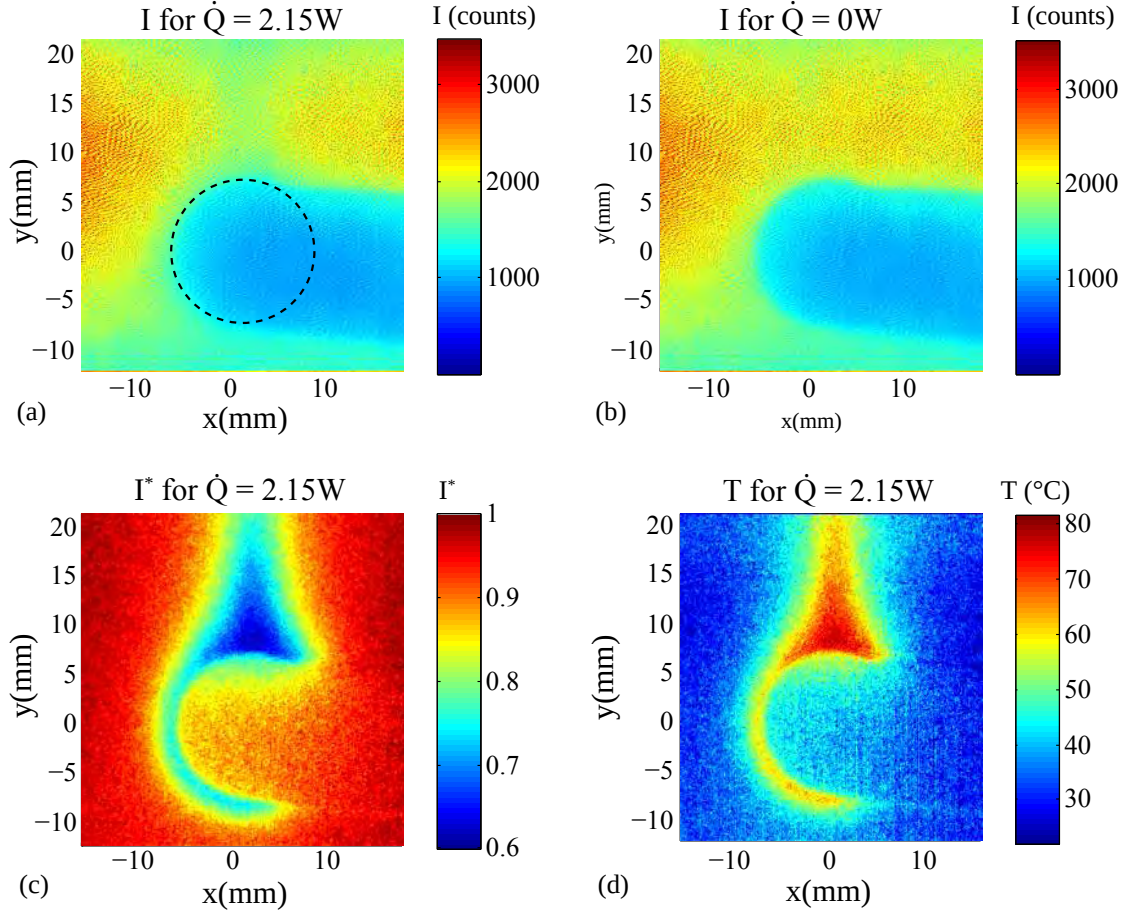
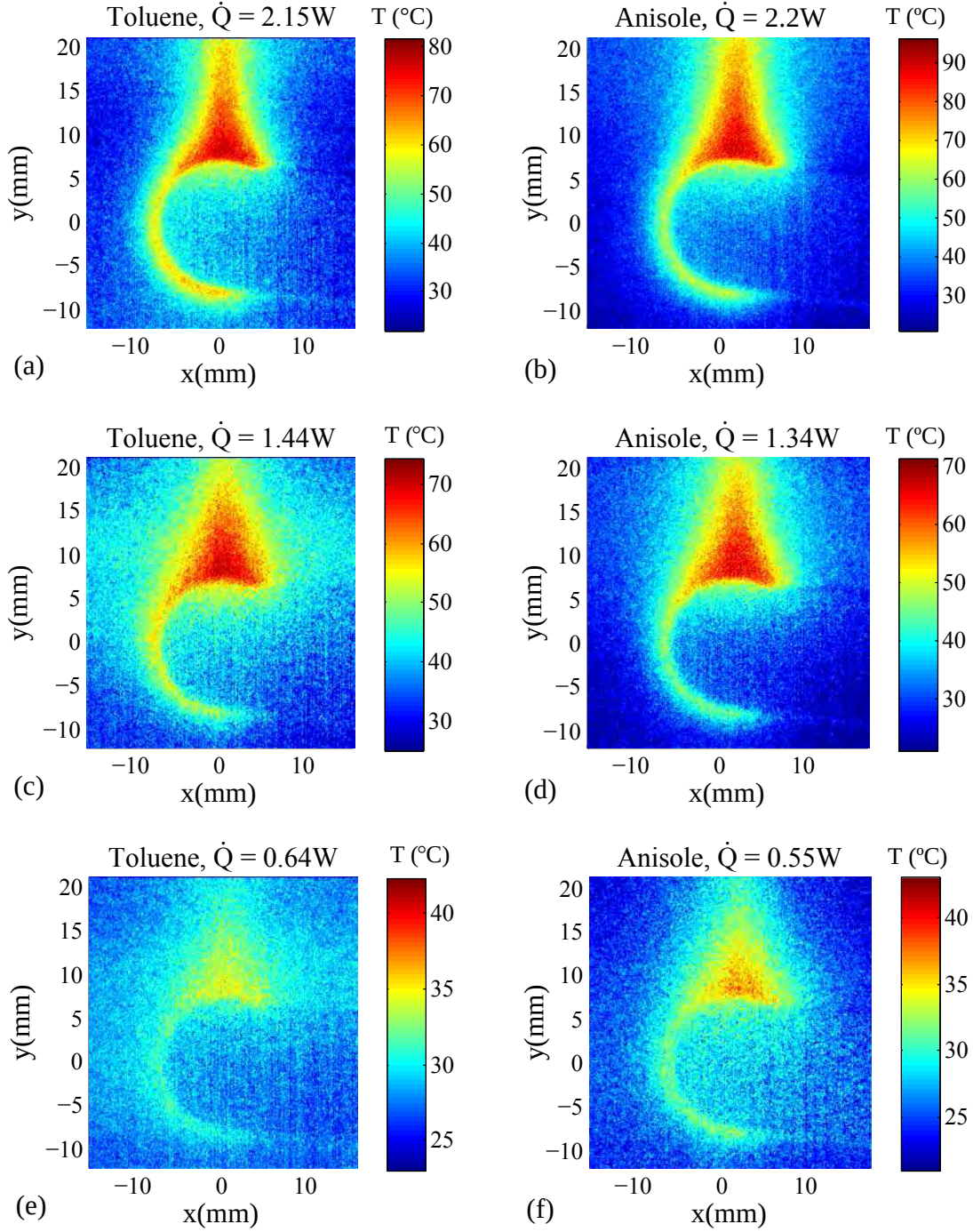


Figure 6.11: Sample images of fluorescence intensity, $I(x,y)$, surrounding a cylinder for (a) a heat input to the cylinder of $\dot{Q} = 2.15\text{W}$ ($Ra = 15200$) and for (b) $\dot{Q} = 0\text{W}$ (i.e. isothermal flow-field where $T(x,y)$ equals a uniform reference temperature, $T_{ref} = 21^{\circ}\text{C}$, at all points). The dashed circle marks the position of the cylinder cross-section. Dividing image (a) by image (b) gives the (c) normalized fluorescence intensity, $I^*(x,y) = I(x,y)/I(T_{ref})$ from which the (d) temperature field, $T(x,y)$, can be determined.

larger temperature differences, a fit from data by Koban et al. [141] and Koch [168] is used as it is found to predict higher temperatures more accurately. The temperature field is shown in fig. 6.11d. The typical heat transfer characteristics of natural convection from a cylinder can be seen here: a boundary layer that extends around the circumference of the cylinder and the convection current or plume on the top surface caused by buoyancy. Note, since the right hand side of the cylinder surface lies in shadow of the laser sheet, the boundary layer cannot be seen but is expected to be symmetric to the left side.

6.2. APPLICATION OF LIF TEMPERATURE FIELD MEASUREMENT: CONVECTION FROM A HEATED CYLINDER



Figs. 6.12a, c & e show temperature fields for progressively smaller temperature differences, $\Delta T = T_s - T_\infty$. This is shown, likewise, for measurements performed using anisole as a tracer in figs. 6.12b, d & f. Note that, depending on the ambient room temperature, the heat input may vary slightly to generate the same (or similar) temperature differences. For temperature measurements using anisole, the calibration fits available [151, 153] do not predict low temperatures with much precision and are more suited for large ranges. Therefore, a two-point calibration between the maximum temperature at the surface and the minimum temperature in the free-stream gas with the minimum and maximum values of I^* , respectively, is used to map $I^*(x, y)$ to $T(x, y)$ for anisole. This is acceptable since the relationship between I^* and T is effectively linear over the temperature range considered in these experiments.

The error of the temperature measurement in figs. 6.12 ranges from 2°C to 10°C for the smallest and largest temperatures, respectively. Table. 6.2 compares this error with previous studies performed at larger temperature scales. Given some of the limitations of the laser-imaging setup, the precision here is quite reasonable. By making refinements to the setup (these are discussed in detail in section 8.1), this error could be reduced significantly. This makes LIF thermometry for low temperature measurement quite promising. Depending on the stability of the heat transfer problem, some authors used single images and others took an average of several images, as shown in table. 6.2. Increasing the number of averages generally increases the precision of the temperature measurements. By using high speed imaging devices that allow a large number of averages to be taken over a short period of time, the error can be reduced considerably whilst maintaining good temporal resolution.

Table 6.2: Comparison of temperature measurement error from this study with that of previous literature. The second column represents the number of images which the measurements are averaged over, from which the error is determined from (1 for single image measurements). The temperature range of each study is included in the third column.

Author	Averages	Range	Error
Current study	10	20 – 95°C	2 – 10°C
Luong et al. [28]	30	25 – 427°C	4 – 7°C
Strozzi et al. [29]	50	27 – 677°C	3 – 35°C
Trunk et al. [34]	1	100 – 600°C	8 – 25°C
Peterson et al. [147]	72	22 – 227°C	5 – 29°C
Zegers et al. [145]	1	27 – 727°C	7 – 12°C
Jainski et al. [150]	1	20 – 227°C	9 – 20°C

It's clear in fig. 6.12 that both tracers appear to be suitable for temperature measurement. Anisole is reported by Faust et al. [149] to have a less sensitive temperature response for low temperature ranges. However, the authors report that its quantum efficiency, Φ , is 60% greater than that of toluene and it also has a much greater absorption cross section. This may explain why there is less noise in fig. 6.13f compared to fig. 6.13e.

It can be seen in fig. 6.12 that for temperature differences as low as $\Delta T = 25^\circ\text{C}$ (6.12e & f), the convection current from the cylinder can be demonstrated. The noise in these temperature field measurements is quite significant. This suggests that the lack of sensitivity in this temperature scale may limit the resolution of temperature field measurements for particularly low temperature convection problems with the current LIF setup. However, the signal to noise ratio can be maximized by using a higher laser pulse energy and more efficient detection optics and imaging devices, which in turn would improve the precision and resolution of the images.

Ensuring a completely oxygen-free environment will also ensure maximum sensitivity at the low temperature ranges. Air pockets may have existed after purging of the container which may cause a small concentration of O_2 . The presence of even a small concentration of O_2 molecules can shift the relationship between I (and hence I^*) and T significantly. This is evident in fig. 6.13, taken from Faust et al. [149], which show plots of normalized fluorescence quantum yield, Φ^* , versus temperature for toluene and anisole in pure N_2 (fig. 6.13a) and air (fig. 6.13b). The sensitivity of Φ^* with temperature, in the range $295\text{K} < T < 600\text{K}$, decreases significantly as the amount of O_2 molecules approaches that of air.

Another method for improving the sensitivity of temperature measurement may be to make use of the temperature-dependent time response of the fluorescence signal. It is known that the fluorescence lifetime decreases exponentially with temperature [141, 152, 153, 156]. This is shown in fig. 6.14a which shows the time-normalized signal (for toluene excited at 266 nm [152]) and how it shortens with temperature (on a logarithmic scale). A method is proposed here which involves setting up the camera gating or exposure such that it captures a later portion of the signal (i.e. it cuts off the front edge of the signal). This is shown in fig. 6.14b. Cutting off the early portion of the signal means that the signal not only decreases in amplitude for increasing temperature but also less of the signal is captured as the temperature increases. Therefore the decrease in fluorescence intensity with temperature

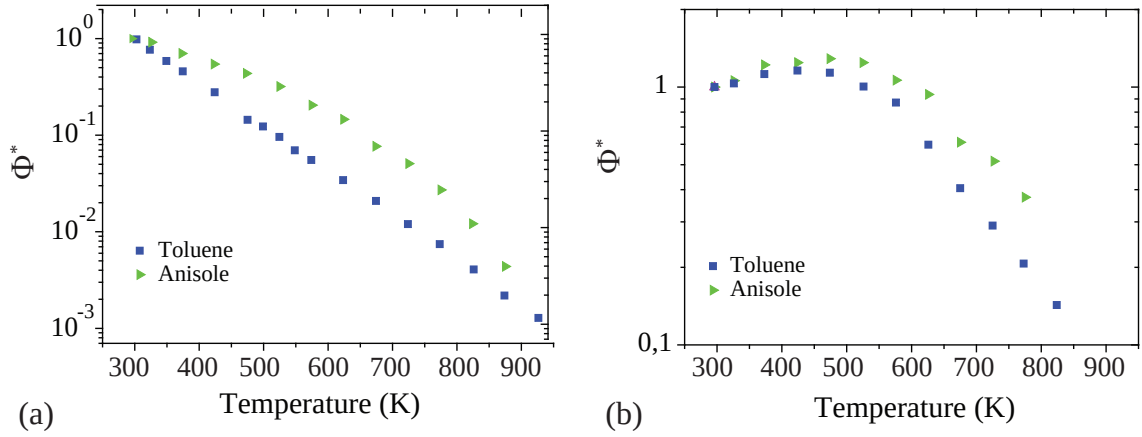


Figure 6.13: Temperature dependence of the (normalized) fluorescence quantum yield, Φ^* , of toluene and anisole for excitation at 266 nm in (a) pure nitrogen and (b) air normalized to the value at the lowest investigated temperature. Figure taken from Faust et al. [149] and data from [153, 156, 167, 171].

should become more pronounced.

To test this method, some toluene LIF measurements were performed for an increased delay to the intensifier gate (exposure time). The gate delay - originally at a value of $gd = 20$ ns (as in fig. 6.12) such that the gate overlaps the full toluene fluorescent signal - was increased to larger values between $gd = 30$ to 120 ns which capture progressively less amounts of the signal. The increase in sensitivity only becomes significant as the delay is shifted towards the trailing end of the fluorescent decay. This is shown in fig. 6.15 which demonstrates the normalized fluorescence intensity, $I^*(x,y)$, for different intensifier gate delays. As the delay

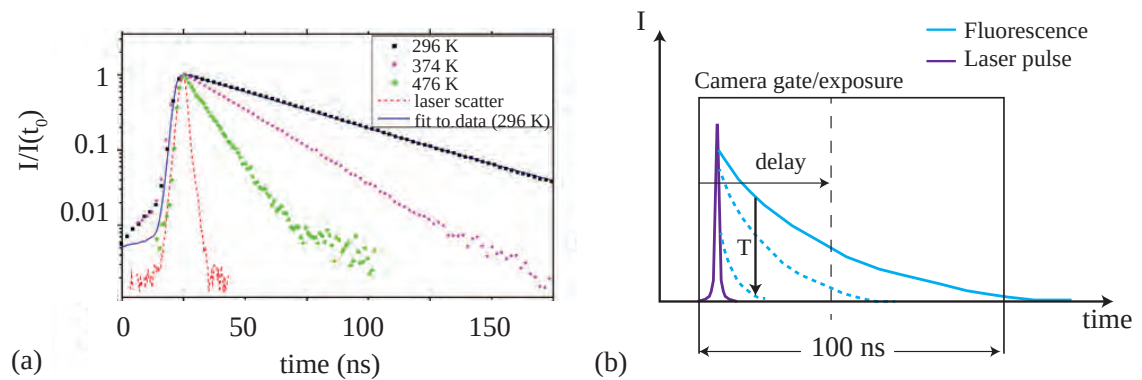


Figure 6.14: Plot of (a) decreasing fluorescence lifetime of toluene with increasing temperature as shown by Faust et al. [152] and illustration (b) of gate delaying method for increasing I vs T sensitivity.

6.2. APPLICATION OF LIF TEMPERATURE FIELD MEASUREMENT: CONVECTION FROM A HEATED CYLINDER

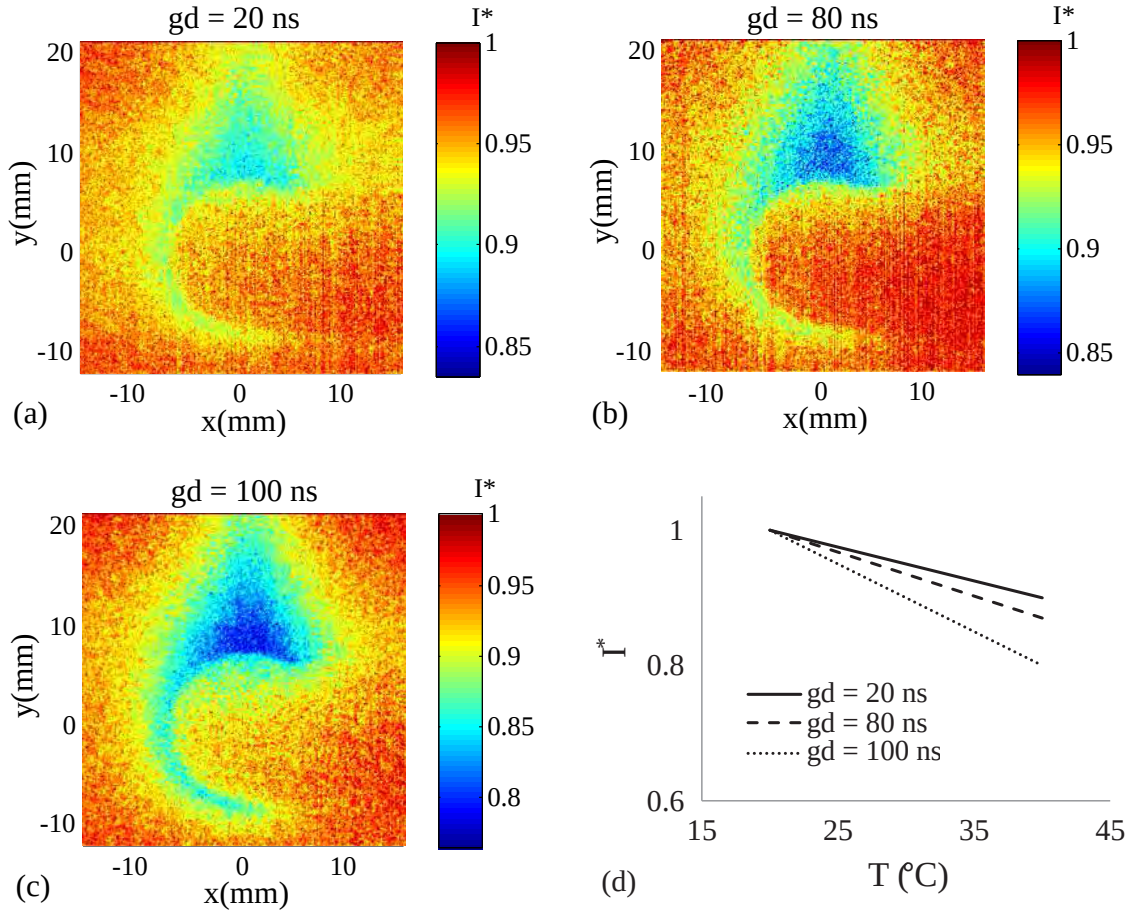


Figure 6.15: Normalized fluorescence intensity, $I^*(x,y)$, surrounding a heated cylinder, for a delay to the intensifier gate of (a) 20 ns (beginning of fluorescence), (b) 80 ns (60 ns from beginning of fluorescence) and (c) 100 ns (80 ns from beginning of fluorescence). The change of I^* with temperature for each gate delay is also plotted in (d), demonstrating an increased sensitivity for longer delays.

is increased, the fluorescence intensity in the boundary layer is clearly decreasing by a more significant amount. The change in normalized fluorescence intensity with temperature (i.e. the sensitivity) increases from 0.004°C^{-1} to 0.008°C^{-1} . This is plotted in fig. 6.15d.

Temperature field measurements are shown in fig. 6.16 for the same intensifier gate delays and for two low heat inputs. The gate delay is 20 ns in figs. 6.16a & b, 80 ns in figs. 6.16c & d and 100 ns in figs. 6.16e & f. It can be seen that the increased sensitivity improves the accuracy of the temperature field measurements. Even for a temperature difference as low as $\Delta T = 11^\circ\text{C}$, although there is a lot of noise, the convection current can be demonstrated for the longest gate delay of $gd = 100$ ns (fig. 6.16f). Note that because the relationship between I^* and T is being altered by imaging only part of the signal, the fit used to calculate $T(x,y)$

for the full fluorescence signal, is not valid. Therefore, a two-point calibration (as performed for anisole) had to be used in order to determine these temperature fields.

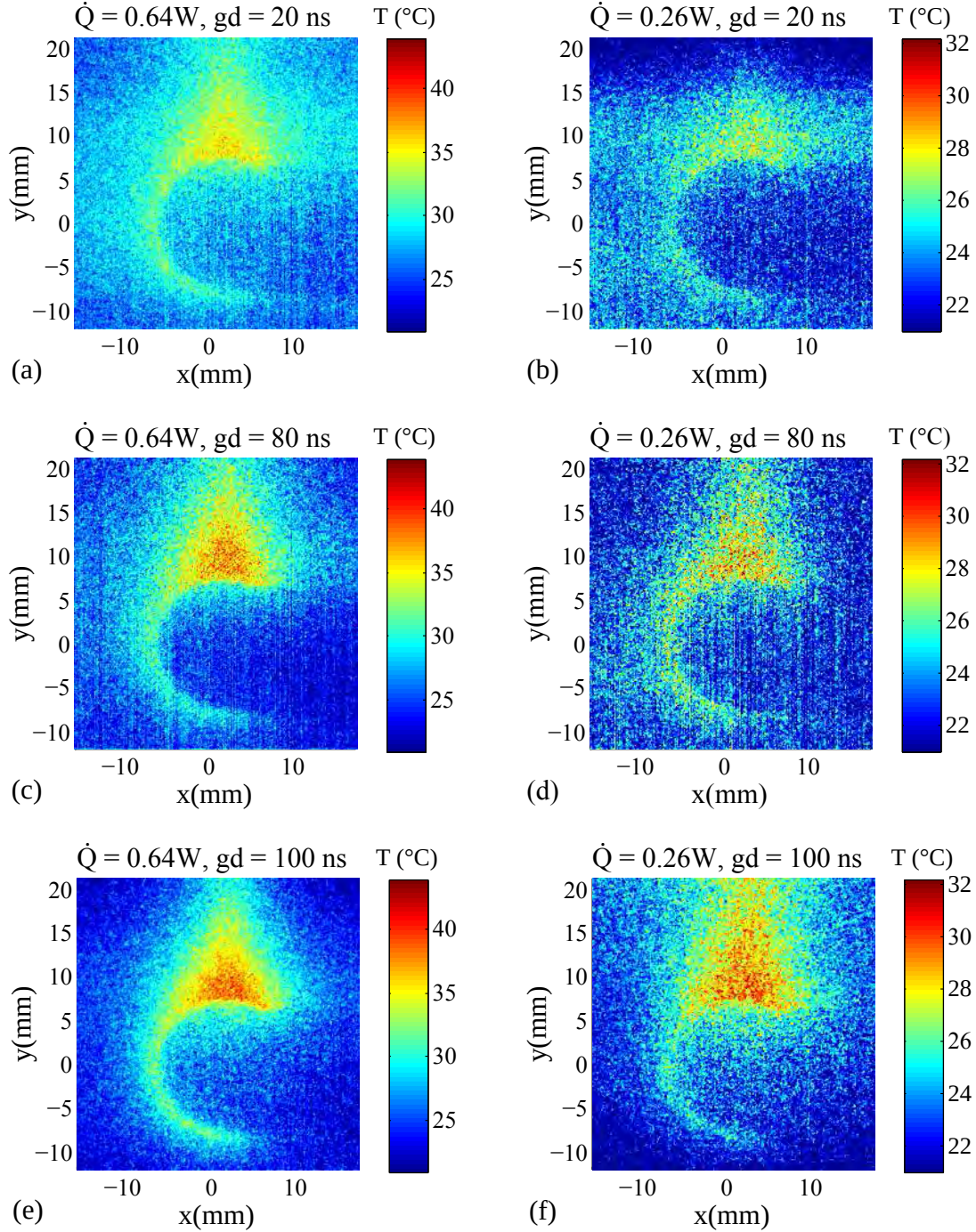


Figure 6.16: Temperature field surrounding a heated cylinder determined from single colour toluene fluorescence, for a delay to the intensifier gate of (a, b) 20 ns (beginning of fluorescence), (c, d) 80 ns (60 ns from beginning of fluorescence) and (e, f) 100 ns (80 ns from beginning of fluorescence). The heat inputs are (a, c, e) $\dot{Q} = 0.64\text{W}$ (or $Ra = 6900$) and (b, d, f) and $\dot{Q} = 0.26\text{W}$ (or $Ra = 3600$).

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Using anisole as a tracer, this method was more difficult to evaluate due to the shorter lifetime of anisole fluorescence [153]. It can be concluded from fig. 6.16, that in convection experiments which involve significantly low temperature ranges and in which the laser pulse energy and imaging efficiency are limited, this novel method may be very useful technique for acquiring more accurate temperature fields. By averaging over a greater number of images, the accuracy can be increased somewhat, as shown in fig 6.17 which shows a temperature field found using 30 averages. Since the iCCD camera frame rate was limited to 0.2 Hz, it was difficult and impractical to use this number of averages throughout the experiments. However, in steady-state experiments where single measurements are required for a specific set of parameters or by using a high-speed camera, this would be a lot more feasible.

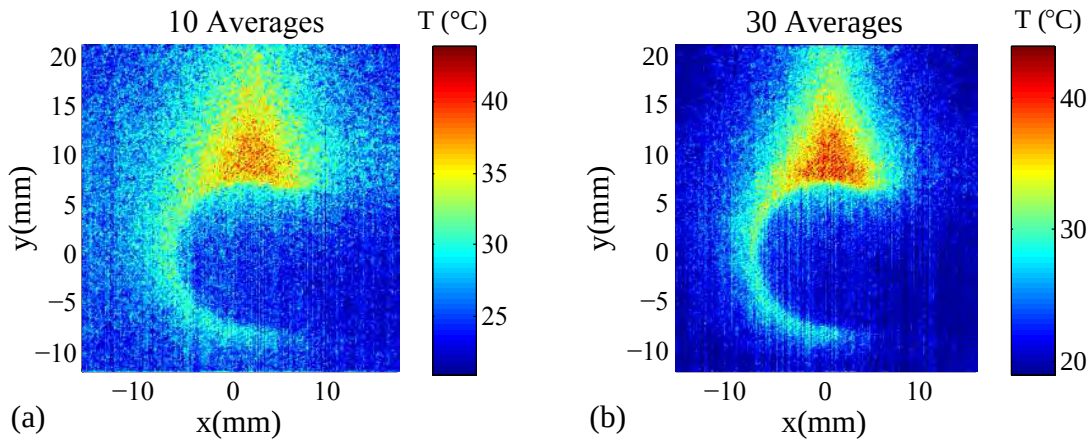


Figure 6.17: Temperature field surrounding a heated cylinder determined from single colour toluene fluorescence by averaging over (a) 10 images and (b) 30 images. These plots were determined for a gate delay of $gd = 80$ ns and a heat input of $\dot{Q} = 0.64$ W (or a Rayleigh number of $Ra = 6900$).

Fig. 6.18 shows a comparison of a LIF temperature field measurement (using toluene) with that of a numerical solution (CFD simulation) of natural convection from a cylinder by Atayilmaz and Teke [172]. The Rayleigh numbers for both temperature fields are different ($Ra = 6900$ and 3600 , respectively) but are of a similar order of magnitude. Some median-filtering is applied to the temperature field in fig. 6.18a. The temperature distributions appear quite similar. However, a noticeable difference can be seen in the boundary layer at the bottom half of the cylinder. The CFD plot in fig. 6.18b demonstrates a temperature close to the surface temp in the thin layer near the surface, unlike the LIF measurements in fig. 6.18a. It is likely that the limited resolution of the imaging setup cannot accurately detect this

thin layer in fig. 6.18a, whereas with a CFD model, the spatial resolution would not be limited. The temperature distributions do, otherwise, demonstrate good qualitative agreement, especially given the small temperature range of the LIF measurements compared to previous literature (see table 3.1).

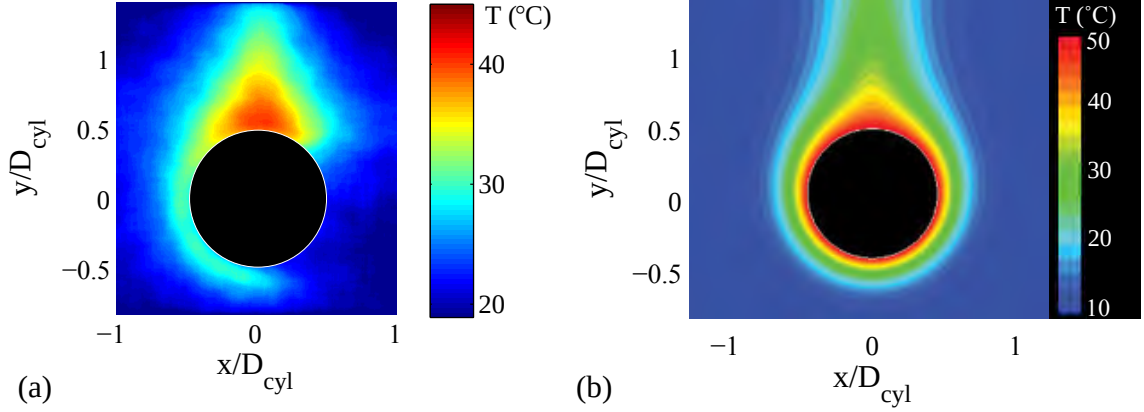


Figure 6.18: Temperature field around a heated cylinder determined using (a) toluene LIF single colour measurement and (b) a CFD simulation [172]. The Rayleigh numbers in (a) and (b) are $Ra = 6900$ and 3600 , respectively.

6.2.2 Two-colour technique

As discussed in section 3.3, a more desirable LIF technique for temperature field measurement is taking the ratio of the fluorescence intensity through two wavelength channels (or ‘colours’) with different responses to temperature. As in section 6.1 the ratio of the (temperature-normalized) fluorescence intensities $I_{blue}^*(x, y; T)$ and $I_{red}^*(x, y; T)$ was measured to give

$$R_I(x, y; T) = \frac{I_{blue}^*(x, y; T)}{I_{red}^*(x, y; T)} \quad (6.7)$$

which is solely a function of temperature. Difficulties in getting good spatial resolution from the camera when using the image doubling system (see fig. 4.8) - due to reflections from the edges of the 50/50 beamsplitter and UV mirror - meant that separate (non-instantaneous) images were taken as in section 6.1 for blue and red channels.

Since the heat transfer problem in consideration is in a steady-state, the temperature field is not expected to be changing significantly with time. Therefore, by taking several

averages, the two-colour technique can still be applied here, though with a loss in temporal resolution. The temporal resolution is greater when taking the same instantaneous images through the blue and red channel, because the fluorescence distribution across x and y would be almost identical, only the magnitudes would be different. Taking images at different instances of time means the fluorescence distribution will not be identical and hence R_I will deviate or fluctuate more across x and y . Sample images of normalized fluorescence intensity, $I^*(x, y)$, are shown in figs. 6.19a & b, for the blue and red channel, respectively. There is a significant amount of noise in the red channel image, fig. 6.19b, compared to that of the blue channel, fig. 6.19a. For the experiments in section 6.1, this did not affect the accuracy of measurements, since spatial averages were typically taken. However, for these measurements, the spatial deviations or variance makes it difficult to outline more precisely the boundary layer and convection current of the cylinder as can be seen, in fig. 6.19b. It is important to note here that the position and size of the field of view, when combining the UV imaging lens with the iCCD camera, were limited. This means the camera position, and hence the image resolution, were not fully optimized.

The resulting signal ratio, $R_I(x, y)$, in the flow field of the cylinder is shown in fig. 6.19c and the temperature field calculated from this is in fig. 6.19d. Note, a two-dimensional median filter was necessary to smooth the plots of $R_I(x, y)$. The resulting temperature field has quite low spatial resolution as a result. The convection plume or current rising from the top of the cylinder appears to have a less symmetric and accurate shape, compared to the temperature fields calculated using the single colour technique (fig. 6.11). It's immediately clear that the two-colour technique is a lot more limited within the temperature range of this study compared to the single-colour technique. Nevertheless, there still is a clear convection current in the temperature distributions in fig. 6.19d. Optimization of the imaging and laser system to ensure maximum signal-to-noise ratio, temporal and spatial resolution, may provide more precise and resolute measurements of temperature fields. Fig. 6.20 shows temperature fields for lower heat inputs. At these lower temperature differences, the noise increases and the temperature field becomes significantly less precise across x and y . The convection current can be just about inferred in fig. 6.20a. The error of the temperature measurement in figs. 6.19 and 6.20 ranges from 3.4 to 15°C for the smallest and largest temperatures, respectively.

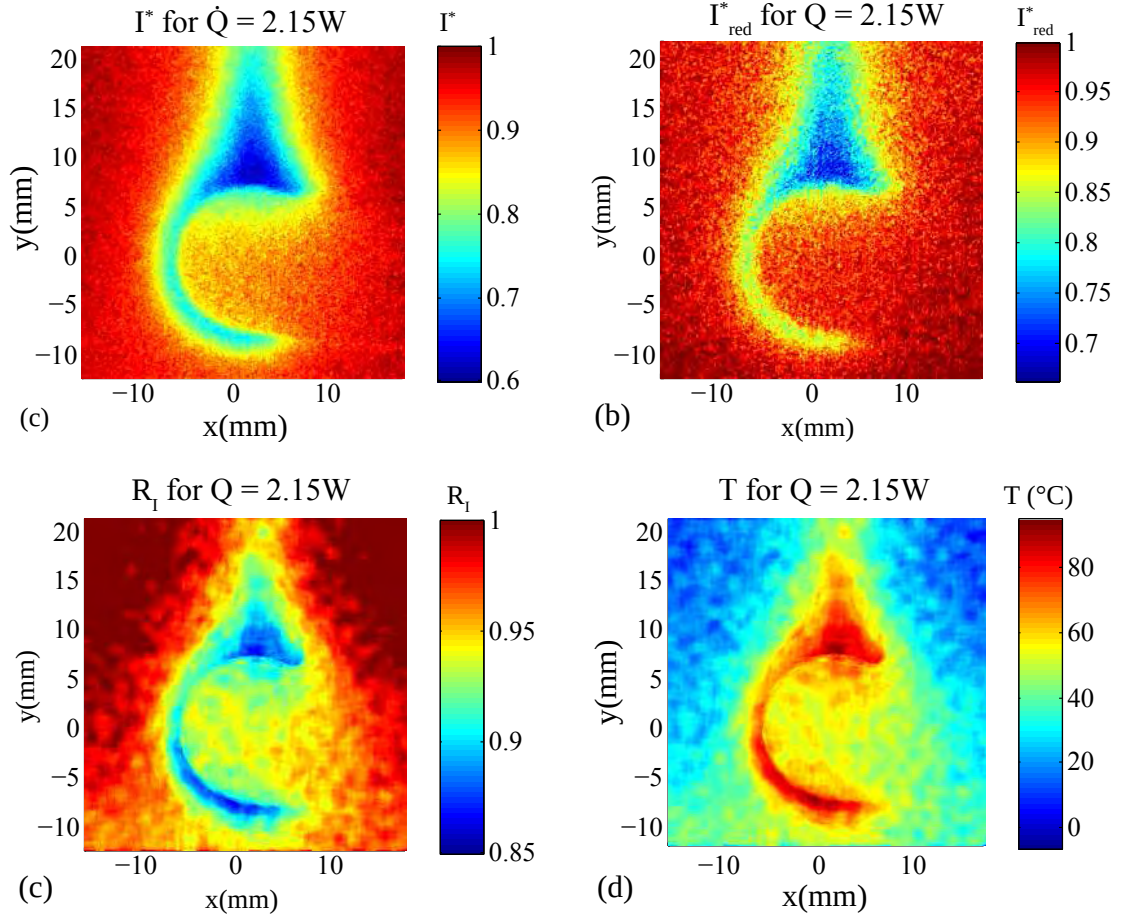


Figure 6.19: Sample measurements using two-colour technique for a heat input to the cylinder of $\dot{Q} = 3.3\text{W}$. The normalized fluorescence intensity, $I^*(x,y)$, for the (a) blue (275 – 375 nm filter combination) channel and for the (b) red (305 – 375 nm filter combination) channel. Dividing the blue channel image, a, by the red channel image, b, gives the (c) signal ratio, $R_l(x,y) = I_{blue}(x,y)/I_{red}(x,y)$ from which the (d) temperature field, $T(x,y)$, can be determined. Note, a 2D median filter is applied to the signal ratio plot. This reduces the spatial resolution in (c) and (d) as a result. The error of the temperature measurement here is up to 15°C .

Though the accuracy of the two-colour technique is rather low here, it is important to note that many steps can be taken to improve the accuracy: Firstly, optimizing the image doubling system (or using two separate cameras for each channel), will increase the temporal resolution. Optimizing the camera sensor selection such that the sensor's quantum efficiency is high in the fluorescence wavelength region will increase the signal-to-noise ratio. Higher transmission filters and imaging lenses will also increase the sensitivity. A more advanced UV lens and optimized focal length will improve the spatial resolution. Finally, using a laser with higher pulse energy would increase the fluorescence signal and hence the signal-to-noise ratio. This study was limited to a pulse energy of 3.6 mJ which is significantly less than

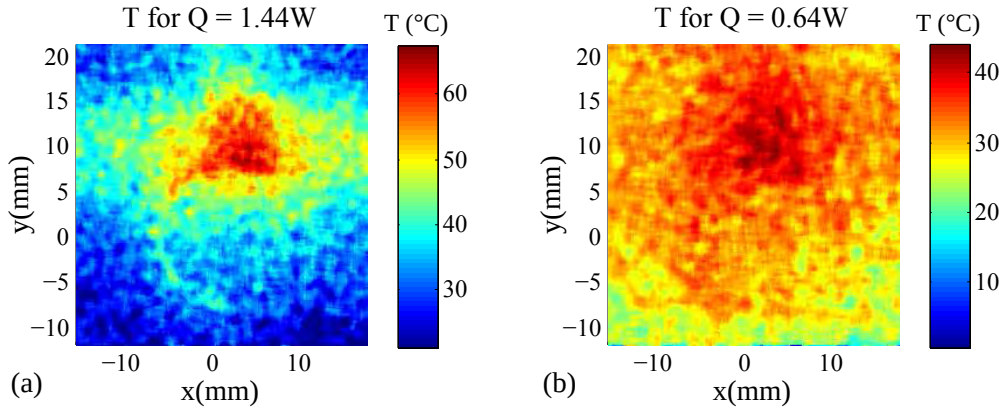


Figure 6.20: Temperature fields calculated from the signal ratio (two-colour technique), R_I , for heat inputs to the cylinder of (a) $\dot{Q} = 1.44\text{W}$ and (b) $\dot{Q} = 0.64\text{W}$. The error of the temperature measurement in (a) and (b) is up to 11°C and 7°C , respectively.

that of most other studies (for example see [28, 29, 31, 33, 148, 150]) where the two-colour technique is applied to much greater temperature ranges. Therefore, while the two-colour temperature measurements are significantly inaccurate here, the images do demonstrate the concept of the technique at low temperatures. It may still be possible to use this technique with an optimized laser-imaging setup.

6.2.3 Local heat transfer analysis using LIF thermometry

While the two-colour technique is quite limited in the temperature range of this study and for the current laser-imaging setup, the single-colour temperature measurements allow for analysis of convective heat transfer phenomena. Fig. 6.21 contains temperature fields for forced convection from a cylinder. The temperature fields shown are for Reynolds numbers of $Re = 530$ (fig. 6.21a & b), 1150 (fig. 6.21c & d) and 1460 (fig. 6.21e & f) and for a constant heat input to the cylinder of $\dot{Q} = 3.4\text{ W}$. The temperature fields in figs. 6.21a, c & e are for toluene and those in figs. 6.21b, d & f are for anisole. The characteristic features of forced convection from a cylinder can be seen in these plots. There exists a thin thermal boundary layer at the front or leading side of the surface, where the fluid flow remains attached to the surface. At the rear side, where the fluid flow becomes separated, the turbulent wake region is represented by wide distribution of temperature due to intense mixing of the fluid.

Fig. 6.21 demonstrates the effect of increasing the Reynolds number (or fluid velocity)

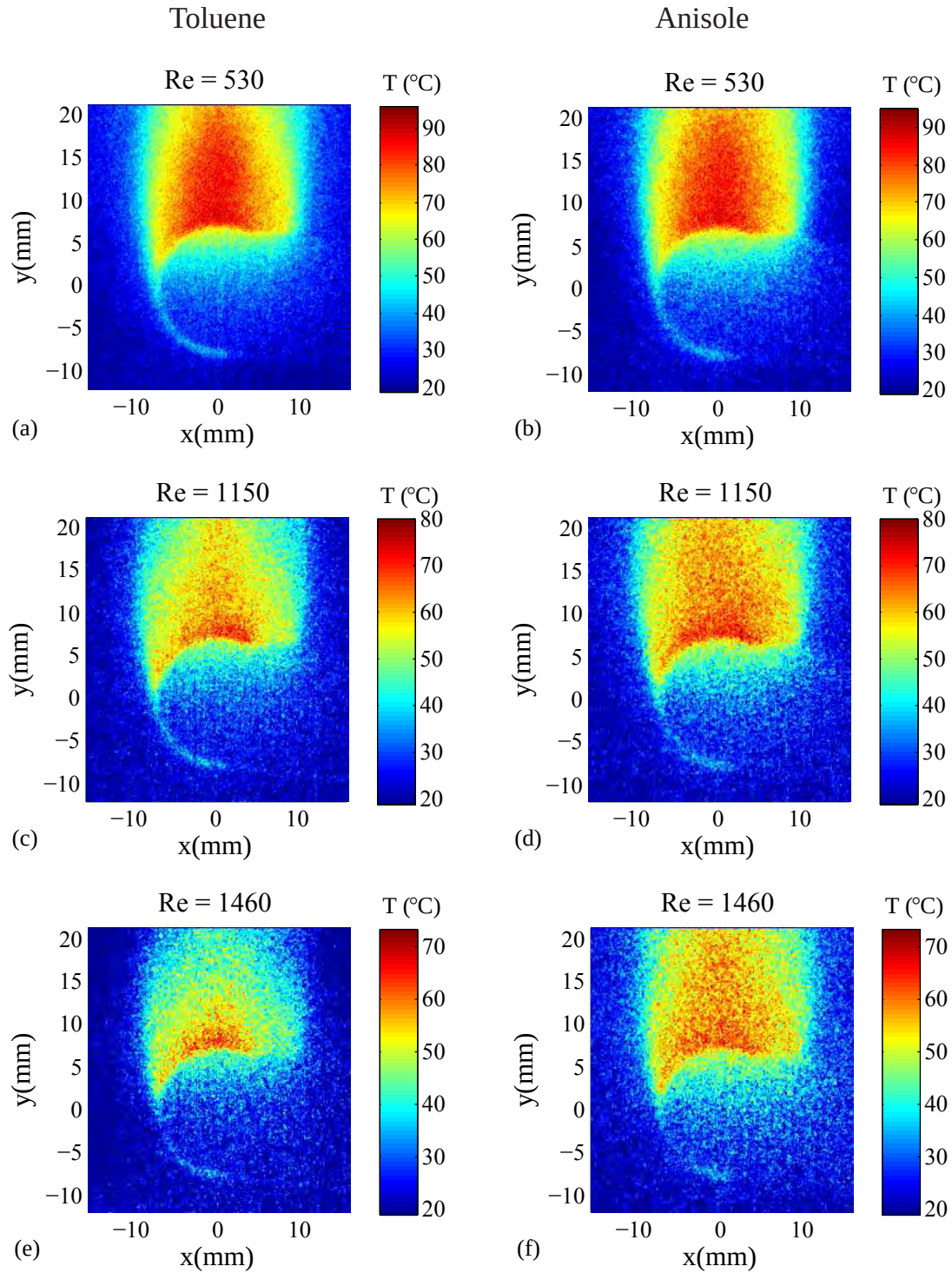


Figure 6.21: Temperature field surrounding a heated cylinder, using single colour LIF thermometry, for various Reynolds numbers, Re . The plots are for a fixed heat input of $\dot{Q} = 3.4\text{W}$. The temperature fields are measured using both (a, c, e) toluene and (b, d, f) anisole as tracer molecules.

of the gas on the temperature distribution around the cylinder. It's well known that higher Reynolds numbers cause the thermal boundary layer to become thinner and increase the rate of convective heat transfer from the surface. This can be seen clearly here by the overall drop in temperature. At the leading edge, the thermal gradient becomes more faint as Re is increased from 530 to 1460. The significant decrease in temperature at the rear side of the cylinder with Reynolds number is due to more intense turbulent mixing.

There appears to be some slight disagreement between the temperature fields determined with toluene (fig. 6.21e) and anisole (fig. 6.21f) as Re is increased. This is possibly because a two-point calibration is not adequate for mapping T to I^* when using anisole as a tracer. This disagreement is also possibly due to the uncertainty of the Reynolds number and heat input settings (see section 5.2) which may have deviated slightly from measurement to measurement. However, the temperature fields of anisole in fig. 6.21, display good sensitivity when compared to toluene (as also previously seen for natural convection in figs. 6.12). Given a more precise calibration, anisole may be a more suitable tracer material for gaseous convective heat transfer experiments.

These local temperature measurements enable detailed analysis of the surface heat transfer. By determining the temperature at the closest point to the surface (i.e. the maximum temperature in the boundary layer), it can be assumed that this is equal to the surface temperature, T_s . The surface temperature distribution along the cylinder circumference (at an angle, θ , from the front stagnation point), $T_s(\theta)$, can then be used to determine the local heat transfer coefficient, $h(\theta)$, such that

$$h(\theta) = \frac{\dot{q}}{(T_s(\theta) - T_\infty)} \quad (6.8)$$

where $\dot{q}$ is the local heat flux at the cylinder surface, $\dot{q} = \dot{Q}/A_s$, which is assumed to be uniform across the surface area, A_s . Distributions of the Nusselt number, $Nu = hD_{cyl}/k$, across the cylinder surface at angles, θ , from the front stagnation point are shown in fig. 6.22. These distributions are for a fixed Reynolds number of $Re = 530$ using both toluene and anisole as a tracer, each for two different heat inputs. The distributions agree quite well with each other, despite the different measurement conditions, illustrating good repeatability.

Also included in fig. 6.22 is the stagnation Nusselt number, $Nu(\theta = 0^\circ)$, calculated from

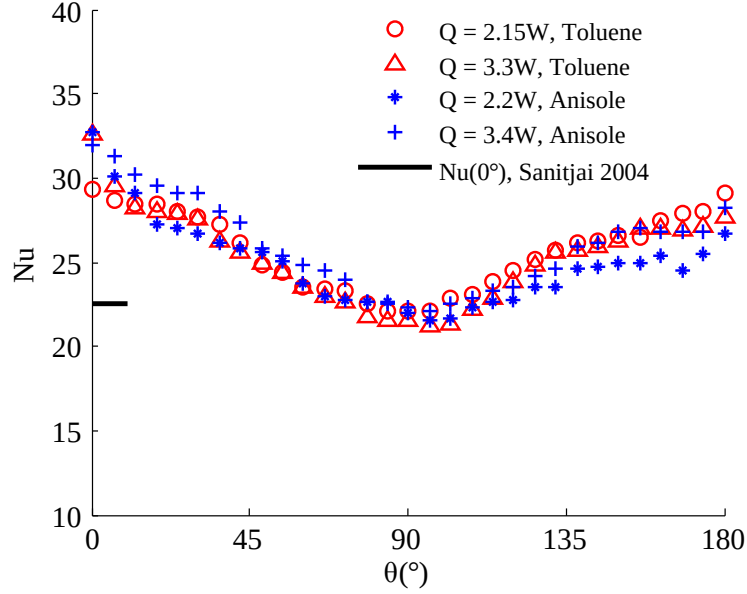


Figure 6.22: Nusselt number, Nu , versus angular position, θ , along the surface of the cylinder (i.e. angle from the front stagnation point) for a fixed Reynolds number of $Re = 530$. This is shown for single colour LIF measurements with both toluene and anisole as a tracer material, each for two different heat inputs. Also included is the stagnation Nusselt number, $Nu(\theta = 0^\circ)$, calculated from a correlation by Sanitjai and Goldstein [173].

a correlation by Sanitjai and Goldstein [173]. There is a significant offset between this value and that of the distributions from this study. A likely explanation for this is the limited resolution of the images at the fluid-surface interface. The temperature is possibly higher in the thin fluid layer infinitely close to the surface which is too small to be accurately detected by the camera. In other words, the number of pixels measuring the fluorescence signal per unit of space in the boundary layer is not sufficient here. Assuming the maximum temperature in the boundary layer is approximately equal to the surface temperature may not be valid in this case.

This suggests a need for optimizing the imaging setup to ensure high resolution in the boundary layer when using LIF thermometry for local heat transfer measurements. With an optimized spatial resolution, the heat transfer coefficient may even be directly calculated by estimating the temperature gradient very close to the surface, such that

$$h(\theta) = \frac{\dot{q}}{(T_s(\theta) - T_\infty)} = \frac{-k \left. \frac{\partial T}{\partial r} \right|_{r=0}}{(T_s(\theta) - T_\infty)} \quad (6.9)$$

where $\partial T/\partial r$ is the temperature gradient in the radial direction from the cylinder surface and k is the thermal conductivity of the fluid (i.e. the toluene or anisole carrier gas, in this case N_2). This would eliminate the need to calculate the heat flux, $\dot{q}$, from external measurements but rather quantify it directly from the temperature images.

The plots in fig. 6.22 do still provide good qualitative information and agree quite well with each other for different heat inputs and tracer materials. The characteristic local heat transfer behaviour of forced convection from a cylinder, is verified here (for instance, see [173, 174]). Maximum heat transfer is exhibited at the stagnation point, $\theta = 0^\circ$. The monotonic decrease of Nu can be seen for $0^\circ < \theta < 90^\circ$ which is due to an increase in the thermal boundary layer thickness, and hence heat transfer resistance, with distance along the circumference. A local minimum is reached at $\theta \cong 90^\circ$, at the point where the flow separation occurs. The intensity of the turbulent mixing in the wake region typically increases towards the rear of the cylinder which results in an increase in Nu from $90^\circ < \theta < 180^\circ$.

Fig. 6.23 shows the same profiles of Nu vs θ for Reynolds numbers of $Re = 530, 1150$ and 1460 and for both toluene (fig. 6.23a) and anisole (fig. 6.23b) excitation (corresponding temperature fields can be seen in fig. 6.21). An increase in the overall Nusselt number across the cylinder circumference with Reynolds number is verified here. The shape of the profiles does not change significantly since the Reynolds number range is relatively small. For the largest value of Re , however, the increase in heat transfer at the rear of the cylinder is slightly more significant. This is most likely due to the increased intensity of turbulent mixing in the wake region which is promoted by an increase in Reynolds number.

Comparing figs. 6.23a & b, it can be seen that the Nusselt number distributions generally agree for the two different fluorescent tracer materials. However, the slight disagreement seen previously between the temperature measurements of both tracers (fig. 6.21), appears to yield higher values of Nu at the rear side of the cylinder for toluene. This effect only occurs for higher Reynolds numbers. Interestingly, as the Reynolds number is increased for both tracers, the stagnation Nusselt number, $Nu(\theta = 0)$, appears to coincide more closely with the correlation by Sanitjai and Goldstein [173] (marked in fig. 6.23 by the solid lines on the left side of the axis).

Despite some inaccuracy, these distributions of local Nusselt number demonstrate how LIF thermometry may be a valuable tool for performing detailed analysis of local convective heat

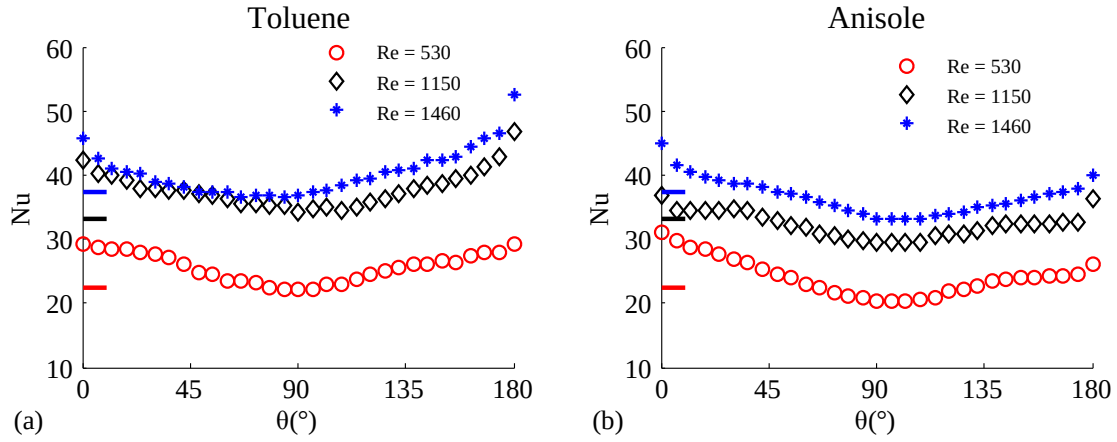


Figure 6.23: Nusselt number, Nu , versus angular position, θ , along the surface of the cylinder for Reynolds numbers of $Re = 530$ (red), 1150 (black) and 1460 (blue), using both (a) toluene and (b) anisole as a fluorescent tracer material. The stagnation Nusselt numbers, $Nu(\theta = 0^\circ)$, calculated for each value of Re using the correlation by Sanitjai and Goldstein [173], are represented by the solid lines (the colour corresponding to the value of Re).

transfer in gaseous flows. By optimizing the laser-imaging setup to maximize the precision and resolution of the temperature fields in low temperature ranges, this technique could be applied to convection problems involving more complex geometries and flow configurations. In section 7.3 of the next chapter, two-dimensional temperature measurements of the flow field of the convective drying experiment are presented. These measurements are used to supplement heat and mass transfer analysis of a drying cube which is performed using the experimental setup described in section 4.1.

Chapter 7

Results and discussion: Heat and mass transfer in forced convective drying

In this chapter the experimental results of the forced convective drying of a porous cube are presented and discussed. Measurements of surface temperature distribution and moisture content versus drying time, for a porous cube, have been carried out for Reynolds numbers ($Re = V_\infty L / \nu$) in the range of 180 – 1000, for a constant free-stream temperature of 59°C. V_∞ is the average upstream air velocity in the direction parallel to the cube surface being imaged by the thermal imaging camera. The experiments were also performed for three different free-stream air temperatures, $T_\infty = 40^\circ\text{C}$, 59°C and 79°C , at a fixed Reynolds number of $Re = 350$. Supplementary measurements of the drying gas temperature field using LIF were also carried out to characterize the local surface heat transfer in more detail.

Section 7.1 presents analysis of the evolution of moisture content, M , and average surface temperature, T_s , with drying time, t . Section 7.2 deals with the local surface temperature evolution and the heat transfer spatial heterogeneities associated with this. Also, included in this section are some LIF temperature field measurements of the gas surrounding the drying cube in section 7.3. Section 7.4 contains heat and mass transfer analysis during the quasi-steady state constant-drying rate period and a discussion of the analogy between heat and mass transfer during this period.

7.1 Evolution of surface temperature and moisture content

Some surface temperature distributions $T(x,y)$ are shown in fig. 7.1 and fig. 7.2 at different points in non-dimensional drying time, $\tau = t/t_e$, during the drying process. These distributions are demonstrated for two different Reynolds numbers, $Re = 300$ (fig. 7.1) and $Re = 800$ (fig. 7.2) for a free-stream temperature of $T_\infty = 59^\circ\text{C}$. Temperature distributions for other parameter settings can be found in Appendix B2. The red dashed square shown in the first distributions of figs. 7.1 and 7.2 represents the averaging border over which the average surface temperature, T_s , is calculated. Since there is some small inevitable displacement of the cube during each experiment, the border is a little inside the cube edges to ensure the border remains within the perimeter of the cube.

It can be seen from the first four distributions in figs. 7.1 and 7.2 for $0 < \tau < 0.51$, that the temperature across the surface $T(x,y)$ remains at an almost constant value since these still lie within the constant-drying rate period. The CRP typically ends at about $\tau = 0.5$ (the critical values of τ where CRP and FRP begin are defined later in this section and illustrated in fig. 7.4). Intuitively, it would be expected that the edges and corners dry at a faster rate than the rest of the surface. This is noticeable here since the temperature gradient at the perimeter starts to thicken slightly. This suggests that the corners of the cube are not completely saturated (i.e. $RH < 100\%$), since their temperature is increasing above the wet-bulb temperature.

The last two distributions at $\tau > 0.51$ show a significant change from the previous four and exhibit quite non-uniform surface temperature distributions, indicating that the FRP has been reached. The increase of the surface temperature occurs at the edges at a faster rate, however, it does not appear to be very symmetric. The asymmetry can be attributed to the development of dry patches caused by invasion percolation (see section 2.2.4). As the cube fully dries out and τ tends to 1, the surface temperature, $T(x,y)$, approaches the surrounding air temperature, T_∞ , at all points. The surface temperature distribution is discussed in more detail in section 7.2.

Fig. 7.3 demonstrates the evolution of moisture content (dry basis - kg of water per kg of dry solid), M , and average surface temperature, T_s , with drying time, t , for various values of Re and T_∞ . Note that each curve is an ensemble average of 3 to 4 repeated experiments at

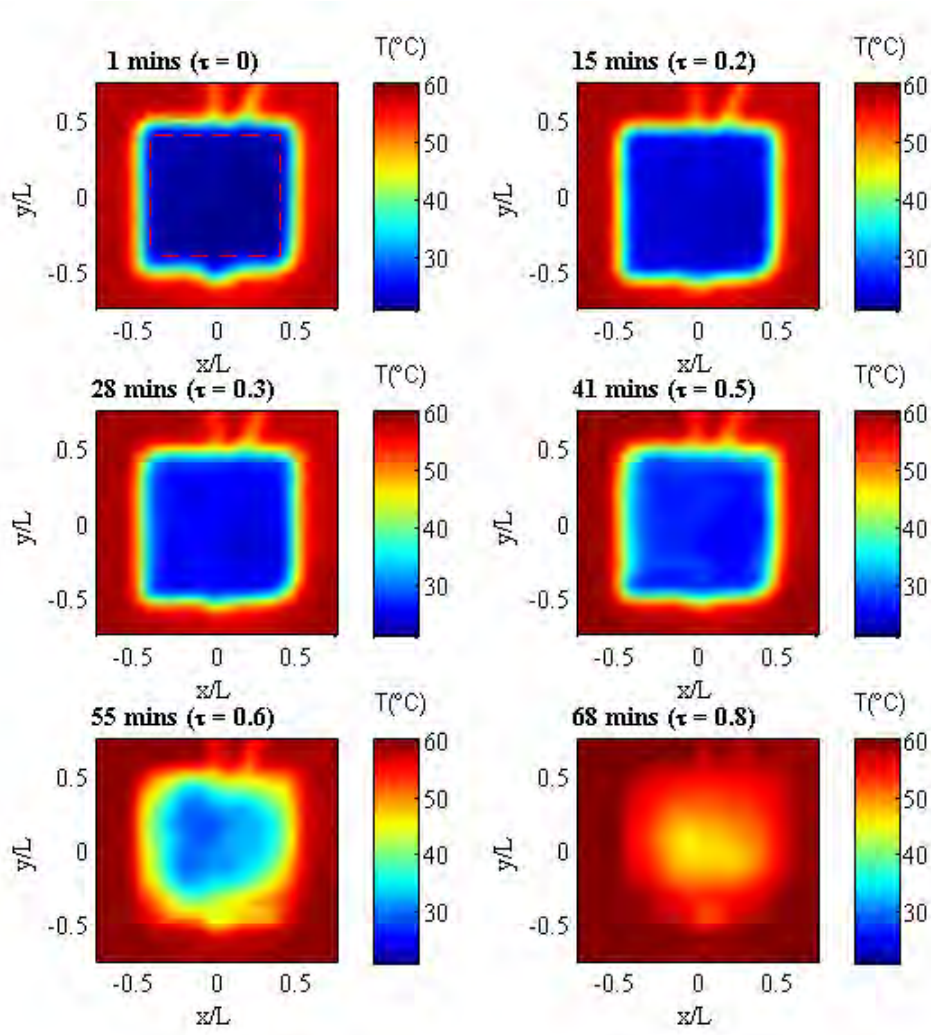


Figure 7.1: Temperature distribution across the cube surface $T(x, y)$ for different non-dimensional drying times, τ , for $Re = 300$ and $T_{\infty} = 59^{\circ}\text{C}$. The red dashed line in the first distribution illustrates the border over which the average surface temperature, T_s , is calculated

the exact same parameter settings. In general, the higher values of Re and T_{∞} lead to faster rates of drying and hence shorter drying times, as expected. For each value of Re and T_{∞} , the curves of M and T_s versus t demonstrate the two distinct stages of drying: a constant-drying rate period (CRP) and a falling-drying rate period (FRP). The CRP can be identified from the linear decrease of moisture content with time in the left hand side of the time axis in fig. 7.3a & b and the almost constant surface temperature with respect to time in the left hand side of the time axis in fig. 7.3c & d. Also verified here, in fig. 7.3c, is the increase in the ‘steady’ or wet-bulb surface temperature with increasing free-stream temperature,

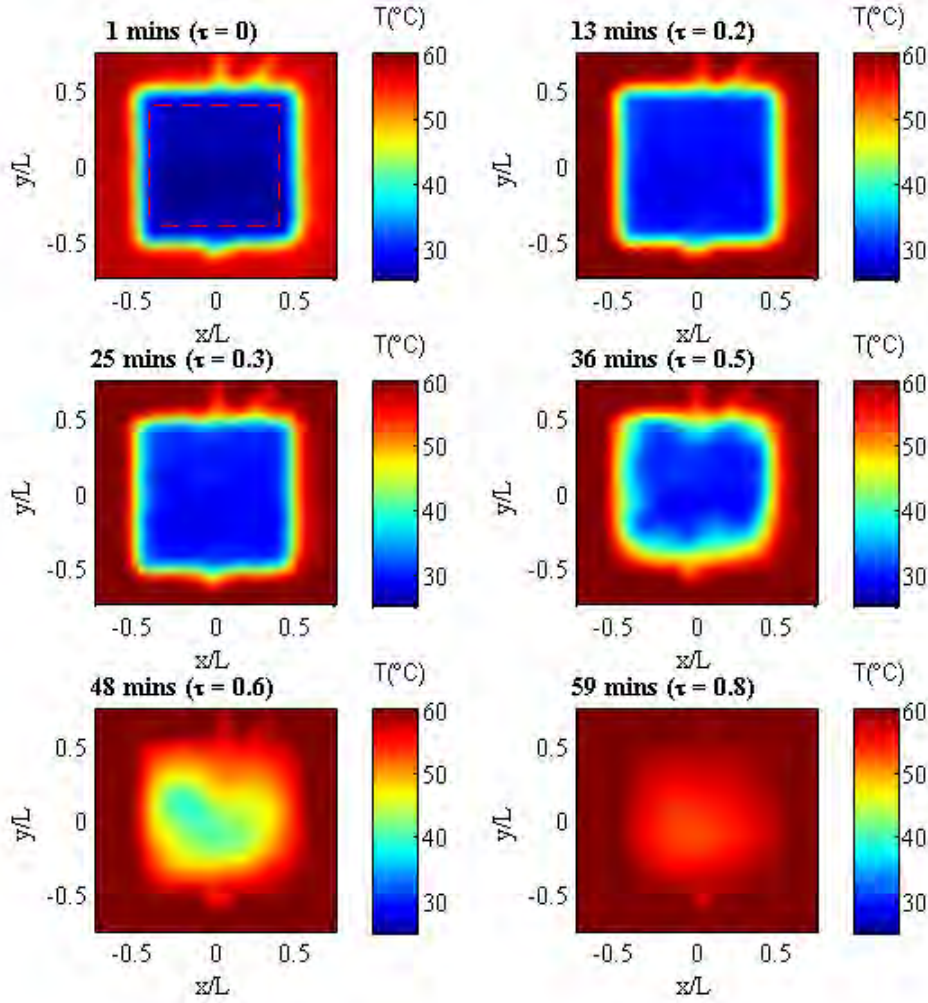


Figure 7.2: Temperature distribution across the cube surface $T(x,y)$ for different non-dimensional drying times, τ , for $Re = 800$ and $T_\infty = 59^\circ\text{C}$. The red dashed line in the first distribution illustrates the border over which the average surface temperature, T_s , is calculated

T_∞ . The pre-heating stage which can be seen at the beginning of the curves in fig. 7.3c & d demonstrates the cube surface being heated to reach this steady temperature. For the lowest free-stream temperature of $T_\infty = 40^\circ\text{C}$, the evaporation appears to cool the cube surface down from its initial temperature since the wet-bulb temperature is lower than room temperature for $T_\infty = 40^\circ\text{C}$.

It can be observed that the surface temperature is not completely uniform with respect to time during the CRP in all cases. Particularly for the larger two free stream temperatures ($T_\infty = 59^\circ\text{C}$ and 79°C), the surface temperature appears to increase with drying time slightly

7.1. EVOLUTION OF SURFACE TEMPERATURE AND MOISTURE CONTENT

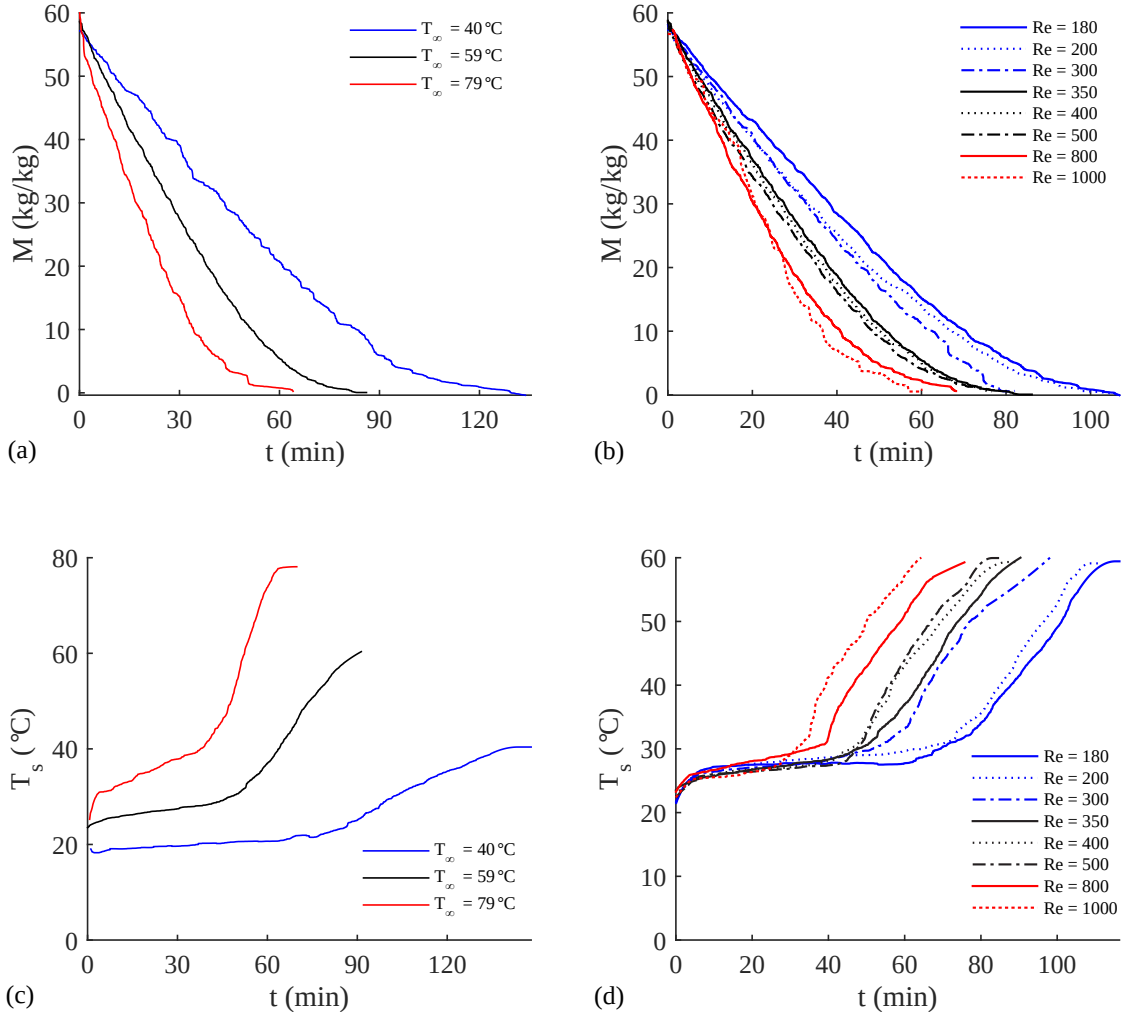


Figure 7.3: Evolution of (a, b) moisture content, M (kg/kg - mass of water per mass of dry solid), and (c, d) average surface temperature, T_s (°C), with drying time t (minutes) for (a, c) $T_\infty = 40, 59$ and 79°C (for a fixed Reynolds number of $Re = 350$) and (b, d) for $Re = 180 - 1000$ (for a fixed free steam temperature of $T_\infty = 59^\circ\text{C}$).

during the CRP. This is most-likely due to the faster drying rate near the corners or surface edges, as shown in fig. 7.1 and fig. 7.2. This results in a temperature above the wet-bulb temperature at each cube corner, which leads to a small amount of sensible heat transfer and hence a slight increase in the average surface temperature, T_s , during the CRP.

The FRP can be identified in the right hand side of the time axes in fig. 7.3 by the significant increase in the surface temperature towards the temperature of the drying air (T_∞) and the decrease of the rate of drying (i.e. non-linear decrease of moisture content with respect to time). The rise in surface temperature during the FRP shows that the whole surface

is no longer saturated and is being heated to the air temperature. At this point, it is likely that the moisture front or plane of saturation is retreating into the core of the cube. The FRP is shorter for higher values of Re and T_∞ , which can be seen in the change of the slopes of T_s and M versus drying time t .

The point where the CRP and FRP begin can be identified more clearly by examining the time-normalized evolution of moisture content and surface temperature. Fig. 7.4 shows the evolution of non-dimensional temperature and moisture content, θ and ϕ , respectively, versus τ for the highest, lowest and intermediate tested values of Re and T_∞ . In general, each experiment has the same characteristic curves for θ and ϕ versus τ . Each curve demonstrates a turning point at the beginning of the FRP where the dimensionless temperature, θ , starts to increase towards unity and where the dimensionless moisture content, ϕ , is in transition from a linear decrease (constant drying rate) to a non-linear decrease (falling drying-rate) with respect to τ .

Critical values of τ where the constant-drying rate period and falling-drying rate period begin (τ_{crp} and τ_{frp}) are indicated in fig. 7.4 by the blue and red dashed lines, respectively. These turning points are determined by examining the rate of change of the average surface temperature curve, dT_s/dt , in fig. 7.3b. This can be approximated from the instantaneous slope, $\Delta T_s/\Delta t$, such that

$$\frac{dT_s}{dt} \cong \frac{\Delta T_s}{\Delta t} = \left(\frac{T_s(t + \Delta t) - T_s(t)}{\Delta t} \right) \quad (7.1)$$

where Δt is a short time interval between consecutive temperature measurements. During the pre-heating period, the slope of the surface temperature curve, $\Delta T_s/\Delta t$, is changing with respect to time but as the CRP begins, it reaches a local minimum. The beginning of the CRP, τ_{crp} (or t_{crp}), is found by solving for the local minimum of $\Delta T_s/\Delta t$ during the first fifth of the drying experiment (i.e. for $0 < \tau \leq 0.2\tau_e$ which is sufficiently long to enclose τ_{crp}). An exponential data fit (using the least-squares method) is fit to $\Delta T_s/\Delta t$ during this period (a temperature increase due to heating typically has an exponential time response). This provides a smooth curve from which the local minimum and hence τ_{crp} can be determined.

The beginning of the FRP, τ_{frp} (or t_{frp}), corresponds to the point where the slope, $\Delta T_s/\Delta t$, starts to change with time again. This is determined as the point in time between the beginning

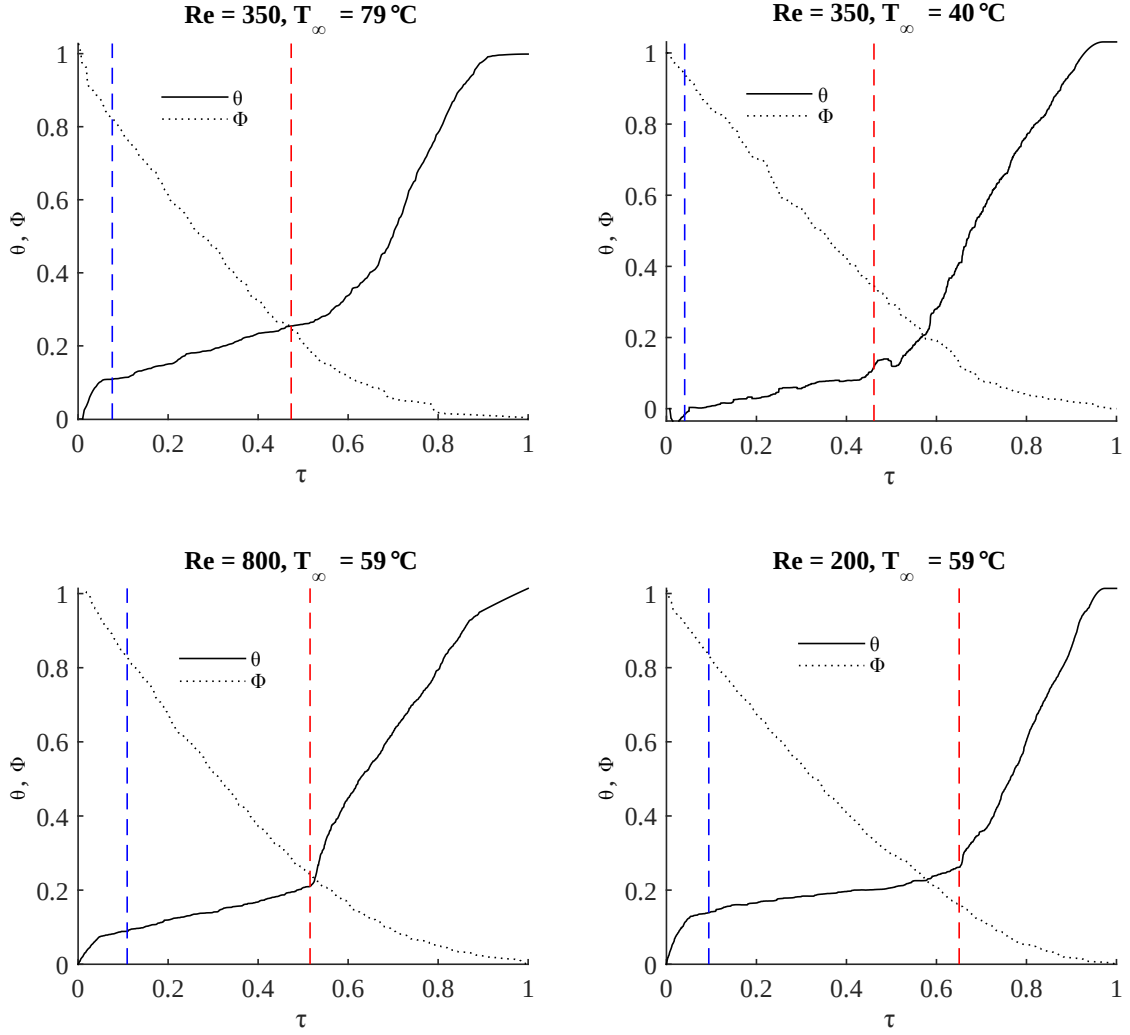


Figure 7.4: Non-dimensional temperature, θ , and moisture content, ϕ , versus non-dimensional time, τ , for: (a, b) $T_\infty = 79$ and 40°C for $Re = 350$; (c, d) $Re = 800$ and 200 for $T_\infty = 59^\circ\text{C}$;

of the CRP and the end of the drying experiment (i.e. for $\tau_{crp} < \tau \leq \tau_e$) where the slope, $\Delta T_s / \Delta t$, first exceeds a threshold value close to zero. This threshold value can be expressed as

$$\Delta T_s / \Delta t \geq 0.02 \quad \text{for} \quad t_{frp} \leq t \leq t_e \quad (7.2)$$

where the value of $\Delta T_s / \Delta t_{frp} = 0.02$ was determined to be the value which coincided most closely with the turning point from CRP to FRP for all experiments. It can be seen in fig. 7.4 that these definitions of τ_{crp} and τ_{frp} represent quite clearly the turning points of the drying process for different values of T_∞ and Re .

7.1. EVOLUTION OF SURFACE TEMPERATURE AND MOISTURE CONTENT

The effect of free-stream temperature and Reynolds number, T_∞ , on the evolution of dimensionless temperature, θ , and dimensionless moisture content, ϕ , can be examined by comparing figs. 7.4a - d. During the pre-heating period ($\tau < \tau_{crp}$), θ increases to an almost steady value for the highest free-stream temperature of $T_\infty = 79^\circ\text{C}$ (fig. 7.4a) whereas for the lowest $T_\infty = 40^\circ\text{C}$ (fig. 7.4b), θ decreases to a negative value since its surface temperature decreases below the initial surface temperature (i.e. $T_s(t_{crp}) < T_s(t_i)$) as seen in fig. 7.3b. Reducing the Reynolds number to a lower value of $Re = 200$ (fig. 7.3c) has the effect of shifting τ_{frp} to a higher value (or increasing the duration of the CRP relative to the duration of the FRP). The reduction in Reynolds number also results in a much more uniform value of θ during the CRP. On the other hand, for a higher Reynolds number of $Re = 800$ (fig. 7.3d), θ is increasing significantly more during the CRP. This suggests that the constant-drying rate period becomes more ‘pronounced’ for lower Reynolds numbers.

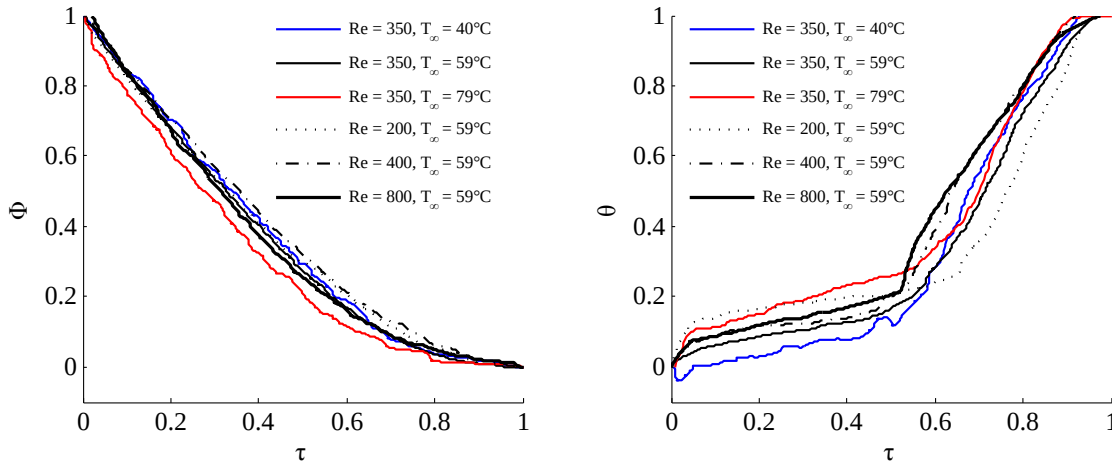


Figure 7.5: Dimensionless moisture content, ϕ (a), and dimensionless surface temperature, θ (b), versus τ for low, intermediate and high values of T_∞ and Re (over the range of values studied).

The curves of ϕ versus τ in figs. 7.4a - d do not differ much with varying free-stream temperature or Reynolds number. This is more clear in fig. 7.5a where they are plotted together. The curves of ϕ versus τ are almost identical, whereas θ versus τ varies more significantly, as seen in fig. 7.5b. This suggests that the evolution of ϕ with τ is independent of external conditions. That is to say, the curve of ϕ versus τ is a characteristic curve for the same average internal conditions of the drying material (e.g. porosity, diffusivity, bulk density) and does not change by varying the Reynolds number or free-stream temperature. It

can be seen in fig. 7.5b, however, that evolution of θ with τ varies more significantly with external conditions.

7.2 Local temperature evolution

The evolution of surface temperature distribution, demonstrated previously in fig. 7.1 and fig. 7.2 highlights a significant change in the heat transfer behaviour as the drying process goes from the constant-drying rate period to the falling-drying rate period. The first four images (from fig. 7.1 and fig. 7.2) are from during the CRP, which is characterized by an effectively uniform distribution of temperature and a period of steady heat transfer. Some very slight thickening of the gradients near the edges during the CRP is also evident, particularly in the fourth distribution for $\tau \cong 0.5$. This suggests that the corners of the cube are not fully saturated during the CRP and there is some sensible heat being transferred by conduction into the cube.

During the FRP (the final two images), the heat transfer is clearly not steady and the temperature distribution is non-uniform. Both fig. 7.1 and fig. 7.2 show an increasing temperature (and hence drying rate) from the centre to the edges. Interestingly, however, the distributions do not show the same exact shape and are somewhat asymmetric. It is possible that this is related to the invasion percolation theory discussed in section 2.2.4 in which a random or asymmetric distribution of wetter and dryer patches is observed [66, 68–70].

To examine the surface temperature heterogeneities during the FRP in more detail, contour plots of the surface temperature are presented in fig. 7.6 and fig. 7.7. These contour plots demonstrate the effect of the edges heating up (and hence drying out) faster than the core ($((\tau - \tau_{frp})/(\tau_e - \tau_{frp}) < 0.8)$) before eventually approaching the free-stream air temperature ($((\tau - \tau_{frp})/(\tau_e - \tau_{frp}) > 0.8)$). Each consecutive plot demonstrates the higher temperature ‘advancing’ inwards. This is characteristic of the retreating moisture front (see section 2.2.2) in which the saturated water recedes into the core of the cube as it dries out.

There is a certain level of asymmetry in the contour shapes in both fig. 7.6 and fig. 7.7 particularly in the early stages of the FRP ($((\tau - \tau_{frp})/(\tau_e - \tau_{frp}) < 0.2)$). There is no clear or consistent pattern when comparing $Re = 300$ (fig. 7.6) and $Re = 800$ (fig. 7.7). The moisture transport mechanism of invasion percolation may be influencing this. The resolution of

7.2. LOCAL TEMPERATURE EVOLUTION

the infra-red camera is limited but with greater resolution, the development of wet and dry patches may be observed.

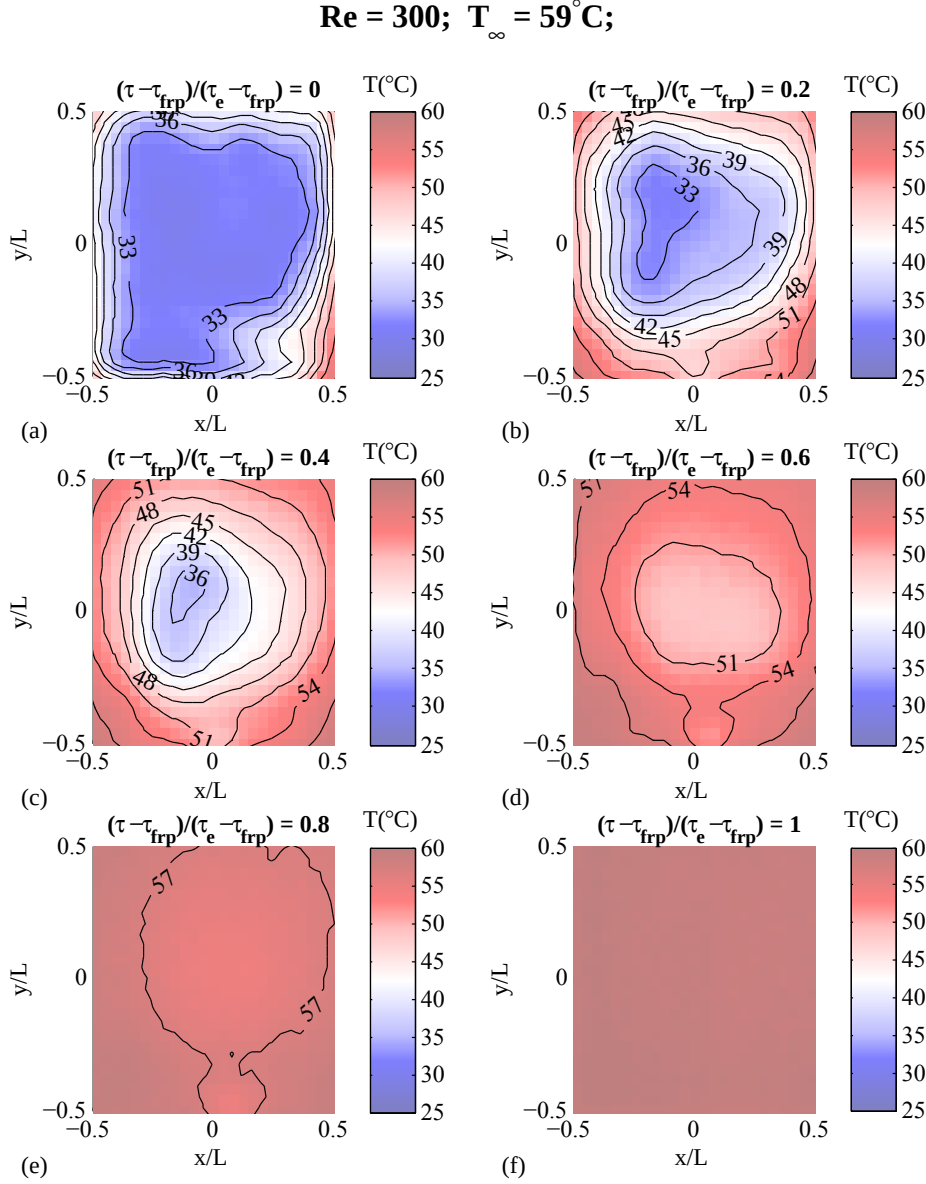


Figure 7.6: Contour plot of surface temperature, $T(x,y)$, at different stages during the FRP ($0 < (\tau - \tau_{frp})/(\tau_e - \tau_{frp}) < 1$) for a Reynolds number and free-stream temperature of $Re = 300$ and $T_\infty = 59^\circ C$, respectively

Figs. 7.8a-f present local temperature profiles along the centre of the cube surface (facing the infra-red camera), for a Reynolds number of $Re = 180$ (a & b), 350 (c & d) and 800 (e & f), respectively, and for a free-stream temperature of $T_\infty = 59^\circ C$. These profiles demonstrate

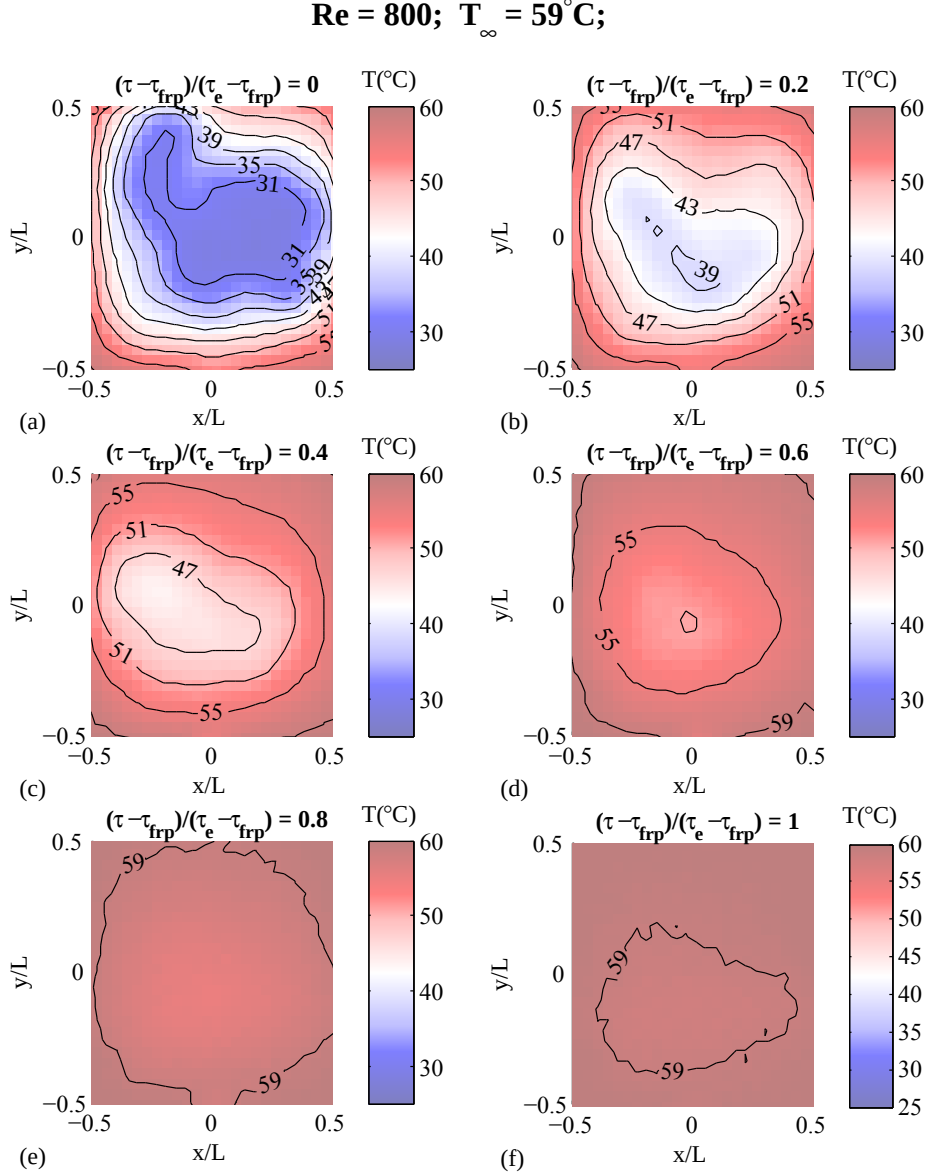


Figure 7.7: Contour plot of surface temperature, $T(x,y)$, at different stages during the FRP ($0 < (\tau - \tau_{frp})/(\tau_e - \tau_{frp}) < 1$) for a Reynolds number and free-stream temperature of $Re = 800$ and $T_\infty = 59^\circ\text{C}$, respectively

the evolution of temperature across the cube surface centre for both $y = 0$ (a,c and e) and $x = 0$ (b,d and f) at different points in time. The slight asymmetry about $x/L = 0$ and $y/L = 0$ is caused by a minor inevitable displacement of the suspended cube from the centre of the camera image over the course of the drying experiment. Some correctional shifts of the pixels have been applied to account for this.

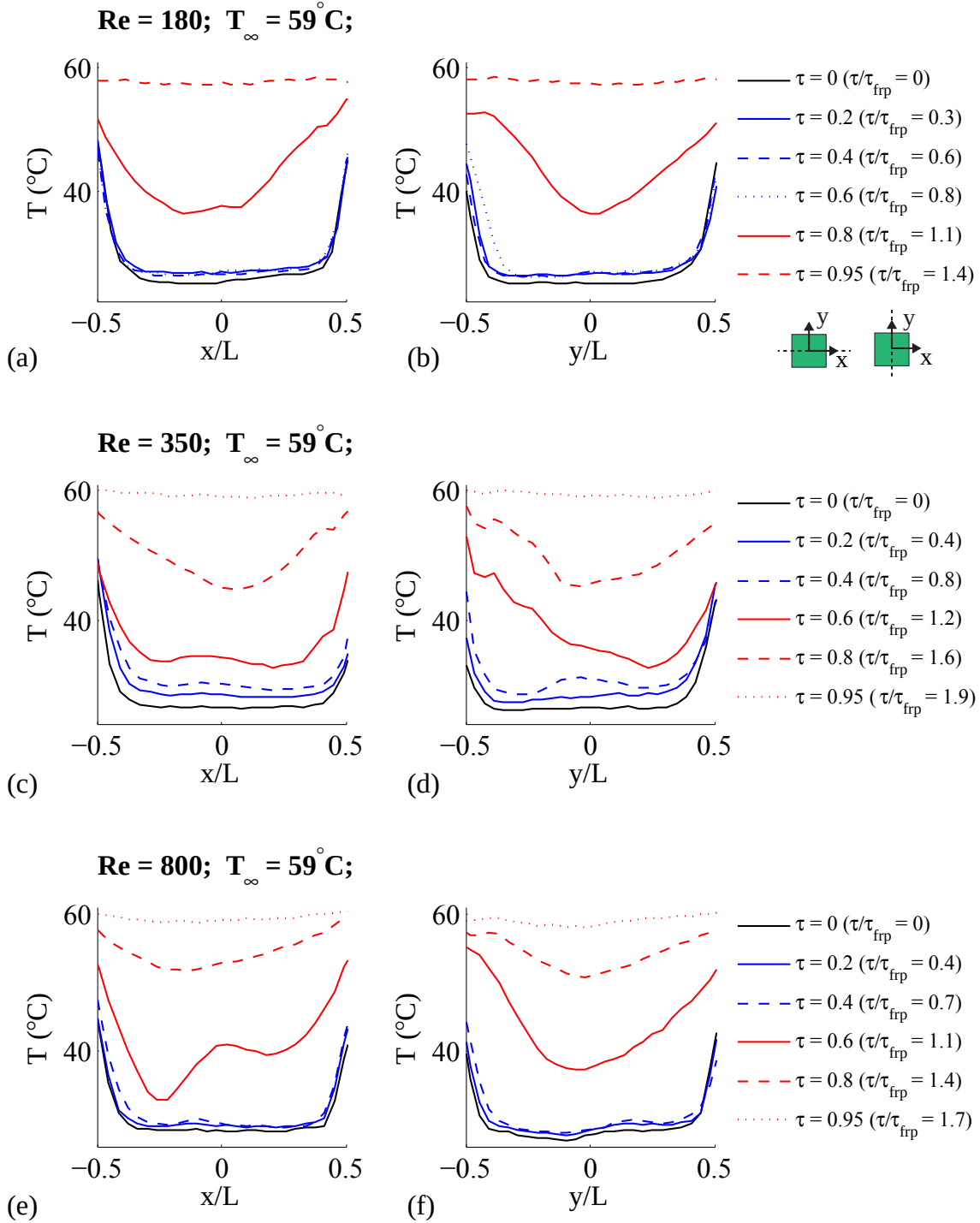


Figure 7.8: Profiles of local temperature $T(^{\circ}\text{C})$ across the centre of the cube surface (facing the IR camera) for (a, c and e) $y = 0$ (x direction) and for (b, d and f) $x = 0$ (y direction), for $Re = 180$ (a & b), 350 (c & d) and 800 (e & f). Each profile is for $T_\infty = 59^\circ\text{C}$. The profiles are colour-coded as per the period of the drying process in which they lie. Black represents the beginning or pre-heating stage ($\tau/\tau_{frp} = 0$), blue represents the CRP ($\tau/\tau_{frp} < 1$); and red represents the FRP ($\tau/\tau_{frp} > 1$)

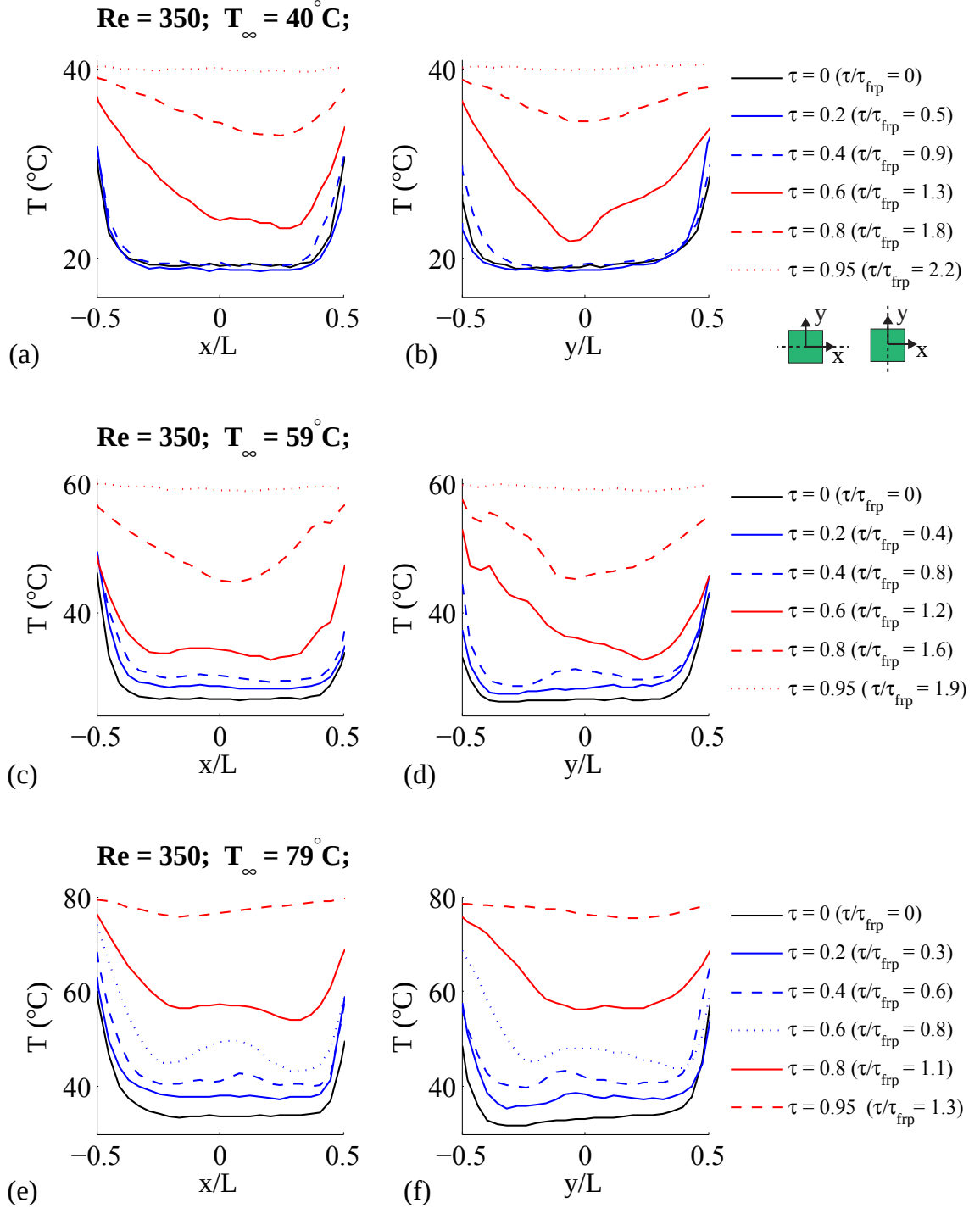


Figure 7.9: Profiles of local temperature $T(^{\circ}\text{C})$ across the centre of the cube surface (facing the IR camera) for (a, c and e) $y = 0$ (x direction) and for (b, d and f) $x = 0$ (y direction), for $T_\infty = 40^\circ\text{C}$ (a & b), 59°C (c & d) and 79°C (e & f). Each profile is for a Reynolds number of $Re = 350$. The profiles are colour-coded as per the period of the drying process in which they lie. Black represents the beginning or pre-heating stage ($\tau/\tau_{frp} = 0$), blue represents the CRP ($\tau/\tau_{frp} < 1$); and red represents the FRP ($\tau/\tau_{frp} > 1$)

At the end of the drying process, the local surface temperature reaches the same temperature as the free-stream air as can be seen in the flat profiles for $\tau = 0.95$. In comparison, at the very beginning of the process ($\tau = 0$), the profiles in fig. 7.8a & b are also flat but at the minimum (or initial) temperature. During the constant-drying rate period ($\tau/\tau_{frp} < 1$), the temperature across the surface remains at a uniform wet-bulb temperature with some minor variation in the y direction (fig. 7.8b). There is also a steep increase or gradient in temperature near the edges of the cube ($x/L = \pm 0.5$ and $y/L = \pm 0.5$) during the CRP, since very close to the edges of the cube, the drying is occurring at a faster rate. As mentioned previously, this is a likely cause of the small amount of sensible heat transfer, which is evident in fig. 7.3c & d from the slight increase in average surface temperature during the CRP.

There is no obvious effect of increasing the Reynolds number on the shape of temperature profiles in fig. 7.8. During the FRP, the profiles appear to be quite quite asymmetric and there is no clear relationship between the profile shapes and the Reynolds number. This suggests that the internal moisture transport mechanisms have a more dominant influence on the surface heat transfer than the transport mechanisms in the boundary layer, during the FRP. The more symmetric profiles in the CRP illustrate an almost steady-state period of heat transfer and hence drying in which the external flow conditions are more dominant. During the FRP, for most of the profiles parallel to the flow (y -axis) direction (fig. 7.8b, d & f), there is a slightly higher temperature near the leading-edge ($-0.5 < y/l < -0.4$) where the thermal boundary layer is thinner, compared to the trailing edge ($0.4 < y/l < 0.5$). This presumably leads to a slightly faster drying rate near the leading-edge as witnessed in many studies involving drying of rectangular drying bodies [19, 20, 24, 76, 77, 79, 82]. In most of these studies, the Reynolds numbers were greater and the bodies were more rectangular in shape (longer side parallel to flow) and so the effect was more pronounced.

Fig. 7.9 presents the same temperature profiles for free-stream temperatures of $T_\infty = 40^\circ\text{C}$ (a & b), 59°C (c & d) and 79°C (e & f) at a fixed Reynolds number of $Re = 350$. As with the Reynolds number, the free-stream temperature has no major influence on the shape of the profiles, which become quite asymmetric and unpredictable during the FRP. One interesting feature of fig. 7.9, however, is how the surface is significantly less isothermal during the CRP for the largest free-stream temperature of $T_\infty = 79^\circ\text{C}$ (fig. 7.9 e & f). This is presumably due to the greater temperature difference between surface and free-stream and hence a faster rate

of heat transfer, which pronounces the effect of faster drying at the cube edges. That is to say, the drying rate becomes more non-uniform across the surface as the free-stream temperature or the thermal load is increased.

This hypothesis can be investigated by trying to quantify the portion of the area of the cube surface that is at a slightly greater temperature during the CRP. It can be assumed that this is the portion of the cube surface which is drying at a faster rate. Fig. 7.10 and 7.11 provide maps, at consecutive points in time before the FRP (τ/τ_{frp}), of the cube surface area where the temperature is more than 10% greater than the mean surface temperature during the CRP. The black area in fig. 7.10 and 7.11 represents this area of ‘drier’ solid, A_{ds} . This area, which is mostly at the surface edges, clearly becomes thicker with drying time during the CRP. Fig. 7.10 and 7.11 are for a Reynolds number of $Re = 800$ ($T_\infty = 59^\circ\text{C}$) and a free-stream temperature of $T_\infty = 79^\circ\text{C}$ ($Re = 350$), respectively. At these conditions, relatively high rates of convective heat transfer, $\dot{Q}_{conv}$, are observed.

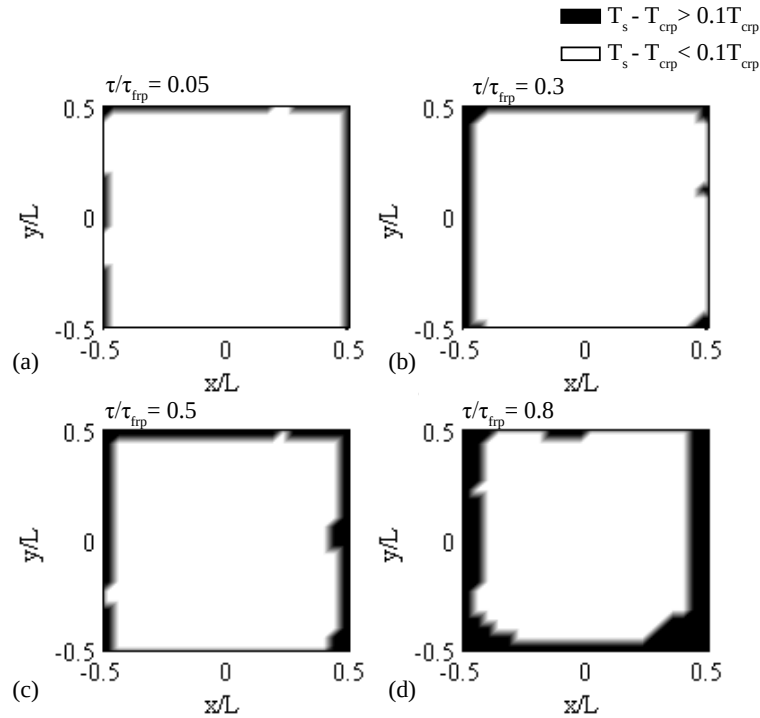


Figure 7.10: Maps of points (x, y) on the cube surface where temperature is more than 10% greater than the mean surface temperature during the CRP, T_{crp} , for a different point in time before the FRP, τ/τ_{frp} . This area is represented by black and corresponds to the portion of the surface which is drying at a faster rate. The rest of the surface which is within 10% of T_{crp} is represented by white. This particular set of maps is for a Reynolds number and free-stream temperature of $Re = 800$ and $T_\infty = 59^\circ\text{C}$, respectively.

7.2. LOCAL TEMPERATURE EVOLUTION

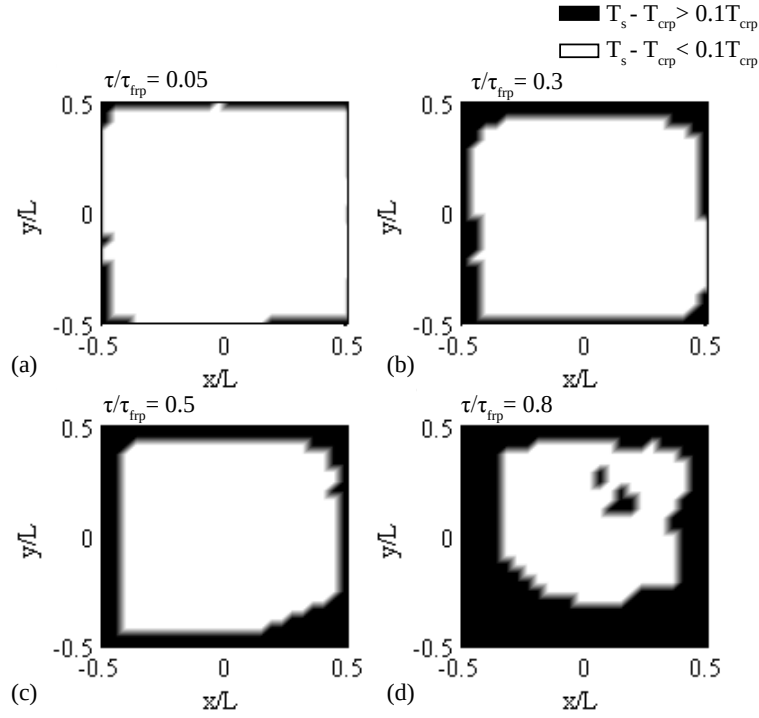


Figure 7.11: Maps of points (x,y) on the cube surface where temperature is more than 10% greater than the mean surface temperature during the CRP, T_{crp} , for a different point in time before the FRP, τ/τ_{frp} . This area is represented by black and corresponds to the portion of the surface which is drying at a faster rate. The rest of the surface which is within 10% of T_{crp} is represented by white. This particular set of maps is for a Reynolds number and free-stream temperature of $Re = 350$ and $T_\infty = 79^\circ\text{C}$, respectively.

The fraction of drier solid across the cube surface, F_{ds} , can be found by dividing A_{ds} by the total surface area. This is plotted versus drying time for four different drying conditions in figs. 7.12a - d. It is difficult to calculate this factor with accuracy as can be seen in the variance of these plots. This is mostly due to slight oscillations of the cube - and hence movement of the cube borders - during each experiment. Also, the limited spatial resolution of images affects the accuracy. It can be seen, however, that there is a clear increase in the fraction of drier solid, F_{ds} , with time. The solid black line in figs. 7.12a - d is a linear fit to the data which gives an approximate estimate of the rate that this is increasing. It appears that F_{ds} is increasing at a faster rate for the largest free-stream temperature, $T_\infty = 79^\circ\text{C}$ (fig. 7.12a) compared to the lowest, $T_\infty = 40^\circ\text{C}$ (fig. 7.12b). Similarly an increase in the Reynolds number leads to an increase in the rate of increase of F_{ds} . This suggests that the edge effects are, in fact, accelerated by more intense drying conditions.

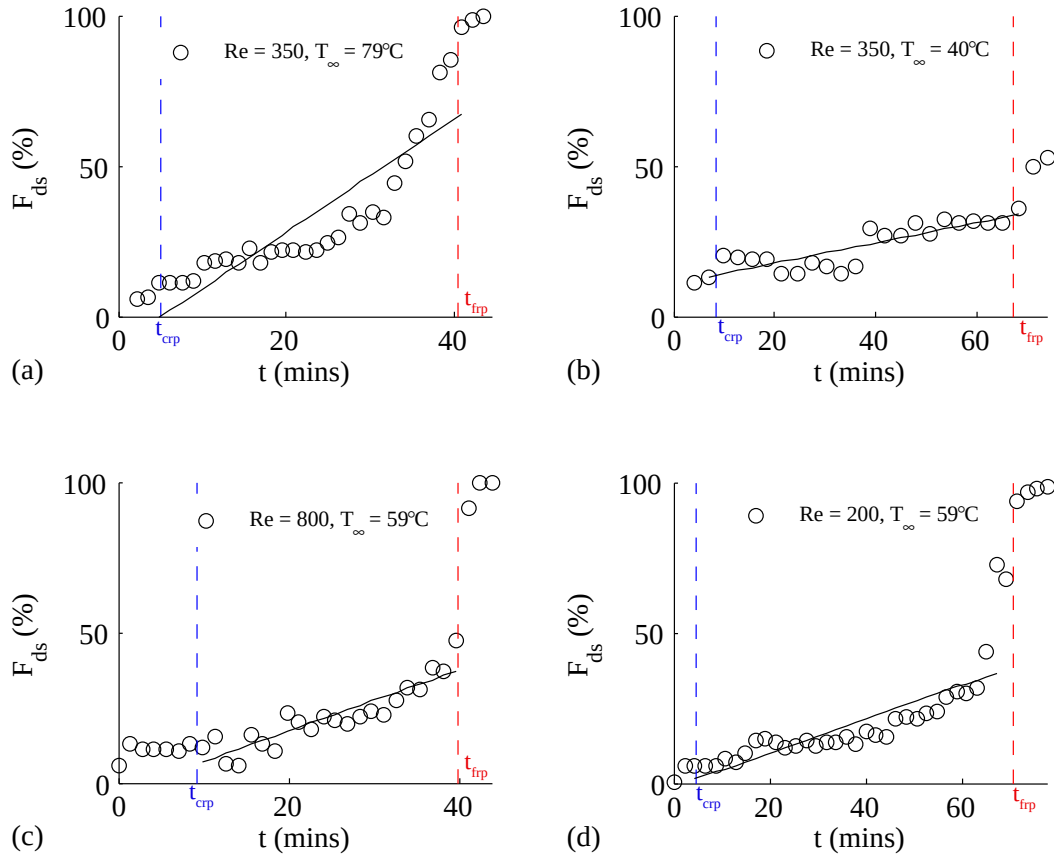


Figure 7.12: Fraction of drier solid across the cube surface, F_{ds} , versus drying time for: (a) $Re = 350$ and $T_\infty = 79^\circ\text{C}$; (b) $Re = 350$ and $T_\infty = 40^\circ\text{C}$; (c) $Re = 800$ and $T_\infty = 59^\circ\text{C}$; and (d) $Re = 200$ and $T_\infty = 59^\circ\text{C}$

7.3 LIF temperature field measurement of drying gas

The LIF thermometry technique (for more detail see sections 4.3 and 6.2), was applied to a drying porous cube of the same material but of slightly bigger side length, $L = 15$ mm (in order to maximize the spatial resolution of the images). In this case, the drying gas was nitrogen with a small amount of toluene or anisole ($w_{tol} = 2.5\%$ and $w_{ans} = 0.7\%$, respectively). The experiment was performed in the LIF flow tunnel illustrated in figs. 4.10 to 4.12 of section 4.3.

Since the experiment took place in a sealed tunnel, the humidity of the incoming or free-stream gas, gradually increases with time as the cube evaporates. This effect is slowed down by a moisture absorbing desiccant inside the container but, inevitably, the free-stream relative humidity or vapour concentration will still increase. Therefore, the wet-bulb temperature will also increase slightly with drying time at the surface of the cube during the CRP. This can be seen in fig. 7.13 which shows the evolution of temperature at the surface of the cube during a drying experiment, measured using a K-type thermocouple (positioned at the surface facing away from the camera). Though there is a constant increase in the surface temperature due to this effect, two distinct periods which resemble the CRP and FRP can still be identified clearly here. The decrease at the beginning of the experiment shows the cube cooling down from the free-stream temperature to the wet-bulb temperature when the water is injected, before it starts to rise again. This is unlike most of the drying experiments performed in the other drying chamber (see fig. 4.2) where the cube is initially soaked in a beaker at room temperature and so it usually heats up (or only cools down slightly) to reach the wet-bulb temperature.

The wet-bulb temperature at the beginning of the CRP in fig. 7.13 is about 30°C . A lower value of approximately 18°C would be expected for a free-stream temperature of $T_{\infty} = 44^{\circ}\text{C}$ and an initial relative humidity close to zero (see, for example, the temperature curve for $T_{\infty} = 40^{\circ}\text{C}$ and $RH_{\infty} = 1\%$ in fig. 7.3 in which wet-bulb temperature is about 19°C). The higher wet-bulb temperature is caused by some free water dropping from the cube surface after injection of the water into the porous cube. This water then evaporates in the tunnel, raising the relative humidity to about $RH_{\infty} = 30\%$. It is difficult to avoid this when trying to ensure the cube is fully saturated. However, in future experiments, injecting different

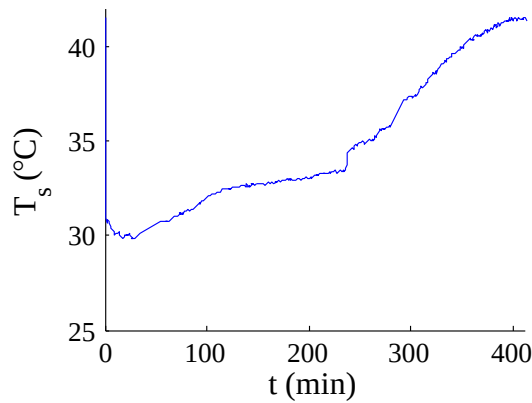


Figure 7.13: Evolution of surface temperature during LIF drying experiment

quantities of water into the cube may provide a more detailed insight. The tunnel was also limited to free-stream temperatures of up to 45°C, since increasing the power of the heating elements beyond this risked melting the PVC piping of the tunnel. This may be improved in future studies by distributing more heaters around the flow loop.

Fig. 7.14 provides temperature fields of the drying gas surrounding the cube, using the single colour LIF technique and toluene as the tracer molecule. The temperature fields are averaged over 10 images. The cube perimeter can be made out in the centre of the image and the shadow to its right where the laser sheet is blocked. Since the noise in these LIF images was particularly significant, a median filter was applied. The plots in fig. 7.14 are not quite as good quality as those for the heated copper cylinder in section 6.2. Attaining accurate and spatially resolved temperature plots was more difficult during the drying experiment for two main reasons: first of all, as mentioned in the previous paragraph, the wet bulb temperature was higher than had been intended, yielding a significantly smaller temperature difference. Secondly, since the porous foam is quite light and soft, there was some difficulty in trying to secure it perfectly orthogonal to the camera and in avoiding displacements due to flow-induced vibrations.

These effects can be seen at the border of the cube, where the signal and hence temperature is apparently very high or low. This is due to some minor displacement of the cube over the time during which a set of images was taken, as the cube vibrates slightly. It is also due to a slight misalignment between the cube positions in the captured image and the reference isothermal image used for normalizing the fluorescence signal ($I^* = I/I(T_{ref})$). Some of

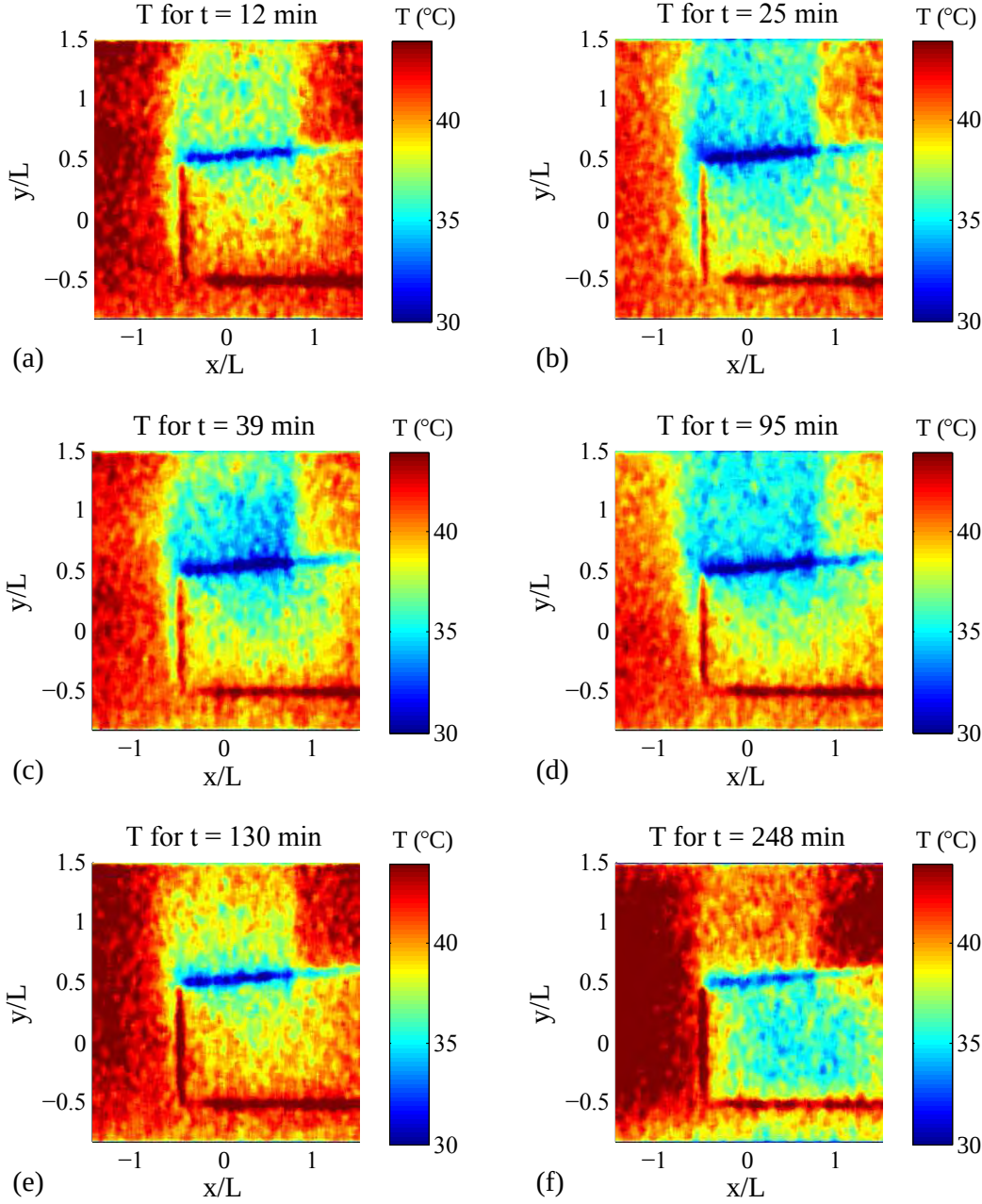


Figure 7.14: Temperature fields of the gas surrounding a drying porous cube at different stages of drying time for a Reynolds number and free-stream temperature of $Re = 530$ and $T_\infty = 44^\circ\text{C}$, respectively.

the fluorescence very close to edges may also be blocked if the edges are not perfectly perpendicular to the camera's line of sight. With some refinement of the cube suspension system inside the test section, an increased free-stream temperature and also optimization of the laser-imaging setup (as discussed in section 6.2), more accurate temperature fields with higher spatial resolution can be produced.

Despite these limitations, the boundary layer and wake region at the rear of the cube can still be identified in fig. 7.14. Along the left side of the cube parallel to the flow, the boundary layer appears to be significantly thin at the bottom corner and then increases in thickness along the cube length. Hence, the local heat transfer is highest at the corner. This agrees well with the observations in section 7.2 where it was shown that the cube edges and corners are drying at a faster rate than the rest of the surface. At the front face, the boundary layer appears to be very thin compared to the sides, where there is presumably high stagnation heat transfer. The cube edges may also be cutting off some of the increased fluorescence signal in the thin boundary layer of the front face. The turbulent wake region can also be seen clearly in fig. 7.14. Overall, the heat transfer appears to be more substantial at the leading side of the cube.

As the drying time increases in fig. 7.14, the thermal gradients start to decrease, since the surface temperature is increasing, as illustrated in fig. 7.13. At the beginning of the experiment, in fig. 7.14a, the temperature gradients appear to be slightly lower. This is because the surface temperature still has not cooled down to the wet-bulb temperature. As the experiment enters the falling-drying rate period, the temperature in the boundary layer and wake region becomes significantly higher. This can be seen in fig. 7.14d, which is approximately at the beginning of the FRP. Then as the drying time increases further, the temperature difference becomes too small to accurately measure.

In summary, these temperature fields clearly show a thinner boundary layer on the front or flow-facing side of the cube compared to the rear side. Therefore, a significantly faster drying rate on the front side of the cube compared to that observed in section 7.2 would be expected. This suggests that the internal mass transfer is more dominant in controlling the surface temperature distribution during the drying process. It may also be that the turbulent mixing in the wake region is particularly intense due to the blunt shape of the cube, meaning the rate of heat transfer and hence drying is quite similar at the rear of the cube.

By determining the rate of mass transfer simultaneously, an energy balance can be used to calculate the local heat transfer coefficients. Furthermore, with enhanced spatial resolution, the local heat transfer coefficients may be directly calculated from the thermal gradient in the boundary layer, as discussed in section 6.2.3. Nusselt number profiles along the surface, similar to those for a cylinder in section 6.2.3, may be used to quantify the distribution of

local heat transfer along the surface as a function of drying time. Local Sherwood numbers may also be inferred by estimating the local vapour concentration from local temperature. This would give a more detailed explanation of how local heat transfer is influencing the drying rate.

7.4 Heat and mass transfer analysis

During the constant-drying rate period, a steady mass transfer rate is observed and the surface temperature is almost uniform, both spatially and temporally. This quasi-steady state period allows for estimation of heat and mass transfer coefficients using equations 4.3 - 4.8 from section 4.1.3. A sample plot of the heat and mass transfer coefficients during this period is shown in fig. 7.15 for an intermediate Reynolds number (over the range of this study) of $Re = 500$ and for $T_\infty = 59^\circ\text{C}$.

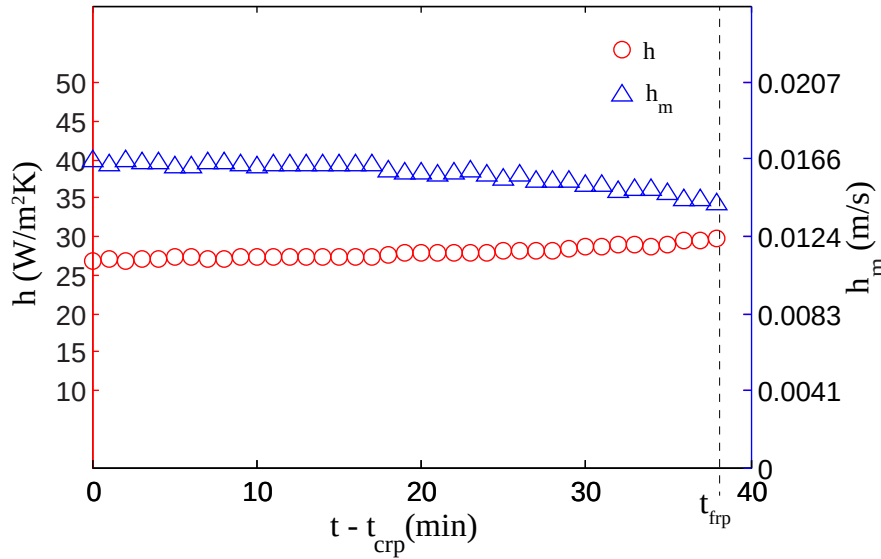


Figure 7.15: Heat transfer (red) and mass transfer (blue) coefficients, h and h_m , respectively, plotted versus the time into the CRP, $t - t_{crp}$.

As demonstrated in fig. 7.15, h increases slightly over the course of the CRP. This is due to the slight increase in surface temperature during the CRP (as discussed in section 7.2 and 7.1), which leads to a decrease in the temperature difference between surface and the free-stream air. Conversely, the mass transfer coefficient, h_m , decreases slightly since the

increase in temperature at the surface leads to an increase in vapor pressure and hence vapor concentration, ρ_{vs} (i.e. more vapour can be held by the air near the surface). This leads to a higher vapour concentration difference between the surface and free-stream air. These changes are not significant, however. In this particular case, both h and h_m change by less than 10% over the CRP. For less intensive drying conditions - lower Reynolds number and air temperature - these values change by even less since there is less sensible heating, $\dot{Q}_{st}$, of the cube.

The average Nusselt number and Sherwood Number over the duration of the CRP, Nu_{av} and Sh_{av} respectively, are plotted versus Reynolds number in fig. 7.16a and free-stream temperature in fig. 7.16b. As expected, both the Nusselt and Sherwood number increase with Reynolds number in fig. 7.16a. The uncertainty range of Nu_{av} and Sh_{av} , as well as Re , are represented by the error bars. Also included in fig. 7.16a is a correlation for the average Nusselt number to a cube by Saha [175] (valid for a Reynolds number range of $50 < Re < 325$). Though the trend with Re shows some variance, the values of Nu_{av} for lower Reynolds numbers ($Re \leq 300$) agree quite well with the correlation by Saha [175]. It appears that for $Re > 300$, the fit would predict slightly lower Nusselt numbers. However, the fit is still likely to lie within the error range of Nu_{av} as Re is increased.

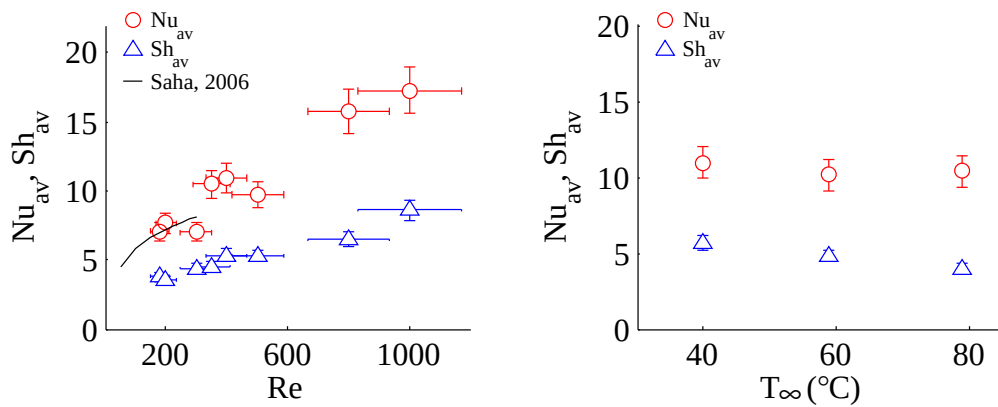


Figure 7.16: Average Nusselt number (red), Nu_{av} and average Sherwood number (blue), Sh_{av} , during the CRP versus (a) Reynolds number, Re and (b) free-stream temperature, T_{∞} . The error bars for Nu_{av} , Sh_{av} , and also Re , are included (error bars for T_{∞} are omitted since its error is particularly small). Nu_{av} is also plotted versus a fit (black line in a) of the average Nusselt number for forced convection to a cube by Saha [175] which is valid for a Reynolds number range of $50 < Re < 325$.

In fig. 7.16b the Nu_{av} and Sh_{av} values are plotted versus free-stream temperature, T_∞ . The Nusselt number, Nu_{av} , as expected remains more or less constant. There is, however, a very slight decrease in the Sherwood number, Sh_{av} , which may be some effect of the the greater sensible heating, $\dot{Q}_{st}$, that occurs for higher free-stream temperatures due to faster drying near the cube edges (see fig.7.9e & f). As a result, the saturation temperature and hence vapour concentration at the surface, ρ_{vs} , is significantly greater leading to a larger concentration difference, $\rho_{vs} - \rho_{v\infty}$ and therefore lower mass transfer coefficient. It is also possible that for higher free-stream temperatures, the assumption of saturated vapour and wet-bulb temperature at the surface is not valid.

The slightly differing trends between the Nusselt number and Sherwood number and the free-stream conditions, suggests that the analogy between heat and mass transfer (Chilton-Colburn analogy) does not hold during the CRP. This analogy can be examined more closely by testing the validity of equation 2.5 (section 2.1.3) for each experiment, by calculating a heat and mass transfer analogy factor C_{hm} , such that

$$C_{hm} = \frac{h}{h_m} \frac{1}{\rho_a c_{p,a}} \left(\frac{Sc}{Pr} \right)^{-2/3} = \frac{h}{h_m} \frac{1}{\rho_a c_{p,a}} Le^{-2/3} \quad (7.3)$$

where $C_{hm} \cong 1$ if the Chilton-Colburn analogy is obeyed. These values are plotted in fig. 7.17a for different Reynolds numbers, Re , and for a range of free-stream temperatures, T_∞ , in fig. 7.17b.

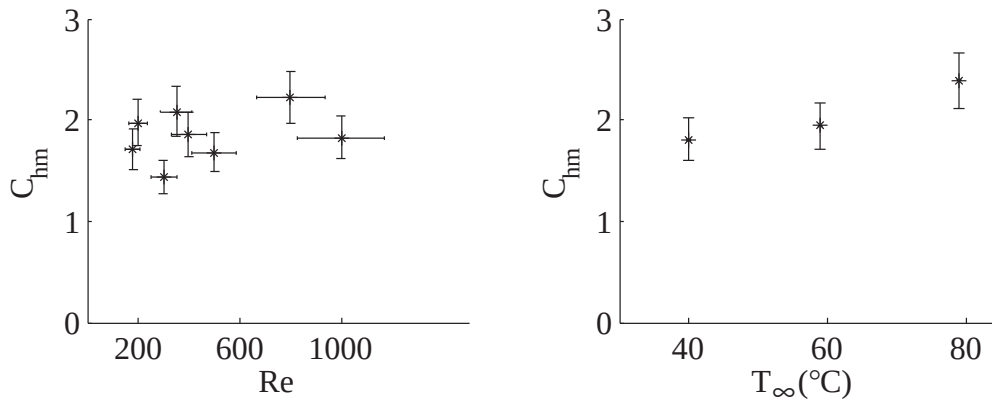


Figure 7.17: Heat and mass transfer analogy factor C_{hm} versus Reynolds number, Re , and free-stream temperature, T_∞ .

There is quite a lot of variance in the values of C_{hm} versus Re , as shown in fig. 7.17a. A relatively high level of variance was expected, given the uncertainty of C_{hm} (see table 5.2). A general increase with Reynolds number, can be identified, to some extent. On the other hand, fig. 7.17b shows a more clear increase of C_{hm} with free-stream temperature. Fig. 7.17 suggests that the analogy between heat and mass transfer does not hold during the CRP for most values of Re and T_∞ . However, for the lower Reynolds numbers and free-stream temperatures, the analogy factor, C_{hm} , is closer to unity. This suggests that the heat-mass transfer analogy is more likely to hold for less intensive drying conditions. It can be seen from the evolution curves of T_s (fig. 7.3c & d in section 7.1) that for the lowest values of Re and T_∞ (180 and 40°C, respectively) the CRP is more ‘pronounced’ than the other curves. That is to say, the surface temperature θ remains more uniform throughout. This suggests a more analogous heat and mass transfer process for lower values of Re and T_∞ . Using the analogy as a means to determine h from h_m , or vice versa, (see for example Kaya et al. [19, 45], Chandra Mohan and Talukdar [20] or Sahin et al. [44]) may only be a valid assumption for modeling low temperature natural or mixed convective drying problems.

A likely explanation for this trend is that the rate at which capillary water is being transferred to the surface, and then evaporated, may not be able to match the rate of heat transfer to the cube. This effect may lead to higher temperature and lower moisture content at the cube corners where the capillary water must travel the greatest distance from the core. This effect may be accelerated at higher values of Re and T_∞ as demonstrated in figs. 7.10 - 7.12.

Therefore, the heat and mass transfer analogy may be lost at higher flow rates and air temperatures due to this effect. It may be possible that at some critical value of Reynolds number or temperature difference, $(T_\infty - T_s)$, the analogy is lost and hence the heat transfer coefficient and mass transfer coefficient are no longer increasing at the same rate with respect to T_∞ or Re . Such critical values will depend on the average internal mass transfer resistance of the drying material, which depends on the properties of the porous material (e.g. permeability, pores size, pore size distribution, etc.). By taking higher resolution images and ensuring more accurate cube dimensions, this could be explored in greater detail and with more accuracy. Nevertheless, this finding demonstrates the value of examining the influence of local heat transfer behaviour on the drying rate.

Chapter 8

Conclusions and further work

This study has explored two distinct aspects related to the drying process: Firstly, an investigation of local heat and mass transfer in the drying of a porous cube has been conducted and, secondly, a demonstration has been provided of the concept of laser-induced fluorescence temperature field measurement in low temperature convective heat transfer problems, such as drying.

An experimental model has been developed which measures the instantaneous moisture content and surface temperature distribution of a drying porous cube, using a sensitive mass measurement system and infra-red thermography. An experimental setup has been designed which investigates the relationship between laser-induced fluorescence (LIF) intensity and temperature for toluene-seeded nitrogen gas, excited with a 266 nm laser sheet and imaged with both CCD and intensified CCD cameras. Upon successfully validating the trends between LIF intensity and temperature, for low temperature ranges, this experimental setup was then extended to a flow tunnel and used to measure temperature fields surrounding a heated cylinder and a drying porous cube. This was also tested using an emerging fluorescent tracer material, anisole. A novel method for improving the sensitivity of the temperature measurements by varying the camera gate or exposure was also proposed. This was found to increase the sensitivity significantly.

The relationship between fluorescence intensity and temperature was found to agree well with previous literature for a significantly smaller temperature scale. The signal ratio of two different spectral regions of the fluorescence was also determined as a function of temperature

and found to agree well with previous literature. Temperature fields, therefore, could be inferred from fluorescence intensity (single-colour technique) and from the intensity ratio of two different spectral regions of fluorescence (two-colour technique). Using the single-colour technique, the temperature field measurements clearly demonstrated the characteristic heat transfer behaviour of convection from a heated cylinder. At lower temperature differences ($10 - 20^{\circ}\text{C}$), the temperature measurements became quite inaccurate, for the given laser-imaging setup. Two-colour measurements were rather inaccurate at low temperatures. However, the temperature distribution could be inferred to some degree from the signal ratio. With an optimized laser-imaging setup, more accurate temperature measurements can be attained.

A constant-drying rate period (CRP) and falling-drying rate period (FRP) was observed in the convective drying experiments. During the CRP, a constant drying rate and an almost constant and spatially uniform temperature was observed. In the FRP, the temperature increased towards the free-stream or dry-bulb temperature and the temperature distribution became non-uniform. This coincided with a decrease in the drying rate as the cube approached equilibrium moisture content. Heat and mass transfer analysis of the CRP demonstrated Nusselt numbers and Sherwood numbers ranging from $7 - 17.25$ and $3.5 - 8.6$, respectively for a Reynolds number range of $180 - 1000$.

Some key findings of this experimental work include:

- Temperature measurement using LIF can be performed at low temperature ranges, particularly with the single colour technique. The error of the temperature measurement ranged from 2°C to 10°C for temperatures of 20°C to 95°C , respectively. This is comparable to the relative error in previous studies where the technique was applied to significantly larger temperature scales.
- A novel method for increasing the sensitivity of LIF temperature measurements by making use of the temperature-dependent time response of the fluorescence signal was demonstrated. Since the fluorescence lifetime decreases exponentially with temperature, varying the gate to cut off more signal with temperature can lead to a more sensitive relationship between fluorescence intensity and temperature. By increasing the gate delay by 80 ns , a 50% increase in sensitivity between fluorescence intensity

and temperature was observed. This technique has not been performed to date and could be particularly useful for improving the quality of temperature fields where the temperature scale is small and hence the temperature-sensitivity is limited.

- It was also shown that the LIF technique could be used to measure local Nusselt number distributions across the cylinder surface.
- In the convective drying experiments, near the surface edges (or cube corners), the rate of temperature increase and hence drying appears to be greater than near the centre. This effect was also observed, but to a much lesser extent, in the CRP. It is hypothesised that this is what leads to a slight increase in temperature over the course of the CRP.
- It was found that as the Reynolds number and free-stream temperature were increased, the analogy between heat and mass transfer during the CRP is less likely to hold. This important finding suggests that for higher air temperatures and flow rates, numerical models must solve for heat and mass transfer coefficients explicitly rather than using the heat and mass transfer analogy. It is likely that this is related to the faster drying rate near the corners. The rate of capillary transport of water to the corners may not be sufficient to match the rate of convective heat transfer and hence evaporation at the corners. Since a larger rate of heat transfer exists for larger Reynolds numbers and free-stream temperatures, this effect is amplified.
- LIF temperature field measurements of the drying cube, showed a much thinner boundary layer at the corner of the leading face of the cube, which agrees with these observations. Furthermore, this demonstrated the value of using LIF temperature field measurements to understand and validate local heat transfer theory.

This study has developed the framework for a novel approach to investigating two-dimensional convective heat transfer in gaseous flows. It has also highlighted the value of understanding the local heat transfer behaviour in drying and identified possible theories as to why the heat and mass transfer analogy can or cannot be assumed in drying. By making refinements to the laser-imaging setup and the convective drying test section, this technique could be used to provide in-depth analysis of local heat and mass transfer for a variety of geometries, drying materials and flow configurations.

8.1 Further work

This study has demonstrated a concept for visualizing and quantifying two-dimensional heat transfer in drying, as well as other heat transfer problems. As the results demonstrated and the error analysis (see section 5), there is scope for refining the accuracy of the experimental setups. In particular, the two colour technique demonstrated quite high inaccuracy, though a temperature distribution could be, to some extent, inferred. While there are several topics that this experimental work could be extended to - and these will be discussed briefly here - the first recommendation for future work would be a general optimization of the LIF system to attain more accurate and more spatially-resolved temperature field measurements. Some refinements to the drying experimental setup are also recommended. The following is a list of various refinements that can be made with a brief description of how this would improve the experimental accuracy and quality of the measurements:

LIF experimental refinements:

- Laser pulse energy - Increasing the pulse energy of the 266 nm beam, by using a higher energy laser, would ensure a larger fluorescence signal, $I(x,y)$, relative to the noise of the imaging device. This would be especially beneficial for the red channel (305 - 375 nm) fluorescence of toluene, which has a relatively low signal.
- Frame rate of imaging device - The frame rate of the iCCD camera was limited to about 0.2 Hz which made it impractical to take large amounts of averages and also limited the temporal resolution of fluorescence images. Increasing this would improve the accuracy. Furthermore, if single fluorescence images had a sufficiently low uncertainty or signal-to-noise ratio, instantaneous temperature fields could be attained. This would reveal interesting information in the unsteady wake region of the drying cube, which could be supplemented with particle-image velocimetry (PIV) measurements of local velocity and vorticity.
- Optimized UV lens - The UV lens used in this study was a combination of an aspheric and concave lens (Edmund Optics NT84-339 and NT48-322) placed inside a lens tube, which have not been fully optimized for the iCCD camera. Higher quality UV

lenses with external control of the focal length would provide greater spatial resolution. Increased aperture size would also allow more fluorescence signal to be captured. Also easier and finer external control of the focal length and aperture would make it is easier to optimize the image.

- Anisole as a tracer - The results revealed that anisole is a suitable tracer and though its sensitivity is less over lower temperature ranges, its fluorescence quantum yield, Φ , ensures a higher signal-to-noise ratio. Another key advantage is the larger wavelength of its fluorescence spectrum. This may make it suitable for detection with CCD cameras, which are significantly less sensitive below 300 nm. Unfortunately, the correct filter selection for anisole was unavailable in this study for two-colour measurements. The red channel spectrum in the case of anisole is approximately > 320 nm at room temperature. Therefore, the quantum efficiency of a CCD camera like that used in this study might be sufficient for imaging this portion of the spectrum. Hence, two-colour LIF could be performed, especially if other sources of uncertainty (e.g. laser energy, optics, etc.) are minimized.
- Use of time resolved fluorescence signals - A novel method was proposed in section 6.2.1 in which the temperature-dependent lifetime of the fluorescence signal is utilized to increase temperature sensitivity. By making a rigorous calibration between fluorescence signal and the intensifier gate delay for different temperatures, this technique could be used to make accurate local temperature measurements, $T(x,y)$, with the increased sensitivity between I and T .
- Greater spectral separation for two-colour LIF - Assuming the fluorescence signal is of sufficient signal-to-noise ratio, by using a higher long-pass filter for the red channel and a narrower band pass filter for the blue channel, the sensitivity of R_I with temperature would be greater [31].
- Use of two cameras - Finding the signal ratio of instantaneous images of fluorescence by using two separate cameras, optimized for each wavelength channel, would reduce the spatial noise considerably.
- Matching the field of view to thermal boundary layer thickness - By imaging over

a finer scale of the solid-fluid interface (or using a higher resolution camera), the thermal boundary layer could be imaged with much greater resolution. This would allow determination of the thermal gradient normal to the surface, $dT/dy|_{y=0}$ (or $dT/dr|_{r=0}$), from which the local heat transfer coefficient could be determined directly. This would eliminate the need to estimate the rate of heat input or heat loss at the heat transfer surface of interest.

- The laser sheet passes across the cylinder (or cube) centre. Due to the finite acceptance angle of the fluorescence optics, light emitted near the cylinder may be collected with reduced efficiency. Inclining the angle slightly between the line of sight of the camera and the front edges may prevent this and lead to improved detection of fluorescence signal in the boundary layer.
- Better control and limitation of O₂ - Including a precision oxygen sensor (which was unavailable) in the flow tunnel or test vessel would ensure zero oxygen and hence maximum fluorescence signal, as well as temperature sensitivity.
- Increasing the drying temperature range - The temperature fields measured for the drying experiment were limited by the small temperature difference. This was due to a limited heat input (since larger heating power risked melting the pvc piping) and also some free water dripping from the cube into the tunnel and causing the wet-bulb temperature at the cube surface to rise. By distributing more heaters around the tunnel (rather than increasing the heating power) and improving the water injection technique, the temperature difference could be increased for the drying experiments, making it possible to infer more accurate temperature fields.
- Higher quality optical elements - The optical materials, such as imaging filters, imaging lenses and the fused quartz tube, could be replaced with higher quality parts that would lead to less optical distortion and an increase in transmission.

Drying experimental refinements:

- Improved spatial resolution - The thermal imaging camera (FLIR A-40) was limited

to a minimum focal distance of 300 mm. This meant that, unless a large cube was used, the resolution was limited to only a small portion of the full camera image. Therefore, the first experimental refinement would be to use the FLIR SC6000 high speed camera, which has a much greater resolution and is not limited in its focal distance. The development of wet and dry patches at the surface, as discussed in section 2.2.4, may be identified if the resolution is sufficient. Given the much smaller field of view, a refined suspension of the cube would be required to further minimize any slight displacements of the cube, as these displacements would be more significant over the smaller field of view.

- More precise cube length - As shown in section 5.1, the cube side length uncertainty leads to significant uncertainty in the cube area, which in turn affects the the heat transfer coefficient. The cubes are typically cut carefully with a fine blade against a square edge and measured several times with a digital vernier calipers. This typically produces a relatively accurate side length. However, relative to the cube length, L , of 10 mm, a smaller level of uncertainty is required, since the surface area depends on the square of L . Making the cube larger lengthens the drying time considerably, which can be impractical. This also requires scaling up of the wind tunnel so that the proximity of walls to the cube is not too great. However, increasing L slightly, for example from 10 to 13 mm, would bring the uncertainty of h down from 17% to 14.5%. A specialized cutting tool could also be designed to ensure maximum precision. There are also some alternative materials with similar properties to the phenolic floral foam, but which can be reused after drying. This would ensure a fixed surface area.

By making some of these refinements, the local heat transfer coefficients would be calculated with more accuracy and a more detailed understanding of the local convective heat transfer behaviour would be provided. This would also allow the LIF thermometry technique to be extended to a wider range of low temperature convection problems.

Aside from experimental refinements, there is a wide variety of aspects to consider and topics which this work could be extended to. The convective heat transfer experiments involving the heated cylinder (see chapter 6.2) could be repeated for an optimized two-colour imaging setup and increased laser pulse energy. This would be performed using both toluene

and anisole and once sufficient accuracy is attained, this could be extended to a refined drying experimental setup. The aim would then be to measure temperature fields surrounding a drying cube for a range of Reynolds numbers (and free-stream temperatures). For the same parameter settings the surface temperature evolution could be studied with the higher resolution IR camera. The ultimate objective here would be to accurately characterize the local heat transfer behaviour of the drying cube as a function of the free-stream conditions. This would provide a base experimental model for forced convective drying. The drying problem could then be varied in terms of its geometry, material and flow configuration. The local heat transfer behaviour could then be compared with that of the base model.

After establishing a more accurate LIF setup, the technique could also be applied to more specific heat transfer problems. For example, it could be applied to study temperature fields in impinging jets. It could also be applied to forced convection from more complex geometries or arrays of geometries such as those found in heat exchangers.

The work could also focus more specifically on the LIF thermometry technique. Many of the refinement suggestions listed above could be explored in more detail. In particular, the use of anisole with the regular CCD camera (PCO Sensicam), with its longer wavelength of fluorescence, could be investigated. Since the CCD camera has a frame rate of up to 8 Hz, a large number of frame averages could be taken for steady-state problems to reduce the uncertainty. Other CCD cameras with comparable sensitivity but high frame rates could be used to image high speed or unsteady heat flows. As noted by Faust [167] intensified cameras often sacrifice some spatial resolution for increased sensitivity. Therefore, another advantage here would be an improved spatial resolution which may be highly valuable for imaging thermal boundary layers.

Publications and Awards

The list below contains publications made and an award received during this study. Many of these are related to convective heat transfer of synthetic jet pairs, an additional study which was undertaken (see Appendix A).

- E. Fanning, T. Persoons, D. B. Murray. Heat transfer and flow characteristics of a pair of adjacent impinging synthetic jets. *International Journal of Heat and Fluid Flow*, 54(1), 153–166, 2015.
- S. Alimohammadi, E. Fanning, T. Persoons, D. B. Murray. Heat Transfer in Adjacent Synthetic Jets. *14th UK Heat Transfer Conference, Edinburgh, 2015*
- E. Fanning, T. Persoons, D. B. Murray. Forced convective drying of a porous cube: Non-intrusive temperature and mass transfer measurements. *Proceedings of ASME International Mechanical Engineering Congress and Exposition, Proceedings (IMECE), Montreal, 15-20 Nov*, Vol 8B.
- 2013 Hartnett-Irvine Award - The Hartnett-Irvine Award is awarded annually by the International Centre for Heat and Mass Transfer (ICHMT) to the best paper on heat or mass transfer presented at a conference, symposium or seminar organized or co-sponsored by ICHMT of the year.
- ¹E. Fanning, T. Persoons, D. B. Murray. Heat transfer and flow characteristics of a pair of adjacent synthetic jets. *8th World Conference on Experimental Heat Transfer, Fluid Mechanics, and Thermodynamics*.
- E. Fanning, T. Persoons, D. B. Murray. Heat transfer characteristics of a pair of impinging synthetic jets: Effect of orifice spacing and impingement distance. *Journal of Physics: Conference Series*.

¹Article for which Hartnett-Irvine Award was received

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Appendices

Appendix A: Side study on the heat transfer performance of adjacent impinging synthetic jets

An additional study of the cooling performance of two adjacent impinging synthetic jets was performed using infrared thermography. The fluid flow was quantified and studied using particle image velocimetry (PIV). The study revealed that reducing the distance between the hot surface and the jet pair orifices lead to a large enhancement in heat transfer, due to a trailing effect of adjacent vortices. Larger orifice-to-impingement distances and orifice-to-orifice distances revealed very different flow fields, leading to different heat transfer mechanisms. More information on this topic can be found in references [176, 177, 179, 180].

Appendix B: Additional information and results from drying experiments

Appendix B1: IR camera correction factor

For most of the duration of a typical drying experiment, there is a high concentration of vapour near the surface. This affects the propagation of infra-red radiation towards the camera which can lead to a small disagreement between the thermocouple at the cube surface and the infra-red camera measurement. See for example fig. 1a which shows a plot of the thermocouple temperature measurement, T_{tc} , and the surface temperature, T_s , at the position of the thermocouple. A clear disagreement is found between both measurements.

For this reason a correction factor is derived to correct for this disagreement. It can be

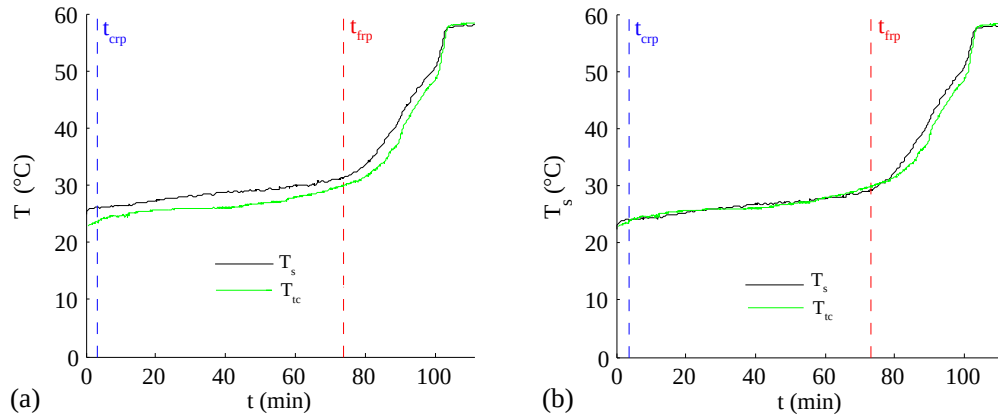


Figure 1: Plot of cube surface temperature measurement from the thermocouple, T_{ic} , alongside that of the IR camera, T_s , (a) without and (b) with the correction factor (equations 1 and 2) implemented. This example is for $Re = 300$ and $T_\infty = 59$.

assumed that at the surface, a thin layer of saturated vapour remains during the constant-drying rate period. This should lead to a constant disagreement, $\Delta T_{vap} = T_s - T_{ic}$, during the CRP which can be seen in fig. 1a. During the falling-drying rate period, the surface is no longer saturated with vapour and this saturated plane retreats inward into the material (i.e. the retreating drying front, see section 2.2.2). The vapour concentration near the surface, during the FRP, is mostly dependent on the internal diffusion of vapour as is the moisture content, M , and so they should decrease at the same rate.

Therefore, a correction factor is defined which is constant or steady-state during the CRP and which decreases as a function of moisture content during the FRP, such that

$$T_{s,corr} = T_s - \Delta T_{vap} \quad \text{for} \quad t \leq t_{crp} \quad (1)$$

and

$$T_{s,corr} = T_s - \Delta T_{vap} \left(\frac{M(t) - M_e}{M_{frp} - M_e} \right) \quad \text{for} \quad t_{frp} \leq t \leq t_e \quad (2)$$

where $T_{s,corr}$ is the corrected temperature measurement of the IR camera. This can be seen in fig. 1b. This correction can then be applied to each pixel which images the cube surface.

Appendix B2: Additional temperature distributions

Below are some additional temperature distributions of the drying cube surface, at different points in time, to accompany those presented in section 7.1. Fig. 2 shows distributions for an intermediate Reynolds number of $Re = 500$ and a free-stream temperature of $T_\infty = 59^\circ\text{C}$. (distributions for $Re = 300$ and $Re = 800$ are found in figs 7.1 and 7.2, respectively). Figs 3, 4 and 5 contain distributions for free-stream temperatures of $T_\infty = 40^\circ\text{C}$, 59°C and 79°C , respectively, for a fixed Reynolds number of $Re = 350$.

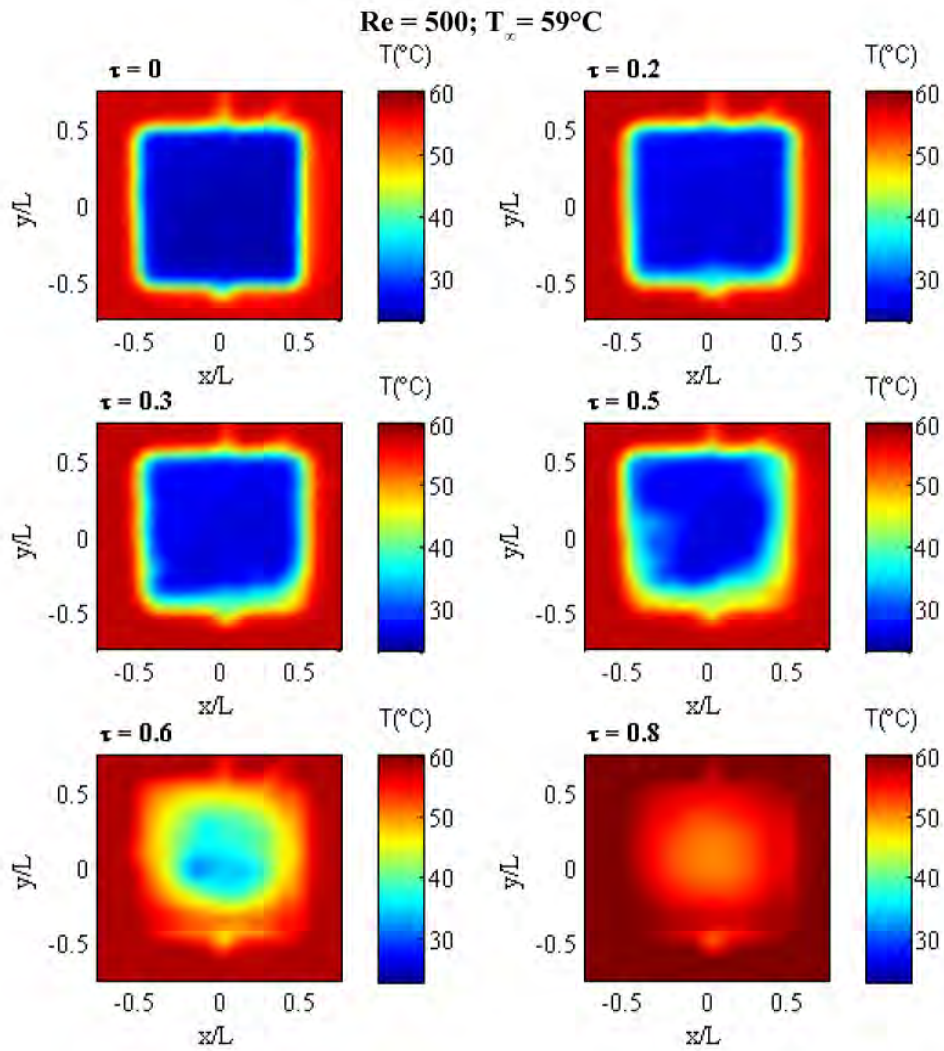


Figure 2: Temperature distribution across the cube surface $T(x,y)$ for different non-dimensional drying times, τ , for $Re = 500$ and $T_\infty = 59^\circ\text{C}$.

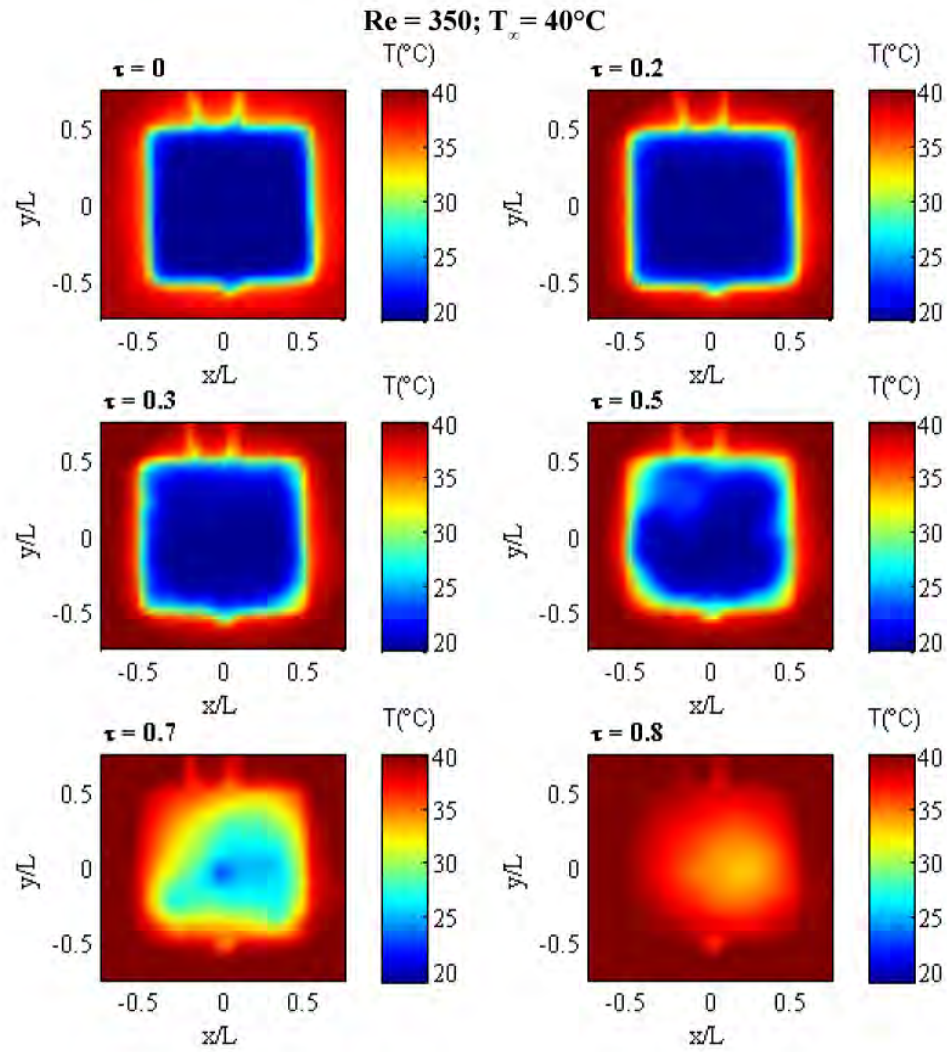


Figure 3: Temperature distribution across the cube surface $T(x,y)$ for different non-dimensional drying times, τ , for $T_\infty = 40^\circ\text{C}$ and $Re = 350$.

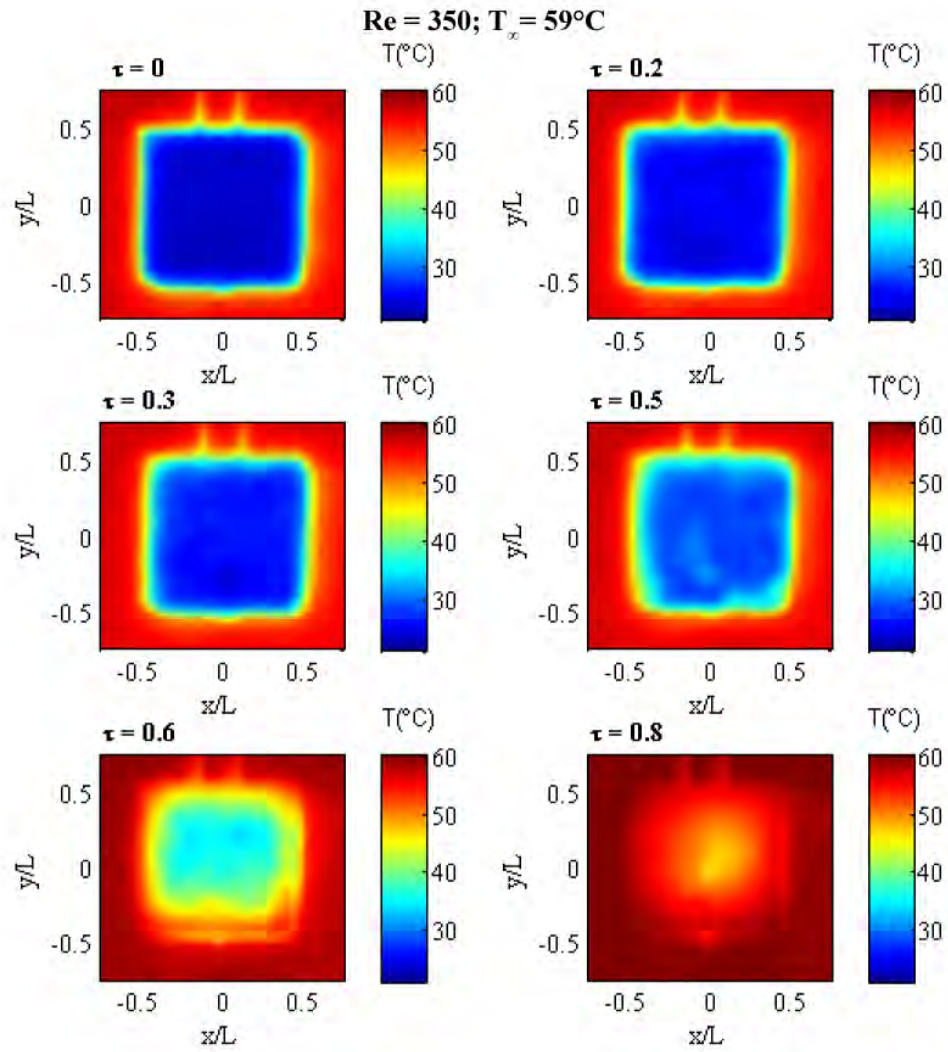


Figure 4: Temperature distribution across the cube surface $T(x,y)$ for different non-dimensional drying times, τ , for $T_\infty = 59^\circ\text{C}$ and $Re = 350$.

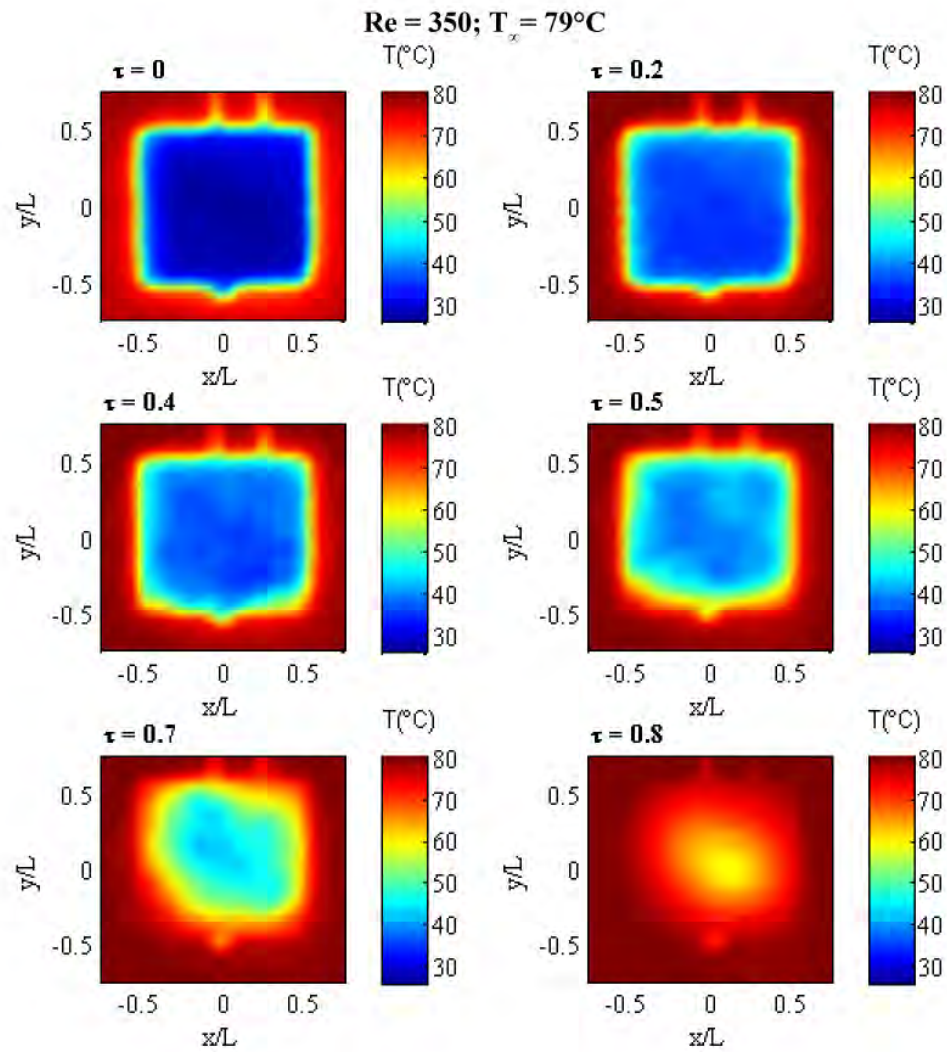


Figure 5: Temperature distribution across the cube surface $T(x,y)$ for different non-dimensional drying times, τ , for $T_\infty = 79^\circ\text{C}$ and $Re = 350$.

Appendix B3: Equipment data sheets

Precision balances KERN KB-N · KB-NM · KBJ



Compact precision balance with large weighing range, as well as EC type approval [M] and automatic internal adjustment

Features

- **KERN KBj with internal automatic adjustment:** Every time the balance is restarted it is adjusted automatically, which guarantees a high degree of accuracy and makes it independent of its location
- Ideal for mobile applications which require verification, such as ambulatory gold and jewellery purchasing
- **User guidance step by step** through Yes/No dialogue on the display
- **Ring-shaped draught shield** standard, only for models with weighing plate size $\square$, weighing space $\emptyset \times H$ 90x40 mm

Technical data

- Backlit LCD display, digit height 9 mm
- Weighing plate dimensions, stainless steel*
 $\square$ $\emptyset$ 81 mm, plastic, with conductive lacquer
 $\square$ WxD 130x130 mm*
 $\square$ WxD 150x170 mm*, see enlarged picture
- Overall dimensions without draught shield
WxDxH 167x250x85 mm
- Net weight approx. 2 kg, for details see the internet
- Permissible ambient temperature
KERN KB-N: 10 °C / 40 °C
KERN KB-NM, KBj: 10 °C / 30 °C

Accessories

- **Protective working cover** over keyboard and housing, standard, can be retrofitted, for models with weighing plate size $\square$ KERN PCB-A02, $\square$ KERN PCB-A04 $\square$ KERN PCB-A05
- **Rechargeable battery pack internal**, operating time up to 15 hours with backlight, charging time approx. 10 h, KERN KB-A01N
- only KB-N: **Rechargeable battery pack external**, operating time up to 15 hours with backlight, charging time approx. 10 h, KERN KS-A01
- **Suitable test weights**, also with calibration certificate, see the internet
- **Suitable printers** see page 157 ff.

STANDARD



Model	Weighing range [Max] g	Readout [d] g	Verification value [e] g	Reproducibility g	Linearity g	Weighing plate g	Options		
							Verification	DAKKS-Calibr. Certificate	
KERN									
KB 120-3N	120	0,001	—	0,001	± 0,003	$\square$	—	963-127	
KB 240-3N	240	0,001	—	0,001	± 0,003	$\square$	—	963-127	
KB 360-3N	360	0,001	—	0,002	± 0,005	$\square$	—	963-127	
KB 1200-2N	1200	0,01	—	0,01	± 0,03	$\square$	—	963-127	
KB 2000-2N	2000	0,01	—	0,01	± 0,03	$\square$	—	963-127	
KB 2400-2N	2400	0,01	—	0,01	± 0,03	$\square$	—	963-127	
KB 3600-2N	3600	0,01	—	0,02	± 0,05	$\square$	—	963-127	
KB 10K0.05N	10000	0,05	—	0,05	± 0,15	$\square$	—	963-128	
KB 10000-1N	10000	0,1	—	0,1	± 0,3	$\square$	—	963-128	
Note: For applications that require verification, please order verification at the same time, initial verification at a later date is not possible. Verification at the factory, we need to know the full address of the location of use.									
KB 650-2NM	650	0,01	0,1	0,01	± 0,03	$\square$	965-216	963-127	
KB 6500-1NM	6500	0,1	1	0,1	± 0,2	$\square$	965-217	963-128	
Automatic internal adjustment									
KBj 650-2NM	650	0,01	0,1	0,01	± 0,03	$\square$	965-216	963-127	

KERN & SOHN GmbH • Ziegelei 1 • 72336 Balingen • Germany • Tel. + 49-[0]7433-9933-0 • Fax + 49-[0]7433-9933-149 • www.kern-sohn.com • info@kern-sohn.com

Figure 6: Datasheet for KernKB-360-3N digital precision mass balance

ThermoVision™ A40M Technical Specifications

Imaging Performance	
Field of view/min focus distance	24° x 18° / 0.3 m
Detector type	Focal plane array (FPA) uncooled microbolometer
Spectral range	7.5 to 13 µm
Spatial resolution (IFOV)	1.3 mrad
Thermal sensitivity @ 50/60Hz	0.08° C at 30° C
Focusing	Built-in focus motor
Image Presentation	
FireWire/Ethernet output	8/16-bit monochrome and 8-bit color
Video output	RS170 EIA/NTSC or CCIR/PAL composite video
Measurement	
Temperature ranges	Range 1: -40°C to +120°C (-40 to +248°F) Range 2: 0°C to +500°C (+32 to +932°F) Optional: Up to +1500°C (+2732°F) Optional: Up to +2000°C (+3632°F)
Accuracy (% of reading)	± 2°C or ± 2%
Measurement modes	Spot, Area, Difference
Automatic emissivity correction	Variable from 0.1 to 1.0
Individual emissivity settings	Individually settable
Measurement corrections	Reflected ambient, distance, relative humidity, external optics. Automatic, based on user input
Supplementary Lenses*	
Field of view/min. focus distance	7" Telescope (7" x 5.3"/4m) 12" Telescope (12" x 9"/1.2m) 45° Wide angle (45° x 34"/0.1m) 80° Wide angle (80° x 60"/0.1m) Close-up: 64/150 mm (FOV=64 x 48 mm at 150 mm); 34/80 mm (FOV=34 x 25 mm at 80 mm) Macro: 50 micron (14.3 to 18.7 mm focus; FOV=14.3 x 10.8 mm at 14.3 mm; FOV=15.1 x 11.2 mm at 18.7 mm; IFOV=45 µm at 14.3 mm; 47 µm at 18.7 mm)
Lens recognition	Automatic lens recognition and measurement corrections

Power Source	
AC operation	AC adapter 110/220 VAC, 50/60Hz (included)
DC operation	8-30V nominal, <6W
Environmental	
Operating temperature range	-15°C to +50°C (5°F to 122°F)
Storage temperature range	-40°C to +70°C (-40°F to 158°F)
Humidity	Operating and storage 10% to 95%, non-condensing
Encapsulation	IP 40 (Determined by connector type)
Shock	Operational: 25G, IEC 68-2-29
Vibration	Operational: 2G, IEC 68-2-6
Physical Characteristics	
Weight	1.4 kg (3.0 lbs)
Size	207mm x 92mm x 109mm (8.1" x 3.6" x 4.3")
Tripod mounting	1/4"-20

User Configuration Table		
TYPE	FUNCTION	REMARK
Digital Input	TTL level • Shutter disable • Store image • Batch enable	Isolation and relay function in external module
Digital Output	TTL level • Spot/Area threshold ALARM • Internal temperature sensor ALARM • V-sync	Isolation and relay function in external module
Analog Output	• Spot/Area out: 0-5V • Internal temperature sensor out: 0-5V	Scaled to T _{low} ~ T _{high} Isolation in external module
Analog Input	• External temperature sensor out: 0-5V	Scaled to T _{low} ~ T _{high} Isolation in external module

CAMERA INTERFACES

Digital I/O ports—jackable screw terminal
3 output/1 input, 1 input/output selectable;
function is user configurable**

Analog I/O ports—jackable screw terminal
2 output/1 input; function is user configurable**

RS-232 (DB-9)—connection to PC
Camera control

DC power in—2-pin jackable screw terminal
8-30V nominal



8-button keyboard

**Ethernet jack (RJ45) or
FireWire jack (IEEE-1394)**

BNC—C-Video (NTSC/PAL)

2.5 mm DC power in
8-30V Nominal; camera needs only one power source

*All attach to standard built-in 24" lens
**See Configuration Table above



The Global Leader in Infrared Cameras

1 800 464 6372

www.flirthermography.com/A40Mdata

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Figure 7: Datasheet for FLIR A40 thermal imaging camera

MASS AND VOLUMETRIC GAS FLOW METERS

For Clean Gases

FMA-1600A Series



- ✓ Up to 18 Hour Battery Charge on Portable “-B” Models
- ✓ Ranges of 0 to 0.5 SCCM Up to 0 to 3000 SLM
- ✓ Reports Mass Flow, Volumetric Flow, Temperature, and Pressure
- ✓ <10 ms Response Time—Field Adjustable
- ✓ 130+ Gas Calibrations Including Pure and Mixed Gases
- ✓ Pressure, Temperature and Volumetric or Mass Flow Simultaneously Displayed
- ✓ NIST 5-Point Certificate Included
- ✓ No Straight Runs of Pipe Required
- ✓ No Warm-Up Time
- ✓ Turndown Ratio of 200:1 (0.5% Maximum Flow)
- ✓ RS232 Standard
- ✓ Custom Live Gas Blend Programming
- ✓ Store Up to 20 User Defined Gas Blends

The FMA-1600A Series mass and volumetric flow meters use the principle of differential pressure within a laminar flow field to determine the mass flow rate. A laminar flow element (LFE) inside the meter forces the gas into laminar (streamlined) flow. Inside this region, the Poiseuille equation dictates that the volumetric flow rate be linearly related to the pressure drop. A differential pressure sensor is used to measure the pressure drop along a fixed distance of the LFE. This, along with the viscosity of the gas, is used to accurately determine the volumetric flow rate. Separate absolute temperature and pressure sensors are incorporated and correct the volumetric flow rate to a set of standard conditions. This standardized flow rate is commonly called the mass flow rate and is reported in units such as standard cubic feet per minute (SCFM) or standard liters per minute (SLM). Standard units include a 0 to 5V output (4 to 20 mA optional) and RS232 communications. The gas select feature can be adjusted from the front keypad or via RS232 communications. Volumetric flow, mass flow, absolute pressure, and temperature can all be viewed or recorded through the RS232 connection. It is also possible to multi-drop up to 26 units on the same serial connection to a distance of 38 m (125'). These flow meters are available in a portable version (“-B” option), the battery charge will last up to 18 hours.



FMA-1603A includes 110 Vac power supply and a 1.8 m (6') cable 8-pin mini DIN connector, shown actual size.

SPECIFICATIONS

Accuracy: $\pm(0.8\% \text{ of rdg} + 0.2\% \text{ FS})$
Repeatability: $\pm 0.2\%$
Turndown Ratio: 200:1
Response Time: 10 ms typical default response time for 63.2% of a step change. A variable register allows response time to be field adjustable to a certain extent via RS232 communications. The primary trade-off for response time is signal noise
Output: 0 to 5 Vdc standard
Operating Temperature: -10 to 50°C (14 to 122°F)
Zero Shift: 0.02% FS/°C/atm
Span Shift: 0.02% FS/°C/atm
Humidity Range: 0 to 100% non-condensing
Pressure (Maximum): 145 psig
Measurable Flow Rate: 125% FS
Supply Voltage: 7 to 30 Vdc (15 to 30 Vdc for 4 to 20 mA output)
Supply Current: 35 mA typical current draw; 100 mA available supply recommended
Cable Connection: 8-pin mini DIN
Wetted Parts: 302 and 303 SS, FKM, silicone RTV, nylon 616, aluminum

D-X

Figure 8: Datasheet for Omega FMA-1613 flow meter

D

Appendix C: Additional information and results from LIF experiments

Appendix C1: Theoretical estimation of fluorescence signal for CCD camera

Since the CCD camera's sensitivity was quite limited, it was important to try and estimate beforehand whether it would be capable of detecting the toluene fluorescence signal. Therefore, using the following formula (taken from [131, 139]), the theoretical fluorescence signal (in electronic counts) could be predicted based on the specifications of the CCD camera and imaging optics and other experimental conditions (e.g. toluene concentration, laser sheet dimensions, etc.):

$$I = \left(\frac{E_0}{H h_{plk} f} \right) (n_{mol} N_{mol}) \sigma \Phi \gamma \quad (3)$$

Equation 3 is a more complex and expanded version of equation 3.1 that takes into account the conditions and specifications of the laser-imaging system. The term in the first set of brackets represents the laser sheet settings. E_0 is the energy (joules) of a laser pulse, H is the laser sheet height (meters), h_{plk} is Planck's constant and f is the frequency (Hz) of the laser light, such that $f = c/\lambda$ (where c is the speed of light and λ is the wavelength). The term in the second set of brackets represents the concentration of the tracer molecule where n_{mol} is the mole fraction and N_{mol} is the number density (molecules/m³) of the tracer. N_{mol} can be approximated from the ideal gas law, $N_{mol} = P/kT$, where P , T and k are the pressure, temperature and Boltzmann constant, respectively. σ is the absorption cross-section and Φ is the fluorescence quantum efficiency of the toluene. These are determined from temperature correlations by Koch [168] and Koban et al. [141], respectively.

The imaging efficiency γ is a product of the different imaging efficiencies and camera settings which affect the signal intensity, such that

$$\gamma = QE_{cam} T_{lens} T_{filter} \frac{\Omega}{4\pi} \frac{H_{pix}^2}{M^2} \quad (4)$$

where QE_{cam} is the quantum efficiency (%) of the camera sensor, T_{lens} is the transmission

(%) of the lens and T_{filter} is the transmission (%) of the filter (and test section quartz window). Ω is the detection solid angle which can be calculated from the focal length, aperture and object distance of the lens. H_{pix} is the sensor pixel size and M is the lens magnification. Subbing all these values into equation 3 gives the fluorescence signal, I , in terms of the number of photoelectrons. This can be converted into electrical counts using the A/D conversion factor which is found in most CCD camera data sheets.

Fig. 9 contains a plot of the theoretical fluorescence signal, I (counts). This is plotted alongside the maximum fluorescence signal (after averaging over several images), I_{max} , from toluene LIF experimental data using the CCD camera. I is calculated using the same conditions and settings from the experimental data. Note I_{max} is plotted here since where the fluorescence is maximum, it is not being affected by laser sheet attenuation or divergence.

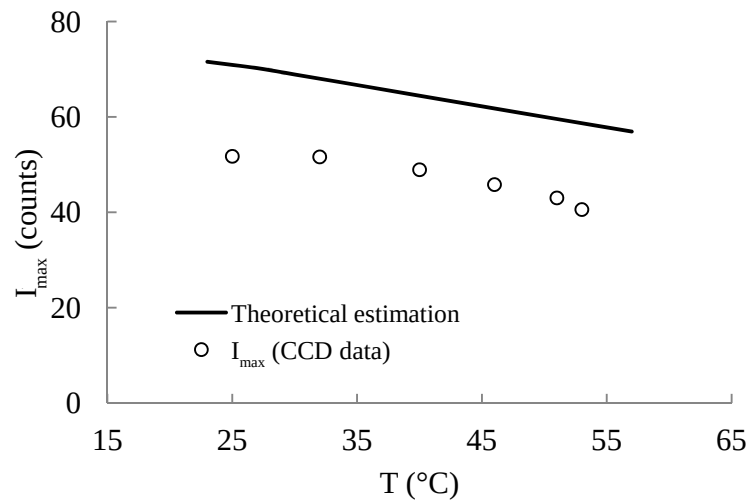


Figure 9: The theoretical fluorescence signal, I (counts), plotted alongside the maximum fluorescence signal (after averaging over several images), I_{max} , from toluene LIF experimental data using the CCD camera. I is calculated using the same conditions and settings from the experimental data. I_{max} is plotted here since where the fluorescence is maximum, it is not being affected by laser sheet attenuation or divergence.

Given, the many values that can increase the uncertainty of the theoretical estimation, fig. 9 shows good agreement with the experimental data. The rate of decrease with temperature is also comparable. Therefore, this approximate prediction is a useful step in the planning and designing of a laser-imaging system for LIF thermometry.

Appendix C2: LIF intensity versus pressure

Fig. 10 shows the fluorescence intensity, I , measured for 3 different pressures, P , above atmospheric pressure. The pressure was set by slowly and carefully venting small amounts of the gas mixture from the test section (see section 4.2.2) in order to reduce the pressure which is recorded from the regulator. Though there are only three data points, there is clearly no increase or decrease of the fluorescence intensity with pressure in fig. 10. This confirms the findings of Strozzi et al. [29] that above 1 bar (atmospheric) pressure, the fluorescence intensity does not change.

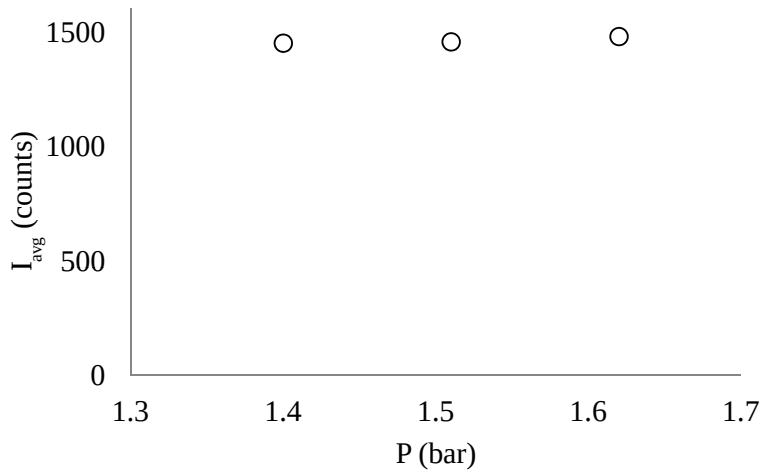


Figure 10: Spatially averaged fluorescence intensity, I_{avg} , versus pressure, P .

Appendix C3: Local LIF signal ratio versus temperature

Fig. 11 demonstrates plots of local signal ratio, $R_I(x, y)$, versus temperature, T , from the LIF thermometry validation experiments 6.1. Note some median filtering is used to produce a smoother plot. For each measurement, the temperature is uniform across x and y resulting in a flat or plain distribution when $I_{blue}^*(x, y)$ is divided by $I_{red}^*(x, y)$, while the overall or average value is decreasing with temperature.

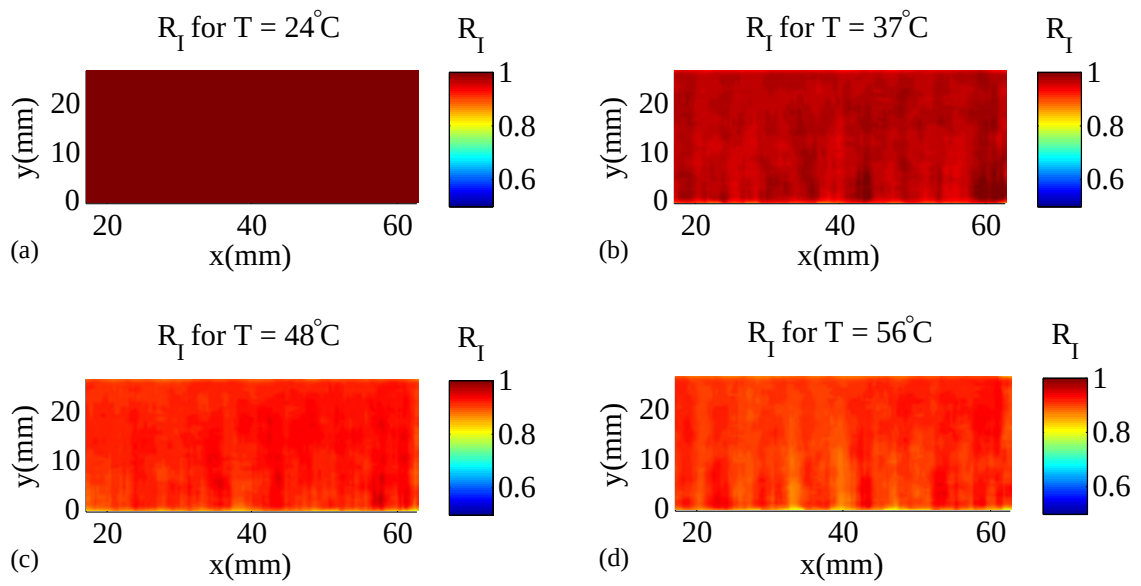


Figure 11: Signal ratio, $R_I(x, y)$, for different temperatures from the LIF thermometry validation experiments 6.1.

Appendix C4: Equipment data sheets

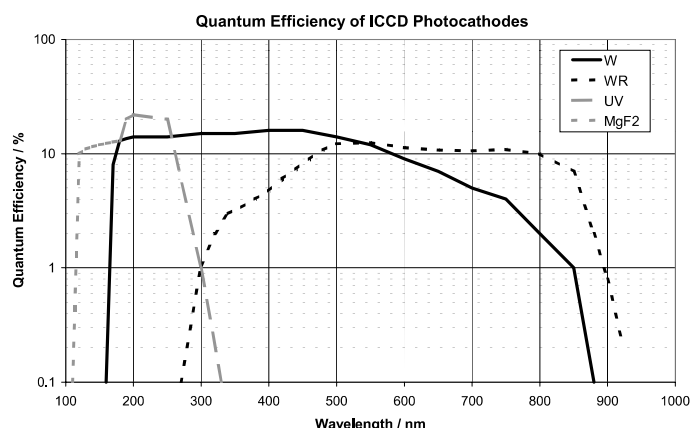


DH520

The DH520 intensified CCD camera is designed for low-light spectroscopy applications that require fast gating. The 1024 x 256 array matches the 4:1 aspect ratio of the spectrograph's focal plane exit, providing maximum use of the CCD's sensitive region. A wide selection of photocathode options provides a range of solutions, whether a broad spectral response or optimization in a particular region is required.

● Sensor	Active Pixels	Ø18 mm	690 x 256
		Ø25 mm	960 x 256
	Effective Pixel Size (µm)	Ø18 mm	26
		Ø25 mm	26
	Active Area (mm)	Ø18 mm	18 x 6.7
		Ø25 mm	25 x 6.7
	Linearity (% , maximum) *1	1	

● Quantum Efficiency *2



Peak Quantum Efficiency at room temperature (%)

Intensifier Model Number		Minimum	Typical
18F-03 (W/HR/P43)	@ 440 nm	13.5	15
18H-03 (W/HR/P43)	@ 440 nm	17	18
18F-04 (W/HR/P46)	@ 440 nm	13.5	15
18F-13 (WR/HR/P43)	@ 520 nm	11	12
18F-23 (UV/HR/P43)	@ 230 nm	10	20
25F-03 (W/HR/P43)	@ 440 nm	13.5	15

Figure 12: Andor DH520-18T datasheet (page 1)

● Image Intensifier Details	Model Number [DH520-]	Intensifier Type				
		18F-03	18F-04	18F-13	18F-23	25F-03
	Diameter (mm)	18	18	18	18	25
	Minimum Optical Gate Width (ns) *3	U 2	2	5	N/A	3
		F 5	5	10	10	7
		H 50	N/A	N/A	N/A	N/A
	Photocathode*4	W	W	WR	UV	W
	Input Window	Quartz*	Quartz*	Glass*	Quartz*	Quartz*
	Spectral Range (nm)	180-850	180-850	250-920	180-320	180-850
	Phosphor	P43	P46	P43	P43	P43
	Phosphor Decay Time (to 10%)	2 ms	200 ns	2 ms	2 ms	2 ms
	Max Gate rep rate (kHz)	25	25	25	25	25
	Max Gain (typical; cts / photoe)*5	1250	500	850	500	850
	Intensifier Tube Resolution Limit (μm) *6	25	30	25	25	35
	EBI (photoe / pix-s)*7	0.02	0.02	0.1	0.01	0.02

*Also available: glass input windows; MgF₂ input windows for operation to below 120 nm; fibre optic windows for easy optical coupling.

● Features & Benefits	TE cooling to -40°C	Negligible dark current, without the aggravation or safety concerns associated with LN ₂			
	High resolution 18 and 25mm intensifier	Highest available intensifier tube resolution			
	Close-Coupled Gating™	Gating as short as 1.2ns gating - the ultimate temporal resolution for both imaging and spectroscopy applications			
	Single photon sensitivity	Pre-requisite for ultra-low light applications			
	25kHz sustained gating rate	Compatible with high speed lasers			
	Software gain control	Ease of use			

● Noise	System Readout Noise *8	Typical	Maximum:
	31kHz pixel readout rate	8	12
	1MHz pixel readout rate	24	35

● Temperature & Dark Current *9	Computer Power Supply		External PSU PS150	
			18mm	25mm
	Air-cooled Temp [Dark Current (typ; e-/pix/s)] (ambient air @ 20°C)	-10 [6.1]	-5 [10.5]	-20 [1.9]
	Water-cooled Temp [Dark Current (typ; e-/pix/s)] (@ 10°C, 0.75 l/min)	-30 [0.6]	-25 [1.1]	-40 [0.15]

● Operating & Storage Conditions	Operating temperature	0 to 30°C ambient
	Relative humidity	< 70% (non-condensing)
	Storage temperature	-25 to +55°C

● Computer Requirements	Minimum:		Also:
	Windows 95/98:	100MHz Pentium + 64Mbytes RAM	
	Windows NT/2000:	100MHz Pentium + 128Mbytes RAM	
	Recommended:		• 32 Mbytes free hard disc
	300MHz Pentium (or better) + 256 Mbytes RAM		

Figure 13: Andor DH520-18T datasheet (page 2)

Specifications

SensiCam High Performance Digital CCD Camera				
Sensor Type	CCD-Interline Progressive Scan with "lens-on-chip"			
Number of Pixels (Resolution)	VGA 640(H) x 480(V)	SVGA 1280(H) x 1024(V)	QE 1376(H) x 1040(V)	
Pixel Size	9.9 μ m x 9.9 μ m	6.7 μ m x 6.7 μ m	6.45 μ m x 6.45 μ m	
Sensor Format	1/2"	2/3"	2/3"	
CCD Temperature	-15°C	-12°C	-12°C	
Full Well Capacity	35,000 electrons	25,000 electrons	18,000 electrons	
Scan Area	6.3mm x 4.8mm	8.6mm x 6.9mm	8.9mm x 6.7mm	
Full Frame Rate	30 fps	8 fps	10 fps	
2x2 Binning Frame Rate	60 fps	16 fps	20 fps	
Readout Noise	13 - 14 electrons @ 12.5 MHz	7 - 8 electrons @ 12.5 MHz	Standard & Low Light Mode High Gain: 4 - 5 electrons @ 16 MHz	Standard Mode Low Gain: 5-6 electrons @ 16 MHz
A/D Conversion Factor	7.5 electrons/count	5 electrons/count	Standard & Low Light Mode High Gain: 2 electrons/count	Standard Mode Low Gain: 4 electrons/count
Quantum Efficiency @ 550nm	42%	42%	65%	
Dynamic Range	12 bit at 12.5 MHz	12 bit at 12.5 MHz	12 bit at 16 MHz	
SNR (dB)	68 dB	69 dB	70 dB	
Spectral Response (monochrome)	290 to 1000 nm	290 to 1000 nm	280 to 1100 nm	
Spectral Response (color)	RGB primary colors	RGB primary colors	N/A	
	VGA/SVGA/QE			
Dark Current	<0.1 electrons/pixel/sec.			
Anti-Blooming	Standard Light Mode: >1000 Low Light Mode: >4 (QE Version)			
Exposure/Delay Time Setting	1ms to 1000 sec			
CCD Quality	Grade 0			
Non-Linearity	<1%			
Binning Horizontal	1 to 8			
Binning Vertical	1 to 32			
Cooling Type	2 stage peltier cooler with forced air			
Lens Mount	C-Mount with adjustable focus			
Blemishes				
Point Defects	0			
Cluster Defects	0			
Column Defects	0			
Dimension	Head: 93(W) x 78(H) x 210(L) mm			
Weight	1.6kg			
Operating Temperature	0 to 40°C			
Storage Temperature	-20 to + 70°C			
Humidity	10 to 90% non-condensing			
High Speed Serial Link				
Standard	10 m coax cable			
PCI Interface Board				
Board	PCI Local bus compatible, revision 2.1, Burst rate 132 Mbytes/s			
Trigger Input	Internal by software, external by TTL level (rising/falling edge); BNC connector			
Power Supply				
Power Supply	110/220 VAC, 50/60 Hz external power supply			
Ordering Information				
The SensiCam High Performance Digital CCD Camera Imaging System comes complete with PCI Interface Board, Standard Coax Cable, Software, Power Supply and Manual				
To order, specify sensor type	VGA Monochrome or Color	SVGA Monochrome or Color	QE Monochrome or Color	



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*Specifications subject to change
with technological advancements.

AL01/03

Figure 14: PCO sensicam datasheet (model used in this study highlighted in yellow)



Solo PIV Products

		Solo I-15	Solo II-15	Solo II-30	Solo III-15	Solo 120XT	Solo 200XT
Repetition Rate (Hz)		15	15	30	15	15	15
Energy^{1,2} (mJ)	532 nm	15	30	30	50	120	200
	266 nm	NA	NA	NA	NA	20	30
Energy Stability² (±%)	532 nm	4	4	4	4	4	4
	266 nm	NA	NA	NA	NA	9	9
Beam Diameter (mm)		3	3	3	4	5	6
Pulse Width³ (ns)		3-5	3-5	3-5	3-5	3-5	3-5
Divergence⁴ (mrad)		< 3	< 3	< 3	< 4	< 3	< 4
Beam Pointing (urad)		< 100	< 100	< 100	< 100	< 100	< 100
Jitter (±ns)⁵		1	1	1	1	1	1

- Optical losses due to optional attenuator will reduce maximum energy by 10%.
- Energy and pulse-to-pulse stability for 98% of shots after 30 minute warm up.
- Full width half maximum.
- Full angle for 86% of the energy, at 1/e² point.
- From Q-switch synch out to light pulse, 98% of 1000 shots.

Physical Characteristics

	Laser Head*			Power Supply		
	Solo I, II, III	Solo 120XT	Solo 200XT	Solo I, II, III	Solo 120XT	Solo 200XT
Length	13.775"/350 mm	22.4"/569mm	24.4"/620 mm	18.15" / 461 mm	19.0"/483 mm	21.2" / 538 mm
Width	7.0"/178 mm	9.24"/235 mm	9.24"/235 mm	7.77" / 194 mm	8.6"/218 mm	10.6" / 269 mm
Height	3.187"/81 mm	4.86"/123 mm	4.86"/123 mm	14.32" / 363 mm	15.0"/381 mm	15.16" / 385 mm
Weight	10 lbs./4.5 kg	34 lbs./15.5 kg	40 lbs./18.2 kg	48 lbs. / 22 kg	53 lbs/24 kg	53 lbs. / 24 kg
Length Umbilical	8 ft / 2.4 m	10 ft / 3 m	10 ft / 3 m			

* Width and height include mounting plate

Operating Requirements

Temperature		50° - 86° F (10° - 30° C)
Relative Humidity		20—80% non-condensing
Voltage		95—240 V, 50/60 Hz
Power	Solo I, II, III	15 Hz-800 watts; 30 Hz-900 watts
	Solo 120XT	1000 watts
	Solo 200XT	1500 watts

Specifications and product offering subject to change without notice.



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Figure 15: New Wave Solo II laser datasheet (model used in this study highlighted in yellow)